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A Study of the Energy Consumption of a Battery Cooling System by Different Cooling Strategies

Justin A. Brumley

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A Study of the Energy Consumption of a Battery Cooling System by Different Cooling Strategies

Justin A. Brumley

**Thesis Submitted
to the Benjamin M. Statler College of Engineering and Mineral Resources
at West Virginia University**

in Partial Fulfillment of the Requirements for the Degree of

**Master of Science in
Mechanical Engineering**

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Department of Mechanical and Aerospace Engineering

**Morgantown, West Virginia
2016**

**Keywords:
Hybrid Vehicles, Energy Storage System Cooling, Hybrid Battery Thermal Models,
Battery Cooling Experimentation**

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Abstract

A Study of the Energy Consumption of a Battery Cooling System by Different Cooling Strategies and Cooling Methods

Justin Brumley

The High Voltage (HV) batteries that are used today in Hybrid Electric Vehicles (HEV), Plug-In Hybrid Electric Vehicles (PHEV), and Electric Vehicles (EV) utilize cooling systems to keep the battery packs within optimal operating temperature ranges. Manufacturers spend a generous amount of money to design these cooling systems to keep the batteries within these safe operating temperature requirements during harsh conditions, such as extreme cold and heat. Desert conditions can reach an average ambient temperature of 40°C. The temperature of the batteries can affect their performance, reliability, and the health of each cell. The Lithium Iron Phosphate (LiFePO₄) battery systems manufactured by A123 Systems Inc. operate between 20-50°C for optimum performance. The thermal distribution among all batteries cells within a battery module is also important. Even a 3-4°C difference in cell temperature can result in reduced performance and potentially damage individual cells. Lithium iron phosphate batteries have a potential safety concern when it comes to temperature; if the temperature is too high, the batteries have a potential to go into thermal runaway and catch fire. This research effort has been conducted at West Virginia University (WVU) to evaluate how different cooling systems compare in cooling batteries during various battery usage cycles. The two systems that were evaluated were a 50/50 ethylene glycol water mixture recirculating coolant system and an R-134A refrigerant system. The research evaluated the impact on battery performance and energy consumption from the system using modeling and simulation. Prototype cooling systems were then fabricated and experiments were conducted using a representative aluminum block to simulate the thermal mass of the battery modules. The experimental data were used to validate the results of the simulation models. Matlab Simulink was used to simulate a PHEV hybrid-electric vehicle and determine the impacts from the thermal cooling system over various drive cycles. The thermal models were validated using experimental bench tests to confirm critical input data and verify that results were valid. The team used this simulation package and model to design the cooling system for the West Virginia University (WVU) EcoCAR 3 PHEV Chevrolet Camaro. Through the results of this thesis it was found that the 50/50 ethylene glycol system would use roughly three times less energy than the R-134a refrigeration system over the aggressive US06 drive cycle.

Dedication

I would like to dedicate this thesis to my family and friends. To my parents, thank you so much for being there for me through school, helping me edit this document, and thank you for helping me to be where I am today. To my friends, thank you for keeping me sane during school and reminding me to have fun every once in a while.

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Nomenclature

$\dot{m}$	Mass Flow Rate (kg/m ³)
ΔT	Change in Temperature (°C)
Δ_{total}	Total Uncertainty
A	Area (mm ²)
Ah	Amp Hours
B	Bias Uncertainty
BatteryCapacity	Battery Capacity (Ah)
BatteryCurrent	Battery Current (A)
BatteryVoltage	Battery Voltage (V)
C	Temperature (Celsius)
C _p	Constant Pressure (kJ/(kg*K))
E	Power (watts)
I	Current (Amps)
K	Temperature (Kelvin)
m	Meter (-)
n	Number of Data Points (-)
P	Precision Uncertainty
Q	Heat Transfer Rate (W)
R	Resistance (ohms)
s	Seconds (-)
SOC	State of Charge (%)

U	Overall Convective Heat Transfer Coefficient (W/m ² K)
V	Voltage (-)
$\delta_{\Delta T_{Air}}$	Perturbed Temperature of Air Uncertainty
$\delta_{\Delta T_{Liquid}}$	Perturbed Temperature of Liquid Uncertainty
δ_A	Perturbed Area Uncertainty
δ_{C_p}	Perturbed C_p Uncertainty
$\delta_{\dot{m}}$	Perturbed $\dot{m}$ Uncertainty
η	Efficiency (%)
σ	Error (-)

List of Abbreviations

AVTC	Advance Vehicle Technology Competition
BCM	Battery Control Module
CAD	Computer Aided Design
CAFEE	Center for Alternative Fuels Engines and Emissions
CFD	Computer Fluid Dynamics
CFM	Cubic Feet per Minute
CSM	Current Sense Module
EDM	Electronic Distribution Module
ESS	Energy Storage System
EV	Electric Vehicle
GHG	Greenhouse Gas
HCFC	Hydro-chlorofluorocarbon
HEV	Hybrid Electric Vehicle
HFC	Hydro-fluorocarbon
HSC	Hardware Supervisory Controller
HV	High Voltage
LFE	Laminar Flow Element
LiFePO ₄	Lithium Iron Phosphate
LV	Low Voltage
MAF	Mass Airflow Sensor
mpg	Miles per Gallon

mpgge	Miles per Gallon Gas Equivalent
MSD	Manual Service Disconnect
OCV	Open Circuit Voltage
PCC	Phase Change Composite
PHEV	Plug-In Hybrid Electric Vehicle
PID	Proportional Integral Derivative
PSAT	Power Simulation Analysis Toolkit
PWM	Pulse with Modulation
SOC	State of Charge
VRB	Vanadium Redox Batteries
W	Watts
Whr	Watt Hours
WVU	West Virginia University

1.0 Introduction

The Advance Vehicle Technology Competition (AVTC) series has been around for over 25 years. This competition is designed to give both graduate and undergraduate students the experience in developing advanced vehicle propulsion and alternative fuel technologies and provide training for the next generation of automotive engineers [1]. EcoCAR 3 is the newest competition in this series where 16 different schools are working to convert a 2016 Chevrolet Camaro into a Hybrid Electric Vehicle (HEV). This competition is four years long and will include the design and manufacturing of a prototype hybrid Camaro. The headline sponsors for this competition are General Motors, the United States Department of Energy, and Argonne National Laboratories. The main objective of EcoCAR 3 is to design a functional performance vehicle that will attract the consumer market while improving the fuel efficiency and reducing criteria emissions and greenhouse gas (GHG) emissions as much as possible. Wallthub released a study in 2015 describing how each state ranked in being eco-friendly and West Virginia was ranked 45th out of the 50 states [2]. The West Virginia University's EcoCAR 3 team wants to show West Virginia and their consumer market that people can have a fun sports car while being eco-friendly.

The West Virginia University (WVU) advanced vehicle team designed a plug-in parallel hybrid electric vehicle architecture. The specific powertrain architecture is referred to as a P3 Hybrid; meaning the electric motor is on the same driveshaft as the engine and is placed somewhere post-transmission. The design includes an internal combustion engine that drives the wheels from the engine bay and an electric motor located under

the vehicle's rear passenger seats. The motor's shaft is at a 90-degree angle to the stock Camaro's driveshaft. The two components that drive the vehicle are a GM 2.4L LEA Ecotec engine that is paired with the stock GM 8L45 transmission and a Parker GVM210-200S electric motor. The Parker motor is coupled to the driveshaft using a Winters differential to transfer power to the main driveshaft of the vehicle. The team wanted 100 percent of the driving torque to be applied to the rear wheels. With the compact design of the new 2016 Chevrolet Camaro, there was limited space to add large and heavy components to the vehicle. However, using Siemens NX Computer Aided Design (CAD) software, the team was able to locate the necessary space to package the components of the hybrid-electric powertrain. In order to power the motor, seven lithium iron phosphate (LiFePO_4) batteries were placed in the trunk of the vehicle. These A123 Systems Inc. (7x15s2p) batteries and supporting components make up the Energy Storage System (ESS). These batteries supply the vehicle with 340 volts and 39.9 amp hours of energy. The block diagram of the vehicle architecture can be seen in Figure 1.

Figure 1: WVU PHEV Power Flow Diagram

A123 Systems Inc. batteries have an optimal operating temperature around normal ambient air temperature found in West Virginia during the summer. According to usclimatedata.com, Morgantown has an average high temperature of 17.1°C throughout the year and an average summer high of 26.6°C [3]. The operating temperature range for a LiFePo₄ battery is between 20-50°C. Regulating the temperature of batteries within this range can be difficult in hot environments such as desert environments where temperatures can reach highs of 51°C. If LiFePo₄ batteries reach a temperature over 60°C, they can go into thermal runaway, in which excessive heat causes more heat generation until the operation ceases or an explosion occurs. Lithium ion batteries have been known to exhibit thermal runaway and explode [4]. To prevent this from happening, cooling is required to ensure the batteries stay in a safe temperature range.

There are two different cooling strategies. The first strategy is passive cooling, in which cooling the component is done by natural convection using ambient air with no control over cooling. The second strategy is active cooling, in which the cooling system controls fans or pumps to keep the components within operating temperature range. The WVU team decided to utilize active cooling, even though the actively cooled systems are more complicated and take up more space than most passive systems. Active cooling uses additional components such as controllers, fans, and pumps whereas passive systems rely on natural convection.

Due to efficiency losses, every component that uses energy generates heat. This makes the use of cooling systems necessary in automotive applications to dissipate this heat generation. This thesis will discuss the difference between the two cooling methods and evaluate which system is more efficient in an automotive application for cooling HEV batteries. The two systems that were evaluated were a 50/50 ethylene glycol system and an R-134a refrigeration system. Experimental results from bench testing were used to verify a representative Simulink model to further improve the accuracy of the full vehicle model and to better predict the vehicle's behavior.

2.0 Background and Literature Review

2.1 Past Coolant Systems

Previous systems of early automobile engines were water cooled, but engineers quickly realized that water would freeze causing engine blocks and other components to break and fail. The first solution to this problem was alcohol [5], which was the first form of antifreeze that could be added to water to bring the freezing point down. Antifreeze is

a term that is used for a chemical additive that could lower the freezing point of the substance and cause it not to freeze. After using alcohol, engineers determined that alcohol increased the corrosion of metal and would cause engines to rust and corrode. In 1856, Charles-Adolphe Wurtz synthesized the antifreeze used today known as ethylene glycol. Ethylene glycol lowered the freezing point, raised the boiling point, and was also used in explosives. Ethylene glycol was not used in automotive applications until 1926, and it is still used today.

Electric vehicle technology has come a long way since 1839 when the first electric vehicle was made by Robert Anderson of Aberdeen in Scotland [6]. Over the last 177 years, vehicles have become more efficient with increased performance. Even though the vehicles are more efficient there are still efficiency losses due to heat. Most mechanical and electrical components used in vehicles today need cooling which can be done many different ways. The most common way components in vehicles are cooled today is by using a 50/50 ethylene glycol/water mixture. Coolants can vary in color depending on their make-up; some of the colors used are orange, green, and pink to signify the different additives used within the mixture. These coolants differ in that each have different types of inhibitors keeping components from rusting and corroding. These inhibitors are typically phosphate, silicate, or borate which forms a barrier coating protecting the components from water corrosion [7]. When mixing glycol with water, a distilled water should be used to insure there is no calcium or mineral mixture that will cause scoring in the components cooling lines that may cause a loss in heat transfer over time.

2.2 Past Refrigeration Systems

In 1940, Packard Company produced the first factory installed air conditioning unit in a vehicle. By 1969 more than half of all new cars had air conditioning. In 1996, auto manufactures were required to switch the air conditioning systems from R12 (Dichlorodifluoromethane also known as Freon-12) to R-134a refrigerant (a refrigerant that is much healthier for the environment). Now, virtually all cars are sold with air conditioning even though Consumer Reports found that using air conditioning on the highway could result in the loss of more than 3 mpg compared to driving with the windows open which had no measureable effect on fuel economy [8].

Several types of refrigerants could be used for an air conditioning unit. The most common refrigerant today is R-134a, a hydro-fluorocarbon (HFC) that does not contribute to the depletion of the ozone layer. Other types of refrigerant are R-410a, R-22, R-407C and R-12. R-410A is another refrigerant that is a HFC that operates at more than 50% higher pressure than R-22 systems [9]. Many commercial screw chillers use R-134a refrigerant because all of the cooling system components have to handle a higher pressure than the normal R-22 refrigerants, but it is more efficient than the R-407C systems. R-22 is very similar to R-407C, but it is a Hydro-chlorofluorocarbon (HCFC) which does impact the ozone layer. R-22 is commonly known by its brand name, Freon. Freon was widely used until 2010 when this product was discontinued due to the impact on the ozone layer. By 2015, R-22 was federally mandated to be discontinued across the United States and is no longer used. After 2015, if a refrigerant system previously using Freon needed to be replaced, it would be replaced with R-407C, the closest refrigerant in thermal characteristics. R-407C is a HFC that closely resembles that of the R-22

refrigerants. R-407C is a high-glide refrigerant that is not as popular in the industry today due to its lower efficiencies [10].

2.4 Literature Review

2.4.1 Introduction

Thermal cooling for battery systems for automotive applications research has been an important topic due to the steady increase of hybrid and electric vehicles being produced and sold in today's market. Research has been done on battery cooling systems from universities around the world including cooling plates, phase change material, and control strategies. The past research has guided the topics of discussion and research basis of this thesis. The design of a battery cooling system includes many steps and each step requires a numerous amount of work. The first step to designing a cooling system is to define the thermal requirements. One way to discover these requirements is by using a simulation model that can output the amount of energy lost to heat. Matlab Simulink was used by the WVU EcoCAR 3 team to model the entire vehicle including the battery system and its thermal characteristics.

2.4.2 Vehicle models and drive cycle reasoning

The vehicle model that was developed by the WVU EcoCAR 3 team was simulated using different "drive schedules" also known as drive cycles. A drive cycle is a speed versus time trace that the simulation model uses to represent a vehicle driving on a road. There are many drive cycles that could be used in designing a vehicle. The EPA uses a 5-cycle driving schedule (FTP (City), Highway, US06 "high speed", SC03 (AC), and FTP @cold Temperatures) to test emissions and fuel consumption during dynamometer testing [11]. Table 1 compares the different drive cycles to one another showing the

different attributes between each cycle. The EPA uses a variety of drive cycles to ensure a broad amount of data is collected from all spectrums. Crain discussed that the EcoCAR 3 program only uses a 4-cycle test which uses a subset of the EPA test cycle to collect emissions and energy consumption during the EcoCAR 3 competition [12]. The 4-cycle test addresses real-world driving conditions without having to directly address A/C use and cold ambient temperatures. Crain also explained the 4-cycle test that was constructed of four separate driving cycles which included HWFET, US06 City, US06 Highway, and the 505 portion of UDDS cycle from federal test procedures. The weight of the 4-cycle drive schedule is 12% HWFET, 29% 505, 14% US06 City, and 45% US06 highway. The US06 drive cycle is one of the more aggressive drive cycles chosen to test the battery system. Graphs of each cycle can be found in 12.0 Appendix C. The 4-cycle test method was embraced in the vehicle simulations to design the vehicle to the needs of the EcoCAR 3 competition.

Table 1: Drive Cycle Experimental Parameter Matrix

Driving Schedule Attributes	Test Schedule				
	FTP (City)	Highway	US06 "High Speed"	SC03 (AC)	FTP @ Cold Temp
Trip Type	Low speeds in stop-and-go urban traffic	Free-flow traffic at highway speeds	Higher speeds; harder acceleration & braking	AC use under hot ambient conditions	City test w/ colder outside temperature
Top Speed	56 mph	60 mph	80 mph	54.8 mph	56 mph
Average Speed	21.2 mph	48.3 mph	48.4 mph	21.2 mph	21.2 mph
Max. Acceleration	3.3 mph/sec	3.2 mph/sec	8.46 mph/sec	5.1 mph/sec	3.3 mph/sec
Simulated Distance	11 mi.	10.3 mi.	8 mi.	3.6 mi.	11 mi.
Time	31.2 min.	12.75 min.	9.9 min.	9.9 min.	31.2 min.
Stops	23	None	4	5	23
Idling time	18% of time	None	7% of time	19% of time	18% of time
Engine Startup*	Cold	Warm	Warm	Warm	Cold
Lab temperature	68-86°F			95°F	20°F
Vehicle air conditioning	Off	Off	Off	On	Off

Other research has been done using custom drive cycles and other simulation tools to further design a vehicle to more specific specifications. Czapinski used finite element analysis and computational fluid dynamics (CFD) to predict temperature distributions in the Embry-Riddle Aeronautical University Saturn Vue [13]. Czapinski used Powertrain Simulation Analysis Toolkit (PSAT) to run electrical calculations to find load cases to test how their 4x25s2p A123 Systems Inc. lithium-ion batteries would perform over specific drive cycles. Czapinski used a drive cycle that resembled a towing scenario. To simulate towing 680 kg at a constant speed of 72 km/hr for 20 minutes up a 3.5% grade. This towing cycle proved to be their worst scenario which created 66.7 W of heat in their ESS system. Using an iterative method in their models they were able to design a cold plate that would sufficiently cool their batteries. The Saturn Vue was run through the towing drive cycle because it could be used to tow small trailers. Czapinski concluded saying it may be beneficial to perform an analysis using time dependent variables to help speed up simulation time since Czapinski had troubles with computation time using CFD and PSAT.

2.4.3 Thermal modeling and battery thermal calculations

The full vehicle model of the Chevrolet Camaro with the A123 batteries was used with the ESS thermal models to simulate how the batteries would perform thermally during operation. These calculations were based on I^2R losses to calculate the amount of heat generation. The model also used initial parameters that were given by A123 Systems Inc. to formulate the model. Schweitzer, Wilke, Khateeb, and Hallaj used a 0-D numerical simulations to model phase change thermal management systems for lithium-ion batteries [14]. A 2.6 Ah battery module, with 10 cells in series 4 cells in parallel, (10s4p)

battery configuration, was used for their experimental validation. They used the I^2R loss calculations to find the irreversible heat generation during charging and discharging. Through experimentation, they found that using the I^2R calculations closely resembled the actual heat generation and could therefore validate that their models were correct. This method was used in the current work since critical information was given to the team for resistance values for the LiFePO_4 batteries.

Baccinno, Marinelli, Norgard, and Silvestro performed experiments to validate their dynamic model on vanadium redox flow batteries (VRB) [15]. Through their experiments, they were able to make their models more precise by finding out that the usable energy for these batteries were much lower than expected. The theoretical energy for these batteries were 320 kWh but they found that the usable energy was only 190 kWh.

Another significant study completed by Cae used modeling to simulate the full vehicle powertrain and thermal management system for their 20 kWh battery pack [16]. This helped clarify how the vehicle would perform over various drive cycles. Cae was able to use a simple thermal model to optimize his control strategies for his cooling loops to be as efficient as possible. Once the implementation of his designs were placed into the car he was able to validate his models by comparing his results to the final product implemented into the vehicle. He found that his model results were conservative and the cooling system ended up performing better than expected. To better predict his thermal system a validation using a small scale setup could have been done in order to not overdesign the thermal cooling loop.

2.4.4 Studies on ethylene glycol

The thermal cooling models used in this work use the coolant properties and heat transfer calculations to simulate the cooling system. There are many types of coolants and there can be different coolant to water mixture ratios. Studies have been performed showing the thermal characteristics and performance with different mixture ratios. Baudot and Odagescu discussed the thermal properties of ethylene glycol [17]. Their results on thermal properties for ethylene glycol were performed using various ratios between 40% and 50% coolant to water mixtures. These percentages were used because lower than 40% ethylene glycol would allow the coolant to freeze in extreme cold conditions and higher than 50% ethylene glycol would be toxic if spilled in the environment. They found that as ethanediol, the systematic name for ethylene glycol, concentration increases the average critical cooling and warming rates decrease, meaning that 50/50 ethylene glycol will not cool as fast as 40/60 ethylene glycol. The matrix for the coolant mixture to water can be seen in Table 2. The 50/50 ethylene glycol solution was chosen for the experimentation because the 50/50 ratio is widely used in vehicles today.

Table 2: Baudot and Odagescu Table for Ethylene Glycol to Water Mixture Matrix [17]

Table 1

Values of T_m , of the parameters (q_{max} , k_4) necessary to evaluate the critical cooling rate v_{ccr} (T_m and q_{max} are given as mean value $\pm$ SE) and calculated values of v_{ccr} and v_{cwr} for all the concentrations of ethylene glycol tested

Ethylene glycol % (w/w)	T_m (°C)	q_{max} (%)	k_4 (°C/min)	v_{ccr} (°C/min)	v_{cwr} (°C/min)
40	-22.35 ± 0.66 ($n = 3$)	21.3 ($n = 1$)	360	569	1.08×10^{10}
43	-26.43 ± 0.48 ($n = 3$)	20.02 ± 2.02 ($n = 3$)	67	105	2.85×10^7
45	-28.47 ± 0.42 ($n = 3$)	15.78 ± 1.98 ($n = 3$)	45	63	1.04×10^6
48	-32.62 ± 0.02 ($n = 3$)	11.45 ± 0.23 ($n = 3$)	14	18	1.08×10^4
50	-36.17 ± 0.46 ($n = 6$)	12.81 ± 0.15 ($n = 6$)	8	11	853

2.4.5 Thermal test bench experimental setup

The validation for dynamic modeling should be done to ensure the results collected are creditable. Liu, Lan, and Chen discussed how they validated their thermal models through experimentation [18]. For their experiment, a 3.2 V 50 Ah Li-ion battery pack (1s5p) was utilized. They used two forms of thermal validation, Thermocouples and a FLUKE Ti25 infrared thermal imaging camera, in their experiments to ensure the 3D computer fluid dynamics (CFD) simulations matched the collected experimental data. A total of 20 K-type thermal couples, with an accuracy of $\pm 1.5^{\circ}\text{C}$, were placed inside and around the battery pack to record not only the outside temperatures but temperatures inside the battery to better understand the distribution of temperature. The infrared thermal imaging camera was used to validate that the temperatures acquired by the thermocouples reading were correct values. Liu, Lan, and Chen recorded the data from these thermocouples once every 10 seconds or 1/10 Hz. Using a smaller frequency of data will simplify post processing over long experimental times but decreases the accuracy of the measurements. The accuracy of the thermocouples was a concern since the range of temperatures being collected were so close. For this paper the experimental design used T-type thermocouples for better precision.

2.4.6 Cold plate design

The design of the cooling plate for the WVU EcoCAR team was done to efficiently cool the batteries while maintaining ease of fabrication. Jarret and Kim discussed how they optimized cooling plates based on dimensions and geometries of cooling paths [19]. They created an optimization algorithm with 18 different geometric design variables to run through CFD simulations. After running through all the possibilities of the 18 different

variables, they found that a cold plate with the lowest pressure drop was able to cool better than those with high pressure drops. They also found that cold plates that have narrow inlet and outlets with a wide channel throughout the cold plate perform better than ones with a very open inlet and exit. Because of the complex manufacturing of this method it was not used in order to simplify the fabrication of the experimental cold plate. The cold plate that was used in the research uses a constant volume channel throughout the cold plate.

Smith, et al. [20] used a method to design a cooling plate for optimal operating conditions by comparing different flow rates and inlet temperatures. They ran one cold plate through multiple iterations using different load conditions by changing inlet temperature and flow rate. From the data collected through the CFD simulation, they plotted the change in temperature vs flow rate and flow rate vs pressure drop through the cold plate. From these two analyses, they found where the two functions converged to find the optimal design conditions for that specific cold plate.

2.4.7 Overall Heat Transfer Coefficient calculations

Modeling a radiator in a simulation is difficult to do without critical information and known parameters. There are multiple ways to find the overall convective heat transfer coefficient (U) values of a component. Onda, et al. [21] performed tests to find the thermal behavior of a small lithium-ion battery during rapid charge and discharging cycles. The battery was suspended in air by the lead wires on the terminals after being heated. The battery was heated past its operating range to 90°C to collect a full range of U values. The temperature was recorded against time until it reached ambient air of 25°C. By using

a calculation not given in his paper, he was able to find the overall convective heat transfer values of the cylinder suspended in air.

2.4.8 Control strategy

The development of a worthy control strategy is also important when designing a cooling system. Oliveira, et al. [22] performed tests on variable speed refrigeration systems that were <1 kW of cooling capacity for food refrigeration. The objective of their research was to keep the system running continuously by varying compressor speed and expansion valve opening size. They found that by keeping the refrigeration system running the whole time and changing the speed of the compressor, they were able to save energy over the thermostatic ON/OFF approach. Increasing the efficiencies and saving more energy relates to fuel economy in miles per gallon (mpg). Significant research has been done to use less energy and increase fuel efficiency in thermal cooling systems. Barr [23] examined two parameters in a cooling cycle to increase the efficiency of the overall system to use less energy. Barr used a variable fan speed and variable speed pumps to better control his cooling system and only use a necessary amount of energy to cool the batteries in his experiment and save fuel. He found that by optimizing the speed of the fan and pumps, he could save eight miles per gallon gas equivalent (mpgge) and improved the vehicle EV range by two miles. MPGGE units are typically used to compare different fuels on an energy equivalent basis. Barr showed that there is refinement that could be done to a control strategy to save energy.

2.4.9 Active vs. Passive Cooling

Pesaran discussed thermal modeling capabilities for hybrid electric vehicles [24]. He discussed how Lithium Ion batteries operate best at temperatures between 25 and

40°C and have a temperature distribution of <5°C between modules. Pesaran used the approach of finite element analysis to predict the temperature distribution to ensure the packs have an even distribution. Pesaran also discussed thermal management using passive and active cooling [25]. He found that battery packs need to be actively cooled and regulated to keep the batteries within the desired operating range. He discussed how having a proper thermal design can give better performance over a passively cooled system especially in extreme hot or cold climates.

2.4.10 Phase change and passive cooling

Some new research has been done on phase change cooling, a passively cooled system. Schweitzer, et al. [14] have performed experimentation and created numerical models to simulate this phase-change composite (PCC) thermal management system with lithium-ion batteries. T-type thermocouples were utilized to monitor the cobalt oxide cathode batteries (10s4p) during discharging and charging. Schweitzer, et al. used C-ratings for declaring the discharge rate that they used. The capacity of a battery has a 1C rating, meaning discharging a battery at a 2C rate will discharge the battery twice as fast. When discharging the batteries at a 2C rating Schweitzer, et al. found that the phase change material expanded the thermal capacity of the battery pack and caused the battery temperature to increase less than a battery without the PCC material. During charging or low discharge rates, of 1C or less, the batteries would not heat up enough for a cooling system to be needed. This method proved to be an efficient way to keep batteries cooler during operation without the use of energy consuming components such as a pump or fan. Li, Qu, He, and Tao found similar results to Schweitzer, Wilke, Khateeb, and Hallaj but used different sets of PCC material to test which system was more efficient

at cooling [26]. They also found that with utilizing T-type thermal couples, that using a copper metal foam that was saturated with PCC resulted in lower and more even battery temperatures then just using only PCC material. They also showed that it would require a load greater then 1C for the PCC to melt.

2.4.11 Conclusion

Previous research has improved the current research below. From prior research, it was shown that extended use of high voltage batteries needed to be actively cooled to help control the temperature of the batteries. There are multiple strategies for cooling a component, one commonly used in vehicles is a liquid cooling system. Past research on coolant has provided evidence that 50/50 ethylene glycol was best to use because it is not toxic if spilled, will not freeze in extreme conditions, will not corrode metal, and is also commonly used in liquid cooling systems. Phase change cooling strategies were also considered, but during extended use, it will allow the batteries to overheat. Studies have shown that thermal models can closely resemble actual systems if validated correctly. To validate the thermal models used in this research, experiments were performed on different cooling systems. T-type thermocouples were used because they are more accurate than K-type thermocouples for the range of temperatures experienced. The data for the experiments was recorded in a 10Hz rate, compared to previous research which used a 1/10Hz rate. This was done to ensure no data or spikes of data were missed. To get data that closely resembled the actual system, tests were performed to find the overall heat transfer coefficient of the radiator so the values could be placed into the model. The approach was used from Onda, et al. [21] for the experiment to find the overall heat transfer coefficient. The control strategy was chosen to be a simple multi-mode on/off

system instead of a complex PID controller because the on/off system could be easily implemented with the equipment in the lab. For the current research, the focus was on small scale testing for simple model validation, prior to integration, using proven methods to help lower fuel economy, and design a proficient cooling system.

3.0 ESS Design

3.1 Enclosure Design

The design of the ESS created its own challenges. In order to fit the large battery modules into the vehicle, work was done to design an ESS that not only fit in the vehicle but could pass the rigorous requirements placed by the EcoCAR 3 competition. Some of these requirements were that the ESS had to fit in a single enclosure, endure a 20g longitudinal load, and be sealed so no gasses expelled from the batteries could reach occupants in the vehicle. The design of the ESS took these requirements into account and was still able to fit two pieces of luggage with a size of 533mm x 178mm x 356mm within the trunk. The cooling system was designed based on the results of this research study, the final design of the ESS utilized a 50/50 ethylene glycol cooling system that was actively controlled.

Aluminum was chosen to fabricate the enclosure due to its lightweight nature, non-corrosive properties, and it is easy to machine. The team designed the ESS structure to enclose all of the components that are part of the ESS. This included the Battery Control Module (BCM), Electronic Distribution Module (EDM), Current Sense Module (CSM), Manual Service Disconnect (MSD), along with the 7 battery modules. Every component

was connected by Low Voltage (LV) wires and High Voltage (HV) Cable inside a single enclosure. A CAD image of the ESS enclosure can be seen in Figure 2.

Figure 2: ESS Outside Enclosure

The Camaro had limited placement options for the ESS system. The easiest place to install the batteries was the trunk of the vehicle, which allowed convenient access and installation. The layout of the battery pack is shown in Figure 3. Siemens NX was also used to provide wire routing through the enclosure. High Voltage (HV) wires require additional volume in the enclosure. Ensuring each wire does not exceed the minimum bend radius and each wire has adequate space is important before manufacturing the product.

Figure 3: ESS Part Callout of Inner Enclosure

3.2 Cooling Plate Design

The design of the cooling system was conducted in parallel with the design of the ESS enclosure. Due to the vehicle's trunk being so small and HV electrical wires potentially being located close to conductive liquid the cooling system was designed to work outside of the ESS enclosure, isolated from the batteries. The cooling plates were designed to not only cool the batteries but to add structural integrity to the enclosure allowing the ESS to be made lighter and more compact. An exploded view of the ESS enclosure can be seen in Figure 4. The cooling plates were used as the base for the ESS enclosure so the ESS could be installed quickly and to reduce weight. This also allows the ESS to be removed without removing the cooling system. The full cooling system can be seen in Figure 5 with the cold plate, piping, radiator, ESS, and reservoirs.

ESS Enclosure

Figure 4: Exploded View of ESS Enclosure with Cooling Plate

Figure 5: ESS Cooling System Overlay with Vehicle

The ESS cooling system uses a single copper line routed throughout the cooling plate to cool the batteries. Cold coolant enters one end of the line and the heated coolant exits at the other end. The cooling plate was designed to allow for even cooling for each

battery module. One line of cool water passes through each battery before warm water comes back through each battery. Battery 7 can be seen in Figure 7 and is the last battery to receive cold water so it's the first battery to touch warm water. This was done to keep each battery as close in temperature to one another as possible by making sure the first battery doesn't get just cold water and the last battery getting hot water all the time. This is essential for keeping the battery cells healthy and to avoid premature failure of a single module. A bleeding valve was placed at the highest point in the system to evacuate air from the system during installation or servicing. An illustration of the bleed valve and port connections can be seen in Figure 6.

Figure 6: Bleeding Port and Port Connections for ESS Coolant System

Figure 7: ESS Cooling Pipe Routing

3.3 Thermal Model Design

The design of the ESS was based on results from the full vehicle Simulink model. The model was constructed using equations to represent the operation of any given component in the vehicle, as well as the ESS and its cooling system. The full vehicle model was used to simulate a real world vehicle and provide data on each included component, giving results on how the vehicle performs over time. By placing all the components into one vehicle model and running the vehicle on a standardized drive cycle, multiple vehicle designs could be made quickly without actually manufacturing the product. By using this method, the team was able to select components that would best

suit the vehicle. For simplicity, only the ESS system, shown in Figure 8, will be explained in further detail. For this study the ESS block from the vehicle model was unitized and modified to include the different cooling systems. Creating subassemblies with a uniform structure makes it easy to generate a library of components which can be easily placed into the model without significant changes.

Figure 8: ESS Block

3.3.1 Model Initiation File

The Simulink model uses a model initialization function that runs the initialization file (11.0 Appendix B) “Battery_Design.m” to get all of the design parameters for all the components in the model. By using an initialization file all of the parameters are in one place and the variables can quickly be changed throughout the model. A naming convention was used to standardize the work that was done in order to make it easy for anyone to understand. An example of this is the variable name Fan_CFM which is the Fan airflow through the radiator with the units of CFM.

3.3.2 Thermal Model Structure

The ESS block is separated by subsystems that represent each part of the Battery System model for the ESS shown in Figure 9. The battery calculations subsystem uses inputs for the current, battery resistance, and battery open circuit voltage (OCV). Battery resistance is a function of battery temperature, voltage, and state of charge (SOC).

Battery OCV is a function of battery temperature, voltage, and SOC. The battery calculation subsystem outputs battery SOC, and battery voltage. The battery lookup tables input current, temperature, and SOC and looks up specific values based off of specifications given by the manufacturer. Further detail will be provided later on the thermal system subassembly. The last system is the Battery Management System which inputs all the variables present and outputs results for post processing, such as battery temperature, battery SOC, battery voltage, and energy consumption just to name a few. Using this structure makes the model easy to understand in collaboration with others.

Figure 9: ESS Layout

3.3.3 Battery Calculations

Other than initial variables that are set by the initialization file, there is only one parameter that is needed for the ESS to run an analysis. Using the current drawn from the batteries along with the initial battery capacity and SOC, the current SOC can be calculated using Equation 1. An illustration of the “Battery Calculations” block can be seen in Figure 10. The logic blocks in the battery calculation subsystem are used to limit the batteries from exceeding 100% SOC and dropping below 20% SOC. A123 Systems Inc. recommends to keep the batteries above 20% SOC to protect the lithium iron phosphate cells.

Equation 1: SOC Calculation

$$\text{SOC (\%)} = \frac{\int \text{Battery}_{\text{Current}} dt (\text{Ah})}{\text{Battery}_{\text{Capacity}} (\text{Ah})} + \text{Initial_SOC (\%)}$$

The battery voltage was calculated using Equation 2. The Battery OCV and Resistance values can be found by using tables that are in the “Battery Lookup Tables” subsystem. By taking in Battery Temperature and SOC, the resistance and OCV can be found. The “Battery Lookup Table” block notation can be seen in Figure 11. This data for the lookup tables was provided by A123 Systems Inc. and will not be discussed in further detail due to non-disclosure agreements. The lookup tables take in Battery SOC and temperature per Lithium Iron phosphate cell and outputs the resistance values that correspond.

Equation 2: Battery Voltage Calculation

$$\text{Battery}_{\text{Voltage}} (V) = \text{Battery}_{\text{Current}} (A) * \text{Resistance (ohm)} + \text{Battery OCV}$$

Figure 10: Battery Calculations

Figure 11: Battery Lookup Tables

3.3.4 Battery Thermal Calculations

To show the thermal characteristics, a subsystem was developed to model the battery system along with the cooling systems. The thermal model subsystem that was used can be seen in Figure 12. This included multiple subsystems and a state flow controller to represent how the batteries would heat and cool. The State flow controller outputs pump flow rate, airflow, pump energy consumption, compressor on/off state, and compressor energy consumption. Rate limiter blocks are used to give realistic start and stop times that the component would take to change states. The subsystem starts by using the “Battery Thermal Block” which calculates the energy the batteries create in heat. The “Battery Thermal Block” is connected to either the 50/50 ethylene glycol or refrigeration systems which are used to cool the battery. The thermal signal that connects each system uses ports that are simply indicated by an “A” or a “B”. The state flow system is used to control parameters inside of the cooling systems.

There are two ways to determine the amount of energy the batteries will generate in heat due to efficiency losses. The first method is to use Ohm’s Law with voltage, current and efficiency (Equation 3) to find thermal energy. The second way is to use Ohm’s Law that uses battery resistance to find the energy lost to heat (Equation 4).

Equation 3: Ohm’s Law using Voltage and Efficiency Loss

$$E = V * I * (1 - \eta)$$

Equation 4: Ohm’s Law using Resistance

$$E = I^2 * R$$

Since the efficiency of the battery system was not provided from A123, Equation 4 was used to calculate the energy created in heat. The Simulink Model for this can be seen

in Figure 13. After using this method, it was found that the batteries were roughly 98% efficient because of the low resistance values. Even though this is very efficient the batteries losses still result an average of 279.1 W of heat that needs to be dissipated on a US06 drive cycle. A gain block was used to alter the amount of battery modules being tested during a drive cycle. For the experimental results the value of this block was 1/7 and during simulation of the ESS system this value was one. This was completed to limit the amount of power being placed in each module to give better control over the simulation.

Figure 12: Thermal Model Block

Figure 13: Battery Thermal Calculation

The energy that is created from the efficiency losses pass through an ideal heat flow source that Simulink translates to a thermal function source. This source has many signals that are carried through the model such as thermal mass, thermal capacitance and temperature signals where it can be cooled or heated using additional systems. The thermal signal is connected through mechanical ports to other subsystems such as the 50/50 ethylene glycol system or refrigeration system. These other systems will use this thermal signal to lower the battery temperature. Sensors such as ideal temperature sensors can be used to detect the temperature at any point within the source. The ideal temperature sensor is in the form of °K. To make reading the values easier this value is changed to °C. The battery thermal block subsystem uses a thermal mass block to give the module weight and thermal capacitance. This block requires two parameters, mass and specific heat. The mass value uses the variable “BatteryTotalMass”, which is the total mass in kg of all 7 batteries, multiplied by the Module gain. Using the “ModuleGain” variable again helps control the amount of batteries being analyzed. The specific heat

takes the heat capacity of a cell in J/K and divides in the mass of an individual cell in kg to give a specific heat of J/(kg*K) a table of these parameters can be seen in Table 3.

Figure 14: Battery Thermal Block

Table 3: Battery Module Subsystem Parameter

Battery Module					
Block Name	Settings tab	Parameter	Variable	Value	Unit
Battery Mass	Parameters	Mass	BatteryTotalMass*ModuleGain	100.8	kg
		Specific Heat	HeatCapacitycell/Mass	1222.9	J/(kg*K)
	Variables	Temperture	Initial_Battery_Temp	293.15	K

3.3.5 State Flow Controller

To control the cooling systems a simple State flow diagram is used. The two control strategies used for this study can be seen in Figure 15 and Figure 16. Every state flow needs an initialization block. For this control method the “off” block is selected. The lines that connect the blocks are used like if statements. From the initial state if the temperature of the batteries increase above 30°C then the system will enter state two and turn the system on with 50% fans. If the temperature increases past 32°C the system will enter state 3 and turn the fans to 100% fan speed thus using more energy. If the system starts to cool below 28°C the system will return to state one and turn off. Each block has set values for each variable being output from the state flow. In the off state each value is set

to zero which represents the component being off. The “on” and “on1” block sets the parameters to represent the cooling system being on with 50% and 100% fan speed. The state flow is also used to collect how much energy the system is using in watts during a simulation.

Figure 15: State Flow Control Strategy One

Figure 16: State Flow Control Strategy Two

3.3.6 50/50 Glycol Blocks

Once the battery block and heat source has been created, additional subsystems can be added to the cooling system (Either by adding additional heat sources such as motors or inverters or adding cooling subsystems). Each cooling system is provided with its own subsystem block which can be taken in and out of the system. The block set is controlled using a state flow diagram. From the controller the parameters “Fan_CFM” and

“Pump_GPM” are used inside of the system to calculate the amount of energy that can be dissipated by the system. The parameters being input to the subsystem are shown in Figure 17.

Figure 17: 50/50 Ethylene Glycol Mixture Cooling

A 50/50 ethylene glycol subsystem is shown in Figure 18 and is composed of four major components. The first component is the cooling plate which is connected directly to the battery through a thermal signal port. The cooling plate is what dissipates energy from the battery block. The second component is the radiator which dissipates the heat from the batteries to ambient air. The third component is the pump which moves the liquid through the system. The fourth component that a 50/50 ethylene glycol system needs is piping or hosing which allows the fluid to pass to each component. The model has sensors throughout the system to monitor the temperature and pressure of each location in the cycle.

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block for pressure drop through the piping. This will also help calculate velocity of the fluid through the pump.

Figure 19: Cooling Pump Block

Table 4: Cooling Pump and Control Subsystem Parameters

Cooling Pump and Control					
Block Name	Settings tab	Parameter	Variable	Value	Unit
Colling Pump	parameters	Characteristic longitudinal	Pump_pipe_Length	0.1852	m
		Pipe cross-sectional area	Pump_CrossSection_Area	7.13E-05	m^2

The cold plate subassembly uses conduction blocks along with mass blocks to accurately represent how the cold plate works. The batteries conduct heat to the copper piping and the aluminum plate itself. The main blocks that create this subassembly with the corresponding parameters can be seen in Table 5. Each of the thermal mass blocks take into consideration the specific heat of the material to add a thermal capacitance to the system. The thermal mass blocks were set using the specific heat for copper which is 360 J/(kg*K) and the thermal specific heat for aluminum which is 910 J/(kg*K). The mass of copper was determined to be 0.53 kg which is based off the length of piping

through the cold plate. The mass of the aluminum plate was 2.04 kg which was based on the geometry and thickness of the cold plate. This is important to add, otherwise the system would not recognize how much material to warm or cool. Conductive heat transfer blocks were used to represent the actual transfer of heat from one material to the other. The three parameters for this block are area of contact, thickness of material, and thermal conductivity of the material. An illustration of this block set can be seen in Figure 20. The orange thermal signal source connects to the battery thermal source. This converts the thermal energy dissipated by the cooling system block into energy taken away from the battery pack.

Figure 20: Cold Plate Subassembly

Table 5: Cold Plate Subsystem Parameters

Cold Plate					
Block Name	Settings tab	Parameter	Variable	Value	Unit
Copper to Aluminum Conduction	Parameters	Area	Coolingplate_Copper_Conduction_Area	0.0448	m ²
		Thickness	Coolingplate_PipeThickness	0.0016	m
		Thermal conductivity	copper_conductivity	400	W/(m*K)
Aluminum to Battery Conduction	Parameters	Area	Coolingplate_Aluminum_Conduction_Area	0.0387	m ²
		Thickness	Coolingplate_Aluminum_Thickness	0.0127	m
		Thermal conductivity	ThermalConductivityAlum	205	W/(m*K)
Copper Piping	Parameters	Mass	Coolingplate_Pipe_Mass	0.5304	kg
		Specific Heat	copper_specific_heat	390	J/(kg*K)
	Variables	Temperture	Initial_Battery_Temp	293.15	K
Cold Plate Aluminum	Parameters	Mass	Coolingplate_Plate_Mass	2.0412	kg
		Specific Heat	SpecificHeatCapacityAlum	910	J/(kg*K)
	Variables	Temperture	Initial_Battery_Temp	293.15	K
Cooling plate Copper	Geometry	Longitudinal length	Coolingplate_Length	1.4986	m
		Hydraulic diameter	Coolingplate_Hydraulic_Diameter	0.0095	m
		Cross-sectional area	Coolingplate_CrossSection_Area	7.13E-05	m ²
	Effects and Initial Conditions	Initial fluid temperature inside the pipe	Initial_Battery_Temp	293.15	K

In comparison, the radiator subassembly acts the same as the cold plate which can be seen in Figure 21. The radiator is represented by a pipe block which transfers the energy from the liquid lines to energy that can be dissipated by the fan that is shown in Figure 22, and radiator fins. The values and variables used for this subassembly can be seen in Table 6 and Table 7. To limit the radiator from using a negative heat transfer value, the parameter “min allowable heat transfer coefficient” was set to zero. This

subassembly uses a controlled convective heat transfer block that takes in a specified overall heat transfer coefficient value (U) for the radiator. The U value is an input from a lookup table shown in Figure 23 and the values for this table are talked about in section (7.1 Radiator Overall Convective Heat Transfer Coefficient Determination). More about how the U values were found and the experimental setup will be discussed in section (3.3.7 Refrigeration System Model). Using the actual U values determined through experimentation for the radiator gives better results to confirm the analysis.

Figure 21: Radiator Block

Figure 22: Radiator Fan Subassembly

Table 6: Radiator Subsystem Parameters

Radiator					
Block Name	Settings tab	Parameter	Variable	Value	Unit
Fluid Path Radiator	Geometry	Longitudinal	Radiator_Length	3.3528	m
		Hydraulic	Radiator_Hydraulic_Diameter	0.0095	m
		Cross-sectional area	Radiator_CrossSection_Area	7.13E-05	m^2
	Effects and Initial Conditions	Initial fluid temperature inside the pipe	AirTemp	293.15	K

Table 7: Fan Subsystem Parameters

Fan					
Block Name	Settings tab	Parameter	Variable	Value	Unit
Controlled Convective Heat Transfer	Parameters	Area	Radiator_Surface_Area	3.3528	m
		Min allowable heat transfer coefficient	0	0	W/(m^2 *K)

Figure 23: Look Up Table for Overall Convective Heat Transfer Coefficient

The temperature and pressure sensors that are placed throughout the system can be seen in Figure 24. This uses a Pressure & Temperature sensor block to translate the Thermal liquid signal into independent signals that can be logged in the Matlab workspace. The use of these probes become helpful when verifying the system with bench testing.

Figure 24: Temperature Probe Block

3.3.7 Refrigeration System Model

One of the main goals of this work is to compare the 50/50 ethylene glycol system to the R-134A refrigeration cycle. This cycle used a subassembly block just like the 50/50 ethylene glycol system. The difference in the refrigeration system is that it uses a compressor instead of a pump, and it has an additional part which is the expansion valve. Since both of these systems use the same cold plate (evaporator) and radiator (condenser), each system uses identical blocks for these components with the exception that each use a pipe block that is for two phase systems rather than single phase. The parameter tables for each subsystem can be seen in the appendix in Table 17 and Table 18. The refrigeration cycle block that is used in the “Thermal System” subassembly can be seen in Figure 25.

Figure 25: Refrigerant Cycle Cooling

Refrigeration systems require an expansion valve. The expansion valve performs as a spray nozzle that separates the high pressure side from the low pressure side of the system. The expansion valve forces the refrigerant to change from a saturated liquid into a liquid/vapor mixture. Once the fluid is through the expansion valve it passes through the evaporator turning the liquid/vapor mixture into saturated vapor or a superheated vapor. The phase change dissipates energy from the batteries to the refrigerant thus cooling the

battery module. The refrigeration system subassembly can be seen in Figure 26 and an example of a refrigeration cycle plotted on an enthalpy curve can be seen in Figure 27. The resemblance between the 50/50 glycol system and the refrigeration system can be seen in the subassembly with the addition of the expansion valve. The expansion valve subassembly can be seen in Figure 28 and the parameters used can be seen in Table 8. The variable expansion valve uses a feedback loop to control how much the valve is opened naturally from thermal expansion depending on temperature during operation.

Figure 26: Inside the Refrigeration Block Subsystem

Table 8: Expansion Valve Subsystem Parameters

Expansion Valve					
Block Name	Settings tab	Parameter	Variable	Value	Unit
Variable Local Restriction	parameters	Cross-sectional area at ports A and B	$\pi \cdot \text{pipe_diameter}^2 / 4$	7.13E-05	m ²
		Characteristic longitudinal length	0.1	0.1	m ²
		Flow discharge coefficient	0.7	0.7	-
		Critical Reynolds number based on restriction diameter	12	12	-

The thermal expansion on the copper capillary tube, used as the expansion valve on the refrigeration system, has a range of opening sizes that vary with the temperature. A study was done to determine the opening size by measuring the capillary cross sectional area at different temperatures. This was done by using a Dino-Lite Pro camera where the diameter of the opening could be measured. By using Dry ice the capillary tube was cooled to -25°C. The opening cross sectional area of the capillary tube at -25°C was on average 0.14mm². The other end of the spectrum was at 16°C where the opening size was on average 0.21mm². The cross sectional area values were taken at incremental temperatures ranging from -25 to 16°C which is well outside of the range, 0 to 10°C, in which the capillary tube will operate. The valve controller block takes in temperature and translates it into the opening size in millimeters for the variable expansion valve block so it can receive the appropriate value for the opening depending on temperature change. The pictures in Figure 29 and Figure 30 show the opening sizes taken with the Dino-lite Pro camera. Figure 29 was taken when the capillary tube was at -15°C and Figure 30 was taken when the capillary tube was at 16°C. In Figure 31 shows a graph of the cross

sectional area with respect to temperature that the model uses to depict the size of opening the capillary tube has.

Figure 29: Capillary Tube -15°C

Figure 30: Capillary Tube 16°C

Figure 31: Cross Sectional Area of Capillary Tube

4.0 Test Bench Hardware

4.1 Hardware Description

Hardware was built to represent both cooling systems and were utilized to perform bench testing. The bench tests were performed in the EcoCAR 3 lab at WVU where the experiment could be environmentally controlled. In order to keep both cooling systems as comparable as possible, the same cooling plate, radiator, and battery thermal mass were used in both setups. The two cooling systems have components that were changed out between tests when going from one system to the other which include the pump/compressor, expansion valve, and tank reservoir. A block diagram of the 50/50 ethylene glycol water coolant setup can be seen in Figure 32. This diagram shows how the system was laid out with hosing and sensors. Thermocouples are placed before and after the radiator and cooling plate which can be used to calculate the amount of energy being dissipated by each component. The mass flow rate sensor is also in line with fluid to measure the flow rate of the water. There are 12 different sensors around the battery module which will be talked about in further detail in section (4.2 Data Acquisition and Instrumentation). The R-134a refrigeration system uses the same thermocouples in line with the radiator and cold plate that the 50/50 ethylene glycol system used to measure the temperatures of the refrigerant. A block diagram of this bench test setup can be seen in Figure 33. The expansion valve is added to this system to give the phase change to the cooling system.

Figure 32: 50/50 Ethylene Glycol Block Diagram

Figure 33: Refrigeration Block Diagram

The cold plate is made of 6061 Aluminum that has cut slots for copper lines to pass coolant through the system to transfer heat from the battery. Aluminum has a thermal conductivity of 167 W/m-K [27] and copper has a thermal conductivity of 401 W/m-K [28]. These materials are easy to acquire and have acceptable heat transfer qualities. A thermal paste was used between the cooling plate and the battery pack along with applied thermocouples attached to components to get temperature readings. The thermal paste that was used was a silicon grease called OMEGATHERM 201 from Omega.com [29].

Figure 34: Cooling Plate with Thermal Paste

The radiator used for the bench testing was the Lytron 6120G1 copper tube-fin radiator which was also used in the vehicle. This radiator rated to dissipate up to 700 watts of energy using maximum flow conditions specified by the manufacturer of 2 GPM [30]. To supply airflow through the airside of the radiator, two 224 CFM fans were mounted to the radiator [31]. A mixing chamber was attached to the other side of the radiator as shown in Figure 35. An 85mm DBC ABACO Performance mass airflow sensor (MAF) was used to measure the airflow through the radiator. This MAF is used for automotive applications where it can be calibrated using ABACO's software. Using this software, it

was possible to calibrate the MAF to the experimental needs of 200 CFM or less. The calibration of the MAF sensor is discussed later.

Figure 35: Lytron Radiator Setup

The battery pack was represented by a 6061 aluminum block that was the same size as the actual battery module (302.9mm X 164.8mm X 243.0mm). The block was made of six different sections as depicted in Figure 36. In between each section, a silicon heating pad was placed. Each pad was capable of producing 100 W maximum, and the actual power was controlled using a potentiometer [32]. The goal of the aluminum block was not to replicate the thermal properties of the complex lithium iron phosphate batteries, but to help validate how both cooling systems behave in the ESS model. The Aluminum blocks were also safer to use because it did not require load banks or other potential loading devices to draw current from the batteries. Another reason for using the aluminum block representation was due to the fact that the A123 battery modules were not available for use during the time of experiments. The model used the current from the motor to decide how much energy would be placed over a drive cycle. The model provided instantaneous current and the amount of heat generated by each battery module which

was approximately 40W. Therefore, the experimental potentiometer was set to a constant 40W for the heating pads to represent the average 40W the model predicted.

Figure 36: Battery Block Representation

The cooling systems require either a pump or a compressor to drive the fluid through the system. In the 50/50 glycol system a pump is used to push the coolant. A Shurflo 2088-343-435 pump capable of providing 3 GPM at 0 kPa was used to supply the flow of coolant. This pump was selected because it was able to provide more than the required flow rate of 2 GPM which was used to overcome the pressure losses in the system. The refrigeration system utilized an R-134a refrigerant compressor from a surplus Sears food storage mini refrigerator. This compressor was powered by a 745W motor. This compressor was chosen because it was already available and was adequate to supply the batteries with sufficient cooling. Since the primary purpose of this test bench hardware was to verify the Simulink model, it is not important that the pump and compressor are not equivalent in performance.

4.2 Data Acquisition and Instrumentation

Data was acquired using two ICP CON DAQ boards and an Arduino board. A software package called Scimitar, which was created by Zachary Luzader, software

engineer at WVU, communicated with the DAQ boards and was used to record all incoming data into one file on a laptop computer. This allowed for efficient synchronization of the data collected. Scimitar was used to manipulate incoming signals for either calibration or calculation. Using the programming features of Scimitar, it was possible to control the pumps and fans through relays on the ICP CON boards. These relays provided a simple controller by turning on and off when a known temperature reached a specified limit.

There were a total of 21 data inputs from the experiment and three digital outputs used to control the relays. The first relay was used for the Pump/Compressor On/Off, while the second and third relay were used for fan speed high and fan speed low. An Arduino was used to control the fan speed by taking in the digital Hi/Low reading and sending out a pulse with modulation (PWM) signal to the fans. The fans could be controlled to give either 50% fan speed or 100% fan speed depending on what the temperature of the batteries required.

A total of twelve thermocouples were used on the battery pack, all of which were type T. The temperature range of these type T thermocouples is -200 to 360°C with an accuracy of 1°C [33]. A diagram of how the thermocouples were placed around the aluminum battery module is shown in Figure 37. The placement of these thermocouples was selected to provide an overall look at how the aluminum block cooled around the perimeter. The number on the diagram corresponds to the input channel to the DAQ board. In addition to these twelve thermocouples, there are six more temperature sensors that record the temperatures of the fluids. There is a thermocouple on the input and output of the cold plate measuring the coolant temperature. This shows the temperature

difference of the fluid and shows how much energy is leaving the batteries by using the laws of thermodynamics talked about later in this paper in section (7.0 Experimental and Model Testing). The radiator uses four thermocouples: one going into the liquid side of the radiator, one going out of the liquid side of the radiator, and the other two measuring the air temperature in and out of the radiator. The thermocouple overall layout for each cooling system was shown in Figure 32 and Figure 33.

Figure 37: Thermocouple Placement on Battery Module

The MAF sensor used for the bench test hardware was made from Abaco Performance. Calibration of the MAF was performed by running air at a known flow rate through the sensor and using the Scimitar software to record and post process the data. The flow rate was measured using a calibrated Laminar Flow Element (LFE). The most

recent calibration for the LFE was performed in August 2014 by Meriam Process Technologies. The MAF calibration was performed over a year later in October 2015 in the Center for Alternative Fuels Engines and Emissions (CAFEE) engine lab at WVU. A photograph of the test setup is shown in Figure 38. A photograph of an LFE can be seen in Figure 62 in the Appendix A. The airflow was recorded at various increments from 0-250 CFM for roughly one minute at each interval. An average was taken at each interval and the calibration graph was generated using a best fit line equation. The best fit equation was used in Scimitar to calibrate the MAF sensor for better accuracy. The calibration graph for the MAF is shown in Figure 39 along with the best fit equation.

Figure 38: MAF Calibration Test

Figure 39: MAF Calibration Chart CFM vs Voltage Reading

To record the liquid flow rate, a Mass Flow sensor from Gems FT-110 173932-C was placed in the system [34]. This flow rate sensor is capable of reading up to 2.7 GPM with an accuracy of +/-3% or +/-0.073GPM. The sensor uses 5 VDC, which is supplied by the Arduino directly. The output of the sensor provides the rate of 12,500 pulses per gallon. In Appendix B, the code that was used to convert pulses to GPM is provided, as well as a photograph of the flow meter shown in Figure 63. For an overall diagram of the entire experiment, Figure 40 can be seen with each part called out. This includes most of the components described in this section. More images of the experimental setup can be seen in the appendix in Figure 62 through Figure 74.

Figure 40: Bench Test Part Callout

1. Aluminum Batteries
2. ICPCON DAQ
3. Volt Meters
4. Power Adjustment for Battery
5. Radiator/fans
6. Arduino
7. AC Compressor
8. Hosing and Plumbing

The bench test systems were calibrated before any tests were performed by ensuring all thermocouples were calibrated to the same environmental conditions. In order to determine the natural convection heat loss occurring in the system, a test was conducted to eliminate potential error during operation. This was done experimentally by

adding heat to the aluminum block without cooling from the cold plate. The experiment started with the Kat's 24100 heating elements inside the aluminum block at 0 watts. Every half hour, 5 watts was added to the aluminum block until it reached a steady state temperature of 35°C [32]. By doing this test, the amount of heat that was lost to ambient air through natural convection could be determined and accounted for by adding this extra energy to the input to the system. It was determined that an extra 90 W was needed to add to the system to account for natural convection depending on environmental conditions in the lab. The environmental conditions in the lab were set to 70°C and could experience temperature differences of $\pm 2^{\circ}\text{C}$.

4.3 System Control Strategy

The performance comparison of the two cooling systems was tested to find which system would work best for the 2016 Camaro's HV battery active cooling system. This system monitors the temperature of the batteries and performs cooling when needed. Since 50/50 ethylene glycol and refrigeration systems are normally used in automotive applications to cool engines and cabin air, both systems were considered for cooling high voltage batteries. The point of this research was to determine which system works better for cooling batteries that operate at temperatures around ambient air and to determine which will be more energy efficient and allow the vehicle to travel farther.

There are multiple control strategies that could be selected to control the battery cooling system, however a simple on/off strategy with multiple modes with varying fan speeds was selected for the test bench to compare the two systems. Even though the on/off strategy has shown to not be as efficient as a proportional integral derivative (PID) control system that Oliveira, et al. [22] discussed, it was used because the implementation

of a PID control strategy would not be possible with the experimental set up. Since the experimental setup was not done to incorporate a PID controller it was not tested in the Simulink model. Additional control strategies could be easily implemented into the vehicle model to increase the effectiveness of the cooling systems. The thermal system controller controls two components of the cooling system, the coolant pump where it can be turned on and off, and the radiator fans. In the test bench setup, the ICPCON 7019z DAQ system was used as the controlling relay board. Scimitar was programmed to control digital outputs on the DAQ board which connected ground to each component turning relays on or off. In the vehicle, the pump and fans are controlled using the Hardware Supervisory Controller (HSC). For the bench test, using the ICPCON allowed a simple way of controlling the system without using the vehicle's HSC and acting as a DAQ at the same time. In the simulation model, there was a state flow model that could be changed and setup for any control strategy that was desired. This feature was used to match the conditions of the bench test to the Simulink model.

The battery system has an important role in the vehicle; due to its chemical makeup, it is also one of the more dangerous parts of the vehicle. The priority level of this system is one of the more important parts of the vehicle's safety. As a result, if the temperature reaches levels that are too high it will cause a forced shutdown and open contactors in the vehicle to keep the batteries from reaching a critical temperature level. The controller that is in the vehicle will have to relay the status of the battery temperature, pump, and fans to the BMS at all times.

The cooling control strategy being implemented in the vehicle was also used to conduct the bench test. This strategy operates the cooling system in three modes. The

initial mode of the system is off. Once the batteries reach a temperature of 30°C, the controller reaches its second mode: to turn the pump/compressor on along with the fans (50% fan speed). If the cooling is sufficient to cool the batteries back to 28°C, then the controller returns to mode one and turns the system off. The system will oscillate between these values just like a cooling system in a house. If the cooling system cannot keep up with operation and the temperature of the batteries increases to 35°C, then the controller will go into mode three where the pump/compressor remains on but the fan increases to 100% to increase efficiency of cooling. Once the temperature decreases below 30°C, it will return to mode two. This control strategy saves energy by running the fans at a lower energy consumption to reduce the amount of energy needed to cool the system. In most cases supplying the fans with only 50% of the energy is sufficient to maintain the batteries at a steady state temperature.

5.0 Cooling System Component Design

The West Virginia University (WVU) advance vehicle team designed a plug-in parallel hybrid architecture vehicle. The P3 architecture used a GM 2.4L LEA engine and a Parker GVM210-200S electric motor that is powered by seven A123 Systems Inc. battery modules, which are mentioned in more detail in section (1.0 Introduction). The seven battery modules are placed inside of a mounting structure referred to in this paper as an enclosure or Energy Storage System (ESS). Because the batteries will heat up during operation, a cooling system is needed to keep the batteries within operating range.

5.1 Cooling System Design Requirements

In order to build a working hybrid vehicle, the ESS coolant system needs to be well designed. Several hardware components are needed for the design of the cooling system. A discussion of the key components selected for the hardware and bench test cooling systems used in this research study is provided below. These components were selected based on availability, price, and performance, as well as conformance to standard requirements. These requirements were derived from the full Camaro vehicle model over multiple drive cycles. For the worst case condition, the ESS creates a maximum of 7658 W of heat generated (Q) on a US06 Drive Cycle but only an average of 279.1 W shown in Table 9. The ESS cooling system was designed to dissipate a minimum of 279.1 W based on the average heat generation from the battery system.

Table 9: Drive Cycle Heat Generation

Test Number	Air Temp	Drive Cycle	Drive Cycle Length (s)	Average Q (W)	Max Q (W)
1	20°C	US06	7813	279.1	7658
2	20°C	UDDS	9590	39.2	1098
3	20°C	HWFET	9180	170.0	3321

5.2 Radiator Selection

The Lytron G120G1 Copper tube-fin heat exchanger was chosen as the radiator for the ESS because it can dissipate 700W which is more than twice the performance needed [30]. This provides more than an adequate margin of safety in the design to ensure that in the hot desert heat, like in Yuma, Arizona, the batteries would not overheat. A graph from Lytron's website can be seen in Figure 41 showing the heat transfer versus

change in temperature for the selected radiator. This radiator was also selected because it came with thermal property information. Having the thermal property specifications from the manufacture helps replicate the radiator performance parameters in simulation models. Another reason why this radiator was selected was because Computer Aided Design (CAD) images were provided and were easy to implement into the full vehicle CAD model needed for installation into the vehicle. While a generic radiator ranges from roughly \$40-\$60, the price of the selected radiator was \$313, however, the information provided with the radiator was worth the expense.

Figure 41: Lytron G120G1 Tube-fin Heat Exchanger [30]

5.3 Pump Selection

The Pump that was chosen for this system was a Shurflo 2088-343-435 pump. The maximum allowable flow rate for the Lytron radiator is 2GPM. The Shurflo Pump can handle up to 3GPM with a maximum pressure of 45PSI. The long coolant lines that go from the front of the vehicle to the back may cause a large pressure drop. To ensure the pump would maintain the 2GPM, a larger pump was selected. The Shurflo pump is also self-priming which allows easier maintenance and serviceability. The price of this pump was \$65.99 which is average compared to most pumps on the market that range from \$50-\$90. It was also readily available so there was no lead time in procurement.

5.4 Pipe Selection

AN stainless steel braided lines were used in the vehicle. These lines are abrasion resistant because of the braiding around the rubber hosing but are also flexible and easily installed in the vehicle. The AN fittings that are attached to the lines are designed to minimize pressure drop through connections. These fittings also are easy to install and disassemble when needed.

5.5 Compressor Selection

The refrigeration system used a 1/25 horse power Sears C-Q30L2A compressor similar to those found in many compact refrigerator applications. The compressor was paired with the expansion valve that was out of the same refrigerator to ensure the capillary tube was the correct size to cool the system. The reason behind using this compressor and expansion valve was because it was readily available and a cheap substitute to buying a system off the shelf. The specification sheet for the compact refrigerator can be found in 13.0 Appendix D.

5.3 Cold Plate Design

In order to cool the batteries a cold plate was placed under the battery. This gives a direct form of contact to transfer heat to the cooling system. The cooling fluid passes through copper piping that was pressed into the cold plate. A custom cold plate was manufactured in the WVU EcoCAR 3 lab. For the experiment the cold plate was designed to fit under the battery to allow the most heat transfer area possible shown in Figure 42. The dimensions of the experimental cold plate were 165.0mm x 303.0mm and the thickness of both the experimental and actual cold plates are $\frac{1}{2}$ " with $\frac{3}{8}$ " outer diameter piping that is pressed into grooves cut $\frac{1}{4}$ " deep in the plates. The cold plate, seen in Figure 43, was used in the actual vehicle and was designed to fit under the ESS enclosure. The upper shelf has the dimensions of 693.3mm X 466.4mm and the bottom plate has dimensions of 323.0mm X 274.0mm which fits under a battery laid on its side.

Figure 42: Cold Plate Used in Bench Test

Figure 43: Cold Plate Used for Full ESS System

6.0 Experimental Test Parameters

The bench test was designed to make it as simple as possible to perform basic operations to validate the vehicle model of the cooling system. The following section describes how the experiment was performed. A single battery module was represented by an aluminum block and heated by using silicon heating elements as described in section (4.0 Test Bench Hardware). The heating elements were supplied with constant power. The power was set and controlled to represent a known thermal load that was determined from I^2R losses found in the full vehicle model in Simulink. The heat generated in the aluminum battery pack by the heating elements is similar to the actual battery modules that will be installed in the car because they heat the battery from the inside out. The DAQ system monitored each test and controlled the pumps and fans depending on temperature of the aluminum battery pack. Each test was run for at least two and a half hours to allow for sufficient amount of time for the aluminum battery system to heat up and cool through multiple cooling cycles. The aluminum battery block takes approximately

30 minutes to an hour to rise enough for the cooling system to turn on. The recognition that the aluminum battery representation is made up of different material than the complicated architecture of the LiFePO_4 batteries was taken into account in the model by adjusting the parameters to match the thermal characteristics of the material being tested. Aluminum was chosen to simplify the fabrication of the bench test and to only help verify the thermal model in Simulink. The model in Simulink uses data from A123 battery systems to represent the batteries.

To compare the Simulink model to the experiment, the parameters in the model were set to be as close as practicable to those in the experimental design bench test setup. The model uses two different subsystems to represent the battery block, see Figure 14, one is the aluminum battery and the other represents the actual A123 battery. The block representation can be switched in and out of the running model as needed. The purpose is to gather results from an actual test and compare it to the one in the model, verifying that the model has been created properly. This was done to speed up the testing time. The real bench test takes over two and half hours to run one test while executing the model takes several minutes to run one test. Once the validity of the models was checked, the results were based from the Simulink model rather than running the physical bench test.

The experimental results were evaluated based on the energy usage from the pumps or compressors that were energized in the cooling system, and the energy that the fans used to cool the radiator. The other variables were mentioned in previous sections, which included the measured temperatures, mass flow rates of the cooling liquid, and mass flow rate of the air through the radiator. All this information was recorded

and used to determine the energy the system used along with key data points for comparison to the model. Once validated, the model was then used to help determine the control strategy used in the vehicle.

The tests were run using basic assumptions for the ambient air conditions. Each test was conducted inside the test lab at approximately 22°C with an atmospheric pressure of 14.59 psi. Through a US06 driving cycle, the batteries generate an average of 40W per module of heat based on the Simulink model. To represent the heat generation of a single battery module, 80 W of power was applied at a constant rate to the aluminum block heating pads. 80 W was chosen because it provides a factor of safety of 2 to ensure the batteries will not overheat in hot conditions. This also sped up the heating process of the batteries so it would take less time to run each experiment. The assumption was made that the heating elements were 100% efficient. Meaning all of the energy used by the heating pads was distributed through the aluminum block. Another assumption was that all energy put into the system was dissipated through the cooling system. Recall as discussed earlier, the natural convection of the aluminum battery block was accounted for and calibrated out.

The experiment only used one battery module where the full ESS system was 7 battery modules. Scaling down the test could impact the overall results by changing the effectiveness of the cooling system compared to what could be found in a complete ESS system. The test was scaled down to keep the experiment to a reasonable size in a working lab. Even though the experiment was scaled down the results are considered suitable because the primary purpose of this experiment was to validate the thermal models that were being used in the full vehicle model cooling system. By building a model

that represents the scaled down bench test the model could be validated. The parameters were then adjusted to represent the full scale version. The mass and material properties in the model for the full scale system was changed to give results of the full thermal system. Another reason to scale the experiment was to determine how two similar cooling strategies compare to one another with different control parameters, thus the full ESS was not required for this purpose. The assumptions were made that if one control strategy would cool the scaled down model then it would comparably cool the full scale model. After confirming that the model is accurate, data could be collected using the thermal model instead of performing experiments.

7.0 Experimental and Model Testing Results

The following section discusses the validation of the model and the results of comparison of energy consumption for the two separate cooling systems performance under three different driving cycles (US06, UDDS and HWFET). Using the data from the experimental bench test the model was calibrated improving the ability to accurately calculate the required energy to cool the system. Using this calibrated model in Simulink the energy consumption for each cooling system could be calculated under the various drive cycles.

7.1 Radiator Overall Convective Heat Transfer Coefficient Determination

The thermal model heavily relies on Simulink's block sets. The energy that the batteries generate is transferred to the cold plate into the radiator and is then dissipated by air flow from the fans. The piping blocks that are found in Simulink's thermal liquids library were used to represent the radiator and the cold plate that heat and cool the fluid.

Mass blocks are added to this system to help account for thermal capacitance in the material of the radiator and cold plate. The amount of energy that the radiator can dissipate depends on liquid flow rate and air flow rate that passes through the fins. The overall convective heat transfer coefficient (U) is directly dependent on the airflow and the difference in temperature between the component being cooled and ambient air.

Based on this dependency, a bench test was performed to find the U value for the radiator that was chosen in order to get a more accurate result then using generic radiator cooling blocks. A simple setup featuring the main components in the test are shown in Figure 44. The test that was performed had the two variables; airflow through the radiator and fluid temperature. Using these two variables, a table of U values was created as shown in Table 11. A water heater was used to add energy into the system so the input temperature could be controlled and held constant.

Figure 44: Overall Heat Transfer Coefficient Bench Test

By measuring the temperature before and after the water heater and the mass flow rate, the energy that was added to the system could be calculated. Equation 5 was used

to solve for the energy transferred into the system. Because the experiment used insulation it was assumed that the system was isothermal and did not lose any heat to natural convection, simply the energy that was put into the system is equal to the energy out of the system. Using this assumption, Equation 6 can then be used to solve for the corresponding U values.

Equation 5: Law of Thermodynamics Equation

$$Q = \dot{m}C_p(\Delta T)$$

Equation 6: Overall Convection Heat Transfer Coefficient

$$Q = UA(\Delta T)$$

The U values that were calculated through this experiment are found in the last column in Table 11 and summarized in Table 10 for different ΔT . The values on the left side in Table 10 that are shown in red were found to be inconsistent and proven to be outliers from a standard deviation analysis and were factored out and discarded. These experimental values were considered to be outliers due to measuring a ΔT smaller than the $\pm 1^\circ\text{C}$ accuracy of type T thermocouples. This small ΔT test was performed to simply see how the cooling system would behave. From the graph in Figure 45 it can be seen the values of U for each valid ΔT measurements, shown in the columns next to the first red column, are similar and do not change with temperature as expected. These are the U values that were incorporated in the Simulink model for the radiator.

Table 10: U Values with Respect to Change in Temperature and Standard Deviation

Airflow CFM	Delta T				Mean	Standard Deviation	Mean Corrected	3*σ Corrected
	0.43	7.14	16.69	27.19				
0.00	136.39	10.98	8.69	10.20	41.57	54.75	9.96	2.85
48.68	220.85	35.21	35.16	36.76	82.00	80.17	35.71	2.23
99.04	86.53	48.83	53.42	51.99	60.19	15.30	51.41	5.76
141.53	1468.97	52.70	65.87	64.20	412.94	609.72	60.93	17.56

Figure 45: Overall Heat Transfer Coefficient Values with Respect to Flowrate

Table 11: U Value Table

	Total Time	Air Radiator Out	Ambient Air Radiator In	Radiator Out Temp	Radiator In Temp	Thermal Bath Out Temp	Thermal Bath Temp	Flowmeter Pulses	Flowmeter Volumetric flow	Maf Cfm	mdot coolant	Q_in Bath	Q_Out Radiator	Surface Area Tube	Q_In Radiator	Surface Area fins	U
	s	°C	°C	°C	°C	°C	°C	pulses	GPM	CFM	kg/s	W	W	m²		m²	w/m²K
1050-150-02	597.6	35.1	19.9	41.0	43.5	33.5	44.3	239.1	1.7	141.3	0.11	1463.3	1116.8	0.096	1116.8	1.142	64.2
1050-100-02	605.0	38.0	19.9	43.1	45.6	34.8	46.6	221.5	1.6	97.6	0.10	1499.1	1078.9	0.096	1078.9	1.142	52.0
1050-050-02	604.3	43.5	20.8	46.1	48.4	35.3	49.2	222.5	1.6	50.3	0.10	1294.2	952.8	0.096	952.8	1.142	36.8
1050-000-02	603.4	26.2	20.9	48.7	49.4	35.6	50.1	229.3	1.7	0.0	0.10	597.7	323.6	0.096	323.6	1.142	10.2
1040-150-02	607.4	32.8	22.8	36.6	38.2	31.4	39.3	246.9	1.8	139.3	0.11	1295.9	756.9	0.096	756.9	1.142	65.9
1040-100-02	597.8	33.4	22.2	36.8	38.3	31.3	39.3	245.8	1.8	101.0	0.11	1156.1	679.1	0.096	679.1	1.142	53.4
1040-050-02	609.4	34.1	22.9	38.0	39.0	31.5	39.7	246.1	1.8	44.7	0.11	761.9	451.6	0.096	451.6	1.142	35.2
1040-000-02	606.3	26.8	23.4	39.1	39.4	32.0	39.7	242.5	1.8	0.0	0.11	307.0	155.5	0.096	155.5	1.142	8.7
1030-150-02	600.6	27.5	22.9	29.1	29.7	27.9	29.8	249.9	1.8	141.0	0.11	323.2	280.3	0.096	280.3	1.142	52.7
1030-100-02	607.1	27.5	22.2	29.0	29.6	26.9	29.7	249.4	1.8	98.9	0.11	364.9	295.8	0.096	295.8	1.142	48.8
1030-050-02	600.0	28.1	22.5	29.2	29.7	26.7	29.8	249.7	1.8	51.1	0.11	280.6	225.0	0.096	225.0	1.142	35.2
1030-000-02	606.0	25.3	23.4	29.9	30.0	26.9	30.2	249.7	1.8	0.0	0.11	155.6	81.3	0.096	81.3	1.142	11.0
1025-150-02	601.8	25.7	25.7	25.8	25.8	26.5	25.7	287.6	2.1	144.5	0.13	30.3	11.5	0.096	11.5	1.142	1469.0
1025-100-02	598.3	24.9	24.9	24.9	24.9	26.5	25.7	288.8	2.1	98.7	0.13	441.9	5.7	0.096	5.7	1.142	86.5
1025-050-02	626.4	24.9	24.9	24.9	24.9	26.4	25.7	289.7	2.1	48.7	0.13	439.1	6.7	0.096	6.7	1.142	220.9
1025-000-02	606.7	25.5	25.5	25.4	25.4	26.5	25.7	289.0	2.1	0.0	0.13	141.6	7.7	0.096	7.7	1.142	136.4

7.2 Model Validation Results for 50/50 Ethylene Glycol System

Using the bench test results it was determined that the cooling model was validated for the 50/50 ethylene glycol system. The data from the bench test were compared to the values that were specified in the models using a constant 80W of heat applied to the system. The measured temperature of the battery in the experimental and the cooling model data is shown in Figure 46. As can be seen in the two graphs there are only some slight differences and the results compare well enough to reasonably conclude that the model performs well in predicting the performance characteristics of the cooling system. This also confirms that the model is considered adequate for use in modeling the full vehicle ESS system.

Both data sets show the fast rise in temperature over the first 1000-1400 seconds reaching the activation cooling temperature of 30 degrees. Once the cooling system was activated a slight dip occurred at approximately 1200 seconds and was seen in both sets of data due to the coolant cycling before it starts to heat up. The maximum battery temperature demonstrated in the experimental data showed roughly 32.8°C at 7000 seconds where the model data shows roughly 32.6°C for the same elapsed time. In this case the temperature did not fully reach steady state over the 2-hour test. Because the temperature value is well below the upper limit of the battery operation temperature of 50°C, there is no danger of overheating. Future test should be analyzed to ensure the batteries remain in operation temperature over longer drive cycles. The approximate 0.5°C difference is well within the accuracy of the thermocouple used for the measurements in the experiment.

Figure 46: Experimental and Simulation Battery Temperature 50/50 Glycol

7.3 Model Validation Results for R-134a Refrigeration System

The results for the refrigeration system were also collected and compared to the Simulink model. The comparison between the two data sets could be obtained but could not verify the model due to the thermocouples only having an accuracy of $\pm 1^{\circ}\text{C}$ and not having instantaneous pressure measurements of the refrigeration cycle. The use of more accurate digital pressure sensors would have benefitted in validating the thermal models. The lack of accurate data that was not obtained during testing was because of equipment that could not adequately measure the parameters needed. Even though the model could not be validated the comparison between the two systems are shown in Figure 47. This comparison was made possible by using the battery temperature properties rather than the refrigerant property data to compare experimental design to the cooling model. The relation between the two was done by varying the unknown initial quality and mass flow

rate values in the initialization file parameters until the model matched that of the experiment. This was done in the attempt to derive the actual values obtained in the experiment and allow the model to be used to prediction the refrigeration system within the vehicle model. Once a set of parameters was found that matched the experimental to the theoretical, the following results were found. Each data set had a fast rise in temperature just like the 50/50 ethylene glycol system did. This makes sense because the initial rise is not impacted by the cooling system. The temperature of the batteries reached the activation of the cooling system at 30°C between 1000-1400 seconds. Both the experimental and theoretical results showed the system oscillating between 30°C and 28°C. The experimental data showed the temperature of the battery overshooting the target temperature by 0.5°C. This could have been caused by the time it took for the refrigeration system to start up and cool down. The experimental refrigeration system was able to cool the battery down enough for the coolant system to shut down in approximately 53 minutes, where the theoretical model took roughly 55 minutes. Comparing the two systems it was reasonable to say that the two systems were equivalent enough to allow the model to be used to compare against the 50/50 ethylene glycol system.

Figure 47: Experimental and Simulation Battery Temperature Refrigeration

7.4 Comparison of Temperature Results between the 50/50 Ethylene Glycol and the R-134a Refrigeration System

Comparing the two cooling strategies shows that the ethylene glycol system was able to control the temperature of the batteries to a steady state, which kept them from overheating, but was not able to reduce the temperature to allow the pump to turn off. The refrigeration system was able to drop the temperature of the battery to the point that the compressor could turn off. By not running a high energy compressor for the duration

of the drive cycle the refrigeration system could potentially use less energy than the 50/50 ethylene glycol system.

7.5 Energy Consumption through Experimental Results

The energy consumption for each component was evaluated by measuring the energy consumed during operation. The power consumption of the pump or compressor was measured while the components were on. The energy consumption for the pump was 14.7W and the amount of energy the refrigeration compressor used while in operation is 87.9W. The energy of the fans was measured at 50% fan speed and 100% fan speed to account for energy saved at lower speeds. The fans at half speed used a power of 21.4W and at full speed they used 57.6W. These values were implemented into the vehicle model to add up the energy consumption that is taken away from the battery. Once the model was validated, the Simulink model executed each cooling system through the series of drive cycles to find the total energy consumption over any given drive cycle. This was done to find the worst case scenario between all the drive cycles and to compare the two systems.

7.6 Model Results for Varying Drive Cycles

The vehicle thermal models created were tested and validated under a constant load. After the model was validated, both cooling systems were run under a series of drive cycle simulations, all set to two and a half hours. During each drive cycle, the model calculated the current drawn from the batteries to run the motor and inverter. The heat dissipation is proportional to the amount of current drawn. To ensure the cooling system is sufficient, the current drawn is multiplied by a factor of safety of 1.5. The US06 drive cycle was one of the most aggressive drive cycles to be analyzed, the batteries would

experience the most heat generation during this drive cycle. As section (4.3 System Control Strategy) discussed, the control strategy had three modes where the fans would either be off, half, or at full fan speed. This was the strategy that was used for the experiment. Once the model was validated, a second control strategy was evaluated to compare if another control strategy would be better or worse. The second control strategy used had two modes compared to the three modes for the first strategy. For the second strategy when the temperature reached 30°C the fan and pump would turn on to full speed. The assumption for this strategy was that it would cause the system to cool much faster therefore using less energy. The results shown in Table 12 compared to Table 13 indicate that increasing the fan speed for both the refrigeration cycle and the 50/50 ethylene glycol cycle does not help with less energy usage over a drive cycle. Instead the energy consumption increases by up to 200% worst case but averaged close to 150%. The additional airflow through the radiator did not make a significant enough change to cool the system fast enough to save energy. For future studies this same model could be utilized to develop more advance control strategies to better optimize the system performance. The UDDS drive cycle only created an average of 39.2 W of energy, which was not enough to heat the batteries to the point where it would need cooling over the two and a half hour drive cycle.

Table 12: 50/50 Ethylene Glycol Strategy Two, Drive Cycle Results

Test Number	Air Temp	Drive Cycle	Drive Cycle Length (s)	Average Q (W)	Max Q (W)	50/50 Ethylene Glycol Cooling system Energy Consumption (Whr)	Refrigeration Cooling system Energy Consumption (Whr)
1	20°C	US06	7813	279.1	7658	45.4	157.2
2	20°C	UDDS	9590	39.2	1097	0	0
3	20°C	HWFET	9180	170.1	3321	8.5	17.5

When the two systems are compared over the multiple drive cycles it can be seen in Table 13 that the water-cooled 50/50 ethylene glycol system used only 38.8 W which is less energy overall to maintain the battery temperature. While at 117.0 W, the refrigeration system used almost three times the amount of energy. Note that in earlier sections it was discussed that the refrigeration system was able to cool the batteries while the water cooled system was only able to control the increase temperature of the batteries. In a very hot condition the refrigeration system would be a better choice in order to cool the batteries. In this case the design for the vehicle could be implemented to have the option to have refrigeration cooled battery system if needed in desert heat however the energy consumption would be much greater and sacrifice vehicle driving distance for this design. If the car is being operated in most climates found in the United States, then the regular water-cooled battery system would be sufficient and require less energy consumption.

Table 13: 50/50 Ethylene Glycol Strategy One, Drive Cycle Results

Test Number	Air Temp	Drive Cycle	Drive Cycle Length (s)	Average Q (W)	Max Q (W)	50/50 Ethylene Glycol Cooling system Energy Consumption (Whr)	Refrigeration Cooling system Energy Consumption (Whr)
1	20°C	US06	7813	279.1	7658	38.8	117.0
2	20°C	UDDS	9590	39.2	1098	0	0
3	20°C	HWFET	9180	170.0	3321	4.2	13.1

An example of the model output calculations, in Figure 48 shows the cooling system performance through a US06 drive cycle, Figure 49 shows the system going through a HWFET drive cycle and Figure 50 plots the same performance through a UDDS drive cycle. The abscissa axis shows the time through the drive cycle in seconds and the ordinate axis shows the vehicle's speed (km/h), fan speed (CFM), and battery temperature (°C). Through the most aggressive drive cycle (US06) the graphs show where the batteries gradually heat up through multiple drive cycles. At the 1000 second mark there is a change in slope of the temperature curve. This is the point where the vehicle turns from charge depleting (only electric motor propels the vehicle) to charge sustaining (the motor and engine propel the vehicle). After this the battery continues to gain heat. Approximately at 4000 seconds the cooling system turns on and stays on for the remainder of the drive cycle, the temperature starts to taper off once the cooling system was activated. Additional tests were conducted which extended the drive time on the US06 drive cycle to discover how the temperature would behave. It was found that the temperature would level out to a steady state at approximately 33°C. The

experimental data for the extended US06 drive cycle can be seen in Figure 51. An analysis was performed to show how the temperature reaches steady state in section (7.7 Experimental Uncertainty Analysis). Looking at the HWFET drive cycle in Figure 49 the cooling system remains off for a majority of the time and doesn't come on until the very end of the drive cycle. This was because the HWFET drive cycle remains at higher speeds and does not use the electric motor as much. Looking at the battery temperature plot there is a flat spot from 2000-2600 seconds because the battery was depleted and needed to be charged. Approximately at the 2600 second mark is when the engine could charge the ESS causing the batteries to heat up again. The UDDS drive cycle in Figure 50 shows that the cooling system does not activate. In the UDDS drive cycle the speed of the vehicle operates in slow short bursts causing low energy consumption from the batteries. Over the two and a half hour drive cycle the batteries only increased to 24°C.

Figure 48: US06 Drive Cycle Speed Vs Time

Figure 49: HWFET Drive Cycle Speed Vs Time

Figure 50: UDDS Drive Cycle Speed Vs Time

Figure 51: Extended US06 Drive Cycle

During the 50/50 ethylene glycol bench test the data resembled a typical first order response where the temperature slowly approached steady state. The Simulink models also showed this curve starting to taper off over the two and half hour constant heat. To prove the ethylene glycol system would level out an analysis was done to ensure the temperature of the batteries would not increase significantly over time. The Simulink

model was ran for a longer period of time extending the drive cycle by 15 minutes. The battery temperature over the extended drive cycle is shown in Figure 52 along with its first derivative. The first derivative showed that the change in battery temperature is slowly decreasing over time towards 0. The initial temperature, final temperature, and predicted temperature at 180,178 seconds can be seen in Table 14. The last slope value was $5.55\text{E-}6^{\circ}\text{C/s}$ in the data set, by linearly extrapolating this last slope value it will take over 50 hours to raise from 32°C to 33°C . 50 hours is insignificant for a hybrid vehicle to travel without stopping and well below the max temperature of 50°C . This proves that, for this research, the 50/50 ethylene glycol system will keep the batteries from overheating over time but will not actually lower the temperature of the batteries.

Figure 52: 50/50 Ethylene Glycol Steady State Test

Table 14: Steady State Test Results

	t_Initial (s)	t_Final (s)	t_180178 (s)
Temperature °C	24.00	32.03	33.03
Slope °C/s	0.00E+00	5.55E-06	"

7.7 Experimental Uncertainty Analysis

An analysis was performed to find the uncertainty of the experiments on the overall heat transfer coefficient values. In an article written in 1988, Moffat [35] discussed how to find the bias and the precision uncertainty in experimental data. The total uncertainty (Δ) of an experiment is found by using Equation 7; the B value is the bias uncertainty and the P variable is the Precision uncertainty.

Equation 7: Total Uncertainty of Experimental Results

$$\Delta_{\text{total}} = \sqrt{B^2 + P^2}$$

The bias uncertainty was found by taking the manufacturer's accuracy specifications for each sensor, then taking the positive and negative perturbation of each variable seen in Equation 5 in section (7.1 Radiator Overall Convective Heat Transfer Coefficient Determination) and in Equation 10. The absolute average of the perturbed values were then used to find the bias uncertainty for the energy placed into the system and the overall heat transfer coefficient, which can both be seen in Equation 8 and Equation 9. The precision uncertainty was not used in this research because only one test for each heat load case was performed.

Equation 8: Qin Bias Uncertainty Equation

$$B_{Qin} = \sqrt{\delta_{Cp}^2 + \delta_m^2 + \delta_{\Delta T_{\text{Liquid}}}^2}$$

Equation 9: Overall Heat Transfer Coefficient Bias Uncertainty Equation

$$B_{\text{Heat Transfer Coefficient}} = \sqrt{\delta_{Cp}^2 + \delta_m^2 + \delta_{\Delta T_{\text{Liquid}}}^2 + \delta_A^2 + \delta_{\Delta T_{\text{Air}}}^2}$$

The bias uncertainty results can be seen in Table 15 and the calculation excel sheet can be seen in the appendix in Figure 75. The test ID number, for the 12 tests, shows which test was executed. Using 1050-150-02 as an example: the 1050 states to what temperature the water heater was set to, for this case it was 50°C, and the 150 indicates the airflow rate that was used for that test, for this example it would be 150 CFM. The airflow rate intervals that were ran were 150, 100, 050, and 000. Each test was executed at steady state for ten minutes at a 10Hz rate. A collection of approximately 6000 data points were collected for each test, the average was taken for each temperature sensor over the 10 minutes. For each test that was run, the energy that was added to the system for the overall heat transfer coefficient experiment had an uncertainty value larger than 55% and as high as 685.8%. As the airflow decreased to 0 CFM the uncertainty increased because the change in temperature approached 0. This was due to the very small change in temperature that was used to calculate the energy added into the system and the use of thermocouples that had an accuracy of $\pm 1^\circ\text{C}$. It was reasoned that since the battery cooling system would operate at temperatures with a small ΔT the experiment should also operate under this condition. Thermocouples were used because it was believed that the accuracy would be sufficient for experimentation, during post processing the uncertainty analysis was run and showed how uncertain the results were. A mathematical analysis was performed to see if the use of thermistors, with an accuracy of $\pm .2^\circ\text{C}$, would help the uncertainty value. The mathematical results showed that by using

thermistors, the Qin uncertainty value decreased from 57.8% to 12.2% uncertainty value. The U value uncertainty also decreased using thermistors, thus proving that for this experiment, it would be beneficial to use more accurate sensors. It is difficult to approximate the convective heat transfer coefficient values at such a low change in temperature, but by increasing the accuracy of the temperature sensors the uncertainty of the experiment could be reduced.

Equation 10: Thermodynamic Equation for Convective Heat Transfer Coefficient

$$\dot{m}C_p\Delta T = UA\Delta T \quad \frac{\dot{m}C_p\Delta T}{A\Delta T} = U$$

Table 15: Uncertainty Analysis on Heat Transfer Coefficient with Thermocouples

	Number of Data Points	Total Time	U	Qin Uncertainty		U Value Uncertainty	
	-	s	w/m ² K	+W	%	+W/m ² K	%
1050-150-02	5976	597.6	64.2	645.6	57.8%	37.6	58.6%
1050-100-02	6050	605.0	52.0	645.7	55.4%	31.4	56.0%
1050-050-02	6043	604.3	36.8	645.3	63.0%	25.0	63.3%
1050-000-02	6034	603.4	10.2	477.6	141.5%	81.4	144.2%
1040-150-02	6074	607.4	65.9	644.6	87.9%	56.8	89.1%
1040-100-02	5978	597.8	53.4	644.4	97.5%	51.1	98.4%
1040-050-02	6094	609.4	35.2	620.9	141.5%	48.5	142.1%
1040-000-02	6063	606.3	8.7	427.1	278.7%	110.4	282.3%
1030-150-02	6006	600.6	52.7	379.4	141.5%	73.1	145.0%
1030-100-02	6071	607.1	48.8	401.3	141.5%	67.5	144.2%
1030-050-02	6000	600.0	35.2	339.1	157.3%	53.8	159.5%
1030-000-02	6060	606.0	11.0	533.8	685.8%	253.3	694.0%

8.0 Conclusion and Future Recommendations

8.1 Conclusion

This thesis presented the thermal design, experimental testing, validation, and model simulations to advance the development of thermal battery cooling design in an automotive application. Validation of the battery thermal model was confirmed using the bench test described in the previous sections. The data showed comparable evidence that the model resembled how the aluminum block would heat up over time, thus proving the model could be used to help design the thermal system. Through the experimentation of the 50/50 ethylene glycol system, the model was also validated by the comparable data that was collected through both methods. The data that was collected through the experimental design for the 50/50 ethylene glycol showed promising results. The refrigeration system needs additional pressure sensors and more precise thermal sensors to indicate which state the refrigerant is in at any given point through the cycle. The overall temperature data from the aluminum block can be used to show how the system cools but additional components are needed to analyze specifics of the cooling system. The experiment showed that the battery representation was able to be cooled using the refrigeration system with reasonable results. This allowed the model to be used to represent the refrigeration system and be used for further data collecting. Through the validated research presented in this thesis, a model of a thermal system could be created from start to finish and integrated into a full vehicle model.

There were two separate cooling methods studied in this research to help compare the performance of different cooling methods. The first method was a 50/50 ethylene

glycol water mixture and the other was an R-134a refrigeration system. Based on the results of this research the 50/50 ethylene glycol mixture was chosen for the WVU EcoCAR 3 team's vehicle. The 50/50 ethylene glycol mixture proved to be adequate for not only meeting all the drive cycle requirements, but was superior over the refrigeration cycle due to its easier maintenance and having a significantly (approximately 3 times) lower energy consumption compared to the refrigeration system. There are positives and negatives to each cooling system. Using a water based coolant for automotive applications has the advantages of easier maintenance and installation on a vehicle since coolant is readily available at many stores and no special tools are required for maintenance. Refrigeration systems require a higher level of understanding and special tools to perform any maintenance on the cooling system. Refrigeration systems also cost more over the water-cooled systems due to the more complex equipment and refrigerant required to use in the system. Refrigeration systems could also leak harmful gases into the cabin of the vehicle and harm the occupants. The one benefit to having a refrigeration system is that in extreme hot weather conditions, the refrigeration system could bring the temperature of the batteries down to a more desirable operating condition. During extreme heat conditions, the refrigeration system would need to be used, because the 50/50 ethylene glycol cooling system would not be able to keep the batteries cool over long drive cycles. The test results also showed that it is better to use a multi-mode control strategy, which uses an average of 150% less energy, over a thermostatic ON/OFF method.

8.2 Future Recommendations

The design of the ESS cooling system was based off the results given from the validated results. This can be used in future competitions and used to help refine the vehicle design. For the future, the testing of variable fan speed control methods, advanced control strategies could be investigated and implemented into the system to help refine energy consumption and efficiencies of the vehicle. This could be done with no modifications to the model and could focus on just the comparison of control methods. Thermal FEA and CFD analysis should be executed for future studies to design cooling plates to ensure the batteries are being cooled evenly.

Future testing and modeling of advanced parameters could be investigated, such as parasitic loads on the system, energy spikes when the system is turning on, pressure loss curves, and efficiencies of components at different operating ranges (pumps, compressors, fans, etc.). The pressure drop could contribute a significant amount of energy consumption on the pumps and future research should focus on modeling it. The testing of more refined specifications will increase the accuracy of the vehicle model and can further the design of the vehicle.

The experimental design for the refrigeration system could be refined to improve performance. The temperature and pressures sensors that were chosen for this system did not have the accuracy required. The tests could be run again using pressure sensors that can be recorded using a DAQ system that has an accuracy greater than ± 2.5 psi and use thermistors (instead of thermocouples) which will give a better accuracy of $\pm 0.2^{\circ}\text{C}$. Another important device that was not used in the refrigeration experimentation was a flowmeter with a minimum operating range of 0.0005-0.01 kg/s, this was due to time

and budget constraints. Designing the refrigeration system is also recommended with parts with specifications instead of using components sitting around the lab. When designing the refrigeration system, it is important to collect as many parameters for the system as possible. Using generic components from unknown manufacturers can cause issues finding important information needed to model the system correctly. Future tests should be done by taking the full ESS system and running the system through specific drive cycles in the vehicle. Data should be collected through these tests to verify the thermal models after installation in the vehicle.

9.0 References

- [1] "avtcseries.org," [Online]. Available: <http://avtcseries.org/>. [Accessed 4 6 2016].
- [2] S. Dorsey, "The Most and Least Eco-Friendly States of 2015," 27 Apr 2015. [Online]. Available: <http://www.triplepundit.com/2015/04/least-eco-friendly-states-2015/>. [Accessed 4 june 2016].
- [3] "usclimatedata," 2016. [Online]. Available: <http://www.usclimatedata.com/climate/morgantown/west-virginia/united-states/uswv0507>. [Accessed 5 July 2016].
- [4] "yourdictionary.com," [Online]. Available: <http://www.yourdictionary.com/thermal-runaway>. [Accessed 4 June 2016].
- [5] J. Laukkoenen, "Crankshaft," 29 Sept 2013. [Online]. Available: <http://www.crankshaft.com/history-of-antifreeze/>. [Accessed 8 June 2016].
- [6] "Hybrid Cars History of Hyrbid Vehicles," 13th June 2011. [Online]. Available: <http://www.hybridcars.com/history-of-hybrid-vehicles/>. [Accessed 10 May 2016].
- [7] P. Fritz, "Machinery Lubrication," 2006. [Online]. Available: <http://www.machinerylubrication.com/Read/841/coolant-fundamentals>. [Accessed 10 May 2016].
- [8] "Automobilemag.com," 24 June 2010. [Online]. Available: http://www.automobilemag.com/features/news/1007_automotive_air_conditoning_history/. [Accessed 20 April 2016].
- [9] R. Thien, "Ac-Heatingconnect.com," 11 Oct. 2012. [Online]. Available: <http://www.ac-heatingconnect.com/whats-the-difference-between-r-22-and-r-410a/>. [Accessed 8 June 2016].
- [10] B. Pwelson, "What HFC refrigerants are used in Commercial Air Conditioning?," 27 Nov. 2013. [Online]. Available: <http://www.ac-heatingconnect.com/what-hfc-refrigerants-are-used-in-commercial-air-conditioning/>. [Accessed 10 May 2016].

- [11] C.F.R., "ecfr.gov," 15 June 2016. [Online]. Available: http://www.ecfr.gov/cgi-bin/text-idx?SID=ba447754d6f766672ab21e5aa4146283&mc=true&node=pt40.33.1066&rgn=div5#se40.33.1066_1815. [Accessed 15 June 2016].
- [12] T. Crain, "Argonne National Labortory Sharepoint account," 21 Oct 2014. [Online]. Available: <https://sharepoint.anl.gov/Projects/ecocar3/students/Year%201/Fall%20Workshop/Presentations/EC3%20Y1%20FW%20EEC%20Deep%20Dive.pdf>. [Accessed 8th June 2016].
- [13] C. R. Czapinski, "THERMAL MODELING OF LITHIUM-ION ENERGY STORAGE SYSTEMS FOR," M.S. thesis, Mechanical Engineering, ERAU, Daytona Beach, FL, 2009.
- [14] B. Schweitzer, S. Wilke, S. Khateeb and S. Al-Hallaj, "Experimental validation of a 0-D numerical model for phase change thermal management systems in lithium-ion batteries," *Jornal of Power Sources*, no. 287, pp. 211-219, 2015.
- [15] F. Baccino, M. Marinelli, P. Nørgård and F. Silvestro, "Experimental testing procedures and dynamic model validation for vanadium redox flow battery storage system," *Jornal of Power Souces*, no. 254, pp. 277-286, 2014.
- [16] M. Cao, Thermal and cooling systems modeling of powertrain for a plug-in parallel-through-the-road hybrid electric vehicle, Detroit: ProQuest, 2014.
- [17] A. Odagescu and V. Baudot, "Thermal properties of ethylene glycol aqueous solutions," *Cryobiology*, no. 48, pp. 283-294, 2004.
- [18] F. Liu, F. Lan and J. Chen, "Dynamic thermal characteristics of heat pipe via segmented thermal resistance model for electric vehicle battery cooling," *Jornal of Power Sources*, no. 321, pp. 57-70, 2016.
- [19] A. Jarrett and I. Y. Kim, "Design optimization of electric vehicle battery cooling plates for thermal performance," *Jornal of Power Sources*, no. 196, pp. 10359-10368, 2011.
- [20] J. Smith, M. Hinterberger, P. Hable and J. Koehler, "Simulative method for determining the optimal operating conditions Simulative method for determining the optimal operating conditions," *Journal of Power Sources*, no. 267, pp. 784-792, 2014.

- [21] K. Onda, T. Ohshima, M. Nakayama, K. Fukuda and T. Araki, "Thermal behavior of small lithium-ion battery during rapid charge and discharge cycles," *Journal of Power Sources*, no. 158, pp. 535-542, 2006.
- [22] V. d. Oliveira, A. Trofino and H. Christian, "A switching control strategy for vapor compression refrigeration systems," *Elsevier*, no. 31, pp. 3914-3921, 2011.
- [23] M. L. Barr, Mississippi State University EcoCAR extended range electric vehicle thermal system, Mississippi State University: ProQuest, 2014.
- [24] a. A. Pesaran, "Battery thermal models for hybrid vehicle simulations," *Journal of Power Sources*, vol. 110, no. 2, pp. 377-382, 2002.
- [25] a. A. Pesaran, "Battery Thermal Management in EVs and HEVs: Issues and Solutions," in *Advanced Automotive Battery Conference* , Las Vegas, 2001.
- [26] W. Li, Z. Qu, Y. He and Y. Tao, "Experimental study of a passive thermal management system for high-powered lithium ion batteries using porous metal foam saturated with phase change materials," *Journal of Power Sources*, vol. 255, pp. 9-15, 2014.
- [27] "asm.matweb," 2016. [Online]. Available: <http://asm.matweb.com/search/SpecificMaterial.asp?bassnum=MA6061t6>. [Accessed 10 May 2016].
- [28] "EngineeringToolBox," 2016. [Online]. Available: http://www.engineeringtoolbox.com/thermal-conductivity-d_429.html. [Accessed 1 May 2016].
- [29] "Omega," 2016. [Online]. Available: <http://www.omega.com/pptst/OT-201.html>. [Accessed 20 October 2015].
- [30] "Lytrondirect.com," Lytron, 2016. [Online]. Available: <http://www.lytrondirect.com/6120g1-copper-tube-fin-heat-exchanger/6120g1/>. [Accessed 10 May 2016].
- [31] "Amazon.com," Sanyo Denki, [Online]. Available: http://www.amazon.com/Sanyo-Denki-SAN-Ace-120mm/dp/B00QWB1X3O?ie=UTF8&psc=1&redirect=true&ref_=oh_aui_detailpage_o06_s00. [Accessed 25 August 2015].

- [32] "Amazon," Kat's, [Online]. Available: http://www.amazon.com/Kats-24100-Watt-Universal-Heater/dp/B000I8VL3O?ie=UTF8&psc=1&redirect=true&ref_=oh_aui_detailpage_o05_s00. [Accessed 25 August 2016].
- [33] "Omega.com," Omega, [Online]. Available: <http://www.omega.com/temperature/Z/pdf/z207.pdf>. [Accessed 12 May 2016].
- [34] "Grainger.com," Grainger, [Online]. Available: <https://www.grainger.com/product/GEMS-SENSORS-Turbine-Flow-Rate-Sensor-1ALZ7>. [Accessed 7th July 2016].
- [35] R. J. Moffat, "Describing the Uncertainties in Experimental Results," *Experimental Thermal and Fluid Science*, vol. 1, no. 52, pp. 3-17, 1988.

10.0 Appendix A

10.1 Appendix A: ESS Simulink Model Blocks

Figure 53: ESS Subsystem

Figure 54: Battery Calculation Subsystem

Figure 55: Battery Lookup Table Subsystem

Figure 56: Thermal Model Subassembly

Figure 57: Compressor Subassembly

Table 16: Compressor Subsystem Parameters

Compressor					
Block Name	Settings tab	Parameter	Variable	Value	Unit
Controlled Mass Flow Rate Source	parameters	Cross-sectional area at ports A and B:	$\pi \cdot \text{pipe_diameter}^2 / 4$	7.13E-05	m ²
		Characteristic longitudinal length:	0.1	0.1	m
Commanded Mass Flow Rate	Main	Gain	commanded_mass_flow	0.0185	kg/s

Figure 58: Pressure Sensor Subassembly

Figure 59: Mass Flow Rate Subsystem

Compressor					
Block Name	Settings tab	Parameter	Variable	Value	Unit
Controlled Mass Flow Rate Source	parameters	Cross-sectional area at ports A and B:	$\pi \cdot \text{pipe_diameter}^2 / 4$	7.13E-05	m ²
		Characteristic longitudinal length:	0.1	0.1	m

Figure 60: Condenser Subassembly

Table 17: Condenser Subsystem Parameters

Condenser					
Block Name	Settings tab	Parameter	Variable	Value	Unit
Evaporator Pipe Test	Geometry	Pipe length	condenser_length	3.5	m
		Cross-sectional area	$\pi \cdot \text{pipe_diameter}^2 / 4$	7.13E-05	m ²
		Hydraulic diameter	pipe_diameter	0.0095	m
	Friction and Heat Transfer	Aggregate equivalent length of local resistances	$0.1 \cdot \text{condenser_length}$	0.35	m
	Effects and Initial Conditions	Phase change time constant	0.1	0.1	s
		Initial fluid regime	Two-Phase Mixture	-	-
		Initial pressure	initial_pressure	0.6171	MPa
		Initial vapor quality	initial_quality_Cond	0	-
		Fluid inertia	Off	-	-

Figure 61: Evaporator Subassembly

Table 18: Evaporator Subsystem Parameters

Evaporator					
Block Name	Settings tab	Parameter	Variable	Value	Unit
Copper to Aluminum Conduction	Parameters	Area	Coolingplate_Copper_Conduction_Area	0.0448	m ²
		Thickness	Coolingplate_PipeThickness	0.0016	m
		Thermal conductivity	copper_conductivity	400	W/(m*K)
Aluminum to Battery Conduction	Parameters	Area	Coolingplate_Aluminum_Conduction_Area	0.0387	m ²
		Thickness	Coolingplate_Aluminum_Thickness	0.0127	m
		Thermal conductivity	ThermalConductivityAlum	205	W/(m*K)
Copper Piping	Parameters	Mass	Coolingplate_Pipe_Mass	0.5304	kg
		Specific Heat	copper_specific_heat	390	J/(kg*K)
	Variables	Temperture	Initial_Battery_Temp	293.15	K
Cold Plate Aluminum	Parameters	Mass	Coolingplate_Plate_Masses	2.0412	kg
		Specific Heat	SpecificHeatCapacityAlum	910	J/(kg*K)
	Variables	Temperture	Initial_Battery_Temp	293.15	K
Evaporator Pipe Test	Geometry	Pipe length	evaporator_length	3.5	m
		Cross-sectional area	pi*pipe_diameter^2/4	7.13E-05	m ²
		Hydraulic diameter	pipe_diameter	0.0095	m
	Friction and Heat Transfer	Aggregate equivalent length of local resistances	0.1*evaporator_length	0.35	m
	Effects and Initial Conditions	Phase change time constant:	0.1	0.1	s
		Initial fluid regime:	Two-Phase Mixture	-	-
		Initial pressure:	initial_pressure	0.6171	MPa
		Initial vapor quality:	initial_quality_Evap	1	-
		Fluid inertia:	Off	-	-

10.2 Appendix A: MAF Sensor Calibration

Figure 62: Laminar Flow Element (LFE)

10.3 Appendix A: Experimental Equipment and Setup

Figure 63: Gems Flow Rate Sensor

Figure 64: Refrigeration Compressor

Figure 65: Test Block for Copper Inlaid Lines

Figure 66: Sacrificial Mini Fridge for Experiment

Figure 67: Refrigeration Pressure Gauges

Figure 68: Refrigeration Compressor

Figure 69: Aluminum Battery System with Thermocouples

Figure 70: ICPCON DAQ Boards

Figure 71: Radiator Setup

Figure 72: Scimitar Program Computer

Figure 73: Arduino Board

Figure 74: Aluminum Block Battery Representation

Calculation of Bias Uncertainty of Heat transfer Coefficient

Variable	Description	Measurement Uncertainty	Units	Measured Value	Units	Perterbed Variables		Perterbed Results			+C _i	-C _i	C _i
						+X _i	-X _i	R _o	+R _i	-R _i			
TIB	Temperature Into Bath	1	°C	41.04	°C	42.04	40.04	1116.78	661.53	1572.03	-455.25	455.25	455.25
TOB	Temperature Out of Bath	1	°C	43.49	K	44.49	42.49	1116.78	1572.03	661.53	455.25	-455.25	455.25
mdot	Flow meter	0.004599	kg/s	0.11	kg/s	0.11	0.10	1116.78	1163.94	1069.62	47.16	-47.16	47.16
C _p	Constant Pressure Value	2.09E+01	kJ/kgK	4180	kJ/kgK	4200.90	4159.10	1116.78	1122.37	1111.20	5.58	-5.58	5.58
Equations:											Total		645.57
Q= m°C _p Δ(T)											Δ/Q=		57.81%

Variable	Description	Measurement Uncertainty	Units	Measured Value	Units	Perterbed Variables		Perterbed Results			+C _i	-C _i	C _i
						+X _i	-X _i	R _o	+R _i	-R _i			
TIR	Air Temperature Into Radiator	1	°C	19.86	°C	20.86	18.86	64.21	68.72	60.25	4.51	-3.95	4.23
TOR	Air Temperature Out of Radiator	1	°C	35.10	K	36.10	34.10	64.21	60.25	68.72	-3.95	4.51	4.23
TIB	Temperature Into Bath	1	°C	41.04	°C	42.04	40.04	64.21	38.03	90.38	-26.17	26.17	26.17
TOB	Temperature Out of Bath	1	°C	43.49	K	44.49	42.49	64.21	90.38	38.03	26.17	-26.17	26.17
mdot	Flow meter	0.004599	kg/s	0.11	kg/s	0.11	0.10	64.21	66.92	61.49	2.71	-2.71	2.71
C _p	Constant Pressure Value	2.09E+01	kJ/kgK	4180	kJ/kgK	4200.90	4159.10	64.21	64.53	63.88	0.32	-0.32	0.32
A	Surface Area X Direction	1.00E-10	m ²	1.14	m ²	1.14	1.14	64.21	64.21	64.21	0.00	0.00	0.00
Equations:											Total		37.59
U= (m°C _p Δ(TB))/(AΔ(TR))											ΔU/U=		58.55%

Figure 75: Bias Uncertainty Analysis Calculations

10.5 Appendix A: ESS Simulink Properties initiation file

Figure 76: Model Properties: Initiation function file callout

11.0 Appendix B

11.1 Appendix B: ESS Simulink Initialization File “Battery_Model”

```
%% Justin Brumley
% Battery design

%% Initialization
addpath([pwd '/Libraries']);
addpath([pwd '/Images']);
addpath([pwd '/Scripts_Data']);
addpath('Cooling');
addpath('lithium_ion_files')

load('Drive_Cycles/EcoCAR_US06Medium.mat');
load('Component_data/Battery_Data.mat');
load ('Component_data/A123_20kW_Justin_oct2015_2.mat');
load ('Component_data/Radiator_h_Values.mat');
load('EcoCAR_US06Medium.mat');
load r134aPropertyTables

pm_addunit('kJ', 1e3, 'J')
pm_addunit('uPa', 1e-6, 'Pa')

cd Libraries
if ~exist('refrigeration_lib', 'file')
    ssc_build refrigeration
end
cd ..

Logging_Sample_Time_s=1;
Solver_Sample_Time = 0.05;
%% Environment:
    %Ambient Air
        AirTemp=20+273.15;                %°K
        environment_temperature=AirTemp;    % K
        natural_convection_coefficient=20;  % W/m^2/K
%% Number of batteries testing
    TotalModules=7;
    NumModules=1;          %How many am I testing?
    ModuleGain=NumModules/TotalModules;
%% Battery Cell/ material Information
    PackArchitecture=[7 15 2];          %[#packs #series #parallel]
    NumPacks=PackArchitecture(1);
    NumCellsPerPack=PackArchitecture(2)*PackArchitecture(3);
    NumCellSeries=PackArchitecture(2);
    NumCellParallel=PackArchitecture(3);
    Capacity= (NDA-DISCLOSED INFORMATION);    %Ah
    NomVolt=(NDA-DISCLOSED INFORMATION);      %V
    NomEnergy=(NDA-DISCLOSED INFORMATION);    %Wh
    EnergyDensity=(NDA-DISCLOSED INFORMATION);    %Wh/kg
```

```

Mass=(NDA-DISCLOSED INFORMATION); %kg
Power=(NDA-DISCLOSED INFORMATION); %W | (@ 25°C, 10 sec,
50% SOC)
DCResistance=(NDA-DISCLOSED INFORMATION); %mohm | <5>
25°C)
HeatCapacitycell=(NDA-DISCLOSED INFORMATION); %J/K
RValue=(NDA-DISCLOSED INFORMATION); %°K/W(From Cell to Surface)
BatteryTotalMass=NumCellsPerPack*NumPacks*Mass;
%Dimensions ( (NDA-DISCLOSED INFORMATION) mm)
CellLength=(NDA-DISCLOSED INFORMATION); %m
CellHeight=(NDA-DISCLOSED INFORMATION); %m
CellThick=(NDA-DISCLOSED INFORMATION); %m
AreaOfContact=(NDA-DISCLOSED INFORMATION); %m^2

Weight=(NDA-DISCLOSED INFORMATION); %kg
InternalResistanceDC=(NDA-DISCLOSED INFORMATION); %mOhm (@
25 °C)
InternalResistanceAC=(NDA-DISCLOSED INFORMATION); %mOhm (@
25 °C)
HeatCapacitymodule=(NDA-DISCLOSED INFORMATION); %J/K
HeatCapacity7x15s2p=(NDA-DISCLOSED INFORMATION); %J/K
%Battery Init
%Battery_Voltage_V=NumPacks*NumCellSeries*3.24; %Battery Voltage
(V)
Battery_Capacity=Capacity; %Battery
Capacity, Ah
Battery_Initial_SOC=.9; %Battery Initial
SOC
Initial_Battery_Temp=40+273.15; %°K
Pump_Consumption=0;
%Vehicle Power Specifications
Contdischarge=17.9292; %A
%% Material Properties:
%Aluminum Properties
DensityAlum=2700; %kg/m^3
SpecificHeatCapacityAlum=910; %J/kgK
ThermalConductivityAlum=205; %W/mK
%Copper Propeties
copper_density = 8940; % kg/m^3
copper_specific_heat = 390; % J/kg/K
copper_conductivity = 400; % W/m/K
%Coolant Properties
CoolantSpecificHeat=4.19; %kJ/kgK
DesiredFlowRate=.13; %kg/s
%% Radiator/Condensor
%Radiator Piping
condenser_length = 3.5; % m
Radiator_Length=3.3528; %m
Radiator_Hydraulic_Diameter=.375*0.0254; % (in*in-m)=m
Radiator_CrossSection_Area=pi*(Radiator_Hydraulic_Diameter^2)/4;
%m^2
Radiator_Surface_Area=1.141675237; %m^2

```

```

    %Fan Properties
    maxCFM=150; %CFM
    Fan_Diameter=4.5; %in
    Radiator_Area=.5*.5; %m^2

%% Cold Plate
    %Pipe
    evaporator_length = 3.5; % m
    Coolingplate_Length=59*0.0254; % (in*in-m)=m
    Coolingplate_Hyrdraulic_Diameter=.375*0.0254; % (in*in-m)=m

Coolingplate_CrossSection_Area=pi*(Coolingplate_Hyrdraulic_Diameter/2)^2;
m^2

    %Copper Piping Mass
    Coolingplate_Pipe_Diameter_OD=.375*0.0254; % (in*in-m)=m
    Coolingplate_PipeThickness=.0625*0.0254; % (in*in-m)=m
    Coolingplate_Pipe_Mass=(pi*(Coolingplate_Pipe_Diameter_OD/2)^2)...
        -(pi*((Coolingplate_Pipe_Diameter_OD...
        -2*Coolingplate_PipeThickness)/2)^2))...
        *Coolingplate_Length*copper_density; %kg

    %Copper Conduction
    Coolingplate_Copper_Conduction_Area=pi...
        *Coolingplate_Pipe_Diameter_OD*Coolingplate_Length; %m^2

    %Aluminum Plate Mass
    Coolingplate_Plate_Mass=2.04116; %kg

    %Aluminum Conduction
    Coolingplate_Aluminum_Conduction_Area=5*12*0.0254^2; % (in*in*in-
m^2)=m^2
    Coolingplate_Aluminum_Thickness=.5*0.0254; % (in*in-m)=m

%% 50/50 glycol Mixture Coolant Information
    %Cooling Pump Data
    Pump_pipe_Length=.18517; %m
    Pump_CrossSection_Area=pi/4*0.009525^2; %m^2

%% Refrigeration Cycle Information (Copyright 2013-2014 The MathWorks, Inc.)
    % Refrigerant:
    initial_pressure =0.617107; % MPa
    initial_quality_Evap = 1;
    initial_quality_Cond = 0;

    % Compressor:
    commanded_mass_flow = 0.00185; % kg/s
    compressor_time_constant = 5; % s

    % Expansion valve:
    min_throat_area =.139; %0.109; % mm^2
    max_throat_area = 0.218; % mm^2
    min_throat_temperature = 248; % K
    max_throat_temperature = 293; % K

% Refrigerant:
    initial_pressure = 0.413; % MPa
    initial_quality = 0.3;

% Compressor:
    commanded_mass_flow = 0.001; % kg/s
    compressor_time_constant = 5; % s

% Expansion valve:
    min_throat_area = 0.109; % mm^2
    max_throat_area = 0.5; % mm^2

```

```

    min_throat_temperature = 293; % K
    max_throat_temperature = 260; % K
% Pipes:
    pipe_diameter = 0.008;           % m    Inner
    pipe_thickness = 0.00058;       % m

```

*** (NDA-DISCLOSED INFORMATION) *** cannot show this information
 Due to NDA agreement with A123 Batteries

11.2 Appendix B: Arduino Fan Controller Coding

// Justin's Battery Fan Controller and Flowmeter DAQ
 #define FLOWSENSORPIN 2

```

volatile uint16_t pulses = 0;
volatile uint8_t lastflowpinstate;
volatile uint32_t lastflowratetimer = 0;
volatile float flowrate;
SIGNAL(TIMER0_COMPA_vect) {
    uint8_t x = digitalRead(FLOWSENSORPIN);

    if (x == lastflowpinstate) {
        lastflowratetimer++;
        return; // nothing changed!
    }

    if (x == HIGH) {
        //low to high transition!
        pulses++;
    }
    lastflowpinstate = x;
    flowrate = 1000.0;
    flowrate /= lastflowratetimer; // in hertz
    lastflowratetimer = 0;
}

void useInterrupt(boolean v) {
    if (v) {
        // Timer0 is already used for millis() - we'll just interrupt somewhere
        // in the middle and call the "Compare A" function above
        OCR0A = 0xAF;
        TIMSK0 |= _BV(OCIE0A);
    } else {
        // do not call the interrupt function COMPA anymore

```

```

        TIMSK0 &= ~_BV(OCIE0A);
    }
}

void setup() {
    Serial.begin(9600);
    //Serial.print("Flow sensor test!");

    pinMode(FLOWSENSORPIN, INPUT);
    digitalWrite(FLOWSENSORPIN, HIGH);
    lastflowpinstate = digitalRead(FLOWSENSORPIN);
    useInterrupt(true);
    int Fan50Pin=7;
    int Fan100Pin=8;
    int FanControlPin=6;
    pinMode(Fan50Pin,INPUT);
    pinMode(Fan100Pin,INPUT);
    pinMode(FanControlPin,OUTPUT);
}

void loop()                // run over and over again
{

    //Serial.print("Freq: "); Serial.println(flowrate);
    //Serial.print("Pulses: "); Serial.println(pulses, DEC);

    // if a plastic sensor use the following calculation
    // Sensor Frequency (Hz) = 7.5 * Q (Liters/min)
    // Liters = Q * time elapsed (seconds) / 60 (seconds/minute)
    // Liters = (Frequency (Pulses/second) / 7.5) * time elapsed (seconds) / 60
    // Liters = Pulses / (7.5 * 60)
    float liters = pulses;
    liters /= 7.5;
    liters /= 60.0;

    Serial.print(pulses);
    Serial.print(" Hz ");
    if(0)
    {
        Serial.print(" ");
    } else {
        Serial.print("? ");
    }
    Serial.print("G\r\n");
}

```

```

pulses = 0;

int Fan50Pin=7; //pin trips to ground when the temp exceeds lower setting
int Fan100Pin=8; //pin trips to ground when the temp exceeds higher setting
int FanControlPin=6; //pwm pin to control fan speed
pinMode(Fan50Pin,INPUT);
pinMode(Fan100Pin,INPUT);
pinMode(FanControlPin,OUTPUT);
if (digitalRead(Fan100Pin)==LOW){
    analogWrite(FanControlPin,255);}
else if ((digitalRead(Fan50Pin)==LOW) & (digitalRead(Fan50Pin)==LOW)){
    analogWrite(FanControlPin,.5*254);}
else if ((digitalRead(Fan50Pin)==HIGH) & (digitalRead(Fan100Pin)==HIGH)){
    analogWrite(FanControlPin,0);
}
delay(1000);
}

```

12.0 Appendix C

12.1 Appendix C: UDDS (505 portion)

Figure 77: UDDS (505 portion) Drive Cycle

12.2 Appendix C: HWFET

Figure 78: HWFET Drive Cycle

12.3 Appendix C: US06

Figure 79: US06 Drive Cycle

12.4 Appendix C: US06 Split up for EcoCAR 3

Figure 80: US06 Drive Cycle for EcoCAR 3

13.0 Appendix D

13.1 Appendix D: Compact Refrigerator Specification Sheet

Figure 81: Compact Refrigerator Specification Sheet

West Virginia University Electronic Thesis and Dissertation

Signature Form

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7/28/16
(date)

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