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Analysis of a Battery Thermal Management System for Electric Vehicles using Heat Pipe Technology

Eoin Guinan[1,2], Joseph Mooney[3], Johnathan Ottman[4], Jeff Punch[1,2], Vanessa Egan[1,2]
[1]School of Engineering, Bernal Institute, University of Limerick
Castletroy, Limerick, Ireland
[2]SFI CONNECT Research Centre
Eoin.Guinan@ul.ie; Vanessa.egan@ul.ie; Jeff.punch@ul.ie;
[3] Department of Mechanical , Manufacturing & Biomedical Engineering,
Parsons Building, Trinity College, Dublin 2, Ireland
MOONEYJ8@tcd.ie
[4]Hosetech Ltd.
Cork, Ireland
johnathanottman@hosetech.ie

Abstract This paper presents an analysis of the performance of a heat pipe assisted battery thermal management system (BTMS) as a means of passive heat dissipation from EV batteries to minimise required pumping power. A BTMS incorporating both standard heat pipes and multi-branched heat pipes (MBHPs) is analysed. The standard heat pipe configuration is analysed experimentally using a custom-built test rig and numerically using thermal equivalent circuit (TEC) models developed using MATLAB's Simscape software. The MBHP setup is analysed numerically having validated the TEC models using the experimental data from the standard configuration. Experimental results showed that the standard heat pipe BTMS provided sufficient cooling at heat loads of <40W. However, at heat loads of 40W the maximum battery temperature marginally exceeded the maximum temperature criteria with a maximum battery block temperature of 42.9 ºC and temperature difference between battery blocks of 4.3 ºC recorded. Due to an increase in heat transfer coefficient with increasing coolant flow rate, the battery block temperature and temperature difference decrease. This decrease in temperature is counteracted by a decrease in the coefficient of performance of the cold plate by approximately 70% as pumping power increases due to increased pressure drop across the cold plate. This showed there are marginal gains to increasing flow rate in the BTMS. Numerical analysis of both setups shows a similar temperature response of the simulated batteries to the experimental setup with analytical results being within 15% the of experimental values. Therefore, a TEC can be used to predict battery temperatures. Numerical results show increased battery temperature for an MBHP setup but kept within the operating range for heat loads of <20W suggesting MBHP's can perform sufficiently for normal battery discharge rates.

***Keywords*:** BMTS for Elective Vehicles, Heat Pipes, Simulink, Thermal Management

1. Introduction

In recent years there has been an increase in demand for electric vehicles (EVs), due to an increased consumer awareness of the contribution of transportation to global warming and climate change. This demand for EVs has been met with an equal demand for increased vehicle performance and driving range which has highlighted the limitations with the thermal management of EV batteries. Thermal management of EV batteries is important for increased vehicle driving range, prolonged battery life and safe operation of batteries. With both high and low temperatures having adverse effects on overall battery performance, a system to maintain batteries within an optimal temperature range is paramount to further advancements in battery and EV technology.

Heat is generated within an EV battery through exothermic and endothermic reactions as well as joule effect heating. This heat generation occurs during charging and discharging of the vehicle where heat generation increases as the charge or discharge rate increases. Lithium-ion batteries used in EVs are particularly sensitive to temperatures above 50 ºC or below -20 ºC with their energy storage capacity, lifetime, efficiency, and charge acceptance being affected [1]. Extreme battery temperatures can cause a thermal runaway phenomenon which permanently damages the battery pack and can potentially lead to fire. The solution to these problems is to implement a battery thermal management system (BTMS) into EV's to actively or passively dissipate heat from the battery packs and maintain an optimal and safe operating temperature. At present, car manufacturers such as Porsche and Tesla have implemented a BTMS into their models using liquid cooling methods to dissipate heat by conduction and forced convection [2]. While this can provide sufficient cooling power, the associated pumping power used to run pumps and fans to pass liquid coolant or air around all the batteries, decreases the range of the vehicle by consuming power from the battery packs which would otherwise be used to drive motors. The use of heat pipes as a means of passively cooling battery packs and dissipating this heat to a cooling source has proven to be a possible means for battery heat dissipation in EVs [3]. By taking advantage of their heat transfer capabilities, the latent heat of vaporization of working fluids and their compact malleable geometries, heat pipes can provide sufficient cooling and temperature uniformity of the battery cells. The passive nature of heat pipes means they have no associated pumping power and therefore require no power to transport heat from the battery packs to a cooling source. This passive heat transport minimises pumping power required for cooling thus minimising the reduction of the vehicle's driving range.

The use of looped heat pipes (LHPs) in a BTMS was studied experimentally by Putra *et al*. [4] for a flat plate LHP. Results showed that if acetone or alcohol were used as working fluids the battery simulator temperature was maintained below 50°C for a heat flux of 1.61W/cm^2. If water was used the battery reached approximately 60°C at start-up which is outside the safe operating temperature of a battery module. Pulsating oscillating heat pipes (POHP) were also investigated by Wei *et al.* [5]. Experimental data showed that an ethanol-water mix of 2:1 and a FR of 30% gave the best thermal performance maintaining temperatures below 46.5°C and differences of 1–2 °C at heat loads of < 60 W.

The use of standard heat pipes in a battery BTMS has also been researched by Smith *et al.* [6]. They integrated flattened and U-shaped heat pipes between simulated battery blocks, where an intermediate condenser plate and secondary liquid cooled plate were used to dissipate heat in two stages using multiple heat pipes. The system maintained cell temperatures below 55 °C with a thermal resistance of 0.075 °C/W at combined heat loads of 400 W. In a similar study, investigating battery thermal management, L-shaped heat pipes with a flattened evaporator section to increase contact surface area were investigated [7]. Results showed that L-shaped heat pipes were capable of maintaining battery temperatures at 41.1 °C for a 4 °C discharge rate, which is within safe operating limits for an EV battery.

Recent literature [8] on heat pipe design has investigated the use of a dual evaporator, T-shaped multi-branched heat pipe (MBHP) to cool multiple electronic components. Results for thermal resistance showed a strong dependence on filling ratio (FR) as values doubled from 0.06°C/W to 0.12°C/W for a 75% to 100% FR. The effect of orientation and copper powder particle size on MBHP performance was also investigated for heat inputs of 10-100W [9]. Results found that a particle size of 75-100 μm gave the best heat transfer rate for all orientations and that a gravity assisted orientation decreased the start-up time of the MBHP due to the gravitational head working with the wick structure to return the working fluid back to the evaporator.

Previous research has shown that there are many viable options for battery thermal management that could potentially be implemented into EV's. While the use of conventional forced convection cooling methods can provide sufficient cooling for batteries the associated power consumption to provide cooling comes at a cost of driving range. Heat pipes offer a passive solution for heat dissipation thus minimising pumping power requirements. A multi-branched heat pipe system, where two evaporators lead to a single condenser, has shown promising results [8] in terms of thermal resistance but also in the design of compact cooling solutions for densely packed heat dissipating components [8]. As such the objectives of the current study are to experimentally evaluate the performance of standard copper powered sintered heat pipes for battery thermal management and to develop and validate a model using Simulink- Simscape to predict the operating temperatures of the simulated batteries for both single and multi-branched heat pipe setups.

2. Experimentation

An experimental test rig was designed to simulate a set of Li-ion cells representing part of a larger electric vehicle battery pack. The test rig was designed to allow for ease of use and minimal footprint for testing in a laboratory environment. Fig. 1 (a) presents an exploded view of the experimental rig while Fig. 1(b) and (c) provide more detail on the heat pipe arrangement in the BTMS for SHP and MBHP setups. The prismatic batteries were simulated using six battery blocks machined from aluminium 1060 with a thermal conductivity and specific heat capacity of 234 W/mK and 920 J/kgK respectively. The heat generated from a typical battery was simulated using 60 W cartridge heaters inserted into each aluminium block with power supplied to the cartridge heaters using a 30V 2A DC bench top power supply. Six sintered, copper powder, copper heat pipes of dimensions 10 mm x 200 mm (Wakefield Thermal 121692_rev1) and power input 70 W were embedded into each battery block (the evaporator) and used to transport the heat dissipated by the cartridge heaters. The condenser ends of the heat pipes were embedded into condenser blocks attached to a cold plate. The whole system as shown in Fig. 1 (b) was embedded in an insulation block to minimise heat losses. A Hubert Ministat 125 cooling circulator was used to circulate coolant (silicon oil) through the liquid cooled cold plate. This simulated the cooling power that would be provided by airflow over radiators in an electric vehicle. Liquid was circulated at 18 ºC using the internal pump and supplied to the cold plate by two 10mm nominal bore hoses. A National Instruments PXIe-1082 electronic instrumentation platform was used in conjunction with K-Type thermocouples and LabVIEW data acquisition software to measure and record temperatures within the system. The temperatures of the surface of each battery, the heat pipe evaporator and condenser and cold plate inlet and outlet coolant temperatures were recorded over time for a given heat load applied the cartridge heaters. The location of each thermocouple is shown in Fig. 1 (c). Heat loads were varied from 10 W to 40 W. An uncertainty analysis [10] was carried out giving an uncertainty of 0.018 ºC/W for thermal resistance and 1.46% for power.

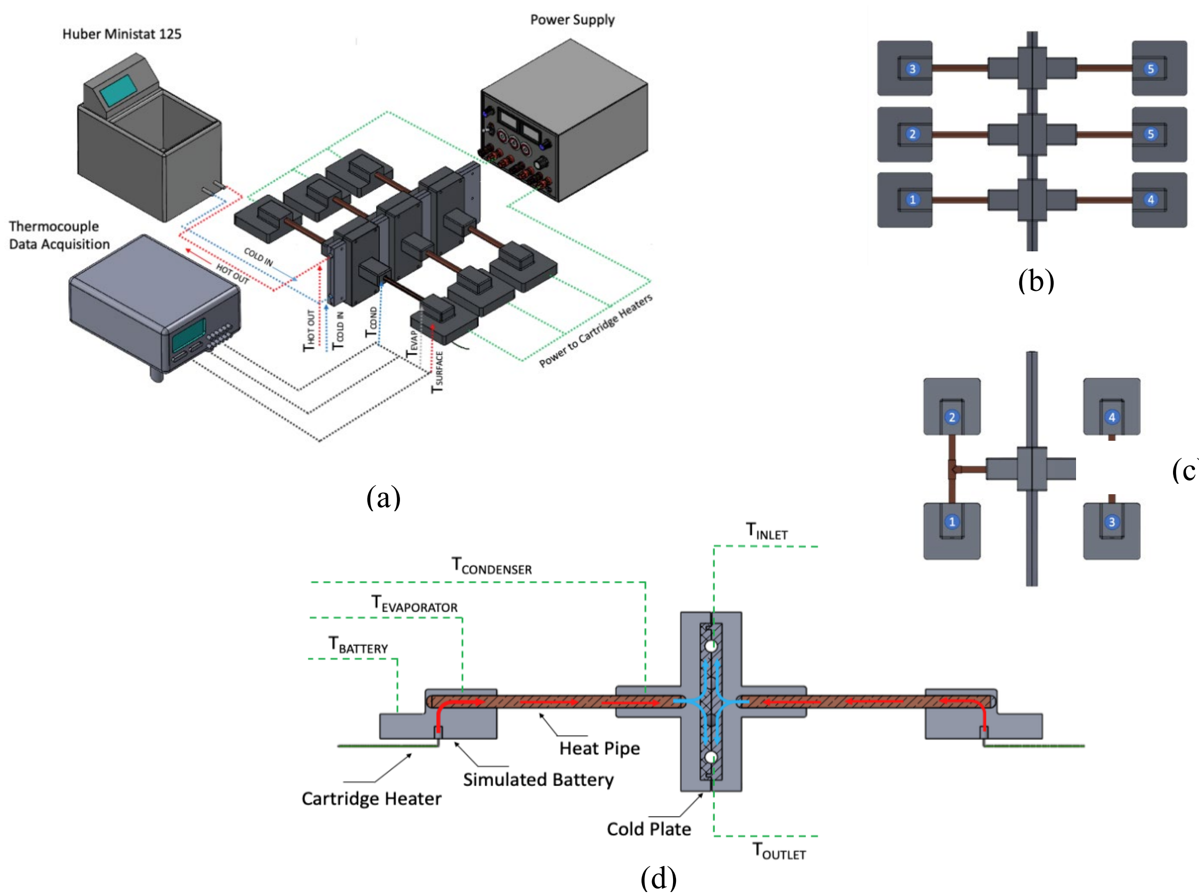


Fig. 1: (a) Schematic of experimental set-up (b) Numbered simulated battery blocks for straight heat pipe setup, (c) Simulated battery blocks for branched heat pipe setup (d) Cross section of BTMS highlighting heat transport from simulated batteries to liquid cooled cold plate showing the location of thermocouples for in each battery, heat pipe along with coolant temperature.

Evaporator and condenser temperatures (T_{evap} and T_{cond} respectively) were used to determine the heat pipe thermal resistance, R_{th} [ºC/W] as given by Eq. 1, where Q is the power supplied to each cartridge heater in Watts.

$$R_{th} = \frac{\Delta T}{Q} = \frac{T_{evap} - T_{cond}}{Q} \quad (1)$$

Effective thermal conductivity (k_{eff}) was determined using Eq. 2

$$k_{eff} = \frac{l_{eff}}{R_{th}A} \tag{2}$$

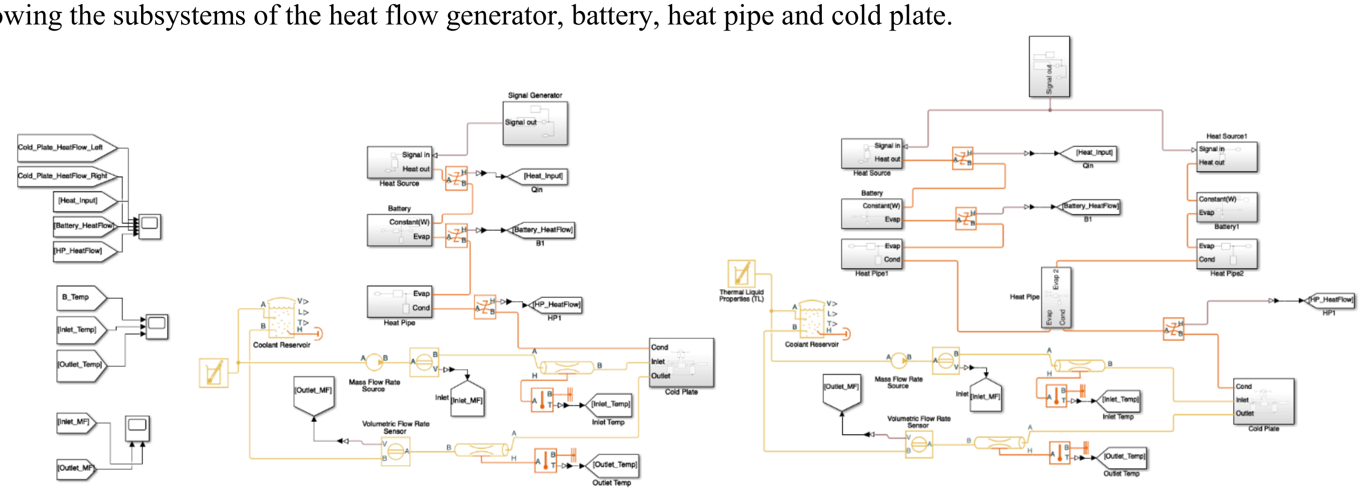
where, l_{eff} is the effective length of the heat pipe and A is the cross-sectional area of the heat pipe.

The Coefficient of Performance of the cold plate is given by equation 3, where Q is the heat dissipated by the coolant, P_{pump} is the hydraulic power, ($P_{pump}= \dot{Q}\Delta P$) of the cold plate system and ṁ is the mass flow rate of the coolant.

$$COP = \frac{Q}{P_{pump}} = \frac{\dot{m}C_p(T_{outlet}-T_{inlet})}{\dot{Q}\Delta P} \tag{3}$$

3. Numerical Analysis

A numerical analysis was carried out on the BTMS using Mathworks Simscape which enables the modelling of physical systems using a block diagram technique simulating physical connections between each block. A simulated system was developed to represent a singular battery in the system and to analyse its response to increasing power input. The system developed simulates heat flow through each component of the system and the subsequent temperature rise in the simulated battery. Heat dissipation by the liquid coolant is also analysed using a thermal liquid circuit connected to a simulated cold plate. Figure 2 (a) and (b) present the block diagram of the simulated single and MBHP systems showing the subsystems of the heat flow generator, battery, heat pipe and cold plate.



(a) (b)

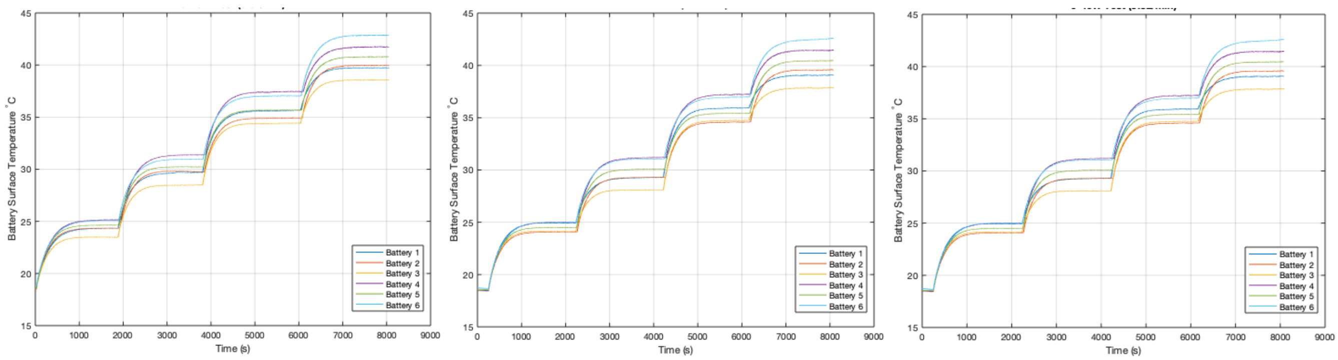
Fig. 2: Block diagram of simulated BTMS system for singular battery heat dissipation by conduction

Increasing power input in the system during increasing discharge rate is simulated using a signal generator that provides timed, increasing step inputs to the controlled heat source. This signal increases the step input by a magnitude of 10 every 2000 seconds to replicate the experimental test. Heat flow through the battery and heat pipe is modelled using conductive heat transfer and thermal capacitance blocks to simulate the thermal resistance and capacitance. A temperature sensor in the battery subsystem plots results for temperature change in the battery over the duration of the simulation. The respective input parameters for each component were determined from material data sheets and measurement of the physical components. Heat dissipation by liquid cooling is modelled using the cold plate subsystem in the model. The subsystem conducts heat from the condenser section of the simulated heat pipe and dissipates it into the thermal liquid circuit in the model. The thermal liquid circuit contains a reservoir and mass flow rate source to circulate liquid to the cold plate subsystem. Temperature and volumetric flowrate sensors are also used in the circuit to determine the temperature rise in the liquid between the cold plate inlet and outlet. The mass flow rate was increased for each simulation by adjusting the input value to the mass flow rate source to replicate each experiment. The simulation results results were compared to experimental results to determine whether the experimental setup can be analysed as a thermal equivalent circuit model for a standard heat pipe setup.

4. Results and Discussion

4.1 Battery Block Temperatures

The recorded temperature response of the battery blocks to increasing heat load and increasing coolant flow rate are shown in Fig. 3 (a), (b) and (c). Temperatures were recorded every second over a period of 8000 seconds and results plotted using MATLAB



(a) (b) (c)

Fig. 3: Battery surface temperature response for increasing heat load at coolant flow rates of (a) 3.5 L/min, (b) 4.5 L/min and (c) 5.5 L/min

It is evident from Fig. 3 that each battery has a similar temperature response to an applied power input. Temperatures rise after applying each incremental power and reach a steady state temperature within approximately 1200 seconds. The temperature difference between batteries should ideally remain at a constant, low value throughout operation. However, it can be seen from Fig. 3 that as the power input increases the temperature difference between batteries also increases. Table 1 presents the maximum temperature among the six batteries at each power as well as the maximum temperature gradient at a given flow rate. According to Pesaran [1] the safe, optimal temperature for the operation of Li-ion batteries is 25-40ºC and a temperature gradient of <5ºC between each cell. It can be seen from Fig. 3 and Table 1 that at power levels of up to 30W these conditions are fulfilled for all flow rates. At loads of 40 W the maximum temperature gradient is not exceeded but the maximum temperature is marginally above the 40 ºC upper limit for all three flowrates. It can also be seen that as the coolant flow rate increases from 3.5 L/min to 5.5 L/min, the maximum temperature and maximum temperature gradient, decrease by 3.6% and 18.6% respectively at heat loads of 40W. An increase in coolant flow rate can be used to reduce battery temperatures and temperature gradients however at an overall level only marginal gains can be achieved as these reductions are negated by a decrease in cold plate performance. An analysis of the current setup using Eq. 3 showed that COP drops approximately 70% as flowrate increases. This finding is similar to that reported by [6], who found that there was an unequal gain in heat transfer for the increased pumping power needed to increase coolant flow rate.

It is also worth noting that the temperature gradient could be a result of thermal resistance variations between each heat pipe setup which includes the battery block, heat pipe and condenser. Unequal thermal resistance between each battery and the cold plate for a constant applied heat load, could result in either higher or lower temperatures in each battery by influencing the temperature difference between two points.

Table 1: Table of values for maximum battery temperature and maximum temperature gradient for increasing heat load and increasing coolant flow rate.

Flow Rate(L/min)	Power (W)	Max. Temperature (ºC)	Max. $\Delta \boldsymbol{T}$ (ºC)
3.5	0	18.7	0.2
	10	25.1	1.6
	20	31.4	2.9
	30	37.5	3.1
	40	42.9	4.3
4.5	0	18.6	0.2
	10	25	1
	20	31.2	3.1
	30	37.2	2.6
	40	42.6	3.5
5.5	0	18.6	0.2
	10	25	1.4
	20	30.9	3
	30	36.7	2.2
	40	41.4	3.5

4.2 Heat Pipe Thermal Resistance and Effective Thermal Conductivity

The thermal resistance of the heat pipes was determined using Eq. 1 where the evaporator and condenser temperatures were measured experimentally. This value was then used with Eq. 2 to determine the effective thermal conductivity of the heat pipes which was then incorporated into the heat conduction model in Simscape. Fig. 4 (a) and (b) present the evaporator and condenser temperatures at a coolant flow rate of 3.5 L/min for power inputs from 10 W to 40 W. At 10 W (Fig. 4 (c)) the difference in thermal resistance between heat pipes was found to be 0.8 ºC/W. An average value of 0.18 ºC/W and 14185.8 W/mK used in Simscape for the thermal resistance and effective thermal conductivity respectively.

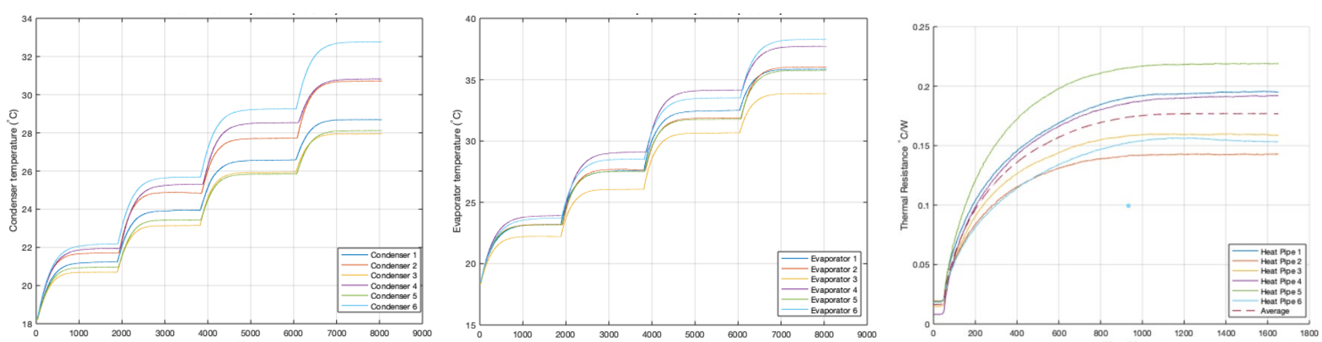
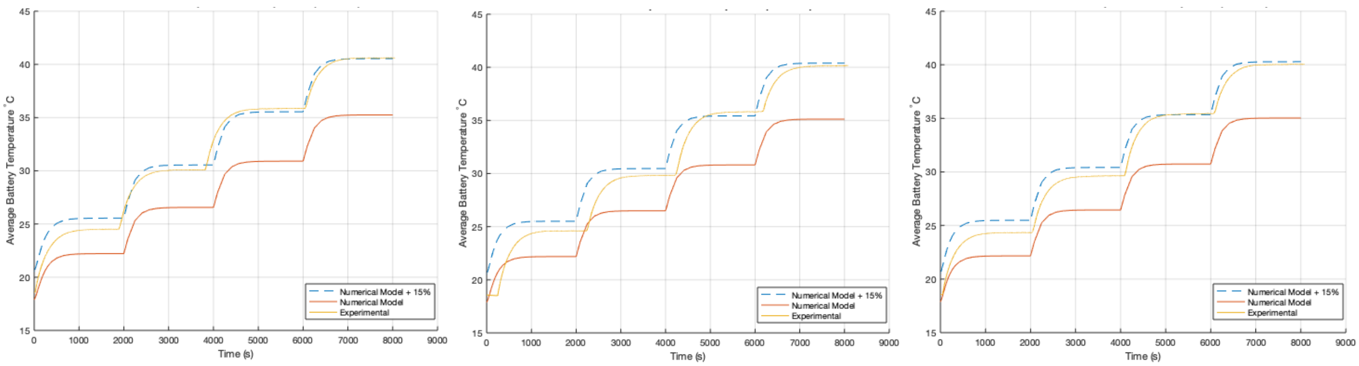


Fig 4: Plot of experimental (a) condenser and (b) evaporator temperatures for power inputs of 10 W- 40 W and coolant at 3.5L/min and (c) heat pipe thermal resistance at 10W

4.2 Numerical Results

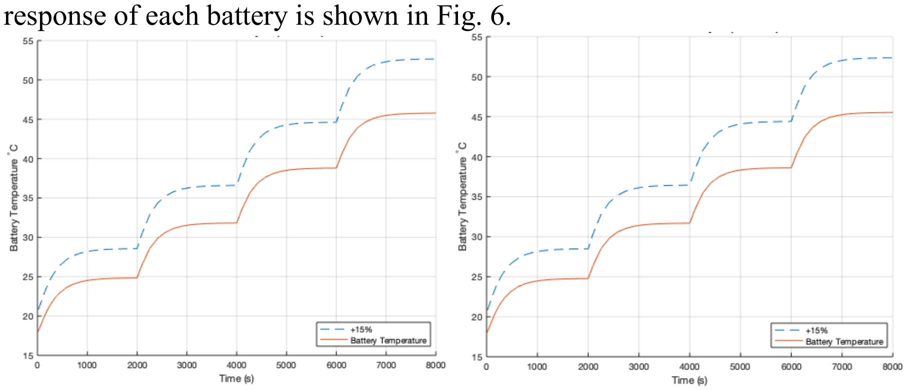
As previously discussed, the numerical model of the BTMS in Simscape used the same cycle as the experimental setup. Fig. 5 compares the experimental average battery temperature with the numerically predicted values for varying flow rates of 3.5L/min, 4.5L/min and 5.5L/min.



(a) (b) (c)

Fig. 5: Average experimental battery temperature compared to numerical predicted temperatures for flow rates of (a) 3.5 L/min, (b) 4.5 L/min and (c) 5.5 L/min.

It is clear from Fig. 5 (a) that the simulated battery in the numerical model exhibits a similar temperature response to the experimental setup but at a lower temperature. It can also be seen from the plot that when an error of 15% is applied to the numerical results, the experimental and numerical steady state temperatures are within +/-1ºC of each other. This trend held for tests with increased coolant flow rate as shown in Fig. 5 (b) and (c). The lower predicted temperatures represent a better performing system compared to the experimental results, but the model does not account for differences in thermal resistance between heat pipes and contact thermal resistance in the evaporators and condensers. Having validated Simscape as a means to model a BTMS, a multi branched heat pipe (MBHP) setup was created in Simscape. Previous work [8] investigating the use of MBHP's for heat dissipation reported thermal resistance and effective thermal conductivities values of 0.12ºC/W and 21.2×10^3 W/m.K. These values were used in the Simscape model for a MBHP structure similar to that shown in Fig. 1 (c). A plot of the temperature response of each battery is shown in Fig. 6.



(a) (b)

Fig. 6: MBHP predicted battery temperatures at coolant flowrates of (a) 3.5L/min and (b) 4.5L/min. 15% error line in blue.

Fig. 6 shows that the temperature response in a MBHP setup is similar to that of the standard heat pipe model. The temperature response was identical for both battery blocks as the power input is dissipated evenly from both branches in the model, hence only one temperature response was plotted. Similar to the results of the standard heat pipe numerical model, a +15% error margin was applied to the results in an attempt to predict how the experimental set up would behave under the same heat load. Results show a 30% increase in maximum temperature in the battery, from 35 ºC to 45.7 ºC., in comparison with the standard heat pipe model numerical results This is expected as the power input is doubled at the condenser section of the heat pipe while using the same amount of cooling power. Using the 15% error line to predict actual battery temperature, it can be seen that the maximum temperature is approximately 52 ºC at a heat load of 40 W, exceeding the set limits. At heat loads of <20 W for normal discharge rates, the battery temperature is within the set limits. Similar to the standard heat pipe model, as flow rate increases, temperatures decrease, but are still above the set limits.

5. Conclusion

An experimental apparatus was design and used to test the viability of heat pipes as a means of passively cooling simulated batteries. A thermal equivalent circuit was developed for single and MBHP configurations and analysed using MATLAB's Simscape. The proposed BTMS operates sufficiently up to power inputs < 40 W under previously set [1]. A maximum battery temperature of 37.5 ºC and a temperature difference of 3.1 ºC was achieved at a power input of 30 W and coolant flow rate of 3.5 L/min. At loads of 40 W the battery temperature marginally exceeds the 40 ºC limit at all three coolant flow rates. As coolant flow rate increased, maximum battery temperature and temperature gradient decreased by 3.6% and 18.6% respectively. However, the coefficient of performance subsequently decreases by 70% due to an increase in pressure drop with flow rate. The thermal equivalent circuits developed in Simscape reflect the temperature response of the simulated batteries in the experimental setup. The model underpredicts the battery temperature for each heat load as it does not account for differences in thermal resistances in the system. When an error of +15% is applied to simulation results the response is within +/- 1ºC of the experimental results. An analysis of a MBHP configuration showed that battery temperature increased in comparison to the standard heat pipe setup for the same heat loads when using the +15% error margin to predict actual battery temperature. This is due to the doubling of the heat load at the condenser section of the heat pipe. Maximum battery temperatures exceed 40 ºC at heat loads > 30 W but at lower heat loads the temperatures are within set limits for Li-ion battery operation according to Pesaran [1]

Acknowledgements

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