ENHANCING ELECTRIC VEHICLE PERFORMANCE: OPTIMIZING BATTERY THERMAL MANAGEMENT WITH COLD PLATE TECHNOLOGY

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ABSTRACT

The performance and driving range of electric vehicles (EVs) are significantly affected by temperature variations in battery systems. Although Li-Ion batteries can operate over a wide temperature range from -20°C to 60°C, optimal performance is achieved when operating in a narrower temperature band of 15°C to 45°C. This study addresses the complex thermo-chemical and thermo-mechanical processes in vehicle batteries to design an optimal cold plate battery thermal management system (BTMS) that maintains the battery temperature within the optimal range. Our methodology integrates both experimental and numerical approaches. First, a lumped model is developed in MATLAB/Simulink to simulate the time-dependent variations in battery current, operating voltage, state of charge, heat generation and cell temperature based on the power drawn from a Li-Ion battery module consisting of 14 series 2 parallel cells with a capacity of 3.3 kWh. Furthermore, a CFD model is developed in ANSYS-FLUENT to simulate novel cold plate channel designs based on the "spider web" design. Experimental validation was carried out at different discharge rates and different mass flow rates. "C-rate" is the battery charge-discharge rate and "1C" is the current that can discharge the battery in 1 hour. The results show that the use of cold plate can reduce the maximum battery temperature by 5°C at 1C rate. Such a temperature reduction underlines the effectiveness of the proposed BTMS in improving electric vehicle performance. This study not only contributes to the advancement of thermal management in electric vehicles, but also lays a foundation for future research in the optimization of electric vehicle battery systems.

Keywords: Electric Vehicles, Battery Thermal Management, Cold Plate Technology, Li-Ion Batteries, Thermal Optimization

1 INTRODUCTION

The expanding adoption of electric vehicles (EVs) marks a pivotal shift towards sustainable transportation. Central to the advancement of EVs is the optimization of their lithium-ion (Li-Ion) battery systems, whose performance and longevity are critically dependent on BTMS. Efficient battery thermal management is essential, as even minor temperature deviations can significantly impact the driving range and safety of EVs. Even though Li-Ion batteries can be operated between -20°C to 60°C, they should be operated in a tight range between 15°C to 45°C to gain optimal performance. Addressing the complex thermal challenges inherent in EV batteries, this study delves into cold plate thermal management strategies a technology pivotal for its capacity to manage complex thermo-chemical and thermo-mechanical processes. Previous research has laid a foundation for understanding various cooling techniques and their efficacy. For instance, Li et al. (2019) designed a BTMS for pouch-type Li-Ion batteries, exploring the influence of cooling plate thickness and air velocity on temperature regulation. Their findings underscored the benefits of increasing plate thickness and air velocity for temperature reduction. Kausthubharam et al. (2023) furthered this research by investigating a mini-channel cold plate, which maintained optimal temperatures even under high ambient conditions and discharge rates, revealing the significance of refrigerant velocity in cooling performance. Extending these insights, Wu et al. (2023) investigated the flow distribution and heat transfer properties of cold plates with differing channel structures, noting how volumetric flow rate and discharge rate are critical to temperature control and BTMS requirements. In addition, Guo et al. (2023) combined electrochemical and thermal modelling to enhance cooling performance in long-term battery cycles, finding that nanofluids with increased nanoparticle concentrations improved cooling efficacy albeit with a pressure penalty. Monika et al. (2021a) reported that enhancing the number of channels in cold plate battery thermal management systems directly correlated with improved cooling performance. However, this cooling efficiency inversely decreased when the system operated under conditions of elevated discharge rates and higher ambient temperatures. Further, Kalkan et al. (2021) sought to enhance the thermal performance of BTMS by innovating a mini-channel cold plate design aimed at reducing temperature disparities across the battery's surface. Their research focused on designing a system that minimized local temperature variations and the overall maximum surface temperature. The study rigorously examined how varying the channel width, the number of channels, refrigerant flow rate, fluid temperature, ambient temperature, and discharge velocity could influence the thermal management capabilities in a liquid-based cold plate BTMS, using a 20 Ah pouch-type cell as the test model. Their findings contribute to a deeper understanding of how these parameters can be manipulated to optimize BTMS performance. Monika et al. (2021b) conducted a numerical analysis to address the temperature gradient issue inherent in conventional straight-channel cold plates by implementing a Multi-Stage Tesla Valve (MSTV) configuration for thermal management in pouch-type batteries. The findings revealed that the MSTV design notably enhanced both flow bifurcation and heat transfer rates, offering a significant improvement over traditional straight-channel configuration. Wang et al. (2021) explored the cooling and thermal equilibrium performance of a pouch-type cell using a spider web model-designed cold plate. Their investigation identified channel width as the primary determinant of cooling efficacy, followed by the number of channels and channel angle, respectively. These parameters were found to significantly influence the thermal management system's performance. Building on these insights, our research aims to enhance EV performance through the design of an optimal cold plate BTMS, with a focus on the demanding WLTP driving cycle conditions. By linking both experimental and numerical studies ranging from MATLAB/Simulink simulations to advanced ANSYS-FLUENT models utilized "spider web" cold plate channel geometry. This design is rigorously assessed for its pressure drop, thermal homogeneity, and overall thermal management efficiency.

2 METHODS

In this study, a Li-Ion Pouch Type battery consisting of 28 cells (14 series and 2 parallel) was used. The specifications of the battery are given in Table 1. (Farasis Energy, 2023a) In the following subsections, the details of the numerical and experimental studies are provided.

Table 1: Specification of battery cells (Farasis Energy, 2022)

Specifications	Value
Mass of battery cell (g)	505
Length of battery cell (mm)	230.5
Width of battery cell (mm)	161
Thickness of battery cell (mm)	6.16
Cell heat capacity (J Kg^{-1} K^{-1})	1040
Nominal capacity (Ah)	29.3
Nominal voltage (V)	3.58
Nominal Energy (Wh)	105

2.1 NUMERICAL STUDIES

The battery thermal management system is modelled with a lumped approach in MATLAB/Simulink. The simplified model extracts the battery-generated heat via conduction and convection. The studies are carried out assuming the input parameters given in Table 2.

Table 2: Input parameters

Parameters	**Value**
Refrigerant Inlet Temperature (°C)	10
Refrigerant Inlet Flow Rate (g s^{-1})	10
Cold Plate Total Channel Area (m^2)	0.217
Ambient Temperature (°C)	25
Initial Temperature of the Battery (°C)	20
Initial SOC Value of the Battery (%)	100

2.1.1 Battery Heat Generation in a Lumped Model

Transient heat transfer within the battery is considered to be lumped to conduct parametric simulations with reasonable computational effort and acceptable accuracy. The mathematical model is developed in the Simulink module of the MATLAB. The assumptions considered while reducing the physical problem into the mathematical model are as follows; All cells in the battery are at the same temperature.

- All cells in the battery are at the same temperature.
- Heat generation inside the battery is uniform and identical in each cell.
- The convection coefficients on the lateral surfaces of the battery module are identical and uniform.
- Heat generation and reactions in the cells are not temperature dependent.
- Heat transfer takes place by convection from the 4 lateral surfaces of the battery. Conduction is defined on the bottom surface. The top surface of the battery is considered to be adiabatic.

Considering the above-listed assumptions, the BTMS design in the lumped model is modelled using the following equations;

$$m_b (c_p)_b \frac{\partial T_b}{\partial t} = Q_b + Q_{loss} \tag{1}$$

where m_b is the battery mass, $c_{p,b}$ is the specific heat capacity of the battery, Q_b is the total amount of heat generated by the battery and Q_{loss} is the total heat dissipated from the battery. Heat generated inside the battery has two components as below,

$$Q_b = Q_{irr} + Q_{rev} \tag{2}$$

where Q_{irr} refers to the conversion of electrical energy into heat energy and Q_{rev} is related to the production and consumption during the charging and discharging processes of the battery. In the lumped model, reversible effects are disregarded, and the irreversible part is defined as follows:

$$Q_{irr} = I^2 R \tag{3}$$

where I is the electrical current, and R is the internal electrical resistance. The battery state of charge

$$SOC(t) = SOC(t_0) \pm \frac{\int_0^t I(t)\, dt}{C_{nom}} \tag{4}$$

where $SOC(t_0)$ indicates the initial state charge of the battery and C_{nom} is the nominal capacity of the battery.

2.1.2 Cold Plate Based Battery Thermal Management System

The term Q_{loss} expressed in Eq. (1) refers to the heat removed from the battery by conduction and convection and is expressed as follows.

$$Q_{loss} = U\, A_s \Delta T_{LMTD} \tag{5}$$

U is the total heat transfer coefficient, A_s is the heat transfer area and ΔT_{LMTD} is the logarithmic mean temperature difference.

$$UA_s = \frac{1}{R_{total}} \tag{7}$$

The thermal resistance is the sum of the convective and conductive components as follows

$$R_{total} = R_{convection} + R_{conduction} \tag{8}$$

$R_{conduction}$ and $R_{convection}$ are defined as below

$$R_{convection} = \frac{1}{h_{ave}\, A_{lateral}} + \frac{t_b}{k\, A_{lateral}} \tag{9}$$

$$R_{conduction} = \frac{t_{wt}}{k\, A_{bottom}} + \frac{t_{CP}}{k\, A_{bottom}} \tag{10}$$

where $A_{lateral}$ is the total lateral surface area of the module, A_{bottom} is the bottom area of the module, k is the thermal conductivity coefficient of aluminium, t_{wt} is the wall thickness of the module, t_{CP} is the top surface thickness of the cold plate and h_{ave} is the average heat convection coefficient. The temperature difference between the cold plate and the battery module is calculated using the logarithmic mean temperature difference formula below.

$$\Delta T_{LMTD} = \frac{(T_b - T_{inlet,f}) - (T_b - T_{outlet,f})}{ln \frac{(T_b - T_{inlet,f})}{(T_b - T_{outlet,f})}} \tag{11}$$

In this equation, T_b is the battery temperature, $T_{\text{inlet},\,f}$ is the refrigerant inlet temperature, $T_{\text{outlet},\,f}$ is the refrigerant outlet temperature. The outlet temperature of the refrigerant from the cold plate is calculated using the equation below.

$$\frac{T_{f,outlet} - T_b}{T_{f,inlet} - T_b} = exp\left(\frac{-h_{ave}\, A_{bottom}}{\dot{m}\, c_{p,f}}\right) \tag{12}$$

In this $\dot{m}$ is the mass flow rate of the refrigerant and $c_{p,f}$ is the specific heat coefficient of the refrigerant.

The thermophysical properties of water are assumed to be constant, and properties are defined at 25 °C. The velocity of the refrigerant is calculated from the mass flow rate. The convective heat transfer coefficient for the fluid was calculated by selecting the appropriate Nusselt correlation for turbulent flow. Internal resistance R_{int} and open circuit voltage V_{OCV} show in Figure 1 are taken from the documents given by Farasis Energy (Farasis Energy, 2023b).

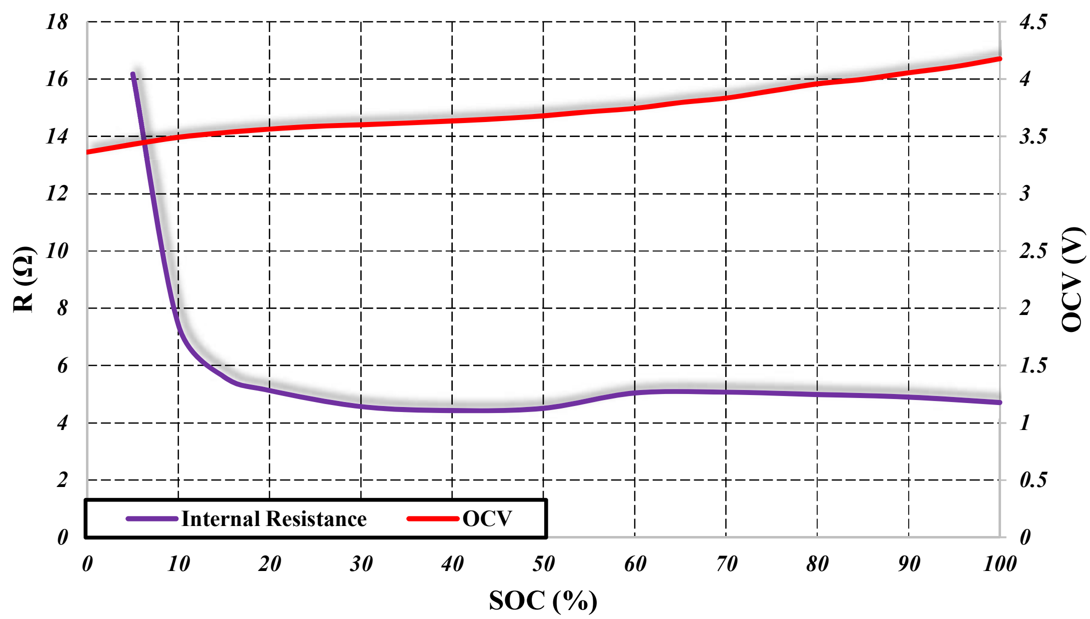

Figure 1: Internal resistance values and OCV values depending on SOC at 20°C

2.2 EXPERIMENTAL STUDIES

In the experimental studies, a load bank was used to control the battery's charging and discharging rates. In the battery's thermal management system, a constant temperature bath with an embedded pump is used to circulate the cooling water at a specified temperature. A rotameter measures the cooling water's mass flow rate, and a spider web structure is mounted at the bottom of the battery for thermal management. T-type thermocouples with a datalogger unit are used to measure the temperature variations at several locations inside the battery and cold plate. The inlet temperature of the water in the system was set to 10°C, and the flow rate was set to 10 g/s. The initial temperature of the cooling coil was 22 °C, and experimental studies were carried out at different C-rates.

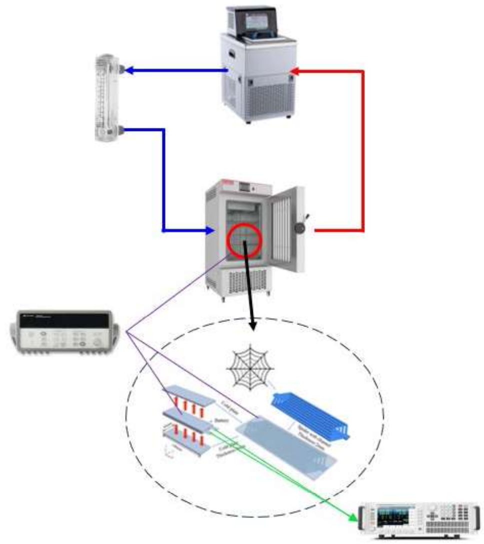

Figure 2: Schematic representation of the experimental setup

3 RESULTS AND DISCUSSION

Batteries heat up during charging and discharging due to reactions occurring in their internal structure. As the charging and discharging rate of a battery increases, the amount of heat increases and the maximum temperature of the battery rises. In order to observe this effect, numerical analyses and experimental studies were carried out at 0.5C - 1C C-rate values. As the C-rate value increases during battery charging/discharging processes, the amount of heat generated in the battery also increases. Accordingly, the maximum temperature of the battery also increases. It is assumed that the initial temperature of the battery is 25 °C and the initial SOC value is 100%. As a result of the numerical analysis, the maximum temperature of the battery increased at a rate of 1C to 32,159 °C at the end of 3200 seconds. When the c-rate was set to 0.5C, the maximum temperature of the battery increased to 26.963 °C at the end of 3200 seconds. As a result of the experimental studies, it was observed that the maximum temperature of the battery increased to 32.529 °C at the end of 3200 seconds at 1C-rate. When the c-rate was set to 0.5C, it was observed that the maximum temperature of the battery increased to 26.884 °C at the end of 3200 seconds. When the same numerical calculations and experimental studies were repeated using a cold plate, the decrease in the maximum battery temperature was noticeable. At 1C, the battery temperature increased to 28,508 °C in numerical calculations and 28,779 °C in experimental studies. At 0.5 C, it decreased to 24,391 °C in numerical calculations and 24,616 °C in experimental studies.

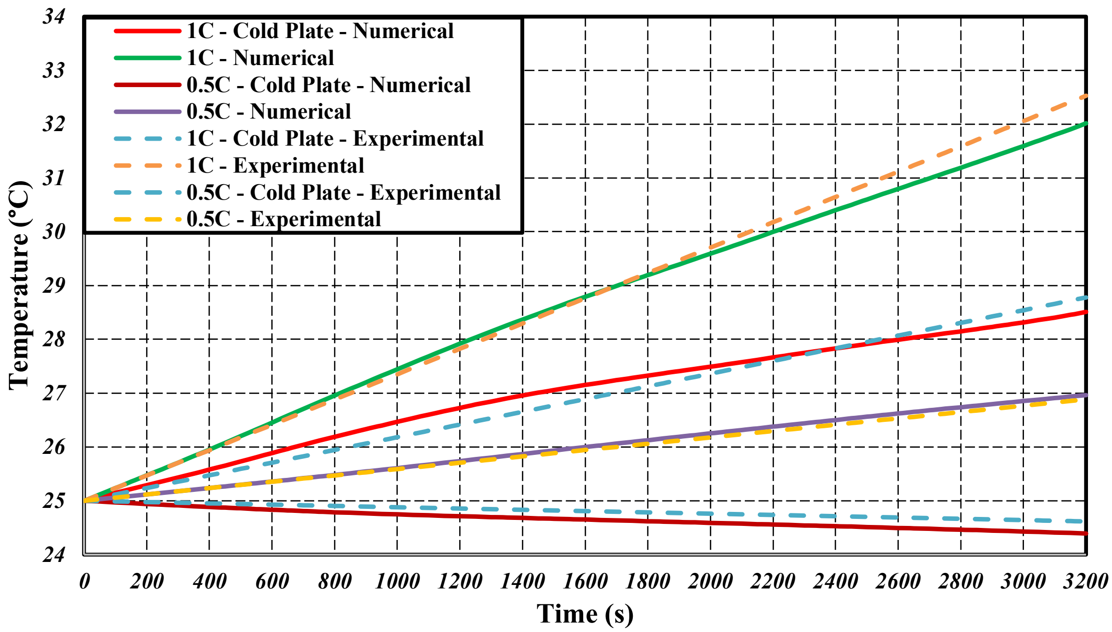

Figure 3: Temperature changes in the battery at 0.5C and 1C c-rates

After the MATLAB/Simulink model was verified with experimental studies, the amount of heat transferred from the coil to the cold plate was calculated as 98.7 W. This heat value was defined as the boundary condition in the cold plate CFD analyses in ANSYS/Fluent program and the temperature distribution of the cold plate was obtained. Thanks to CFD analyses, the pressure drop of the cold plate and the convection coefficient 'h' in the flow volume were calculated as 91.14 $Wm^{-2}K^{-1}$.

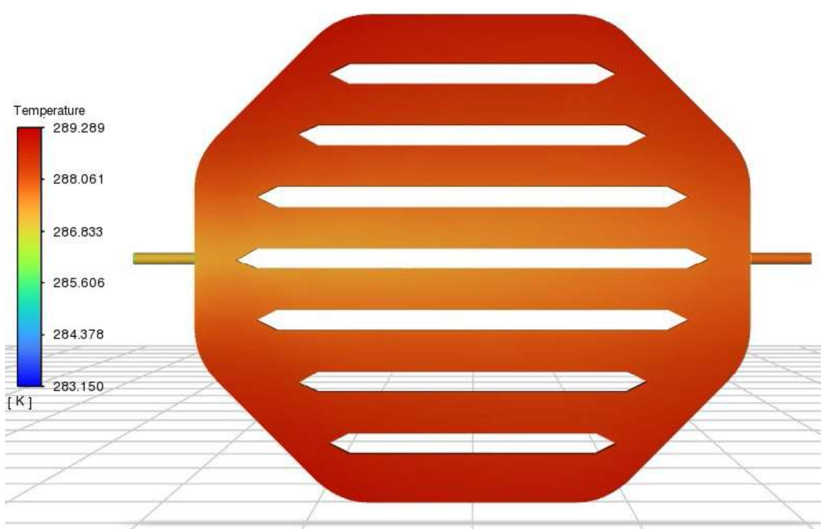

Figure 4: Cold plate flow volume temperature distribution

4 CONCLUSIONS

In this study, a cold plate based battery thermal management system for Li-Ion pouch type batteries is proposed. A cold plate designed in a spider web model is used. The battery is designed using a lumped model in MATLAB/Simulink software package. The effects of different refrigerant inlet temperatures, refrigerant flow rate and battery charge/discharge ratio parameters on the maximum battery temperature and heat rise in the battery are investigated. By decreasing the refrigerant inlet temperature, an increase in cooling performance occurs due to the increased temperature difference. It is observed that the increase in the charge/discharge ratio increases the temperature rise of the battery and increases the maximum cell temperature. It is observed that the cold plate reduces the maximum temperature of the battery by 10.956% in numerical studies and 11.526% in experimental studies at 1C rate. At 0.5C, it was observed that the temperature decreased by 9.539% in numerical studies and 8.437% in experimental studies. It was found that the cold plate cooled the battery by an average of 10.114%. After the verification of the MATLAB/Simulink model, the amount of heat transferred to the cold plate at 1C rate was calculated. The calculated heat load was defined as a boundary condition in the cold plate CFD analyses. In this way, the temperature distribution of the cold plate, pressure drop and convection coefficient of the refrigerant were obtained. In the rest of the study, the cooling performance of the system will be analysed with cold plates of different designs and under different ambient conditions.

NOMENCLATURE

m_b	battery mass (kg)
$c_{p,b}$	battery specific heat capacity ($J\ kg^{-1}K^{-1}$)
T_b	battery temperature (K)
Q_t	total amount of heat generated by the battery (W)
Q_{loss}	total amount of heat transferred by the cooling system (W)
Q_{irr}	irreversible heat generation (W)
I	electrical current (A)
R	Internal electrical resistance (Ω)
U	total heat transfer coefficient ($W\ m^{-2}K^{-1}$)
R_{total}	total thermal resistance ($K\ W^{-1}$)
$R_{convec.}$	convection thermal resistance ($K\ W^{-1}$)
$R_{conduc.}$	conduction thermal resistance ($K\ W^{-1}$)
h_{ave}	convection heat transfer coefficient ($Wm^{-2}K^{-1}$)
A_s	convection surface area (m^2)
$\dot{m}$	mass flow rate of fluid ($Kg\ s^{-1}$)
$T_{f,inlet}$	inlet temperature of the fluid (K)
$T_{f,out}$	outlet temperature of the fluid (K)

Subscript

LIB	Lithium-ion batteries
EV	Electric vehicles
BTMS	Battery thermal management system
SOC	State of charge

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