Passive Electric Vehicle Battery Thermal Management Solution

Literature Review:

The price of electric vehicle lithium ion batteries has been steadily going down over the past few years [1], however, even at a cost of $200 per kWh, battery components remain a significant factor contributing to the total price of an electric vehicle. As a critical component of an EV, batteries should be operated efficiently, and parameters affecting their performance and reliability should be controlled to optimize utilization.

Using accelerated life cycle tests, I. Bloom et al. [2] found that the operating temperature of a battery had a significant effect on its power fade over time. Cell capacity also degrades at a faster rate with a high operating temperature [3], P. Ramadas et al. [4] reported a capacity loss of %70 at 55° C compared to a %30 capacity loss at room temperature over 800 charge cycles. The rate of battery electrical impedance growth over time was also found to increase with higher operating temperatures in the same study. The impedance is directly tied to the internal losses which are converted to heat inside the battery. The increase in impedance has a negative effect on the efficiency of the battery, and if the generated heat is not managed well, thermal runaway can cause the battery to catch fire [5].

Thermal management is needed to optimize the efficiency, reliability, and safety of an EV battery, and in order to develop an effective EV battery thermal management solution, an accurate model capable of predicting its heat generation is required.

D. Bernardi et al. [6] developed a thermo-electric model based on the energy balance equation inside a battery. The model assumes a uniform temperature distribution inside the battery, and it has terms to account for electrical work, chemical reactions, heat transfer to the surroundings, mixing, phase changes, and changes in the heat capacity of the system. A simplified form of the ODE that omits mixing and phase changes is often used [7, 8].

The accuracy of thermoelectric models can be improved by using them with finite element analysis to get the effect of uneven temperature distribution throughout the battery [9, 10]. While the analysis done by A. De Vita et al. [10] assumed a constant heat transfer coefficient inside the battery, the study by C. R. Pals et al. [9] used a location variant thermal conductivity, H. Maleki et al. [11] developed an empirically fit state of charge dependent thermal conductivity model, and found that the conductivity changes by a factor of about 10 depending on the direction.

Battery heat generation models combined with models describing their internal thermal conductivity can be used to design and optimize thermal management solutions for EV batteries. Multiple studies have looked into direct air cooling with no fins for electric vehicle batteries [12, 13]. Seham Shahid et al [12]. Used CFD and heat transfer analysis combined with experimental data to study the effect of adding airflow redirection plenums in multiple configurations on the cooling performance and the temperature distribution of cylindrical batteries. The study reported that adding those plenums made the airflow more turbulent and improved the temperature uniformity of the cell by about %39. Heesung Park [13] did a similar analysis on prismatic cell lithium ion battery packs, the study found that small changes like adding pressure relief ventilation and designing the airflow manifold with a taper can significantly impact forced air battery cooling performance.

Maan Al-Zareer et al. [14] evaluated using a vapor compression cycle with R134a to cool a battery pack and improve its power output. The analysis used a one-dimensional electrochemical model combined with a three-dimensional heat and mass transfer model. The design with fully submerged cells managed to limit temperature rise to 1.6° C. Jushua Smith et al. [15] looked into using heat pipes inserted between prismatic battery cells to extract heat and move it to a liquid cooled plate away from the battery. The system was able to dissipate 50 W of heat load from each cell.

Multiple air cooling and liquid cooling methods were compared directly using the same model in two different studies [16, 10]. Armando De Vita et al. developed a thermal model of a battery and used it with 3-D CFD analysis to compare the performance of direct air cooling to liquid cooling on the same battery pack with different discharge conditions. Unsurprisingly, liquid cooling performed better in all of the conditions tested. The superiority of direct liquid cooling in terms of heat extraction performance was confirmed again in the study by Defan Chen et al. [16] where direct air cooling was compared with fin cooling, indirect liquid cooling, and direct liquid cooling. The study found that direct air cooling used the most amount of power, fin cooling added the highest amount of weight, indirect liquid cooling had the highest temperature variation within the cell, while direct liquid cooling performed better in terms of the amount of extracted heat. The author commented on the impracticality of direct liquid cooling design however, and recommended indirect liquid cooling as a more practical alternative.

Objectives and Expected Outcomes:

This study aims to optimize the design and evaluate the effectiveness of a passive fin cooling solution for electric vehicle battery packs. A model of a single prismatic cell of a lithium ion battery will be developed. After validating the battery model, a fin model will be developed and integrated with that of the cell. The integrated fin single cell model will be expanded to a full battery pack simulation using symmetry conditions. The fin length will be optimized to achieve a low battery temp rise, and the effectiveness of the cooling solution will be assessed at different vehicle speeds using peak and mean temp rise as cooling performance metrics.

Modeling Approach:

To estimate the heat generation term inside the battery, the thermo electric model developed by D. Bernardi et al. [6] will be used:

$$\dot{q}_T = I\,(V - U^{avg}) + I.T.\frac{\partial U^{avg}}{\partial T} = I^2 R_{int} + I.T.\frac{\partial U^{avg}}{\partial T} \quad (1)$$

Where:

$\dot{q}_T$: Internal heat generation   I: Current delivered   V: Voltage U^{avg}: Open-circuit potential   T: Cell temperature R_{int}: Internal resistance

The rightmost term $\left(I.T.\frac{\partial U^{avg}}{\partial T}\right)$ is the reversible entropic heat, and the derivative there is assumed to be constant in this study. The direction dependent thermal conduction and specific heat parameters are estimated using values from the study done by Maryam Ghalkhani et al. [17]. A set of transient three-dimensional differential equations was written as a MATLAB function, which was then called using ode15s to get the temperature distribution as a function of time.

A 3D transient model of the fin covering the front and back of the cell was also developed. The fin model is implemented as another set of differential equations nested inside the battery cell function.

Symmetry boundary conditions were used to cut the size of the cell and fin model nodes in half in the x direction, and then in half again in the z direction (Figure 1). The symmetry conditions also enable expanding the solution to a full battery pack made from many cells stacked in the x and z directions.

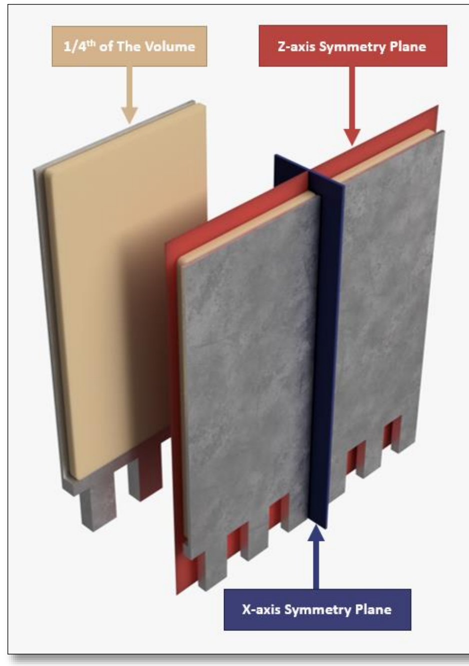

Figure 1: Cell and Fin Model Symmetry Planes

Initially, the air speed under the vehicle was assumed to be the same as that of the car, this assumption was later replaced with a Couette flow assumption. However, both of those assumptions gave unrealistic flow velocities that were not higher than the vehicle speed. The relative air speed under the car should be higher than that of the car since part of the air volume in front of the car needs to flow under it, and since the flow area under the car is smaller, the air velocity needs to be higher. The correlation between the relative air speed under the vehicle and the vehicle speed depends on the exterior design.

This study used a Nissan leaf as a reference vehicle since it uses prismatic cells similar to the ones modeled, and because it does not rely on active cooling methods to thermally manage the batteries. To get the correlation between the speed of the vehicle and the relative air speed under the car, a 3D model of the car was imported into ANSYS fluent, and a CFD analysis was carried out at vehicle speeds ranging from 0 to 35 m/s. The average relative air speed in an area matching that of the pack 5 cm from the underside of the vehicle was calculated at each vehicle speed. A linear fit was used to get a correlation between the air and vehicle speeds after (Figure 2). <u>Note: ANSYS was only used to get that correlation, and the remainder of the analysis was done using MATLAB.</u>

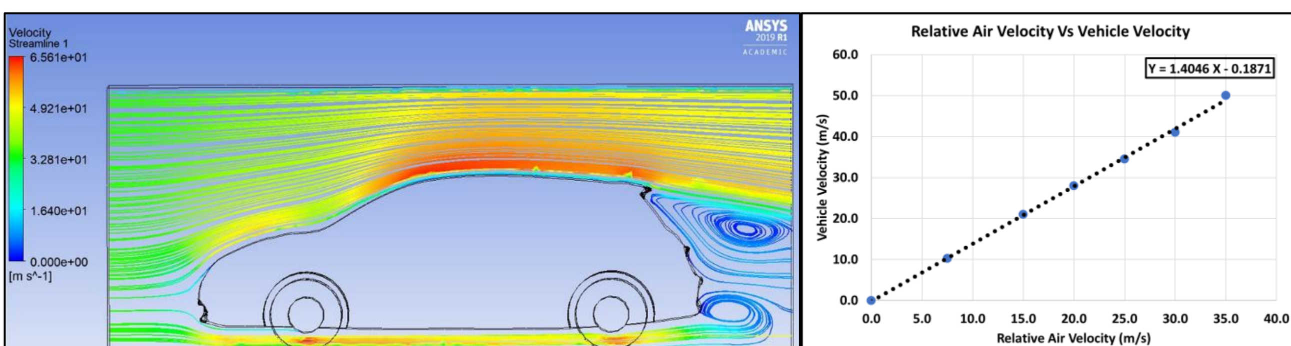

Figure 2: Velocity streamline around Nissan Leaf Moving at 35 m/s (left), Vehicle Velocity vs Air Velocity fit (right)

The convective heat transfer through the bottom of the fin was calculated using a model developed by P.Teertstra et al [18]. The model is intended for use with plate fin heat sinks, and it is based on an equation for Nusselt's number as a function of the fin geometry:

$$Nu = \left(\left(\frac{Re_b^*.Pr}{2} \right)^{-3} + \left(0.664\sqrt{Re_b^*}.Pr^{\frac{1}{3}} \sqrt{1 + \frac{3.64}{\sqrt{Re_b^*}}} \right)^{-3} \right)^{-\frac{1}{3}}, \qquad Re_b^* = Re.\frac{b}{l}, \qquad h = \frac{Nu.k}{b} \tag{2}$$

where: Re_b^*: *Channel Reynold number*, h: *convective heat transfer coeff.*, k: *conductive heat transfer coeff.*, b: *distance between fins*, l: *Fin length*

The total convective heat transfer is calculated using the fin efficiency (η_f) equation for a rectangular straight fin:

$$\eta_f = \frac{\tanh(m.L_c)}{m.L_c}, \qquad m = \sqrt{\frac{2.h}{kt}}, \qquad L_c = L + \frac{t}{2} \tag{3}$$

where: h: *convective heat transfer coeff*, k: *conductive heat transfer coeff*, t: *fin tip thickness*, L: *fin height*

The mechanical power used by the vehicle is calculated as the sum of power from drag, friction, and gravitational potential assuming a constant road grade (Slope):

$$MechanicalPower = Power_{drag} + Power_{friction} + Power_{Gravity}$$

$$Power_{drag} = V.\left(C_d.A_f.\rho.\frac{V^2}{2} \right), \qquad Power_{friction} = V.(m.g.RollingResistance), \qquad Power_{Gravity} = V.Grade.(m.g) \tag{4}$$

where: V: *vehichle velocity*, C_d: *drag coeff.*, A_f: *Car frontal area*, ρ: *air density*, m: *car mass*, g: *gravatational const.*, *Grade*: *Road slope*

The mechanical power is divided by a fixed battery to wheel vehicle efficiency and it is then converted to a current to be drawn through the battery at its nominal voltage of 360 V. The current calculated is divided up to the cells connected in parallel, modifying their heat generation term.

A function that wraps over the entire code and returns the max and mean cell temp rise was used to optimize the fin design, with the intent of keeping temp rise low while minimizing the fin cross sectional area. This was done to reduce added weight and lower the impact on the total vehicle drag coefficient. The parameter to be optimized was set as the fin height, and a cost function that penalizes area while rewarding a lower temp rise was used to get an optimal solution. The optimization was carried out at a vehicle speed of 80 MPH, which represents the worst case in terms of internal heat generation. The optimization study used grid search with a fixed discrete step size of 1mm.

Results and Validation:

To setup the battery cell model for validation, it will be set for an adiabatic transient battery test. To do that, the boundary heat transfer rates in all axes is set to zero, and a fixed current is drawn through the cell. The transient battery ODE is integrated numerically over a fixed amount of time, and the mean temperature as a function of time is calculated. The slope of the temperature was compared to published experimental and model-based results done by Defan Chen et al [16]. The internal electrical resistance and entropic loss coefficient were tuned to get the adiabatic test results as close as possible to those of the study, and the resulting tuned cell model was used for validation without changing those parameters.

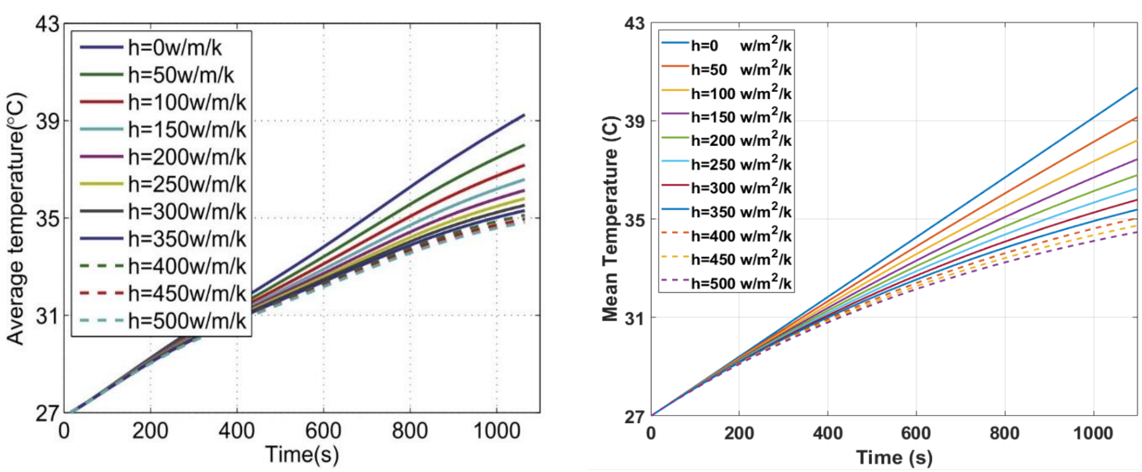

Figure 3: Results by Defan Chen et al [16] (Left) compared to the MATLAB ODE Simulation results (Right)

To validate the combined fin and battery cell model, the fin parameters and equivalent heat transfer coefficient was modified to match the cases tested in the same study [16]. The cell internal resistance and entropic loss coefficient were maintained at their previous values. A transient test calculating the average cell temperature as a function of time was run for multiple convective heat transfer coefficient values ranging from 0 to 500 $\frac{w}{m^2.k}$. Figure (3) shows the results of those tests compared to those of the study.

Since numerical data was not provided by the study, the results can only be compared visually. Looking at the intercept at time = 1000s, the total temperature rise relative to ambient (27 C) is within %10 of what was observed in the study for all convective heat transfer rates. The mismatch is likely due to the assumed constant state of charge, and the less accurate directional heat transfer rates (since those were set based on another cell model by Maryam Ghalkhani et al. [17]).

Equation set (4) was used to get the mechanical power used by the vehicle at different speeds. The drag coefficient, frontal area, and mass were set to those of a Nissan leaf vehicle. The road grade was set to 3%, and the rolling resistance was set to 0.015. The power was divided by an assumed vehicle efficiency of 60% and converted into a cell current assuming a 96 series 2 parallel battery pack cell configuration (40kWh Nissan Leaf Pack).

An optimization study that uses grid search (discrete parameter sweep) was used to find an optimal fin height that provides sufficient cooling while minimizing impact on weight and drag. The cost function weights were modified to give a solution with a low average and peak cell temp rise. Figure (4) shows a plots of the cost function, the mean temp rise, and max temp rise as a function of fin height at 80MPH vehicle speeds:

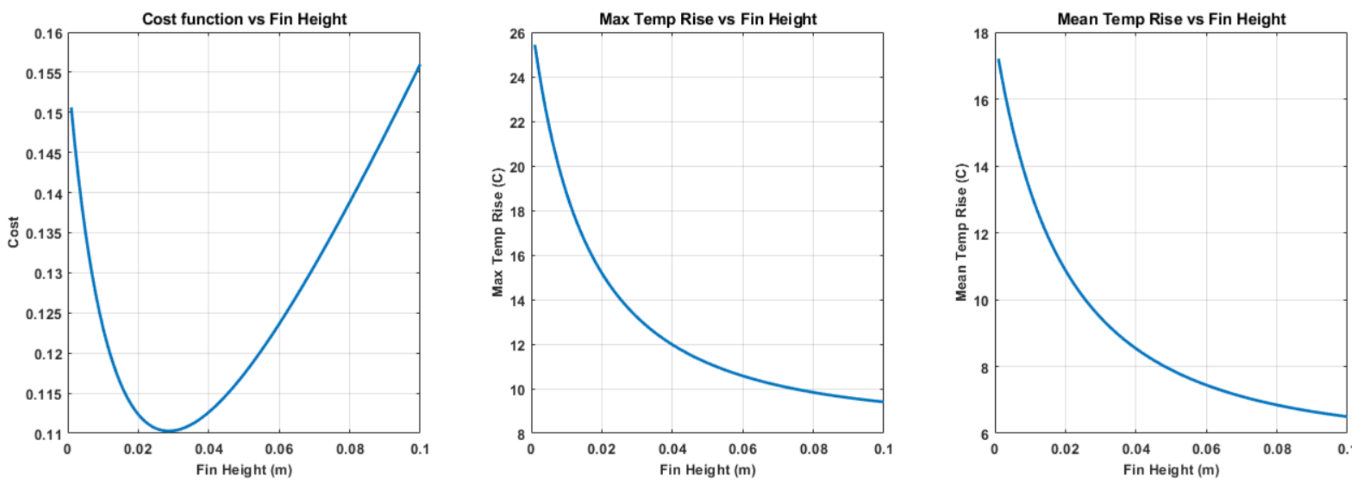

Figure 4: Fin height vs cost, peak temp rise, and mean temp rise at 80 MPH vehicle speed

The optimal fin height used was 29 mm and it achieved a peak temp rise of 13.3 C. The fin height was set to its optimized value for the remaining parts of the analysis.

The equivalent fin heat transfer coefficient was calculated using equation sets (2) and (3) at the corresponding relative air speed derived from the CFD linear fit. Figure (5) shows the cell current, fin efficiency, and the convective heat transfer coefficient at different vehicle speeds:

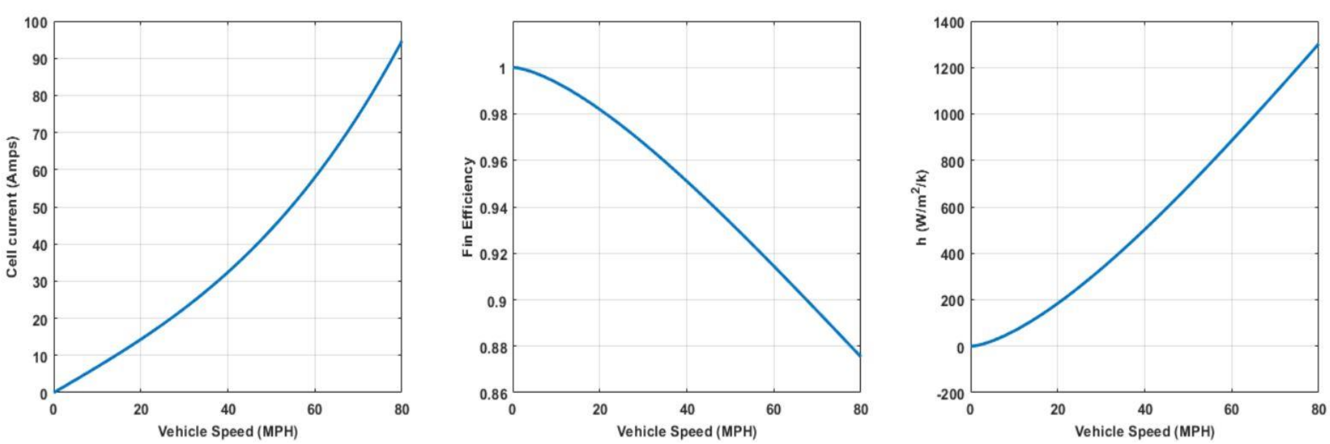

Figure 5: Cell Current, Fin Efficiency, and Convective Heat Transfer Coefficient as a function of Vehicle Speed

The following analysis was done at three vehicle speeds (50, 65 and 80 MPH) each with its corresponding convective heat transfer rate, and cell discharge current as an input to the battery ODE solver. The simulations were all carried out at an initial cell temperature distribution matching ambient at 25 C. ODE15s was used to integrate the differential equation set. Figure (6) shows the average temperature of the cell and fin jacket as a function of time at three different speeds:

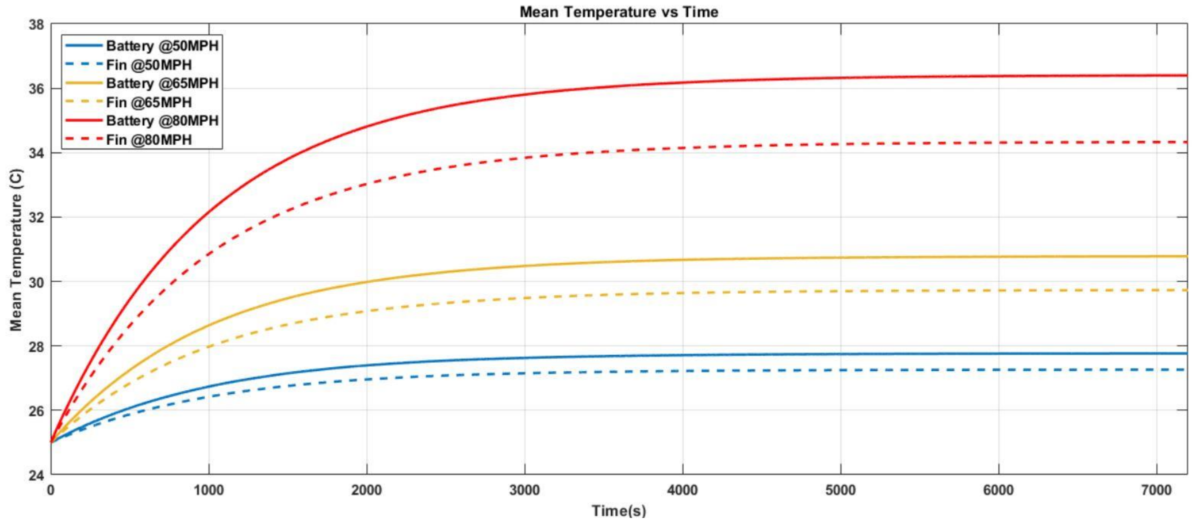

Figure 6: Mean Battery Cell and Fin Temperature at Different Vehicle Speeds over 2 Hours

Figure (7) shows the peak temperature of the cell and fin jacket as a function of time at three different speeds:

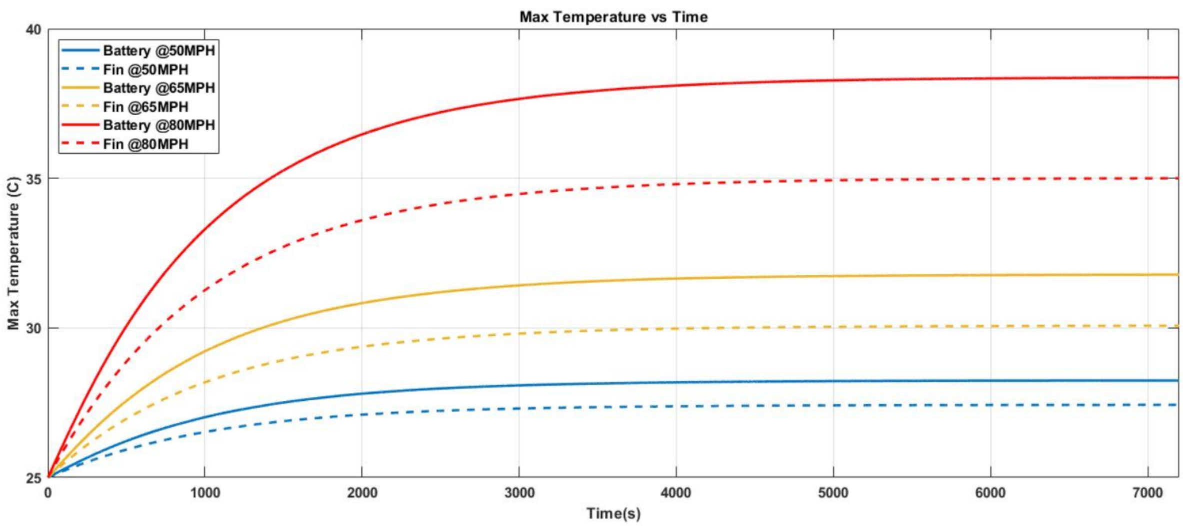

Figure 7 Peak Battery Cell and Fin Temperature at Different Vehicle Speeds Over 2 Hours

Figure (8) shows the total power dissipated out of the battery pack at speeds of 50, 65 and 80 MPH over a 2-hour run:

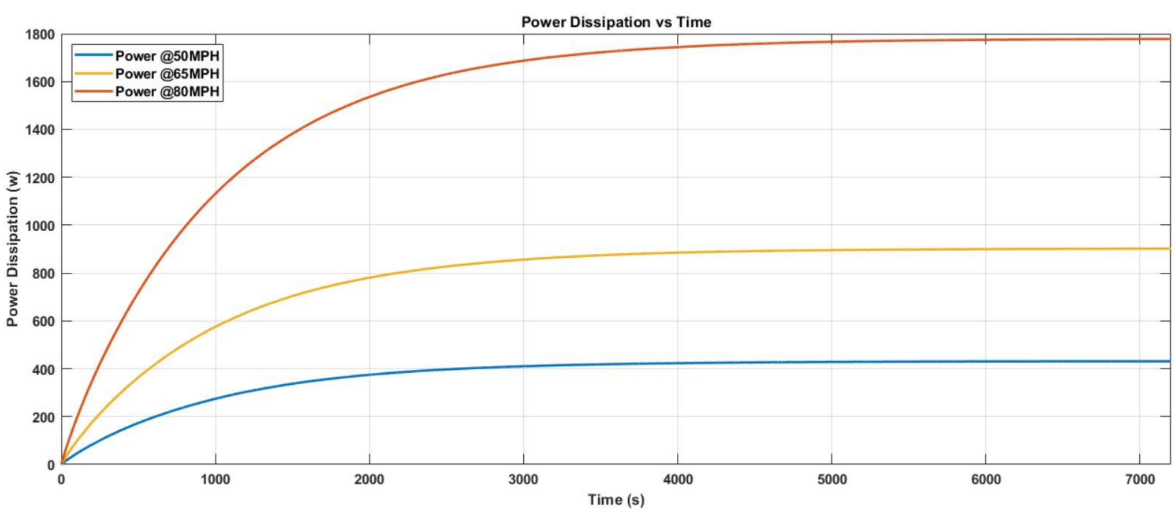

Figure 8 Peak Battery Cell and Fin Temperature at Different Vehicle Speeds Over 2 Hours

Figure (9) shows the temperature Z-Y distribution inside the cell and fin jacket. The fin jacket covering the side of the battery is the leftmost layer in the plot, the right side of the battery is not shown or modeled in the ode as it is symmetric:

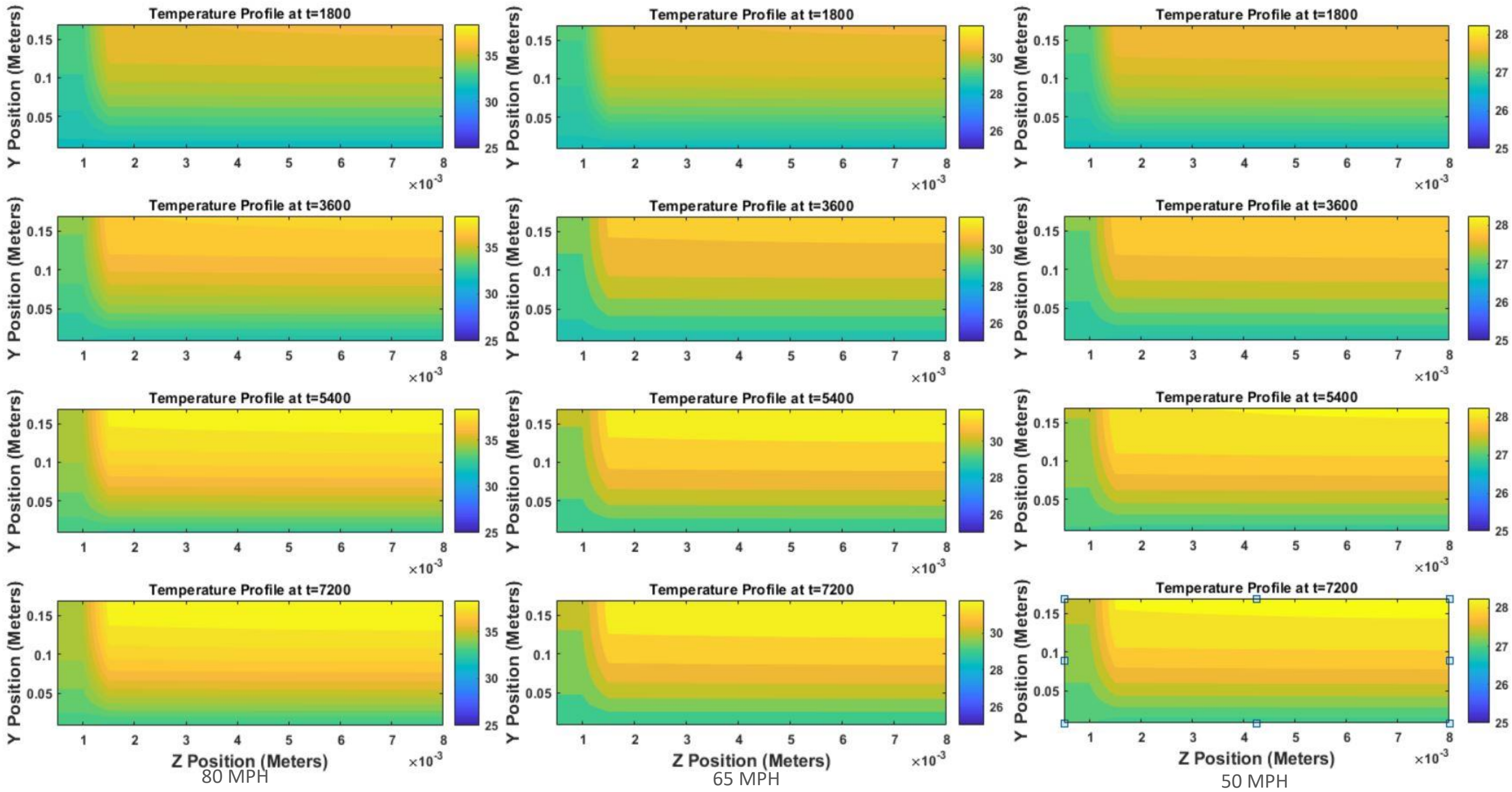

Figure 9: Temperature Z-Y Gradient at different vehicle Speeds and at different Times

Discussion:

The results of this study show that a passive cooling solution for an electric vehicle that relies on a fin under the vehicle is possible. The battery cell had an average temp rise approaching 11.5 degrees C with a 94 A discharge current (at 80 MPH), compared to a temp rise of less than 5 degrees that could be achieved on the same cell with the same current using water/glycol liquid cooling or mineral oil jacket cooling [16]. Those numbers can be improved by increasing the height of the fin, but that has a negative effect on the efficiency of the vehicle, as it increases drag, and adds to the total weight. From figure (4) we can see that even if the fin height was 0.1 m long, which is more than 2/3rd of the height of the vehicle, the mean temp rise would still be higher at 6.6 C than what can be achieved using active cooling.

The cooling solution performed much better at slightly lower speeds where the power drawn was significantly lower, and the convective heat transfer rate was still sufficiently high. Figure (5) shows a much higher sensitivity for the discharge current to the speed compared to the convective heat transfer rate, which shows that a lighter city electric vehicle with a lower top speed can rely on a passive cooling solution. Another option that would improve the effectiveness of this cooling solution would be a more parallel battery cell configuration. Doubling the number of parallel cells would cut the cell discharge current in half, which would reduce the internal resistance heat generation term to one fourth its value. Battery cells are not cheap or weightless however, so doing this is likely more expensive and less efficient than adding an active cooling solution. Another issue with passive fin cooling is the resulting uneven temperature distribution inside the battery. Figure (9) shows a temperature difference of more than 5 degrees in the 80 MPH test.

Charging speed will definitely be negatively impacted too. Since the vehicle is stationary when charging, cooling the cells to offset the heat generated from the charge current will rely on natural convection only, which would either result in long charge times, or very high cell temperatures. Running the vehicle at lower ambient has the opposite problem, the battery will cool down to much lower temperatures than optimal, reducing the battery's power output in the case of no active heating, or significantly impacting the range if active heating is used.

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