THERMAL ANALYSIS AND MANAGEMENT OF LITHIUM TITANATE
BATTERIES FOR USE IN BATTERY AND
HYBRID ELECTRIC VEHICLES

by

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ABSTRACT

Lithium-titanate batteries have become a viable option for automotive energy storage due to their long lifetime, good energy density, and ability to withstand large charge/discharge currents. Normal vehicle operation exposes the battery to significant current demands which can cause substantial heat generation. Therefore, the battery pack requires an active thermal management system in order to maintain a safe operating temperature and prevent battery degradation. The goals for this research are to analyze the temperature rise and heat generation in a battery pack of 50 Ah lithium-titanate cells, and to design and evaluate cooling systems that maintain safe operating temperatures without consuming excessive parasitic power.

A liquid crystal thermography technique was devised to measure the temperature rise of the cells under various charge/discharge current cycles. Two cooling systems, one water-cooled and the other air-cooled, were designed and implemented, and their performance was evaluated. While both systems performed effectively, the air-cooled system is more efficient as it consumes less parasitic power. Finally, the experimental data were employed within a numerical model to accurately characterize the spatially-varying heat generation source term within the cells.

Ultimately, it has been shown that ambient air can be used as the active fluid to adequately cool the cells through proper heat exchanger design. The results from the numerical model to characterize the heat generation term will be useful in future thermal modeling and management studies of lithium-titanate cells.
Chapter 1

INTRODUCTION

1.1 Introduction

The growing popularity of battery electric vehicles (BEV) and hybrid electric vehicles (HEV) has created a demand for efficient, lightweight, durable, and safe batteries. Lithium-ion batteries are preferred for EVs as they offer high energy density and good cycle life. In addition, they are generally safe and do not suffer from memory effects. In EV applications, these batteries need to accommodate high currents over many charge and discharge cycles. In order to achieve the voltage necessary to power an EV, many cells must be connected in series to create a battery pack that will likely be confined to an internal compartment within the vehicle. When in use, each cell will generate heat throughout its volume due to loss mechanisms arising from the cell’s internal chemistry and ohmic resistance. The battery’s large current demand, and the proximity of individual cells to one another within the battery pack which is itself situated in a confined region, make thermal management a high priority concern for the proper functioning and safety of EVs.

Battery technology has long been considered a viable option for on-board electrical energy storage. Lead acid batteries are the oldest form of rechargeable cells with the first developments dating back to the mid 1800’s. It was not until the 1970’s,
however, that advancements in gel electrolytes enabled lead-acid battery operation in various orientations. Lead acid batteries suffer from low energy-to-weight and energy-to-volume ratios, but offer a low-cost solution to providing high surge currents that are required to start a traditional internal combustion engine. The Nickel-Cadmium composition represents another option for secondary batteries. Ni-Cad batteries were in development at the beginning of the 20th century but did not become feasible for EV’s until the mid-1990’s when improvements in manufacturing techniques reduced their cost. These batteries offer a longer lifetime than lead acid cells, and can accommodate high, continuous discharge rates. Unfortunately Ni-Cad batteries suffer greatly from memory effects and thermal runaway which limits their effectiveness in real world applications. Another option for electrical energy storage is the Nickel Metal-Hydride cell which was developed in the late 1970’s and early 1980’s but was not commercially available until the 1990’s. This technology offers high energy density and a nearly constant voltage during discharge, but experiences high self-discharge rates and fewer life cycles than previous technologies, especially in high temperature climates. Nickel Metal-Hydride cells also do not allow extremely high charge and discharge rates which limits their use in EV’s that require this characteristic.

The most recent advancements in battery technology have led to the development of Lithium-Ion cells which supply good power density and do not suffer at all from memory effects. In addition, certain Li-ion chemistries can accommodate high currents which make them ideal for rapid charge and discharge. Some
chemistries, however, experience thermal runaway at high temperatures which can lead to cell degradation and ultimately combustion. Current research with this technology is aimed at alleviating these concerns and reducing their high cost. Ultimately, the goal of this technology is to provide a battery that can meet all the specifications required in a demanding automotive application [1, 2].

Recent advances in Li-ion technology have led to the development of lithium-titanate batteries which, according to one manufacturer, offer higher energy density, more than 2000 charge/discharge cycles (at 100% depth-of-discharge), elimination of thermal runaway, and a life expectancy of 10-15 years [3]. Hence, lithium-titanate cells have been selected as the candidate for the battery pack for the Phase 3 and 4 buses of the University of Delaware’s Fuel Cell Bus Program. It is anticipated that these batteries will deliver superior performance at higher efficiency and significantly lower weight than the Nickel-Cadmium batteries that are used in the earlier Phase 1 and 2 buses in our program. Prior to the implementation of lithium-titanate technology in our buses, it was necessary to gain a good understanding of their thermal behavior under typical bus operating conditions. Accordingly, the research reported in this thesis has focused on lithium-titanate cells manufactured by Altairnano. These cells have a maximum voltage of 2.79 V, a 50 Ah capacity, and can accommodate currents up to 400 A.

All batteries undergo heat generation during charging and discharging due to three primary reasons: ohmic, entropic and concentration (species transport) losses. Previous research has explored each aspect of battery heat generation in great detail.
In smaller batteries it is important to characterize the effect of mixing and phase change reactions as ohmic resistance is less significant due to minimal current demands [4-6]. Large cells, such as the 50 Ah lithium-titanate variety chosen for this research, have been shown to exhibit thermal behavior that is dominated by ohmic resistance and entropic heat generation. The most common representation of this behavior takes the form shown in Equation 1.1.

\[
q = I^2 R - IT \left( \frac{\partial V_{oc}}{\partial T} \right) \quad (1.1)
\]

Here, the first term represents Joule heating and the second term represents the effect of entropy [7]. In Equation 1.1, \( I \) and \( R \) represent the current and internal resistance of the cell, respectively, \( T \) is the cell temperature, and \( V_{oc} \) is the open-circuit voltage of the cell. Another important consideration for high-current battery systems is the ohmic loss mechanisms at the interface between cell terminals. Depending on the nature of the contact between the terminals and the resulting contact resistance, significant heat can be produced at the terminals as well. All cell chemistries, including lithium-titanate, have recommended operating temperatures in order to protect the internal component materials. Thermal management, therefore, is an important consideration for overall system design especially when large currents are expected.

Without an active cooling system, the cell temperature within the lithium-titanate battery pack can potentially exceed the maximum operating temperature of
55°C during normal operation. Hence, implementing an effective cooling system is an essential requirement for this type of battery. According to the manufacturer, thermal runaway is not a concern with these cells below 250°C due to the use of nanostructured electrode materials [3]. Additional benefits from good thermal management of lithium-titanate cells include improved electrochemical performance, better charge acceptance, higher power and energy capacity, and improved cycle life [3].

The overall objective of this work is to characterize the temperature rise due to heat generation during charge and discharge cycling in a lithium-titanate battery and design and implement efficient cooling systems to maintain battery temperatures below maximum operating temperatures. In addition, this research has employed experimental data to characterize the nature of heat generation within the cells. One of the requirements for our experimental program was a method to measure the temperature over the entire surface of the cell in a convenient and accurate manner. We chose a method based on thermochromic liquid crystals for this purpose as described next.

1.2 Thermochromic liquid crystal-based technique for battery temperature measurement

A technique based on thermochromic liquid crystals was devised to instantaneously measure the temperature field over the entire surface of the battery. This measurement technique was subsequently used to analyze and evaluate two
active cooling systems designed to maintain an internal cell temperature of less than 50°C which is below the maximum operating temperature of 55°C as specified by the manufacturer.

Preliminary tests revealed that the cells do not generate heat evenly throughout their volume. With reference to Figure 1.1, it can be easily understood that the current flux within the cell increases steadily from the edges of the cell to its terminal, implying a greater heat generation as the terminal is approached. For this reason, a system for measuring the temperature of the cell over its entire surface had to be devised. In many cases an infrared camera is used for this type of surface measurement [8]. This solution, however, could not be used in our application due to the particular construction of the cell. Each cell has a loose-fitting aluminum jacket around it that does not make complete contact with the cell itself. Acquiring an infrared image of the cell without pressing the jacket against the cell’s interior will not yield an accurate surface temperature measurement due to inadequate thermal contact between the cell and its jacket. In order to ensure proper thermal contact while also allowing visual access to the front of the cell, the cell would have to be compressed between transparent plastic or glass panels. Unfortunately, such materials act as infrared blockers and hence would contaminate the infrared signal from the surface of the cell. While infrared technology is extremely useful in some circumstances, our unique needs precluded its use in our study.
Another option for measurement of surface temperatures is discussed in [9] which involves fixing thermocouples to the surface of the cell at different locations. This method also presents problems in our application. The first issue is physically adhering the thermocouples to the cell. We require the complete insulation of the front and back face of the batteries to simulate conditions in the center of the stack. Placing multiple thermocouples on one of these faces would create a small air gap between the cell and the insulation layer thereby reducing the effectiveness of our insulation. More importantly, thermocouples are point wise measurement devices and would restrict our ability to determine the entire surface temperature distribution of the cell. While thermocouples provide very accurate temperature measurements, our
application would require a large array of thermocouples further complicating the insulation process.

To bypass these issues, we employed thermochromic liquid crystal (TLC) thermography. TLCs are organic molecules that change color with temperature. They are characterized by two temperatures: Red Start, which is the temperature at which color play commences, and Bandwidth which is the temperature range over which the TLC transits the color spectrum from red (cold) to blue (hot). TLCs can be ordered with customized red starts and bandwidths. They are available as adhesive-backed sheets from Hallcrest, and can be applied to the desired surface very easily. Since the TLC strips are very thin, they respond rapidly to temperature changes of the substrate. In order for TLCs to provide accurate quantitative temperature information over extended two-dimensional domains, they must be properly calibrated to create a mapping between color and temperature. Temperature visualization with TLCs has been practiced for many years, and several papers can be found in the literature describing the most effective way to perform liquid crystal thermography [10-12]. Following the procedures outlined in these papers the TLC layer can be used to determine surface temperature for time-varying applications. TLC technology coupled with thermistors can provide the entire temperature field of the cell without compromising insulation. Therefore, TLC thermography provides an effective method to evaluate the thermal performance of the battery system under various operating conditions, and to evaluate the effectiveness of battery cooling systems.
1.3 Active cooling approaches

With an accurate method of temperature measurement implemented, two active cooling systems were designed and tested. The first system tested used chilled water as the active fluid, circulating it through channels that were welded to aluminum plates and spaced intermittently between each cell. While this method was shown to be extremely effective in laboratory testing conditions, certain limitations must be considered prior to a practical application. Batteries such as these lithium-titanate cells most frequently appear on HEV’s or BEV’s where energy efficiency is a critical design element. A traditional liquid-cooled system must incorporate both a chiller and a pump in order to cool the liquid after each cycle and circulate it through the cooling channels. This process consumes a significant amount of energy and ultimately decreases the overall efficiency of the vehicle. In addition to parasitic loss mechanisms the weight of the fluid and system components will also decrease energy efficiency by increasing vehicle curb weight.

The desire to reduce the energy overhead imposed by the battery cooling system on the vehicle led to the second cooling system design which employed ambient air instead of water as the active fluid. In such an open-loop system, cool air could be drawn in from the ambient, circulated across the face of the batteries, and then expelled from the system. Using ambient air as an active fluid eliminates the need for on-board chiller unit and coolant storage, resulting in reduced parasitic losses and lower curb weight. However, air has a significantly lower thermal conductivity and heat capacity than water, which severely inhibits its ability to effectively transfer heat.
To overcome this limitation, a method for improving the effective heat transfer coefficient was developed. A thin layer of aluminum open-cell foam material was placed between adjacent cells in the battery pack, and cooling air was forced through the foam. The foam draws the heat away from the cell and transfers it to the air stream that is forced through the porous medium. This cooling system was tested in laboratory conditions to characterize its effectiveness and real world applicability.

1.4 Numerical modeling

The final goal of this work was to develop a model for the two-dimensional heat generation source term within the cell. The extensive experimentation in this study yielded significant temperature data that could be used to develop such a numerical model. Although thermal models of batteries have been developed for many different geometries and cell properties [13-18], experimental studies of the thermal characteristics of batteries are scarce. Resolution of this source term is necessary to construct an accurate finite difference model of the cell. The model can then be used to design and evaluate a multitude of thermal management systems in software prior to their construction. The extensive experimental procedures completed in this research supplied adequate information to enable the characterization of battery thermal behavior through an iterative finite difference method.
1.5 Organization of the thesis

Ultimately, the goal of this research is to explore the realistic potential of various active cooling systems and perform a study on the thermal behavior of lithium-titanate batteries. The thesis is organized into the following chapters. Chapter 2 describes the procedures used to calibrate the TLCs, and the protocols used to conduct temperature measurements of the cells under various charge and discharge currents. Next, the design and demonstration of the effectiveness of the water-cooled system is presented in Chapter 3, and the air-cooled system in Chapter 4. Chapter 5 focuses on the numerical methods employed to determine the two-dimensional heat generation within the cell by matching the numerical results with the experimental data. Finally, Chapter 6 presents conclusions from the thesis and directions for future work.
LIQUID CRYSTAL THERMOGRAPHY

As stated in Chapter 1, we selected thermochromic liquid crystal thermography to map the surface temperature of the lithium-titanate cells during charge-discharge cycles. It is first necessary to perform an accurate calibration of the thermochromic liquid crystal strips before they can be used to acquire temperature data in a given application. This chapter describes how the TLC strips were implemented for thermal analysis and management of the lithium-titanate battery, and the procedure that was followed to calibrate them prior to data acquisition.

2.1 Test Stand

To simulate cells in the center bank of a battery pack on an EV, a test stand was designed and fabricated to house a stack of five cells insulated on the front and the back. A solid model of the 5-cell stack is shown in Figure 2.1 in a simplified version of the test stand. Each cell is 247 mm by 228 mm, and is 13 mm thick. The cell has two rectangular tabs (38 mm by 76 mm) that serve as its terminals.
Figure 2.1 Solid model of the test stand for temperature measurement in battery experiments.

Figure 2.2 Schematic view of the front cell of the stack. Five TLC strips with temperature ranges in 5 degree increments were attached to the surface of the cell as shown. Thermistor locations on the tabs are also indicated.
In order to understand the thermal behavior of the stack, we focused on the cell in the front of the stack. Figure 2.2 is a detailed solid model of the first cell. Five TLC strips (each 228 mm long and 11 mm wide) were attached to the surface of the cell symmetrically about its vertical axis as shown. The strips were arranged such that temperature range of each strip increases in 5-degree increments from left to right. Five separate temperature ranges were chosen to obtain a more vivid color play and improve the resolution and accuracy of our temperature measurements. This arrangement allowed us to monitor the surface temperature of the cell within the combined bandwidth of 25-50°C. In order to replicate the actual conditions within the battery pack in an EV, the front and back faces of the stack were insulated. In particular, the front of the cell was insulated by employing a double-paned, Plexiglas window because it permitted visual access of the TLC. Temperature visualization with TLC provided valuable insights into the nature of heat generation in these cells.

The positive and negative terminals are marked on the top of the cell in Figure 2.2. When the cells are connected to one another in series, the tabs from one cell are clamped to those of the adjacent cells. This type of connection is not ideal and results in heat generation at each junction due to contact resistance as follows:

\[ q = I^2 R_{con} \quad (2.1) \]

where \( q \) is the heat generated, \( I \) is the current through the terminals, and \( R_{con} \) is the contact resistance. Heat is also generated within the body of the cell due to its internal
resistance to current flow, and due to the electrochemical reactions that occur during charge and discharge. The current densities are much higher in the upper region of the cell because all of the current generated across the face of the cell during discharge has to converge at the terminals where it is collected. Similarly, during charging, all of the current supplied to the cell travels through the tabs before it gets distributed throughout the entire volume of the battery. This leads to much higher heat generation rates and higher temperatures in the neighborhood of the tabs, whereas the temperature decreases towards the lower half of the cell (positive $dT/dy$) due to lower heat generation. During charge and discharge experiments, it was apparent that the heat generation is uniform in the $x$-direction but noticeably changes in the $y$-direction. Therefore, the chosen arrangement of the five TLC strips allows us to monitor the variation of the temperature of the surface of the cell as a function of the vertical coordinate and time. Also note that thermistors are placed at each tab as shown in Figure 2.2 to monitor its temperature as well.

2.2 Calibration of TLC strips

In order to obtain reliable and accurate temperature data from the TLC strips they must first be carefully calibrated. These strips are extremely sensitive and therefore all variables within the environment must be carefully controlled. Any changes in the illumination intensity, illumination angle, or orientation of the camera relative to the region of interest can affect the accuracy of the final data. Because the actual color reflected by the TLC depends both on the view angle, it is necessary to
include the \((x, y)\) coordinate of the TLC in the calibration process. As a result, the mapping between the TLC’s color and its temperature is a function of the \((x, y)\) coordinate of the TLC on the surface of the cell.

![Diagram of the heater block used for TLC calibration](image)

**Figure 2.3** Solid model of the heater block used for TLC calibration. The aluminum block is 50 mm thick to allow for uniform heat distribution, and includes three slots for thermistors.

The calibration process consists of heating the TLC to a uniform temperature, and acquiring a color image. The process is repeated over the entire temperature range in 1-degree increments. Figure 2.3 shows a solid model of the aluminum heater block used for TLC calibration. The black section in the center represents the five TLC strips. Three grooves were cut on the right side to accept thermistors. The thermistors were epoxied to inserts which were threaded directly into the aluminum block allowing accurate surface temperature measurement at three locations in the \(y\)-
direction. The aluminum block is heated by a strip heater with a controller to regulate the amount of heat being generated. Strip heaters, however, do not generate heat evenly. Therefore, the strip heater was bonded to the back of a 50 mm thick aluminum block. The high thermal conductivity of aluminum allows the heat to diffuse evenly in the $x$-$y$ direction and present an isothermal condition for the TLC. The temperature readings of the three thermistors were closely monitored during calibration to ensure isothermal conditions.

The calibration is conducted by slowly increasing the amount of heat generated by the strip heater. When each of the three thermistors reads to within ±0.2°C of an integer value between 25 and 50°C, an image is acquired. These images are stored for subsequent processing using a Matlab code. Each TLC strip is divided into 26 regions vertically (see Figure 2.4) to account for the $y$-variation in illumination and view angle. In addition, since each of the five TLC strips is calibrated separately, the sensitivity in the $x$-direction is also accounted for. Color data were recorded for each temperature value in the calibration process. Hue values are then generated from the color data. Hue values are especially useful for TLC thermography as they increase monotonically with temperature.
Figure 2.4 TLC strips were divided into 26 regions to account for the y-variation in illumination and view angle. The sensitivity in the x-direction is also accounted for as each of the five strips is calibrated separately.
Figure 2.5 Variation of color with temperature, in 1°C increments, for the rightmost TLC strip whose color bandwidth is 45-50°C. The changes in hue are significant enough to be discriminated with image processing techniques and accurately correlated to temperature.
Figure 2.6 Calibration data for the 45°-50°C TLC strip. The $x$-axis represents the 26 regions (top to bottom) of the TLC strip, and the $y$-axis is the hue value for each of those regions. The slight fluctuations in the data indicate that careful calibration is necessary for accurate results.

To appreciate the range of color play in the TLC strips, the actual images of the rightmost strip (45°-50°C) during calibration are displayed in Figure 2.5 in 1°C increments. The corresponding hue values for these images, spanning the 26 vertical regions, are shown in Figure 2.6. The change in color from red to blue as the temperature transits the bandwidth is very apparent in Figure 2.5. For example, the leftmost TLC image for 45°C displays a reddish-yellow tone, making its color quite distinct from the greens and blues seen for higher temperature values. This visual observation is confirmed in Figure 2.6 as the jump in hue values is greatest between 45°C and 46°C. Most importantly, however, these images prove that hue can provide...
an accurate relationship between color and temperature during TLC calibration. Recording standard RGB values for these colors is also an option; however, these would not yield a monotonic variation as temperature increases. Instead, each image would have a unique set of coordinates that described the relative contribution of the colors red, green and blue. The nonmonotonic variation of these RGB values would complicate the calibration process of the TLC’s. Therefore, when employing this thermal mapping technique it is more convenient to first convert the RGB values into the HSV (hue, saturation, and value) color space and utilize hue values to develop an accurate correspondence between temperature and color.

Another important observation from Figure 2.5 is the variation in color with the vertical coordinate. For example, the leftmost strip appears more reddish-yellow at the bottom than at the top, which is reflected in the slight dip of the 45°C curve in Figure 2.6. This variation in hue is not due to temperature variations since the entire strip is exactly at 45°C. Instead, this variation in hue is due to the change in the angle of the reflected illumination into the camera lens. This effect has been discussed earlier in this chapter, and again confirms the need to include the spatial location of the TLC strips during calibration.

A second Matlab code was written to convert TLC color images acquired during battery experiments into temperature values using the calibration data. The program first converts the color image into hue values for each of the 26 regions. Next, it compares the hue value for a given region with the hue values from the calibration data files, and calculates the corresponding temperature by interpolation for
each region. In this way, temperature measurements are obtained from the entire surface of the cell covered by the TLC strips.

In order to instill confidence in the results derived by the image processing technique, a method for verifying the proper calibration of the TLC strips was employed. To accomplish this, a thermocouple was fixed to the center of the front-most cell where the TLC strips are located. A test was run in which the temperature of the stack was slowly increased with an image being recorded once per minute. The temperature at the location of the thermocouple was allowed to reach 50°C such that the calibration could be verified across the entire active temperature range of all TLC strips. The time-varying thermocouple output was compared to the results from the image processing technique for Region 13 (which is co-located with the thermocouple) to estimate the RMS error in TLC thermography. This process confirmed that the TLC strips measured the temperature of the cell’s surface to an accuracy of ±0.5°C.

In addition to TLC image processing, a LABVIEW program was used to record thermistor data from each of the cell’s terminals with respect to time. This enabled the mapping of the entire temperature field of the cells during testing. After developing this method for accurately measuring cell surface temperature, we next turned our attention to designing and implementing active cooling systems. The next two chapters will describe in detail two cooling systems for the cells, one using water, and the other using air as the active fluid. The TLC temperature measurement technique was employed to quantify the effectiveness of each system. In addition,
knowing the steady-state temperature distribution of the cells is important information
to characterize the spatial variation in heat generation by the cells as described in
Chapter 5.
Chapter 3

WATER-COOLED DESIGN

As stated in Chapter 1, two active cooling approaches were explored in this work. The first method was a liquid-cooling method, and its design and performance are described in this chapter. The design consisted of 4.76 mm thick aluminum cooling plates inserted between adjacent cells. The in-plane dimensions of the aluminum plate were identical to those of the cell. A 6.35 mm ID aluminum pipe was welded to the top edge of each cooling plate. The entire stack comprising five cells and four cooling plates was compressed before testing to ensure good thermal contact between the cells and the plates. Manifolds were constructed for both inlet and outlet so that the coolant flow (water) was evenly distributed between each cooling channel. The inlet manifold was supplied by a chilled water line in the lab. Both the inlet flow temperature and flow rate were held constant during the experiments. Custom thermistor mounts were fabricated to monitor the inlet and outlet temperatures on one of the channels. This data was useful in calculating the total heat extracted from the cells. Figure 3.1 shows the experimental layout.
Figure 3.1 Experimental layout of the cooling system for the stack of five cells. The inlet and outlet manifolds distribute water evenly between each of the four cooling channels while the thermistors provide water temperature readings.

In order to test these cells in the lab, a power source capable of supplying 400 A charging current was acquired. The cells were discharged through a bank of 24 0.6-ohm resistors wired in parallel. Each resistor was connected to a relay that could be switched on or off depending on how much current the user wants to draw. At full resistance it was possible to draw over 300 A of discharge current from the cells. The same LABVIEW program that recorded thermistor data was also used to regulate the current. This program allowed the user to activate the desired number of resistors for discharging, or to enter the current value for charging. The fuel cell hybrid bus in use on the University of Delaware campus [19] only demands charge and discharge currents up to 300 A. Hence, our experiments were held under this current limit.
3.1 Testing Protocol

The first test was conducted at 300 A. The cells were subjected to 5-minute charge and discharge cycles until the 50°C threshold was reached, or 90 minutes had elapsed. The test was conducted first without activating the cooling system, and then repeated with a coolant water flow rate of 0.0315 kg s\(^{-1}\) per channel. The surface temperature variation of the cell was recorded by acquiring TLC images at one-minute increments for the duration of the trials. These images were stored for post-processing with the Matlab code. The TLC data along with the thermistor data provided temperature versus time curves that were used to quantify the effectiveness of the cooling system. The same tests were repeated at 200 A, and at 100 A since these belong in the range of currents experienced by the cells under normal operating conditions. At these lower current levels the tests were only conducted for 60 minutes as a steady state was reached more rapidly.

A second set of tests was conducted to determine the total heat generation in each cell during charge and discharge cycles by measuring the temperature rise of the coolant as it flowed through the stack. The previous tests employed a high flow rate of water that resulted in a temperature rise that was too small to be accurately resolved by the thermistors. Hence, the flow rate was decreased to 0.0063 kg s\(^{-1}\) per channel in the second set of tests. These tests used the same charge cycles and currents as the previous set of three tests.
Safety is of utmost importance when working with such large currents. During testing, the portion of the room containing the experimental setup was isolated to prevent accidental entry by lab personnel and eliminate the potential for injury.

3.2 Results and Discussion

Each curve on the following temperature versus time graphs represents the temperature at specific locations on the cell. Region 3 is at the top of the cell near the terminals and region 24 is close to the bottom. Tab 5 is the thermistor data from the positive interconnect on the front cell.

3.2.1 300 A Charge and Discharge Cycling

Figure 3.2 presents temperature data for the 300 A test with and without cooling. This plot clearly demonstrates the need for a cooling system as the temperature rises rapidly in the absence of cooling. In fact, the test without cooling had to be terminated at 30 minutes to prevent cell damage because the threshold temperature of 50°C had already been reached. Also note that the top of the cell (Region 3) is always hotter than the bottom (Region 24). This is due to the higher heat generation at the top of the cell as well as the heat generated due to contact resistance at each terminal’s connection.

The temperature profiles are dramatically different when the cooling system is in operation at 0.0315 kg s$^{-1}$ of water flow per channel. Figure 3.2 shows that the threshold temperature of 50°C is not reached even after 90 minutes of testing. The
temperature plots indicate that the system has reached a steady state around 45°C, something that was not observed in the absence of cooling. It is also important to realize that the two curves from the top (Region 3) and bottom (Region 24) of the cell present virtually the same temperature versus time profile up to about the 14-minute mark with cooling. This implies that the higher heat generation in Region 3 is countered by a more effective heat removal due to its proximity to the cooling channel. Beyond 14 minutes, the two curves diverge and maintain about a 5 degree difference at steady state. It is also interesting to note that the top of the cell is now cooler than the bottom. This proves that the cooling system design is quite effective in removing heat from the region where it is most intensely generated.

Another interesting phenomenon seen in Figure 3.2 is the periodic fluctuations in the temperature profiles. These fluctuations are more evident in the curves with cooling, and are strongly correlated with the periodicity of the charge and discharge cycles. Discharge cycles generate more heat due to the entropy term and variations in cell internal resistance as explained below. The heat generated in the cell can is expressed in the form shown in Equation 1.1. The first term on the right side of the equation refers to the heating due to ohmic resistance. The second term is the entropy term. Discharge currents occur when $I > 0$, whereas the term $\frac{dV_{oc}}{dt}$ is always negative. Consequently, the second term on the right hand side is positive during discharge and negative during charge. The change in sign of the second term during charge/discharge cycling accounts for the periodicity in heat generation. In addition,
cell internal resistance is both a function of battery state of charge and direction of current flow. This phenomenon is discussed in detail in Section 4.3. These two factors lead to a strong dependence of heat generation on charge/discharge cycling and hence the wavy appearance of the temperature versus time profiles in Figure 3.2.

Figure 3.2 Temperature versus time for the 300 A test with and without cooling. Maximum temperatures are reached very quickly without cooling, proving the need for an active cooling system. At a coolant flow rate of 0.0315 kg s$^{-1}$ steady state is reached around 45°C.

The effect of the cooling system can be visually inferred by examining Figure 3.3. The image on the left shows the temperature distribution at 18 minutes into the
test without cooling, while the image on the right is at 34 minutes into the test with cooling. It is seen that the fourth strip from the left is the active strip, indicating that the cell temperature is in the 40-45°C range in both views. As stated earlier, the red end of the color spectrum corresponds to cooler temperatures whereas the green and blue shades represent the warmer temperatures in the range. These images provide visual confirmation of the data in Figure 3.2. The image at the left indicates that the upper half of the cell is warmer than the lower half. Conversely, the image on the right indicates that the upper half of the cell is cooler than the lower half. The two temperature distributions are almost exact inverses of one another demonstrating the impact of cooling, and the ability of the cooling system to remove heat from the cell.

Figure 3.3 Left: temperature visualization 18 minutes into the 300 A test without cooling. Right: the same image 34 minutes into the test with cooling. The activation of the cooling system causes the temperature profile to invert.
3.2.2 200 A Charge and Discharge Cycling

The next test was run at 200 A with the same coolant flow rate of 0.0315 kg/s. Figure 3.4 depicts the temperature data both with and without cooling for 5-minute charge and discharge cycles at 200 A. In the absence of cooling, the cell temperature climbed continuously and approached the threshold temperature of 50°C at about 60 minutes. It is also clear that the tab temperature remains hotter than the surface temperature of the cell at all times without active cooling. All three temperature traces show a continuously increasing trend even at the end of the testing period, implying that a steady state has not been reached. This indicates that an active cooling system is necessary even when cycling at a lower current value of 200 A. Also note that the top of the cell (Tab 5 and Region 3) is always hotter than the bottom (Region 24).

When the cooling system is activated the temperature profiles reach a steady state at less than 35°C. As for the 300 A case, we can see that the lower half of the cell (Region 24) is warmer than the upper half (Region 3), while Tab 5 is the coolest point. Although the temperature at the bottom of the cell still increases during discharge, the cooling system is able to completely absorb nearly all of that heat when the cell is charging. The temperature of the tab appears to be trending toward a steady state at under 28°C. With all parts of the cell approaching a steady state well under 55°C, it is clear that this particular cooling system is effectively managing the heat generated by the cell at 200 A.
Figure 3.4 Temperature versus time for the 200 A test with and without cooling. The threshold temperature of 50°C is reached in 60 minutes with no cooling, indicating the need for a cooling system even at 200 A. With the cooling system in operation at 0.0315 kg s\(^{-1}\) a steady state is reached below 35°C.

It is interesting to note in Figure 3.4 that the tab temperature profile does not show a pronounced wavy appearance unlike the profiles for Regions 3 and 24. This is because the tab temperature is dominated by its local heat generation, which is purely due to contact resistance and therefore independent of the direction of current flow.

Figure 3.5 shows the temperature distribution from the second TLC strip (30-35°C) with and without cooling at 200 A. The image on the left is at 16 minutes without cooling, and the image on the right is at 47 minutes with cooling. Comparing these with the images from Figure 3.3 shows the difference between 300A and 200A battery
cycles. The images on the right were taken within two minutes of each other into the test but they are very different. At 200 A the active thermochromic strip is 30-35°C while at 300 A the active strip is 40-45°C. Also, the temperature gradient is stronger at 300 A in the absence of cooling. Figure 3.5 (left) shows mid-range yellow and green shades at the top (32-33°C), but no indication of blue. In contrast, see Figure 3.3, the top of the cell is nearly at the blue limit of the strip, nearing 45°C.

![Figure 3.5 Left: temperature visualization 16 minutes into the 200 A test without cooling. Right: the same image 47 minutes into the test with cooling. As with the 300 A test, the cooling system effectively removes heat from the top of the cell where it is most intensely generated.](image)

### 3.2.3 100 A Charge and Discharge Cycling

The next test was run at 100 A (Figure 3.6). At such a low current much less heat is generated. Even after one hour of 100 A charge and discharge cycles in the
absence of cooling, the temperature only climbs to 32°C. However, it is possible that
the temperature may exceed the 50°C threshold in a longer test. With the cooling
system operating at 0.0315 kg s\(^{-1}\) per channel the color play on the TLC strips was
never activated, indicating that no region on the cell ever exceeded 25°C. Moreover,
the thermistor on the cell’s positive terminal (Tab 5) clearly shows a steady state
temperature of about 19°C which is well under the threshold temperature.

It is also important to note the temperature distribution in the cell (Figure 3.7).
Both panels in Figure 3.7 show TLC images without cooling at 11 minutes (left), and
54 minutes (right) into the 100 A test. At the beginning of the test the top is hotter
than the bottom resulting is a slight, positive, vertical temperature gradient. As time
elapses the heat is able to distribute itself evenly throughout the cell, creating a more
uniform temperature profile as shown in Figure 3.7. Such an even distribution is
possible because the cell does not generate as much heat at 100 A as it does at higher
currents. The slight differences in temperature profiles of regions 3 and 24 in Figure
3.6 at later times can be attributed to the margin of error of the TLC temperature
measurement.
Figure 3.6 Temperature versus time for the 100 A test with and without cooling. At such a low current, the threshold temperature is not approached even after one hour of operation. However, there is no indication of steady state without cooling and the temperature may eventually exceed 50°C. With a coolant flow of 0.0315 kg s$^{-1}$, Tab 5 reaches a steady state below 20°C, and the TLC strips are never activated.
Figure 3.7 Left: temperature visualization 11 minutes into the 100 A test without cooling. Right: 54 minutes into the same test also without cooling. Note that there is a slight (positive) temperature gradient at 11 minutes but by the end of the test the temperature profile appears to be nearly uniform. Lower currents cause a smaller heat generation and therefore more even temperature distribution. The actual temperature values for these cases are shown in Figure 3.6.

### 3.3 Total Heat Flux Measurements

For the next set of tests the coolant flow rate was reduced to 0.0063 kg s$^{-1}$ per channel to increase the temperature rise of the coolant from inlet to exit. As a result, the change in coolant temperature could be adequately resolved by the thermistors whose least count is ±0.2°C.

The heat extracted by the coolant is given by:

\[
Q_{\text{channel}} = \dot{m}C_p(T_{\text{outlet}} - T_{\text{inlet}}) \tag{3.1}
\]
where $Q_{\text{channel}}$ is the heat extracted by the cooling channel, $\dot{m}$ is the coolant mass flux, and $C_p$ is the specific heat of water. At steady state, the heat balance for the cell can be expressed as:

$$Q_{\text{cell}} = Q_{\text{NC}} + Q_{\text{channel}} - Q_{\text{tabs}} \quad (3.2)$$

where $Q_{\text{cell}}$ is the heat generated by each cell, $Q_{\text{NC}}$ is the heat lost due to natural convection from its exposed surfaces, and $Q_{\text{tabs}}$ is the heat generated due to contact resistance at the tabs. Note that Equation 3.2 is only valid at steady state because the amount of heat stored in the aluminum plates and the cell itself has not been included in the analysis. The heat generated in the tabs is estimated as:

$$Q_{\text{tabs}} = I^2 R \quad (3.3)$$

To find the contact resistance, a current was run through the cells and the voltage drop was measured across the tabs:

$$R = \frac{V}{I} \quad (3.4)$$

The average contact resistance was computed as 2.4E-5 Ω. To find the heat lost by natural convection we use:
\[ Q_{NC} = Ah(T_{cell} - T_{ambient}) \]  \hspace{1cm} (3.5)

where \( A \) is the exposed surface area, and \( h \) is the heat transfer coefficient, estimated at 5 Wm\(^{-2}\)K\(^{-1}\) [20].

The energy balance was performed by running the same charge and discharge cycles as before at 100, 200, and 300 A. The data in Figure 3.8 were generated from Equation 3.2.

Figure 3.8 Total heat removed \((Q_{\text{channel}} + Q_{NC})\) versus time for 100, 200 and 300 A cycles. These curves approach steady state in a manner similar to the temperature versus time graphs.
The heat generated by the cell during 300, 200 and 100 A charge cycles is determined following the procedure outlined in Equations 3.1-3.5. Note again that this analysis is only valid at steady state. Table 3.1 lists the steady-state values of $Q_{\text{channel}}$, $Q_{\text{NC}}$, $Q_{\text{tabs}}$, and $Q_{\text{cell}}$ for the three current values tested.

Table 3.1 Values of heat generation and removal rates computed from Equations 3.2-3.6 for the three current demands at steady state.

<table>
<thead>
<tr>
<th>Current (A)</th>
<th>$Q_{\text{channel}}$ (W)</th>
<th>$Q_{\text{NC}}$ (W)</th>
<th>$Q_{\text{tabs}}$ (W)</th>
<th>$Q_{\text{cell}}$ (W)</th>
</tr>
</thead>
<tbody>
<tr>
<td>100A</td>
<td>16.1</td>
<td>0</td>
<td>0.2</td>
<td>15.9</td>
</tr>
<tr>
<td>200A</td>
<td>38</td>
<td>1</td>
<td>1</td>
<td>38</td>
</tr>
<tr>
<td>300A</td>
<td>94.8</td>
<td>4.3</td>
<td>2.2</td>
<td>96.9</td>
</tr>
</tbody>
</table>

For 300 A the time period from 40-45 minutes is examined because the cell temperature remains nearly constant. At 200 A the average heat removed from 50 to 55 minutes is used for the same reason. For 100 A charge cycles, the heat lost to natural convection can be ignored because the cell temperature never exceeds room temperature. These data were examined in the 30-35 minute time period. As expected, Table 3.1 indicates that the heat generation rate within the cell increases with the charge/discharge current to a maximum value of 96.9 W at 300 A. Yet, the cooling system developed here is able to restrict the cell temperature to within the desired limit.
3.4 Conclusions

By developing a method to determine the entire surface temperature field of a lithium-titanate cell we were able to draw important conclusions and develop a path forward for further research. First, these cells do not generate heat evenly throughout their volume. Instead, a higher amount of heat is generated in the regions of the cell closest to the terminals. Thermochromic liquid crystal thermography clearly revealed the temperature gradients across the face of the cell. Battery temperature measurements were conducted at 100, 200, and 300 A charge/discharge cycles. At 300 A, the cell temperature rises very rapidly proving that an active cooling system is definitely required. At 200 A, although the temperature rise is slower, an active cooling system is still required. At 100 A, the heating rate is small enough that active cooling may not be required. The actual heat generated by the cell during each of these current cycles was also experimentally measured as 16, 38 and 97 W at 100, 200 and 300 A, respectively through the energy balance conducted in Equations 3.1-3.5.

We also determined that the liquid-cooled system is a viable option for the thermal management of the cells. During the 300 A test the cooling system was able to restrict the steady state cell temperature below 50°C. In actual driving conditions the current demand is rarely this high. The cells will most often be functioning in the 100-200 A range. Our cooling system was easily able to cool the batteries at these currents and approach a steady state well below the maximum operating temperature. Liquid cooling, however, may not be the best option as it requires a chiller to cool the liquid and a pump to circulate it through all of the cooling channels. With EVs
carrying hundreds of cells, the parasitic power required to drive such a system would be substantial. Hence, the next chapter describes our effort to design, implement, and evaluate an air-cooled system that can effectively manage the thermal environment of the cells with minimal parasitic losses.
Chapter 4

AIR-COOLED DESIGN

In the previous chapter, a liquid-based system was shown to be a viable option as an active cooling system for lithium-titanate batteries. It was also realized that any liquid-cooled system potentially requires a substantial amount of power to operate. Traditional liquid-cooled designs incorporate a closed loop system in which the active fluid would absorb heat from the batteries and reject it to the ambient by means of an on-board chiller unit. In addition, a pump would also be required to circulate the fluid at the desired mass flow rate. The resulting parasitic power losses would reduce the overall efficiency of the EV.

As previously stated, the purpose of this research was to design a cooling system to effectively manage the heat generated by the cells during normal vehicle operation while minimizing the parasitic losses. With this goal in mind, the implementation of an air-cooled system was explored. Such a system could, in theory, use ambient air to cool the cells during operation, eliminating the need for a closed loop system. The obvious concern with such an open-loop system is its loss of effectiveness when operating in hot ambient conditions. This situation would require consideration during the design stage of the air-cooled system.
4.1 System Design and Metal Foam Specifications

The liquid-cooled system described earlier provided valuable data that proved useful in the design of an air-cooled system. At 200 A of charge current, the heat generated by each cell is calculated as approximately 40 W, which served as a suitable target for assessing the feasibility of various designs. These values replicate actual operating conditions in which the batteries are operating under large current demands. It was apparent that simply creating a gap between adjacent cells and blowing ambient air through it would not provide an adequate heat transfer coefficient to effectively remove the heat from the cells, especially when hot ambient conditions are considered. For this reason the design had to incorporate a method to increase the effective heat transfer coefficient across the face of the cells. Accordingly, a heat-exchanger plate was designed that incorporated a panel of open-cell aluminum foam placed between adjacent cells. To manage the direction of air travel and to protect the surface of the battery from damage, an aluminum encasement was constructed to house the porous medium. This design is shown in Figure 4.1.
Figure 4.1 Exploded view of the assembly. The black panel represents the aluminum foam while the red arrows mark the direction of air flow. The air inlet is located at the top to match the region of the cell with the highest heat generation rate. The aluminum foam is sandwiched between aluminum sheets (1.27 mm thick) to complete the encasement.

In order to distribute the air evenly between cells, a manifold was constructed from aluminum box channel. The manifold was fitted with a thermistor and pressure gauge to monitor the temperature and pressure of the incoming air. A rubber air hose was used in conjunction with barbed pipe fittings to connect the manifold to each cooling plate. The air flow rate was controlled with a regulator on the compressed air line in the lab and monitored with a rotameter installed upstream of the manifold. Given the design constraints of our application it was necessary to select aluminum foam with the optimal specifications. Open cell porous media are defined by their material of construction, relative density, and pore size defined in terms of pores-per-inch (PPI). Based on physical space limitations of the battery compartment the gap
between each cell must be 9.5 mm. Accordingly, the optimal pore size was determined to be 1.3 mm, thus allowing seven pores to exist across the gap between adjacent cells. In order to determine the optimal density the chart in Figure 4.2, published by ERG Duocel, was used.

![Flow Resistance of ERG's Reticulated Metal Foams](image)

Figure 4.2 Pressure gradient vs. velocity for aluminum foam with 20 pores per inch and various densities [21]. These data were used to determine the power required to drive the blower at the desired flow rate.

The data in Figure 4.2 led to the conclusion that a relative density of 8% was optimal for our application. With these specifications, a high flow rate could be achieved to increase the effective heat transfer coefficient while minimizing the pressure drop across the foam thus decreasing the power required to operate the system. Prior to construction, calculations were conducted to confirm that the
theoretical level of cooling was sufficient for effective thermal management of the
cells in a worst case scenario. These heat transfer calculations are outlined next.
First, the required theoretical heat transfer coefficient, $h$, was calculated as

$$h = \frac{q}{A \Delta T} \quad (4.1)$$

where $q$ is known to be 40 W at 200 A, $A$ is the surface area of the cell, and $\Delta T$ is the
temperature difference between inlet air and the surface of the cell. To obtain a
conservative estimate, $q$ is set to 50 W, and $\Delta T$ to 10°C. The surface area of the cell
is equal to

$$A = 0.20 \times 0.22 \times 2 = 0.088 \text{ m}^2 \quad (4.2)$$

where the multiplier of 2 accounts for the surfaces on both sides of the cell. Equations
4.1-4.2 then yield a heat transfer coefficient of 56.5 W m$^{-2}$ C$^{-1}$. Finally, the procedure
outlined in [22] was followed to ensure this value could be achieved. The calculations
are shown below in Equations 4.3-4.5.

$$h = \sqrt{3\pi} \left( \frac{k_s}{a} \right) \left[ CPr \left( \frac{k_f}{k_s} \right)^{1/2} \left( \frac{a\nu_f}{\rho_f} \right)^{3/2} \frac{1}{\left( \frac{a}{v_f} \right)^2} \right] \quad (4.3)$$

where $k_s$ and $k_f$ are the thermal conductivities for the solid and fluid, respectively, $a$ is
the pore size, $Pr$ is the Prandtl number, $V_f$ is the fluid velocity, $\nu_f$ is the kinematic
viscosity of the fluid, $\rho$ is the relative density of the metal foam, $d$ is the predicted diameter of the spinels that connect each pore, and $C$ and $n$ are constants based on Reynolds number and the geometric characteristics of the porous media. The following equations describe relative density, $\rho$, and the estimation for the spinel diameter, $d$.

$$\rho = \frac{\rho^*}{\rho_s}$$  \hspace{1cm} (4.4)

where $\rho^*$ is the density of the porous media, and $\rho_s$ is the density of a solid block of the same material and volume.

$$d \approx a \left(\frac{2}{\sqrt{3\pi}}\right) \rho^{\frac{1}{2}}$$  \hspace{1cm} (4.5)

where $a$ is pore size, and $\rho$ is relative density as defined in Equation 4.4.

The results in Table 4.1 are for two different airflow rates, each yielding heat transfer coefficients within an acceptable range. This reference [22] also states that the theoretical values for heat transfer coefficient are often significantly higher than measured values due to simplifications in the model. In order to obtain accurate values for pressure drop across the porous media and corresponding parasitic power loss, the pressure of the inlet air was measured for each flow rate experimentally.
Table 4.1 Pressure drop, heat transfer coefficient, and parasitic power required to achieve the two prescribed air flow rates per heat-exchanger plate assembly.

<table>
<thead>
<tr>
<th>Q(mls⁻¹)</th>
<th>ΔP(Pa)</th>
<th>h(Wm⁻²K⁻¹)</th>
<th>Parasitic Power(W)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1100</td>
<td>5171</td>
<td>90.44</td>
<td>5.69</td>
</tr>
<tr>
<td>800</td>
<td>3447</td>
<td>80</td>
<td>2.76</td>
</tr>
</tbody>
</table>

Parasitic power was computed using:

\[
Power = Q \times ΔP \quad (4.6)
\]

where \( Q \) is volumetric flow rate, and \( ΔP \) is pressure drop across the metal foam. With the theoretical results showing promising values for both the heat transfer coefficient and parasitic power consumption, construction of the air-cooled system commenced.

4.2 Fabrication

The testing of the air-cooled system was conducted with the same experimental set up as described in Section 3.1. Thermochromic liquid crystal thermography was again employed to determine the surface temperature distribution of the front-most cell in the 5-cell stack. A cooling plate was placed in between adjacent cells, leading to the assembly shown in Figure 4.3. Each cooling plate consisted of a sheet of aluminum foam within an aluminum encasement as described earlier in Section 4.1. Aluminum foam with 20 PPI and 8% relative density was custom ordered from ERG Duocel to fit the dimensions 4.77 mm x 177 mm x 222 mm. The front and back
aluminum plates that sandwich the foam measured 1.27 mm thick. This plate thickness was selected to minimize thermal resistance and to promote ease of construction. The entire assembly required complete air sealing in addition to mechanical robustness so TIG welding was chosen to fasten the design together. This is a tedious process and therefore would not be a reasonable choice for full-scale production. However, this option sufficed for the small number of plates needed in this project.

Figure 4.3 Solid model of the battery pack with cooling plates inserted between adjacent cells in the testing stand.
In order to promote efficient heat transfer between the cooling plates and the cells a method for compressing the assembly of cells was created. This fixture needed to be very rigid and cover the entire surface area of the cell while still allowing optical access to the TLC strips via the camera. To accomplish this, a 12.7 mm thick steel cover plate was used that included a milled viewing window, and eight holes for threaded rods. This design is shown in Figure 4.4. The cross section of the assembly of cells and cooling plates is shown in Figure 4.5 with the compression plates at the front and the back. The insulating material on the front of the stack is the doubled paned Plexiglas fixture that was used in the water-cooled system. Plywood was used as the insulating material on the back of the stack.

Figure 4.4 Experimental setup showing the compression arrangement. The steel compression plate has two windows milled into it for visual access to the TLC strips. Compression springs inserted over lengths of threaded rod are used to adjust the compression force exerted on the cells.
Figure 4.5 Schematic of the cross section of the entire assembly. The front of the stack is on the right side, as indicated by the location of the TLC strips.

Die springs were employed to accurately control the amount of compression applied to the stack. The springs slide over each of the eight threaded rods that are situated along the left and right borders of the compression plate as shown in Figure 4.4. The force exerted by each spring can be determined as the product of its deflection, $x$, and the spring constant, $k$: 
Springs were ordered with a spring constant of 174.8 N/cm with a maximum force of 266.4 N, which when multiplied by a total of eight springs, supplied adequate compressive force for this application. The springs also allowed for the thermal expansion of the batteries and cooling plates during charge/discharge cycling. This eliminates the development of uncontrolled thermal stresses on the cells as they approach their maximum temperature and avoids the possibility of permanent cell damage. In compliance with specifications from Altairnano the applied compression force on the cells was kept below 1776 N. A study was also conducted to assess the effect of cell compression on steady state temperature and cell thermal behavior. The details of this study can be seen in Appendix A.

A full schematic of the completed testing system is shown in Figure 4.6. With the air cooling assembly constructed a testing procedure could be implemented to quantify the effectiveness of the system.
Figure 4.6 Layout of the air-cooled system showing coolant flow configuration and measurement devices for battery testing.

4.3 Battery Testing and Behavior

Through initial testing it was discovered that the thermal behavior of the battery was highly dependent on its state-of-charge. Figure 4.7, acquired from Altairnano, describes internal impedance versus state-of-charge (SOC) for both charge and discharge currents. Figure 4.7 reveals that during low SOC (high states-of-discharge) the cell’s internal resistance during discharge cycles is approximately 50% higher than during charge cycles. On the other hand, the internal impedance of the cell is essentially independent of current direction during high SOC.
Figure 4.7 The difference in internal impedance during charge and discharge cycles is drastic at low states-of-charge. Therefore, battery thermal behavior is a direct function of state-of-charge.

The primary mechanism of heat generation is due to the battery’s internal resistance as given by Equation 2.1. Due to the linear dependence of power on internal resistance, at low SOC the heat generated during discharge is expected to be 50% higher than the heat generated during charge. In contrast, the heat generation is expected to be independent of current direction at high SOC. In order to illustrate this phenomenon experimentally a trial was conducted at 175 A at both low and high SOC (Figure 4.8). As SOC varies linearly with cell voltage, the battery SOC % was determined as follows.

\[
SOC = \frac{(V_{Avg} - V_{MIN})}{(V_{MAX} - V_{MIN})} \times 100 \quad (4.8)
\]
where $V_{\text{Avg}}$ is the average cell voltage for all the cells in the stack, $V_{\text{MIN}}$ is 1.81 V and $V_{\text{MAX}}$ is 2.79 V.

Figure 4.8 Temperature from the TLC strips (left axis), and battery SOC (right axis) are plotted against time for charge-discharge cycling at 175 A with a coolant airflow rate of 950 mls$^{-1}$ per heat-exchanger plate. The oscillation of the temperature curve is only apparent at low SOC (up to 90 min), clearly demonstrating the effect of SOC on internal resistance and ultimately battery thermal behavior.

The cells were operated using 5-min charge and discharge cycles for the duration of the experiment. The results from this trial are displayed in Figure 4.8. It is seen that the temperature profile during charge-discharge cycling exhibits large periodic fluctuations only at low SOC (up to 90 min) for which the internal resistance during discharge is substantially higher than during charge. It is also interesting to
note that the temperature fluctuations are perfectly negatively correlated to the SOC fluctuations. As the battery SOC climbs (i.e. charging) the cell resistance is low, and the heat generation rate is smaller than the heat removal rate, due to which the cell temperature falls. During discharging, the battery SOC drops while the heat generation rate climbs faster than the heat removal rate leading to a positive temperature excursion. At 90 min, the battery SOC is increased to values exceeding 50%, and the temperature fluctuations vanish as expected because the internal resistances during charge and discharge converge to the same value at high SOC as indicated in Figure 4.7, and the heat removal rate matches the heat generation rate. Such steady-state conditions are preferable for battery thermal analysis, and hence, all further testing was conducted at high SOC.

Under typical operating conditions the batteries experience current demands between 100 and 200 A. For this reason these two current values were selected for comprehensive testing. The testing procedure outlined in Chapter 3 was also applied to the air-cooled system.

### 4.4 Results

The first test was run at 200 A and used an airflow rate of 1100 mls$^{-1}$. This flow rate was chosen because it was the maximum rate available in the lab and hence provided the highest heat transfer. The h values and pressure drop for this flow rate are taken from the first row of Table 4.1. Figure 4.9 shows the temperature profile for Region 9 (the center of the cell) in addition to inlet and outlet temperature data.
Battery SOC is also shown on the right axis of Figure 4.9. It is seen that the cooling system converges to a steady-state differential of 10°C between the outlet and inlet air temperatures.

Figure 4.9 Temperature at the inlet and outlet thermistor and the temperature at region 9 as measured by the TLC along with battery SOC as a function of time for charge-discharge cycling at 200 A with a coolant airflow rate of 1100 mls$^{-1}$ per heat-exchanger plate. Operating at high SOC suppresses periodic temperature fluctuations leading to consistent steady-state temperature profiles.

Figure 4.9 demonstrates that an air-cooled system utilizing aluminum foam to increase the effective heat transfer coefficient can be used to successfully manage the heat generated by the cells during normal vehicle operation. The outlet air temperature is consistently smaller than the TLC temperature measured at Region 9 at
steady state because outlet air temperature is measured at the bottom of the cell, while Region 9 is at the center of the battery. The y-gradient in the heat generation source term leads to this temperature distribution. The maximum temperature achieved in this trial is under 35°C, which is well within the safe operating range prescribed by the manufacturer. Given the 10°C temperature rise of the cooling air at steady-state, it can be conjectured that this cooling system could limit the battery temperature to 50°C even at ambient operating temperatures as high as 40°C. These results support the conclusion that the air-cooled system is an effective solution for maintaining a safe cell operating temperature even under very hot ambient operating conditions with current demands of 200 A.

In addition, another trial was conducted with a reduced coolant airflow rate of 800 mls⁻¹ per heat-exchanger plate. Obtaining steady-state temperature data at two different airflow rates will generate the required information to formulate and validate the constitutive form of the heat generation rate in the battery by comparing the temperature predictions from a numerical model of heat flow in the battery with the experimental values as will be explained in Chapter 5. The calculated values for pressure drop and parasitic power consumption for this case are listed in the second row of Table 4.1. Due to the decreased heat transfer coefficient, the steady-state temperature of the cell is expected to increase. These results are shown below.
Figure 4.10 Temperature and battery SOC as a function of time for charge-discharge cycling at 200A with a coolant airflow rate of 800 mls$^{-1}$ per heat-exchanger plate. Note that the steady-state temperature rise of the incoming air is 14.5°C as compared to 10°C for the flow rate of 1100 mls$^{-1}$.

It is important to note that the steady-state differential between the outlet and inlet air temperatures of 14.5°C indicates that a flow rate of 800 mls$^{-1}$ per heat-exchanger plate can effectively reject the heat generated by the batteries in ambient conditions up to 35°C. Thus, we arrive at the obvious conclusion that the reduced effective heat transfer coefficient associated with the smaller airflow rate lowers the maximum allowable ambient operating temperature.

The next test demonstrated the full effectiveness of the cooling system at 100 A of charge-discharge cycling, therefore the original mass flow rate of air, 1100 mls$^{-1}$ was employed. Current demands of this magnitude replicate conditions at the lower
end of vehicle operation. The results from this trial are displayed in Figure 4.11. As before, Equation 4.8 was used to calculate the battery SOC. Note that the temperature from Region 9 does not appear until nearly 40 minutes into the test because it is below the lowest bandwidth TLC strip which can only register temperatures higher than 25°C. The results from this test further confirm the feasibility of using ambient air as the active fluid in battery cooling systems.

![Figure 4.11 Temperature and battery SOC as a function of time for charge-discharge cycling at 100 A with a coolant airflow rate of 1100 mls⁻¹ per heat-exchanger plate. The system is shown to converge to a steady-state differential of only 4°C between the outlet and inlet air temperatures.](image)

Figure 4.11 Temperature and battery SOC as a function of time for charge-discharge cycling at 100 A with a coolant airflow rate of 1100 mls⁻¹ per heat-exchanger plate. The system is shown to converge to a steady-state differential of only 4°C between the outlet and inlet air temperatures.
4.5 Data Analysis

The steady-state temperature results obtained in Section 4.4 allow a complete thermal analysis of the air-cooled system by applying an energy balance to the cells. The following calculations are presented for the trials that employed an airflow rate of 1100 mls\(^{-1}\). The energy balance calculation is represented by Equation 3.3. The value of \(Q_{\text{tabs}}\) is known at both 100 and 200 A from Section 3.3. In determining the value of \(Q_{\text{NC}}\) it is important to consider the primary heat removal mechanism. The location of the cooling channel in the water-cooled system resulted in intense heat removal directly at the top the cell. This condition led to tab temperatures that did not rise significantly above ambient conditions. The air-cooled system employed a different flow configuration that resulted in higher tab temperatures and therefore greater heat removal by natural convection, given by Equation 3.6.

The contribution of the heat exchanger plate was calculated using Equation 3.2. The results from this analysis are shown in Table 4.2.

Table 4.2 Results from the energy balance analysis giving the relative magnitudes of heat generation and removal terms.

<table>
<thead>
<tr>
<th>Current (A)</th>
<th>(Q_{\text{plate}}) (W)</th>
<th>(Q_{\text{NC}}) (W)</th>
<th>(Q_{\text{tabs}}) (W)</th>
<th>(Q_{\text{cel}}) (W)</th>
</tr>
</thead>
<tbody>
<tr>
<td>100A</td>
<td>6</td>
<td>1</td>
<td>0.2</td>
<td>6.8</td>
</tr>
<tr>
<td>200A</td>
<td>16</td>
<td>5</td>
<td>1</td>
<td>20</td>
</tr>
</tbody>
</table>

In high-capacity, high-current cells such as the 50 Ah lithium-titanate batteries studied here, the dominant mechanism of heat generation is ohmic heating due to the electrical...
resistance as described by Equation 2.1. Therefore, the values of $Q_{cell}$ in Table 4.2 can be compared directly to the value of heat generated by ohmic losses. Altairnano specifies the internal resistance of a single cell to be 0.00055 $\Omega$. An inspection of Figure 4.7 yields a similar value, and most importantly, shows that the value of internal resistance is independent of current direction at high SOC. Equation 3.3 then yields heat generation values of 22-24 W and 5.5-6 W for current values of 200 A and 100 A, using internal resistance as 0.00055 and 0.0006 $\Omega$ respectively. Considering the uncertainty in value of $h$ for natural convection, these values closely match the results shown in Table 4.2, confirming the validity of the energy balance.

A comparison of heat generation values listed in Table 4.2 and the corresponding values estimated in Table 3.1 for the water-cooled system reveals a discrepancy. While the values for $Q_{cell}$ in Table 4.2 for 100 A and 200 A are 7 W and 20 W, respectively, the corresponding values for the water-cooled system were estimated at 15.9 W and 39.5 W, respectively. The reason for this discrepancy is attributed to the difference in the battery SOC during the water-cooled and air-cooled tests. Figure 4.8 indicates that fluctuation in cell heat generation during charge vs. discharge is largely governed by the difference in the internal resistance of the cell at low SOC rather than due to entropic considerations. The trials for the water-cooled system were conducted at low SOC, which led to pronounced fluctuations in heat generation, and therefore, in the resulting cell temperature as well. The energy balance performed with the water-cooled data pertained to the case of heat generation at low SOC, where the internal resistance of the cell is substantially higher. Figure 4.7
indicates an average electrical resistance value of 0.00095 Ω at low SOC which leads to ohmic heat generation of 9.5 W and 38 W for 100 and 200 A, respectively. These values are similar in magnitude to the $Q_{cell}$ estimates presented in Table 3.1. Ultimately, the experimental trials using both cooling systems have approximately quantified the heat generated by the cells at both low and high SOC’s. These results are essential in designing a thermal management system that is effective under all operating conditions.

4.6 Conclusions

The design, construction, and evaluation of a cooling system for 50 Ah lithium-titanate cells that utilizes ambient air as the active fluid has been described. The key component of the cooling system is a heat exchanger based on open cell aluminum foam to increase the effective heat transfer coefficient. Experiments indicate that the temperature rise of the battery is restricted to just 10°C above ambient even under charge-discharge cycling at 200 A. The steady-state temperature results shown in Figure 4.9 and Figure 4.10 provide encouraging data that suggest that an air-cooled system is a realistic solution for thermal management of these batteries. Moreover, calculations for parasitic power consumption indicate that the energy overhead of such a cooling system is very small which increases overall vehicle efficiency. The airflow rate and parasitic power consumption on a per-plate basis are 1100 mls$^{-1}$ and 5.69 W, respectively.
The obvious benefit of the air-cooled system is the elimination of on-board coolant storage and a chiller unit leading to saving in vehicle weight and parasitic power consumption. For example, the current cooling system on the University of Delaware’s fuel cell bus consists of a 400 W pump and a 1600 W chiller with a 50% duty-cycle during normal ambient conditions. Therefore, the average parasitic loss due to battery thermal management amounts to 1200 W, with a maximum draw of 2000 W. We can compare these values to the conclusions drawn in Table 4.1 by multiplying the computed value by 300 which represents the number of heat exchanger plates required by the UD fuel cell bus. These values are 1707 W and 828 W for 1100 mls⁻¹ and 800 mls⁻¹ respectively. Thus, the reduced power consumption for the air-cooled system is encouraging even before considering the benefits of curb weight reduction.

The air-cooled system, however, can benefit from further improvements. The first improvement involves the manufacturing process of the heat exchanger plates. While the current design employed TIG welding, alternative manufacturing techniques would become necessary to enable mass production. Second, the design evaluated here utilized a laboratory compressed-air line; for vehicle applications, an on-board compressor would need to be selected and evaluated. Another option for providing the desired airflow rate is a vacuum pump which would draw ambient air through the cooling plates from a downstream location.

The next chapter will utilize the experimentally acquired temperature data to determine a function that describes the spatially-varying heat generation term within
the cells. An accurate model for the heat generation term would allow one to design a
more robust cooling system for other configurations of this battery pack and predict
temperature rise for battery packs subjected to different environmental boundary
conditions.
Chapter 5

CHARACTERIZATION OF HEAT GENERATION SOURCE TERM USING A
FINITE DIFFERENCE SCHEME

5.1 Introduction

The experimental observations described in Chapters 3 and 4 indicated that lithium-titanate batteries do not generate heat uniformly throughout their volume. Figure 5.1 clearly demonstrates this phenomenon. Non-uniform heat generation is mainly due to the configuration of the cell itself. The location of the positive and negative terminals at the top of the cell implies that the current entering and leaving the cell experiences its highest flux density towards the top of the cell than at the bottom. According to the law of ohmic heating as seen in Equation 2.1, this naturally leads to elevated levels of heating near the cell’s tabs.

Figure 5.1 visually demonstrates the higher density of current distribution and collection in proximity to the terminals. While this is easily understood, it would be desirable to quantify the variation of the heat generation source term with position. Initial testing revealed that heat generation was nearly constant in the x-direction. Therefore it was decided to characterize the heat generation term as a function of the vertical coordinate (y-direction) within the cell. This would allow one to perform
accurate thermal modeling of lithium-titanate batteries and hence help design the most efficient cooling strategies for them.

![Figure 5.1 Schematic of current flow within the cell with arrow length corresponding to current density magnitude. Current density is larger near the tabs leading to higher ohmic heating towards the top of the cell.]

The method used here to characterize the heat generation term exploits the temperature data collected during charge/discharge cycling in Chapter 4. A finite-difference model is constructed for a unit cell consisting of a single lithium-titanate cell attached to a metal-foam heat exchanger plate as shown in Figure 5.2. The model accounts for all of the thermal resistances encountered as the heat generated within the cell travels laterally to the heat-exchanger plate and is then removed by the air. Next, a
guessed parabolic profile for the heat generation term is inserted into the model and the steady-state temperature distribution on the surface of the cell is obtained. The predicted temperature distribution is then compared with the measured temperature distribution for the same experimental conditions, and the heat generation profile is adjusted until the error between the predicted and measured distributions is minimized.

The following sections describe the numerical model, governing equations and boundary conditions, the finite-difference scheme, and the results.

5.2 Numerical Model

In order to apply a two-dimensional finite-difference method, a cross section of the front cell and heat exchanger plate assembly is modeled as shown in Figure 5.2. The numerical domain is a composite consisting of three layers: the cell, the Al plate, and the small air gap between them. Heat is removed via the heat exchanger plate from the left side of the battery, while all other surfaces are assumed to be thermally insulated. These conditions closely approximate the actual experimental parameters for the air-cooled system from Chapter 4. The finite-difference scheme was implemented in Matlab to obtain a steady-state temperature distribution throughout the numerical domain.
Figure 5.2 Numerical model for characterizing the heat generation term within the lithium-titanate cell. The right edge of the full cell is where the TLC strips are located, therefore the steady state temperature distribution ($T_{ss}$) is known on that face. Heat is removed by air flowing through the Al foam heat exchanger plate to the left of the Al plate; all other surfaces are assumed to be thermally insulated.

5.3 Governing Equations and Boundary Conditions

The steady-state heat conduction equation was solved within the numerical domain to obtain the complete thermal mapping of the cell. The equation was solved only in $x$ (along the thickness of the cell) and $y$ (the vertical coordinate). Temperature variations in the out-of-plane direction are ignored. The governing equation is shown below.
Here $k$ is thermal conductivity of the corresponding material in each layer. It is assumed to be independent of $x$, $y$, and temperature. $G$ represents internal heat generation in Wm$^{-3}$ and is assumed to only vary in the $y$ direction.

The boundary conditions are as follows:

Right surface insulated: \[ \frac{\partial T}{\partial x} = 0 \]

Top and bottom surfaces insulated: \[ \frac{\partial T}{\partial y} = 0 \]

Left surface convection: \[ \frac{\partial T}{\partial x} = \frac{h(\Delta T)}{k} \]

The governing equation is discretized as follows for implementation within the finite difference code:

Internal Nodes:

\[
k \left[ \frac{(T_{i+1,j} + T_{i-1,j} - 2T_{i,j})}{(dy)^2} + \frac{(T_{i,j+1} + T_{i,j-1} - 2T_{i,j})}{(dx)^2} \right] + G_i = 0 \quad \text{(5.2)}
\]

where $i$ and $j$ represent nodes in the $y$ and $x$ directions, respectively, as shown in Figure 5.3. Equation 5.2 can be rearranged to isolate $T_{i,j}$ on the left hand side:
\[ T_{i,j} = \frac{1}{2} \left( \frac{(dx)^2(dy)^2}{(dy)^2+(dx)^2} \right) \left[ \frac{G_i}{k} + \frac{T_{i+1,j} + T_{i-1,j}}{(dy)^2} + \frac{T_{i,j+1} + T_{i,j-1}}{(dx)^2} \right] \] (5.3)

Equation 5.3 was then applied to all the internal nodes within the finite difference code. The domain boundaries corresponding to the mesh in Figure 5.3 include four edges and four corner points. The finite difference treatment for a selection of these is shown below.

**Insulated corner node: i = 1, j = 12**

\[ \frac{T_{i-1,j} - T_{i,j}}{dx} = 0 \] (5.4)

\[ \frac{T_{i,j+1} - T_{i,j}}{dx} = 0 \] (5.5)

**Insulated edge nodes: i = 2-25, j = 12**

\[ \frac{T_{i,j+1} - T_{i,j}}{dx} = 0 \] (5.6)

**Convection surface: i = 1-26, j = 2**

\[ \frac{T_{i,j-1} - T_{i,j}}{dx} = \frac{h(T_{air,i,j-1} - T_{i,j})}{k} \] (5.7)

It is important to note that the equations corresponding to the Al plate and the air gap did not contain a heat generation term.
A finite difference mesh was created to solve the equations along with the prescribed boundary conditions. Temperature data were acquired with the help of the TLC strips at 26 equispaced locations along the vertical coordinate as described in Chapter 4. This number was therefore a natural choice for the number of nodes along the vertical axis. The $x$-axis was divided into 12 nodes: eight for the full cell, two for the contact resistance region, one for the aluminum plate, and one for the convecting air. The $y$-axis contains cell elements of equal length ($dy$). The $x$-axis, however, uses two different lengths ($dx$) values as the thickness of the layer for contact resistance is very small in comparison to the other layers. Figure 5.3 depicts the mesh used in the finite difference method; note that the nodes are not drawn to scale.
Figure 5.3 Mesh configuration used in the finite difference method. The node spacing along the $x$-direction is smaller for the contact resistance layer than the other layers. The nodes are equispaced along the $y$-axis.

The initial temperature of the entire mesh, with exception of the left boundary, was set at 22°C to represent ambient thermal conditions. The air temperature on the left boundary, $T_{\text{AIR}(i,1)}$, is prescribed by interpolating between the inlet and outlet air temperatures which are measured using thermocouples as described in Chapter 4. The interpolation is required because the Al foam heat exchanger plate is completely
sealed making it impossible to measure the actual air temperature distribution \( T_{\text{AIR}(i,1)} \). The interpolation method is explained next.

The cooling air is introduced at the top and travels downward through the heat exchanger plate. Due to the intense heat generation at the top of the cell, air temperature is expected to rise significantly upon entering the heat exchanger plate leading to a high magnitude of \( dT/dy \). As the air flows downward through the aluminum foam, it passes through cell regions that experience decrementally smaller levels of heat generation leading to a lower magnitude of \( dT/dy \). Therefore, the air temperature profile is assumed to be parabolic in \( y \), increasing rapidly at the top of the cell and reaching a maximum at the bottom of the cell; thus, the air temperature gradient is matched to the local level of heat generation. The vertical extent of the cell is divided into 26 regions, with Region 1 located at the top, and Region 26 at the bottom. Hence the expression for the parabola that describes air temperature is expressed as:

\[
T_{\text{AIR}(i,1)} = -A(i - 26)^2 + C \quad (5.8)
\]

\( C \) is the measured outlet air temperature, and \( A \) is obtained by setting the air temperature in Region 1 to the measured inlet air temperature.

As stated in Section 4.4, 200 A tests were conducted at airflow rates of 800 and 1100 mls\(^{-1}\). In addition, we will examine the results from the 100 A test at 1100 mls\(^{-1}\). The
temperature profiles and equations corresponding to these three cases are shown in Figure 5.4 and Equations 5.9-5.11, respectively.

![Figure 5.4](image)

**Figure 5.4** The prescribed temperature profile of the air as it flows through the porous Al foam heat exchanger for the three experimental trials. The temperature gradient is higher at the inlet (Region 1) due to more intense heat generation near the terminals. Inlet and outlet air temperatures were obtained from thermocouple measurements.

\[
200 \text{ A, 800} \text{ } ml/s^{-1}: T_{\text{AIR}_{(L1)}} = -.0242(i - 26)^2 + 38 \quad (5.9)
\]

\[
200 \text{ A, 1100} \text{ } ml/s^{-1}: T_{\text{AIR}_{(L1)}} = -.0176(i - 26)^2 + 33.5 \quad (5.10)
\]

\[
100 \text{ A, 1100} \text{ } ml/s^{-1}: T_{\text{AIR}_{(L1)}} = -.0057(i - 26)^2 + 26 \quad (5.11)
\]
The heat flux leaving the Al plate, $Q_{out}$, is dictated by the local air temperature value $T_{AIR}(y)$ and an effective heat transfer coefficient that incorporates the porous morphology and thermal conductivity of the aluminum foam. The heat transfer coefficients employed in the code were determined by conducting the following heat balance:

$$\dot{m}C_p(T_{outlet} - T_{inlet}) = h \sum_{y=1}^{26} A(T_{cell}(y) - T_{air}(y))$$  \hspace{1cm} (5.12)

Here $\dot{m}$ is the mass flow rate of air, $C_p$ is the specific heat of air, $T_{outlet}$ and $T_{inlet}$ represent the exit and inlet air temperatures, respectively, $h$ is the effective heat transfer coefficient, $A$ is the area of the corresponding region of the cell, $T_{cell}(y)$ is the temperature at the left edge of the Al plate, and $T_{air}(y)$ is the air temperature profile as shown in Figure 5.4. Due to the small thickness of the cell and its high thermal conductivity, it is assumed that the temperature at the left edge of the Al plate is approximately equal to the measured temperature on the surface of the cell. Equation 5.12 is solved to obtain the value of the effective heat transfer coefficient, $h$.

Comparing the experimentally determined values in Table 5.1 to the theoretical values in Table 4.1 reveals a discrepancy. The experimentally determined values are lower by 51% and 58% for the 1100 mls$^{-1}$ and 800 mls$^{-1}$ flow rates, respectively. This difference is attributed to the simplifications employed in [22] while deriving the theoretical values.
The material properties and geometric parameters used in the solution of the governing equation are listed in Table 5.1. The values for contact resistance were estimated using the data from [23].

Table 5.1 Material properties and geometric parameters employed in the finite difference solution.

<table>
<thead>
<tr>
<th>Units</th>
<th>Material/Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Thermal Conductivity (Wm⁻¹K⁻¹)</td>
<td>Air</td>
<td>0.02</td>
</tr>
<tr>
<td></td>
<td>Aluminum</td>
<td>200</td>
</tr>
<tr>
<td></td>
<td>Battery</td>
<td>100</td>
</tr>
<tr>
<td>Length (m)</td>
<td>dx</td>
<td>0.0017</td>
</tr>
<tr>
<td></td>
<td>dy</td>
<td>0.0091</td>
</tr>
<tr>
<td></td>
<td>dair</td>
<td>0.0001</td>
</tr>
<tr>
<td></td>
<td>Total X</td>
<td>0.0242</td>
</tr>
<tr>
<td></td>
<td>Total Y</td>
<td>0.2286</td>
</tr>
<tr>
<td>Heat Transfer Coefficient (Wm⁻²K⁻¹)</td>
<td>hₘ₈ 1100 mls⁻¹</td>
<td>44.26</td>
</tr>
<tr>
<td></td>
<td>hₘ₈ 800 mls⁻¹</td>
<td>33.84</td>
</tr>
<tr>
<td>Contact Resistance (m²KW⁻¹)</td>
<td>R</td>
<td>4.53E-04</td>
</tr>
</tbody>
</table>

The numerical solution can now proceed to quantify the function that governs the heat generation source term within the lithium-titanate cell.

5.4 Characterization of Heat Generation G(y)

As previously discussed, heat generation is not constant along the y-axis of the cell. For this reason, a function that describes the distribution of heat generation per unit volume along the y-axis was developed. From initial testing it is known that the cell generates heat most intensely near the terminals (see Figure 5.1). A simple argument suggests that the heat generation term cannot be linear in y. If it is assumed
that the reaction rate within the battery electrodes is uniform across the plane of the cell, then the number of electrons generated per second per unit cell area will also be uniform across the plane. All of these electrons will stream from or towards the positive or negative terminals, respectively. At the bottom of the cell, the flux of electrons will be small, but the flux will increase linearly with $y$ as the terminals are approached. Since heat generation goes as $I^2$, this simple model suggests that the function for heat generation should be quadratic in $y$. Accordingly, the heat generation source term can be expressed as:

$$G(y) = a(y - b)^2 + c \quad (5.13)$$

The constant $b$ in Equation 5.13 is set to 26 which ensures that the local value of heat generation is the smallest at the bottom of the cell. The constants $a$ and $c$ fully describe the nature of the heat generation term; they are determined by running the finite difference scheme and searching for those values of $a$ and $c$ that minimize the error between the predicted and measured temperature fields on the right face of the cell. In order to conduct this search in a methodical manner, the variable $a$ needs to be prescribed carefully. This was accomplished by writing a searching function in Matlab. The first step in this process was to specify an operating range for the search by introducing the variable $p$ defined as:
Here $G(1)$ and $G(26)$ are the heats generated in the top-most and bottom-most mesh elements, respectively. Equation 5.13 indicates that $G(26) = c$. For example, if $p$ is equal to 10, then the heat generation at the top of the cell will be 10 times that at the bottom. The program then calls for inputs to define the range and interval over which $p$ will vary so that many different heat generation source terms can be tested. A similar input is required for the variable $c$ as shown below.

\[ p = \frac{G(1)}{G(26)} \quad (5.14) \]

\[ p = p_{\text{min}}; p_{\text{int}}; p_{\text{max}} \]

\[ c = c_{\text{min}}; c_{\text{int}}; c_{\text{max}} \]

Once these variables are defined, the searching function was employed to determine the value of $a$ that creates a parabola with its minimum equal to $c$ in Region 26, and its targeted value, $p \times G(26)$, in Region 1. This process is repeated for each combination of $p$ and $c$ values.

Figure 5.5 shows six different heat generation profiles obtained by inserting their corresponding $a$ and $c$ values into Equation 14. The ranges specified for this particular search are:

\[ p = 2; 2; 6 \]

\[ c = 20,000; 20,000; 40,000 \]
Note that the value of $a$ is unique for each heat generation source term as the shape of the parabola varies according to the values of $p$ and $c$.

![Graph showing heat generation vs. cell region for different $a$ and $c$ values](image)

Figure 5.5 Heat generation vs. cell region for different $a$ and $c$ values as determined from Equation 5.13. The $p$-values were chosen as 2, 4, and 6 for each case.

After setting up the numerical scheme and modeling the heat generation source term in this manner, the program was used to cycle through various values of $p$ and $c$ to determine the steady state temperature distribution on the right face of the cell for each heat generation source term. The system was assumed to have reached steady state when the temperature at any node changed less than 0.00001°C between iterations. Then, the program compared the rightmost column ($T_{i,j=12}$) to the
experimentally measured steady state temperature profile on the face of the battery by computing the RMS error:

\[ RMS_{err} = \sqrt{\frac{\sum_{i=1}^{26} (T_{i,j=12} - T_{ss}(i))^2}{26}} \] (5.15)

The code stores the RMS error values for each heat generation source term in a matrix that is defined by both \( c \) and \( p \)-values. In addition, it also records the total heat generation and \( a \)-values in similar matrices. The entire process can be visualized in the flow chart shown in Figure 5.6.

![Flow Chart](image)

Figure 5.6 A flow chart that describes the methodology used to arrive at a heat generation source term that produces the correct steady state temperature distribution. A similar process is applied for all three experimental cases to generate a complete set of numerical data and validate the initial projected heat generation source term.
The heat generation source term that minimizes the RMS error between the predicted and measured steady state temperature distributions can be identified by examining the RMS error matrix. In this way we can determine an accurate heat generation source term for the batteries. First, the heat generation source term function is determined for the 200A, 1100mls\(^{-1}\) test. This same function is then applied to the 200A, 800 mls\(^{-1}\) test. If the heat generation term is accurate, then it should also minimize the error between the predicted and measured temperature profiles for this second case. The source term is also modified and applied to the experiment conducted at 100A and 1100mls\(^{-1}\). The resulting temperature data is then checked against the actual experimental conditions. Such a comparison would help to validate the empirically derived heat generation source term, \(G(y)\).

5.5 Results and Discussion

Figure 5.7 shows a contour plot for the RMS error for the 200 A, 1100 mls\(^{-1}\) test. Note that there appear to be multiple minima in the contour plot. Each local minimum represents heat generation source terms that produce temperature profiles that closely match the experimental profile. However, the absolute minimum is identified by the red circle in Figure 5.7. By examining these points and locating the absolute minimum we arrive at the temperature profile that most precisely replicates the
experimental data. Therefore the heat generation source term can be correctly characterized.

Figure 5.7 Contour plot of the RMS error for the 200 A trial using a flow rate of 1100 mls$^{-1}$. The minimum RMS error value is circled.
Figure 5.8 shows a comparison of the surface temperature profiles that were measured for the 200 A, 1100 mls$^{-1}$ trial, and the corresponding numerical solution. The noise in the experimentally measured profile is consistent with the ±0.5°C error incurred during TLC thermography as discussed in Section 2.2. The heat generation source term that was obtained for this case is shown in Figure 5.9.
Figure 5.9 Numerically obtained heat generation source term vs. cell region for 200 A, 1100 mls\(^{-1}\).

The following table shows the relevant values associated with the above results.

Table 5.2 The values associated with the heat generation source term that is fit to the 200 A, 1100 mls\(^{-1}\) trial.

<table>
<thead>
<tr>
<th>Trial</th>
<th>200 A 1100ml/s</th>
</tr>
</thead>
<tbody>
<tr>
<td>RMS Error</td>
<td>0.152</td>
</tr>
<tr>
<td>Watts Generated</td>
<td>22.78</td>
</tr>
<tr>
<td>a-value</td>
<td>132</td>
</tr>
<tr>
<td>c-value</td>
<td>2500</td>
</tr>
<tr>
<td>p-value</td>
<td>34</td>
</tr>
</tbody>
</table>
Note that the heat generated at the top is 34 times the heat generated at the bottom. This, by itself, is useful in designing the system. The value for RMS error was computed using Equation 5.15 while $a$, $c$ and $p$-values represent parameters of the heat generation source term as defined in Equations 5.13 and 5.14. The total heat generated from the cell was obtained as:

$$
\int_{1}^{26} A(a(y - 26)^2 + c) \, dy
$$

(5.16)

where $a$ and $c$ define the heat generation source term and $Ady$ is the elemental volume of each battery node.

Next, this heat generation source term, described by parameters $a$ and $c$ from Table 5.2, is applied to the 200 A, 800 mls$^{-1}$ trial. The resulting steady-state temperature profile is shown in Figure 5.10 along with the experimental data at that flow rate.
Figure 5.10 Comparison of the numerical and measured temperature profiles for the 200 A, 800mls$^{-1}$ trial. The reasons for the mismatch in the two profiles are described in the text.

Figure 5.10 shows a clear mismatch between the experimental and numerical results. The mismatch is due to simplifications in the numerical model. The effect of these simplifications is explained next with the help of Figure 5.11.
Figure 5.11 A cross sectional view that highlights the difference between the actual experimental conditions and the simplified model. The contribution of natural convection is ignored in the numerical model, thus skewing the resulting temperature profile for the 200A, 800mls$^{-1}$ case.

For the lower flow rate of 800 mls$^{-1}$, the heat removal by forced air convection is reduced which increases the steady state temperature distribution within the cell. Most importantly, the battery terminal temperature ($T_{\text{Tab}}$) increases greatly as can be confirmed by comparing Figures 5.8 and 5.10. Consequently, the resulting heat loss from natural convection also increases. Although heat removal by natural convection is not included in the numerical model, it is not significant for the higher airflow rate of 1100 mls$^{-1}$, and hence the numerical model is adequate for that case. However, it
creates a noticeable discrepancy when the heat generation source term for the 1100 mls\textsuperscript{-1} case is directly applied to the 800 mls\textsuperscript{-1} trial as seen in Figure 5.10. In order to replicate the experimental profile more accurately for the 800 mls\textsuperscript{-1} case, the heat generation source term at the top of the cell would have to be augmented to account for the additional heat loss due to natural convection.

Finally, the heat generation source term was also applied to the trial at 100 A and 1100 mls\textsuperscript{-1}. The law of ohmic heating in Equation 1.1 indicates that the heat generated at 200 A is four times larger than that at 100 A. Therefore, the magnitude of the heat generation source term for 200 A was decreased by a factor of 4 before applying it to the 100 A case. This reduced heat generation source term is shown in Figure 5.12. Figure 5.13 shows a comparison of the numerical and experimental temperature profiles for this case.
Figure 5.12 The projected heat generation source term for the trial at 100 A, 1100 mls⁻¹. According to the law of ohmic heating, the heat generation for this case is reduced from the original predicted source term (Figure 5.9) by a factor of 4.
Figure 5.13 shows that the experimental data are slightly higher than the values predicted by the numerical model. To explain this discrepancy, the energy balance conducted in Table 4.2 is examined to determine the total experimental value of heat generation per cell. The 100 A trial indicates that approximately 7 W of heat is produced at this current. However, the heat generation source term shown in Figure 5.10 only produces 5.68 W of total heat. These values indicate that while the law of ohmic heating predicts a 75% decrease in heat generation from 200 A to 100 A, the experimental values show only a 64% reduction. Hence, the generation source term
from the 200 A trials, shown in Figure 5.9, was instead reduced by a factor of 3 producing the temperature profile shown in Figure 5.12.

Figure 5.14 Experimental and numerical results for a factor of 3 reduction in the heat generation source term. This adjustment is justified based on the experimental results, and produces a steady state temperature profile that closely matches the experimental data.

These results suggest that the law of ohmic heating is not solely responsible for the internal heat generated by the cell, especially at lower currents. The methods of heat generation within cells are discussed in detail in Section 1.1. At higher charge/discharge currents, it is expected that the heat generated by the batteries would
be dominated by ohmic heating. This behavior is evident in the trials for 200 A. However, when the total current is reduced to 100 A, ohmic heating does not account fully for the total heat generation within the cell as shown in Figure 5.10, Figure 5.11, and the experimental results from Section 4.5. Therefore, a more accurate thermal model at 100 A features a heat generation source term that is reduced by a factor of 3, rather than 4.

The results presented here have successfully demonstrated an empirical method for characterizing the heat generation source term within the cells. In addition, this term was applied to two other sets of experimental conditions to verify its accuracy. Discrepancies in the validation studies were shown to arise from model simplifications and internal cell behavior. Ultimately, the simplified finite difference method offers a reasonable method for determining the correct heat generation source term which can be usefully employed for thermal analysis and management of lithium-titanate cells.

5.6 Conclusions

A finite difference method was implemented to solve the heat conduction equation within a numerical domain consisting of a single cell attached to an Al foam heat exchanger plate with appropriate initial and boundary conditions. This numerical scheme was used to empirically determine the spatially-varying heat generation source term by employing a search procedure that minimized the error between the numerically predicted and experimentally measured temperature profiles on the surface of the cell. In order to ensure that this methodology yielded accurate results,
the heat generation term obtained by fitting data for one case was applied to two other
different experimental cases. The findings show that the numerically obtained heat
generation source term provides a reasonable estimate of cell’s thermal behavior.
Discrepancies between the predicted and measured temperature profiles were
satisfactorily explained.

Ultimately, it was shown that the above finite difference approach to quantify
the heat generation term as a quadratic function is indeed a good approximation of cell
behavior. The assumption of a quadratic heat generation source term was confirmed
and the coefficients for this function were obtained. These results follow the logic that
the concentrated current densities near the terminals will lead to intense heat
generation at the top of the cell. The numerical exercise presented here builds
confidence that this model can be effective in predicting temperature distributions in
various cell cooling designs. Hence, it constitutes a useful design tool whereby
alterations in cooling system designs can be evaluated numerically prior to
construction. Coupling this ability with the promising results shown in Chapter 4 can
lead to a complete design and analysis of an air-based thermal management system for
50 Ah lithium-titanate batteries. Eventually, a system of this nature can be
implemented in future designs of BEVs and HEVs.
Chapter 6

SUMMARY, CONCLUSIONS AND FUTURE WORK

This research has provided a comprehensive understanding of the thermal behavior of 50 Ah lithium-titanate cells in addition to proving the effectiveness of two cooling systems under laboratory conditions.

Battery and hybrid electric vehicles impose a challenging set of demands on the battery systems used for energy storage such as long life cycles, high charge/discharge rates, good energy density, and consistent performance throughout their lifetime. The lithium-titanate batteries studied here possess a good combination of these parameters and thus represent an attractive solution for automotive energy storage. The large current demands of these platforms, such as the fuel cell buses operated by the University of Delaware, lead to significant heat generation and make thermal management of batteries a high-priority concern. Another important design consideration is the minimization of parasitic power consumption by the battery cooling system in order to increase the overall vehicle efficiency.

At the outset, it was necessary to develop a technique to accurately map the surface temperature distribution of the lithium-titanate cells. Chapter 2 discusses in detail the design and implementation of thermochromic liquid crystal thermography to the battery stack. This technique exploits the change in color with temperature to instantaneously map the temperature over extended 2D domains and resolve spatial
temperature gradients. After acquiring and processing a series of calibration images, an image processing algorithm was designed in Matlab to convert color to temperature. The ability to determine cell temperature profiles not only provided a measure of effectiveness for the two cooling systems studied here, but also enabled a numerical method to characterize the heat generation source term to model the heat generated during the operation of the batteries.

With a method of determining cell temperature in place, cooling system design and fabrication could commence. The first, a water-cooled system, is described in Chapter 3. This design circulated chilled water through cooling channels that were welded to the edges of aluminum plates placed between each cell in the stack. The water-cooled system was extremely effective in dissipating heat even at 300 A of charge/discharge current. However, the laboratory provided convenient access to a chilled water line which would not be true in practical applications. On-board systems would require a coolant pump as well as a chiller unit to reject the heat to the environment. These two devices would consume significant parasitic power and therefore decrease overall vehicle efficiency. In addition, their added weight would increase the overall vehicle weight and result in further loss of efficiency.

Hence, we explored the possibility of using ambient air as the coolant fluid in Chapter 4. It became clear that simply circulating ambient air over a bank of cells would not provide the required heat removal rate. Therefore, a heat-exchanger plate incorporating open-cellular aluminum foam was designed to increase the effective heat transfer coefficient. The porous medium was housed in an airtight aluminum
encasement which controlled the direction of air flow. Forcing air through the aluminum foam heat exchanger was found to remove heat effectively. This system was tested using charge/discharge cycles at 100 and 200 A. The 200 A trials were conducted at two different flow rates. This set of experiments proved that ambient air could be employed as the coolant for thermal management of 50 Ah lithium-titanate cells through proper heat exchanger design.

Finally, Chapter 5 discusses the development of a numerical model to characterize the non-uniform heat generation source term of the lithium-titanate cell. The numerical model includes one cell and heat-exchanger plate and solves the steady-state heat conduction equation with appropriate boundary conditions. Based on the data from previous experiments, we have assumed that this source term is quadratic in the vertical coordinate. A search procedure is then employed wherein the heat generation function coefficients are varied systematically until the predicted temperature on the cell surface matches the measured temperature for the same experimental conditions. In this way, we empirically determined the correct heat generation source term for the cells. Validation against experimental data from the three different trials (combinations of current and airflow rates) provided additional confidence in our modeling results. The characterization of this heat generation term provides a more complete understanding of battery thermal behavior, and will allow future cooling system designs to be tested through numerical modeling prior to construction.
As previously discussed, this research has provided a unique experimental study of various active thermal management systems. The cell geometry enabled the utilization of liquid crystal thermography to provide a novel means of acquiring active surface temperature measurements of batteries. The complete thermal mapping of the cells enabled us to conduct a thorough quantitative assessment of each system's effectiveness. Through rigorous testing it was proved that proper implementation of open cell aluminum foam can increase the effective heat transfer coefficient at the battery-foam interface to enable the use of ambient air as the active fluid. By monitoring the conditions of the flowing air it was shown that this system can operate successfully in ambient conditions up to 40°C and reduce parasitic system losses. Ultimately, it has been proven that an air cooled battery thermal management system is a viable option in future BEV and HEV’s and even offers numerous advantages over traditional designs. A numerical method was also applied to this unique geometry to characterize the nature of heat generation within the cell. This information can prove valuable in future thermal management system designs for lithium-titanate cells.

6.1 Path Forward

This research has provided significant insight into the thermal behavior of lithium-titanate cells in addition to cooling methods that may prove beneficial to future designs. However, before a system level design can take place, further research must be conducted to quantify the exact level of cooling necessary for various current
demands. For example, our fuel cell buses are equipped with sensors that monitor and record the operating conditions of its various subsystems, including current demand on the battery stack. This information would enable us to characterize the exact current magnitudes throughout the entire drive cycle of the vehicle, especially during rapid acceleration and regenerative braking. These time-varying current levels can then be programmed on the battery stack in the laboratory and tested against various active cooling systems. The use of TLC thermography will provide a means to characterize the effect of real drive cycle conditions on the battery’s thermal response.

These tests would aim to prove that the maximum available airflow rate is not always necessary to maintain a cell temperature of under 55°C for an entire drive cycle. In order to reduce parasitic losses even further, a thermostat control could be employed to measure the temperature of the battery stack and adjust the air flow rate accordingly. Extensive testing would have to be conducted in laboratory conditions prior to vehicle installation to ensure that the thermostat operates reliably and safe operating temperatures can always be maintained.

In addition, further adjustments to the air-cooled design would be necessary before a full scale build could commence. The vehicle would require an air blower which would increase parasitic losses and vehicle curb weight. Other methods of providing the necessary airflow rate could also be explored. For example, a vacuum pump may suffice to draw coolant air through the stack; the pump would be located downstream of the heat exchanger plates and pull air through the channels rather than forcing it from an upstream location.
The method of constructing the heat exchanger plates should also be addressed. The small number of plates needed for our experiments allowed some flexibility in the manufacturing technique. Therefore, we used TIG welding to construct the aluminum encasements that housed the porous media. For a typical bus, up to 300 heat exchanger plates would be needed due to the increased number of cells in the battery stack, essentially eliminating welding as a fabrication method. Other forms of sealing the fixture, such as silicone gaskets, could be explored in laboratory conditions prior to implementation on a vehicle. Ideally, the thermostat control system would be tested after these adjustments were made such that any undiagnosed issues can be corrected.

Ultimately, we have addressed and solved many issues regarding thermal management of batteries for use on BEV’s and HEV’s. Through the use of a numerical model we were also able to characterize the heat generation source term that will assist in future battery thermal analysis. Traditional systems have used a liquid as the active fluid for thermal management, however, the use of air has not been explored in great detail. Future work in this field can build on the results from this research to obtain a system level design that both improves vehicle efficiency and supplies sufficient cooling to the battery stack. The calculations for parasitic power consumption in conjunction with experimental results provide encouraging insight that suggests that utilizing ambient air for cooling can provide a safe, reliable, and efficient method for lithium-titanate battery thermal management on electric vehicles.
REFERENCES


[3] A. Gotcher, Nanostructured electrodes: experts have recognized for year the advantages of lithium ion technology over traditional power batteries, particularly those based on lead acid chemistry, Advanced Materials and Processes. 163.12 (2005) 32-38.


Appendix A

THE EFFECT OF COMPRESSION ON CELL BEHAVIOR

A design of experiments approach to testing was employed to determine the effect of three parameters—current magnitude, air flow rate, and cell compression on battery thermal behavior. Specifically, we were interested in understanding the factors that influenced the magnitude of the cell’s temperature fluctuations during charge/discharge cycling. The results from this set of experiments are shown in the table below. It is evident from the table that while current magnitude and flow rate have a great effect, the contribution of cell compression is negligible.
Table A.1 The results from a series of tests to determine the effect of compression, flow rate and current on both the average steady state temperature and the magnitude of fluctuations. These results confirm that the contribution of compression is negligible.

<table>
<thead>
<tr>
<th>Run</th>
<th>Compression (lbf)</th>
<th>Air Flow (cfm)</th>
<th>Current (A)</th>
<th>Inlet Air T</th>
<th>SS Avg T</th>
<th>Fluctuation</th>
<th>Delta T</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>200</td>
<td>6.6</td>
<td>175</td>
<td>24.3</td>
<td>37.44</td>
<td>1.175</td>
<td>13.14</td>
</tr>
<tr>
<td>2</td>
<td>400</td>
<td>6.6</td>
<td>175</td>
<td>24.86</td>
<td>37.92</td>
<td>0.8</td>
<td>13.06</td>
</tr>
<tr>
<td>3</td>
<td>200</td>
<td>9.4</td>
<td>175</td>
<td>24.24</td>
<td>33.24</td>
<td>0.62</td>
<td>9</td>
</tr>
<tr>
<td>4</td>
<td>400</td>
<td>9.4</td>
<td>175</td>
<td>23.93</td>
<td>32.98</td>
<td>0.48</td>
<td>9.05</td>
</tr>
<tr>
<td>5</td>
<td>200</td>
<td>8</td>
<td>100</td>
<td>24.8</td>
<td>29.03</td>
<td>0.98</td>
<td>4.23</td>
</tr>
<tr>
<td>6</td>
<td>400</td>
<td>8</td>
<td>100</td>
<td>23.8</td>
<td>27.93</td>
<td>0.97</td>
<td>4.13</td>
</tr>
<tr>
<td>7</td>
<td>200</td>
<td>8</td>
<td>250</td>
<td>25.09</td>
<td>43.98</td>
<td>1.33</td>
<td>18.89</td>
</tr>
<tr>
<td>8</td>
<td>400</td>
<td>8</td>
<td>250</td>
<td>25.31</td>
<td>44.7</td>
<td>1.01</td>
<td>19.39</td>
</tr>
<tr>
<td>9</td>
<td>300</td>
<td>6.6</td>
<td>100</td>
<td>24.29</td>
<td>28.67</td>
<td>0.13</td>
<td>4.38</td>
</tr>
<tr>
<td>10</td>
<td>300</td>
<td>9.4</td>
<td>100</td>
<td>24.87</td>
<td>27.6</td>
<td>0.68</td>
<td>2.73</td>
</tr>
<tr>
<td>11</td>
<td>300</td>
<td>6.6</td>
<td>250</td>
<td>25.03</td>
<td>46.68</td>
<td>0.84</td>
<td>21.65</td>
</tr>
<tr>
<td>12</td>
<td>300</td>
<td>9.4</td>
<td>250</td>
<td>25.54</td>
<td>42.48</td>
<td>0.77</td>
<td>16.94</td>
</tr>
<tr>
<td>13</td>
<td>300</td>
<td>8</td>
<td>175</td>
<td>23.52</td>
<td>33.32</td>
<td>0.667</td>
<td>9.8</td>
</tr>
<tr>
<td>14</td>
<td>300</td>
<td>8</td>
<td>175</td>
<td>23.68</td>
<td>33.29</td>
<td>0.639</td>
<td>9.61</td>
</tr>
<tr>
<td>15</td>
<td>300</td>
<td>8</td>
<td>175</td>
<td>24.18</td>
<td>34.19</td>
<td>0.685</td>
<td>10.01</td>
</tr>
</tbody>
</table>

Eventually, we concluded that the magnitude of the fluctuation in cell temperature during charge/discharge cycling was not due to the amount of cell compression as initially suspected, but rather was due to the variation of cell internal resistance as a function of state of charge and current direction. This effect can be seen in Figure 4.7 and is discussed in detail in Section 4.3.