

Thermal Analysis of a Li-ion Battery System with Indirect Liquid Cooling Using Finite Element Analysis Approach

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ABSTRACT

The performance and life of Li-ion battery packs for electric vehicle (EV), hybrid electrical vehicle (HEV), and plug-in hybrid electrical vehicle (PHEV) applications are influenced significantly by battery operation temperatures. Thermal management of a battery pack is one of the main factors to be considered in the pack design, especially for those with indirect air or indirect liquid cooling since the cooling medium is not in contact with the battery cells. In this paper, thermal behavior of Li-ion pouch cells in a battery system for PHEV applications is studied. The battery system is cooled indirectly with liquid through aluminum cooling fins in contact with each cell and a liquid cooled cold plate for each module in the battery pack. The aluminum cooling fins function as a thermal bridge between the cells and the cold plate. Cell temperature distributions are simulated using a finite element analysis approach under cell utilizations corresponding to PHEV applications. Both single cold plate cooling and dual cold plate cooling were analyzed. Influences of the cold plate surface temperature gradients on the cell temperature distributions were also studied. The studies show that dual cold plate cooling has about twice the cooling capacity of single cold plate cooling and that the dual cold plate configuration can also provide cell terminal and busbar cooling.

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INTRODUCTION

Lithium-ion (Li-ion) batteries are becoming widely used rechargeable energy storage systems for EVs, HEVs and PHEVs. Differently from EVs or HEVs, PHEVs have two operation modes: a charge depleting (CD) or EV mode and a charge sustaining (CS) or HEV mode. The battery pack capacity is determined by the range of the CD mode. During CD mode operations, the state of charge (SOC) of the battery cells typically drops from 90% to 25% (i.e. the usable capacity $\Delta\text{SOC} = 65\%$). If the electrical energy consumption at the CD mode may be assumed to be about 250 ~ 300 Wh/mile for passenger cars, then the pack capacity must be in a range 15 ~ 20 kWh to reach a 40 mile distance under the CD mode. Considerable heat can be generated in the battery cells at the end of the CD mode as a result of high discharge/regen pulse currents and high cell internal resistances which increase with decreasing SOC. Because a battery pack for PHEV applications operates at its full capacity only at the CD mode, the maximum battery heat generation is encountered at the end of the CD mode. The more heat accumulated in the cells, the higher their temperatures.

Battery packs for PHEV applications are often assembled with high power pouch cells in order to achieve a high

packing efficiency. As shown in [Figure 1](#), a pouch Li-ion cell is structured with multiple thin battery layers (with dimensions on the order of 10^2 microns) arranged in parallel in the cell. Each of the thin battery layers consists of two electrodes (anode and cathode), a separator and two current collectors (copper for the anode and aluminum for the cathode). The electrodes and the separator are all porous media with the pores filled with the electrolyte, as illustrated in [Figure 2](#). The positive and negative terminal tabs of the cell can be designed on the same end or opposite ends. If the thickness of a single battery layer may be assumed to be 200 μm , then a pouch cell with a 10 mm thickness may contain up to 50 thin battery layers. For high power pouch cells, the cell thickness is much less than the cell width and height.

The maximum temperature and the maximum temperature difference in a cell are crucial factors to the cell durability and safety. To achieve optimal performance for a given battery pack, working temperatures of the cells in the pack should be controlled to within a proper range. Temperature distributions in the cells should also be as uniform as possible to ensure the durability and safety of the battery pack under specified duties and ambient conditions. Good thermal management of a battery pack depends on good

understanding of the thermal behavior of the cells in the pack under various pack operation and cooling conditions. Battery thermal modeling can play an important role in the pack cooling system design. In this study, a theoretical investigation will be reported on thermal behavior of a Li-ion battery system with indirect liquid cooling under operation conditions simulating PHEV applications.

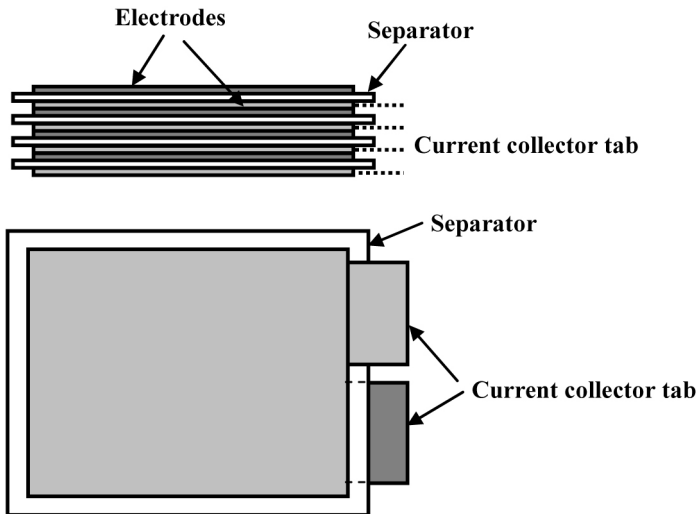


Figure 1. Illustration of the layered structure for a pouch Li-ion cell.

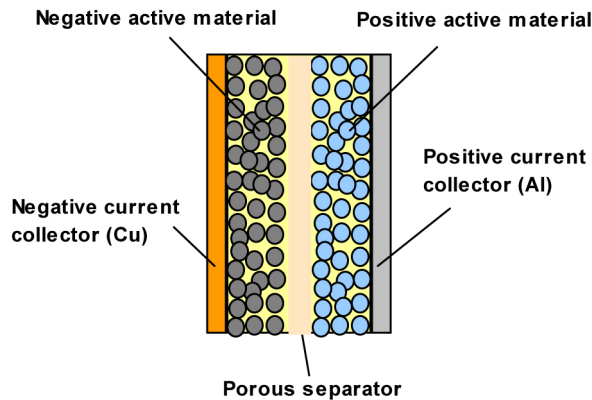


Figure 2. Illustration of the structure for a single battery layer in a pouch Li-ion cell.

BATTERY SYSTEM DESCRIPTION

The battery pack under study is for PHEV applications. It consists of 96 high power Li-ion pouch battery cells (specifications shown in Table 1). The 96 cells in the pack are stacked in eight modules, with each having 12-cells. All the cells are connected in series, giving a 20 kWh capacity for the pack. The battery pack is assumed to have an indirect liquid cooling system where each of the eight modules is cooled with liquid cooled cold plate (hereafter, cold plate for short). The coolant flows to all the cold plates are assumed to be in parallel, making all the modules thermally symmetric.

Each cell in a module is cooled via a 1.5 mm thick aluminum cooling fin which is in contact with a cold plate for the module, i.e. the cooling fin functions as a thermal bridge between the cell and the cold plate. The cell temperatures can be controlled by changing the cold plate surface temperature via varying the coolant flow rate and/or coolant temperature. The cell is assumed to be in close contact with the aluminum cooling fin. The side surface of the cell that is not in contact with the cooling fin is thermally insulated from the next cooling fin by an elastomeric thermal pad. In the thermal analysis of a cell, it is neither necessary nor practical to model details of the multiple thin battery layers in a cell. Instead, the cell may be simplified and modeled as a continuous heterogeneous solid medium, in which heat conduction takes place [1,2]. Figure 3 shows a simplified model for a basic cooling unit in a module: it consists of a cell, an aluminum cooling fin and a thermal pad. In this simplified model, the electrothermal behavior of the cell is characterized with two equivalent electrodes. Figure 4 shows a sectional view of the cell arrangement in a module and cell electrical connections: each cell terminal tab is connected to an opposite terminal tab of the adjacent cell. The aluminum cooling fin for each cooling unit is designed such that it can be attached to one cold plate from the bottom of the cell or two cold plates from two sides of the cell.

Table 1. Cell Specifications

Nominal capacity	58 Ahr
Nominal voltage	3.6 V
Internal resistance	< 1.0 mΩ
Max discharge rate (<10 sec)	15C
Operation temperatures	-15 to 50 °C
Mass	1450 gram
Thickness x Width x Height	10.5 x 250 x 227 (mm)

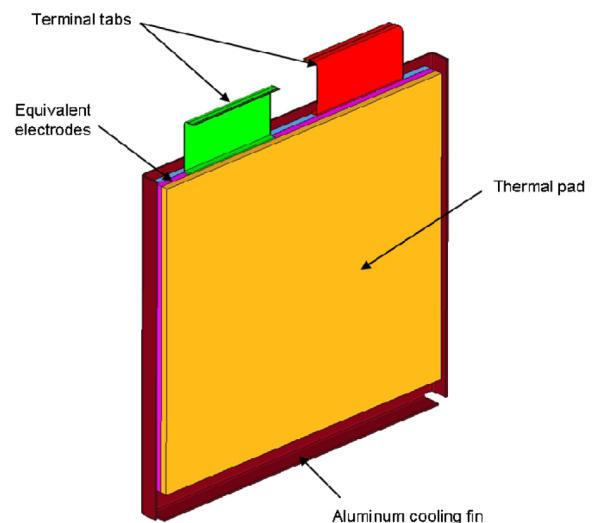


Figure 3. Simplified model for a basic cooling unit in a battery module.

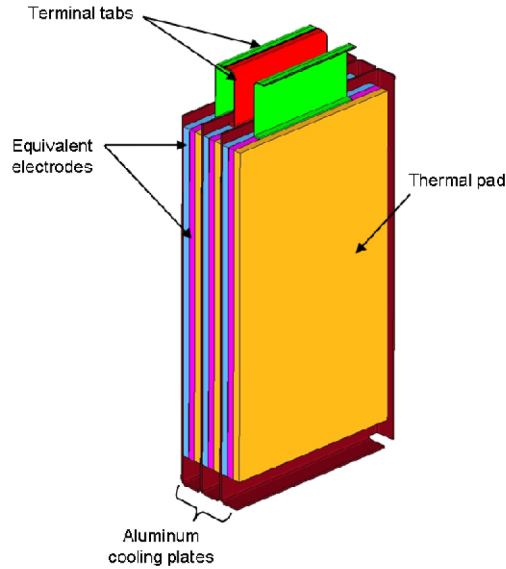


Figure 4. Sectional view of cell arrangement and electrical connections in a module.

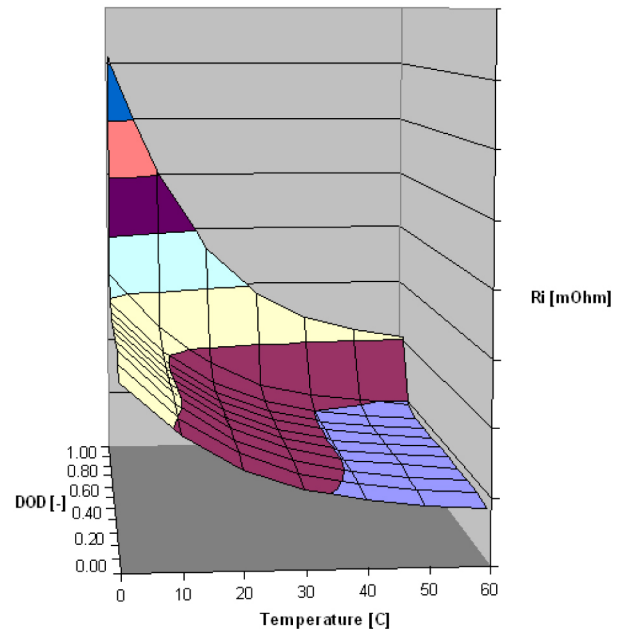


Figure 5. Internal resistance of a Li-ion cell with temperature and depth of discharge (DOD).

BATTERY COOLING MODEL

FINITE ELEMENT MODEL FOR CELL HEAT TRANSFER

Based on the theories developed by Bernardi et al. [3], Newman and Thomas-Alyea [4] and Thomas et al. [5], the rate of heat generation in a Li-ion cell q may be expressed in the internal resistance R_i as

$$q = I^2 \times R_i - I \times \left(T \frac{\partial U}{\partial T} \right) \quad (1)$$

where I is the total volumetric current density, R_i is the internal resistance, T is the local cell temperature and U is the open circuit cell voltage determined by the difference in the open circuit potentials at the positive and negative electrodes. In Eq.(1), the first term is the heat generation due to the internal resistance (irreversible heat dissipation) and the second term is known as the reversible heat resulting from changes in the open circuit potentials with respect to the temperature at the two electrodes. It is assumed that the cell temperature is governed by the irreversible heat in a cell, i.e. by the cell current and the internal resistance. Figure 5 shows dependence of the internal resistance on the temperature and depth of discharge (DOD = 1 – SOC) for a Li-ion cell.

Changes in the cell temperature due to the cell discharge or charge activities can be characterized with energy balance on a unit cell volume as

$$\rho C_p \frac{\partial T}{\partial t} = \nabla \cdot (k \nabla T) + q \quad (2)$$

where ρ , C_p and k are the local density, heat capacity and thermal conductivity of the cell medium, T is the temperature and t is the time. Because of the cell's layered structure, thermal behavior of the cell should be characterized with the effective thermal properties. Each of the battery layers with the same thermal properties may be treated as a component of the heterogeneous solid medium. The composite local volumetric heat capacity may be expressed as

$$\rho C_p = \frac{\sum_i \rho_i C_{p,i} V_i}{\sum_i V_i} \quad (3)$$

where the subscript i indicates the properties for the component i . Thermal conductivities at the component interfaces should be determined based on connections of the components. For series connections, the composite thermal conductivity is given as

$$k = \frac{\sum_i L_i}{\sum_i (L_i / k_i)} \quad (4)$$

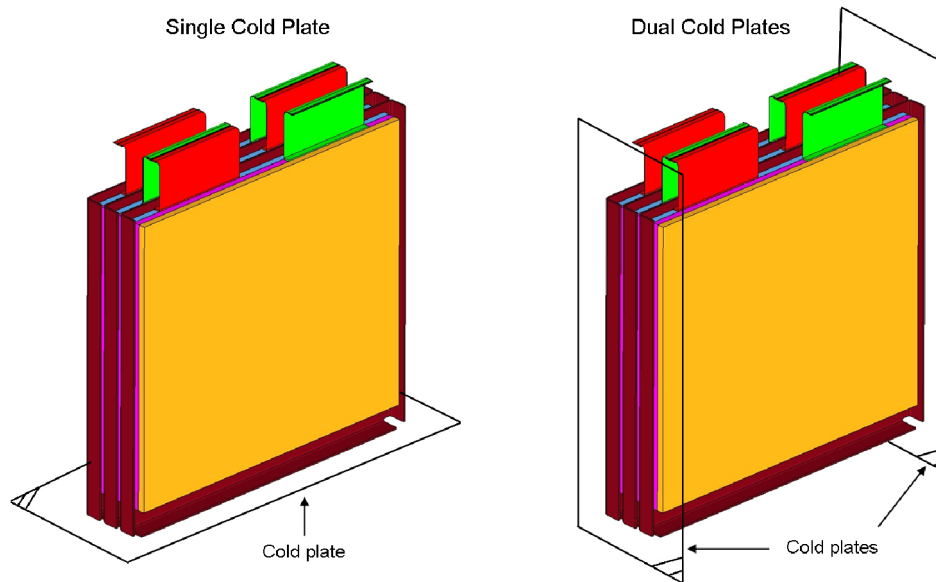


Figure 6. Three-unit battery models with single and dual cold plate cooling.

For parallel connections, the expression for the composite thermal conductivity becomes

$$k = \frac{\sum_i L_i k_i}{\sum_i L_i} \quad (5)$$

In Eqs. (4) and (5), L_i and k_i are the thickness and the thermal conductivity for the component i respectively. Similarly, the composite electrical conductivities can also be expressed by Eqs.(4) and (5) in characterizing the electrical field of the cell. For a given cell with properly defined boundary conditions for heat transfer, Eq.(2) can be solved using finite element analysis (FEA) approach. In this study, the electro-thermal behavior of the battery cells and the module will be characterized using the FEA tool ABAQUS [6]. Since all the cooling units in a module are thermally independent, a full model for a 12-cell module is not needed for the study of the thermal behavior of the battery cells. However, the model must contain a minimum of three cells to evaluate temperatures of the cell terminal tabs properly. Figure 6 shows the 3-cell FEA model developed for analyzing the battery thermal behavior. As aforementioned, the cells are simplified with the equivalent electrodes. The techniques for the battery modeling are the same as in the authors' previous SAE papers [1,2,7]. The cells can be cooled either with a single cold plate or with dual cold plates. It is assumed that the cooling fins and the cold plate are in perfect contact, i.e. the contact thermal resistance is negligible in comparison to the temperature gradients in the cooling fins.

MODEL VALIDATION

Before it was used to characterize the cell behavior under different thermal boundary conditions, the battery cooling model developed was evaluated via comparison of the

predicted temperatures with the measured temperatures for a 3-cell module with single cold plate cooling. The cells tested had slightly different capacity and dimensions from those used in this study. Thus, the FEA model was modified to match the cell and cooling unit geometries, electrical and thermal properties, terminal connections and cooling condition in the test setup as shown in Figure 7. The test module was thermally insulated from the air in the test chamber and the cell heat could be dissipated only through the cold plate. In the test, only temperatures of the middle cell were measured because it had better thermal insulation than the two end cells. Thirteen thermocouples (TC1 to TC13 in Figure 7) were placed on the thermal pad side cell surface. During the test, the cold plate surface temperature was maintained at 25 °C approximately. The test was conducted under a constant 2C discharge rate from a fully charged condition (DOD = 0) to 90% depth of discharge (DOD = 0.9).

Figure 8 compares the measured and simulated temperatures at the end of discharge (DOD = 0.9). For an easy comparison, the measured cell temperatures are presented in a contour plot with 0.5 °C intervals between two contour lines. The measured maximum cell temperature was 38.5 °C at the cell edge connected to the positive terminal tab. It is seen in Figure 8 that the simulated cell temperatures reasonably match the measurements in the maximum cell temperature and cell temperature distribution. This indicates that the model can reasonably characterize the thermal behavior of the cell under study.

FEA SIMULATION

HEAT TRANSFER BOUNDARY CONDITIONS

The surface temperature of the cold plate varies with the cold plate design and the coolant flow through it. Figure 9

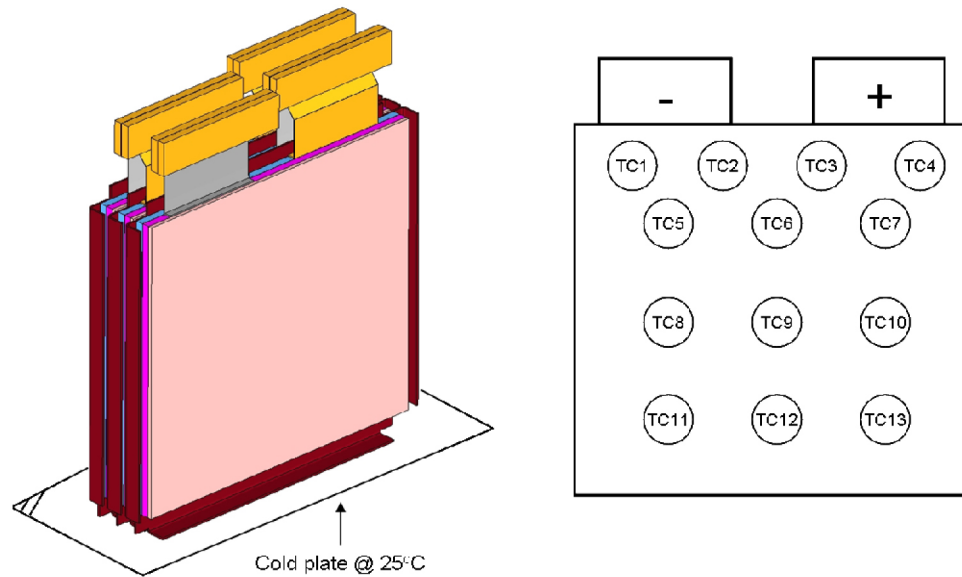


Figure 7. FEA model simulating the test setup and thermocouples on the cell for temperature measurement.

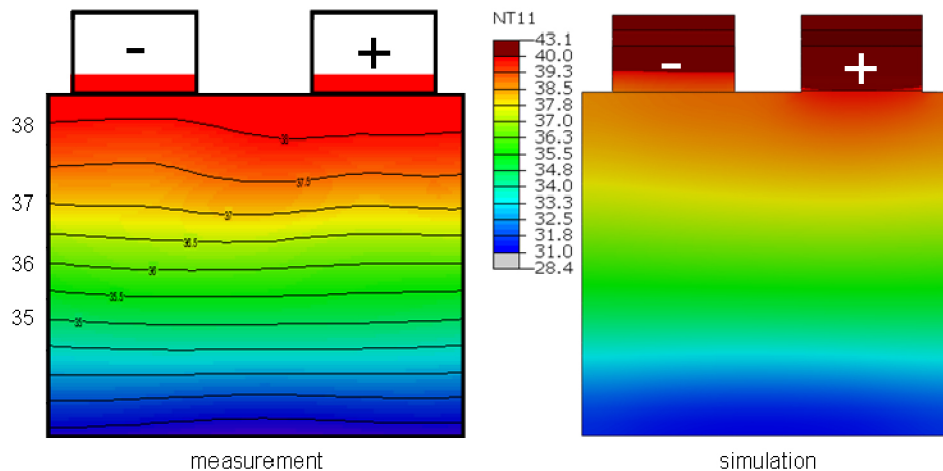


Figure 8. Measured and simulated cell temperatures under 2C discharge with single cold plate cooling.

shows the performance of a commercial cold plate (with a tube-plate structure and dimensions for the cooling surface = 300 mm × 180 mm) that can be used to cool the 12-cell module under study. It is seen that the maximum differential temperature on the cold plate surface varies with the cell heat rejection and the coolant flow. In practice, a conductive thermal interface material (typically 1 mm thick) is applied between the cooling fins and the cold plate, which not only lowers the contact thermal resistance but also improves the temperature distribution at the interface. Because the temperature drop over the thermal interface material is small, it was not considered and a perfect contact was assumed between the cooling fins and the cold plate. For simplicity, the influence of the cold plate structure on the cooling performance was neglected and the maximum differential surface temperature of the cold plate was assumed to be due

largely to the variation of the coolant temperature in the cold plate. The influence of the coolant flow direction was considered by the gradients of the cold plate surface temperatures: low temperature indicating the coolant inlet side and high temperature indicating the coolant outlet side. Three temperature boundary conditions (BC) for the cooling fins (or the cold plate surface) were considered for both the single cold plate and dual cold plate cases As shown in Figure 10 (for single cold plate) and Figure 11 (for dual cold plates): BC1 is a uniform temperature distribution with a constant temperature 35 °C; BC2 is a linear temperature distribution with the cold plate surface temperature varying from 37°C (at the coolant outlet side) to 35°C (at the coolant inlet side) across the cooling fin surface in contact with the cold plate; BC3 is similar to BC2 but with an opposite temperature gradient due to having an opposite direction for