Nanofluid-filled heat pipes in managing the temperature of EV lithium-ion batteries

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Abstract. This paper presents the numerical study on the usage of Al₂O₃ nanofluid-filled heat pipes in controlling the temperature of lithium-ion batteries, specifically for the application in powering the electric vehicle (EV). The heat pipes thermal management system (HPTMS) was modelled as a solid continuum body with high equivalent thermal conductivity. Thermal resistance network model was used to determine the equivalent thermal conductivity. The effect of heat inputs and the nanoparticle volume concentration on the performance of the HPTMS were studied. The simulation results were validated against previous experimental works and showed good agreement with each other. Results also indicate that by using 1.5%vol Al₂O₃ as the working fluid in the heat pipes, the battery surface temperature and the total thermal resistance can be reduced by as much as 4.44°C (7.28%) and 15%, respectively.

1. Introduction

Lithium ion batteries (LIB) are the main component of an electric vehicle (EV) as they store and provide energy source for the vehicle to function. They have been widely used, not only as the energy source in EV, but in various consumer products. It has been demonstrated that operating the batteries at temperature of above than 50°C would reduce their recoverable power and capacity, charging efficiency and their lifecycle [1-2]. In order to properly manage and control the temperatures of the batteries within the required safety limit, an effective battery thermal management system (TMS) should be implemented.

The TMS that could be employed can be in the form of air cooling, liquid cooling, phase-change cooling or a combination thereof. One of these methods is heat pipes TMS, an innovative method that uses the concept of phase-change cooling. Heat pipes initially used in automotive application as the thermal management for electronics components in the vehicles such as car and aircrafts [3-4]. It has started to gain momentum to be used as part of the battery TMS when it firstly experimentally investigated by Rao et. al. [5]. They found that heat pipes are capable of controlling the battery temperature to below 40°C provided that the heat generation of the battery is less than 20 W.

Different types and shapes of heat pipes used as the battery TMS has been investigated by several researchers. Tran et al.[6-7] studied the application of tube heat pipes and flat heat pipe. Wang et al. [8] investigated the application of L-shaped heat pipes for LIB under ‘off-normal’ conditions. Liu et al.[9]
focused on the application of an innovative form of heat pipe, the ultra-thin micro heat pipe (UMHP) as part of the battery TMS.

Majority of the HPTMS employ heat pipes that are exposed to water cooling at the condenser section, but others conducted studies to investigate the HPTMS for different method of condenser cooling; air cooling [6-7] and wet cooling (water spray) [10]. The use of heat pipes in combination with other cooling methods has also been the focus of a few investigators. In a study by Ye et al. [11], heat pipes with cooling plates were used to manage the battery temperature. TMS that employed heat pipes and PCM were investigated by Yamada et al. [12] and Hata et al. [13].

All these studies use different types of heat pipes that operates using the same working fluid, which is water. There are no studies conducted on using nanofluids as the heat pipes’ working fluid. Hence it is the objective of this work to investigate the feasibility of using nanofluids-filled heat pipes as the battery TMS. This work focuses on the usage of Al₂O₃ nanoparticle at different volume concentration with water as the base fluid.

2. Simulation Setup

The setup of the HPTMS were adopted from the authors’ previous work [14]. The heat pipes were embedded on an 8-mm aluminium plate, which then placed between two proxy battery cells (Figure 1). The heat generated by the battery cell were represented by the heat input provided by cartridge heaters (stainless steel, \( d = 10 \text{ mm}, L = 100 \text{ mm} \)) slotted in the middle of each of the proxy cell. The detail of the LIB cell that is being considered in this work is shown in Table 1. The dimension of the heat pipes are shown in Table 2. The condenser of the heat pipes is cooled by circulating water.

### Table 1. Details of the LIB

| Property | Value |
|----------|-------|
| Chemistry | LiFePO₄ |
| Capacity | 10 Ah |
| Voltage | 3.2 V |
| Shape | Pouch |
| Height | 140 mm |
| Width | 86 mm |
| Thickness | 12 mm |

### Table 2. Dimension of the Heat Pipe

| Property           | Value (mm) |
|--------------------|------------|
| Outer diameter, \( d_o \) | 6.0 |
| Inner diameter, \( d_i \) | 5.4 |
| Evaporator length, \( L_e \) | 100 |
| Condenser length, \( L_c \) | 150 |
| Total length, \( L \) | 300 |

2.1 Simulation Model

The HPTMS was modelled as a solid continuum body with high equivalent thermal conductivity. To reduce model complexity and computing resource, the model was simulated using a quarter symmetry simulation. The computational domain is shown in Figure 2, consists of unstructured mesh with 92,984 number of elements. In this work, steady state thermal analysis was conducted for the model using ANSYS Mechanical module.

Three boundary conditions were specified for the quarter model; internal heat generation for the solid body of the cartridge heater, external convection on the wall of the condenser and adiabatic conditions for all the other wall surfaces. The internal heat generated by the heater is calculated from the heat input supplied to the heater from the variable AC controller, in the range of \(10 \text{ – } 30 \text{ W} \) for each heater.

The convection heat transfer coefficient, \( h_{ext} \) is determined from the Churchill & Bernstein correlation for the Nusselt number:

\[
Nu_t = 0.3 + \frac{0.62 \cdot Re^{0.5} Pr^{rac{1}{3}}}{1 + \left( \frac{0.4}{Pr} \right)^{rac{2}{3}}} \left[ 1 + \left( \frac{Re}{282000} \right)^{\frac{5}{6}} \right]^{rac{5}{2}}
\]  

(1)
2.2 Thermal Conductivity Model

Heat pipes have higher thermal conductivity than a solid copper pipe. Their thermal conductivity can be as high as 90 times that of a solid copper pipe [15]. Thus, specifying the thermal conductivity of the heat pipes the same as that for copper would be inaccurate. Hence, in this work, thermal resistance network analysis was used to determine the equivalent thermal conductivity of the heat pipes.

The thermal resistance network for the heat pipes is shown in Figure 3. To simplify the network, the existence of sintered wick on the heat pipes are ignored. The descriptions of each of the resistance are explained in Table 3. The convection heat transfer coefficient for the evaporation or boiling at the evaporator can be found from Imura et. al [16].

\[
h_{\text{evap}} = 0.32 \sigma^{0.4} \left[ \frac{\rho_f^{0.65} k_l^{0.3} c_{p_l}^{0.7} g^{0.2}}{\rho_v^{0.25} h_{fg}^{0.4} \mu_l^{0.1}} \right] \left( \frac{P_{\text{sat}}}{P_{\text{atm}}} \right)^{-0.3} \]

(2)

The heat transfer coefficient for the condensation process on the internal surface of the condenser section is calculated from Jouhara and Robinson [17].

\[
h_{\text{cond}} = 0.85 \cdot \text{Re}_f^{0.1} \cdot h_N \cdot \exp \left[ -0.00067 \cdot \frac{\rho_l}{\rho_v} - 0.14 \right]
\]

(3)

where \( \text{Re}_f = \frac{4 \cdot Q}{\pi \cdot d_i \cdot h_{fg} \cdot \mu_l} \) and \( h_N = 1.47 \cdot \text{Re}_f^{-1/3} \left[ \frac{\rho_l (\rho_l - \rho_v) g k_l^2}{\mu_l^2} \right]^{1/3} \).

The total thermal resistance is determined from:
The equivalent thermal conductivity of the heat pipes can be calculated from:

\[ k_{eq} = \frac{L_{eff}}{R_{tot} \cdot A_{cs}} \]  

(5)

**Table 3.** The description and the equations for the resistance components

| Label | Description | Equation |
|-------|-------------|----------|
| \( R_1 \) | Radial conduction resistance at the evaporator | \( R_1 = \frac{\ln\left( \frac{d_i}{d_o} \right)}{2 \pi k_p L_e} \) |
| \( R_2 \) | Evaporation resistance | \( R_2 = \frac{1}{h_{\text{evap}} \pi d_i L_e} \) |
| \( R_3 \) | Vapor space resistance | \( R_3 = \frac{T_{\text{sat,e}} - T_{\text{sat,c}}}{Q_{\text{latent}}} \) |
| \( R_4 \) | Condensation resistance | \( R_4 = \frac{1}{h_{\text{cond}} \pi d_i L_c} \) |
| \( R_5 \) | Radial conduction resistance at the condenser | \( R_5 = \frac{\ln\left( \frac{d_o}{d_i} \right)}{2 \pi k_p L_c} \) |
| \( R_6 \) | Axial wall conduction resistance | \( R_6 = \frac{L_{eff}}{k_p A_{cs}} \) |
| \( R_7 \) | External convection resistance | \( R_7 = \frac{1}{h_{\text{ext}} \pi d_o L_c} \) |

**Figure 3.** The thermal resistance network for the HPTMS.

2.3 Nanofluid Thermophysical Properties

The required thermophysical properties of the Al₂O₃ nanofluid for this study are the effective thermal conductivity, density \( \rho_{nf} \), specific heat capacity \( c_{p,nf} \) and viscosity \( \mu_{nf} \). To characterize the phase change phenomena occurring in the heat pipes, the latent heat of vaporization (LHV) for the nanofluid is also needed. Table 4 outlines the equations used to determine the required properties.
Table 4. Equations for the thermophysical properties of Al₂O₃ nanofluids

| Properties                        | Equations used                                                                 |
|----------------------------------|-------------------------------------------------------------------------------|
| Nanofluid Effective Density      | \( \rho_{nf} = (1 - \phi) \rho_{bf} + \phi \rho_{np} \)                      |
| Nanofluid Specific Heat Capacity | \( c_{p,nf} = \frac{(1 - \phi)c_{p,bf} + \phi(c_{p,np})}{\rho_{nf}} \)        |
| Nanofluid Effective Thermal Conductivity | The Maxwell model: \( k_{nf} = k_{bf} \cdot \left[ \frac{k_{np} + 2k_{bf} + 2(k_{np} - k_{bf})\phi}{k_{np} + 2k_{bf} - (k_{np} - k_{bf})\phi} \right] \) |
| Nanofluid Effective Viscosity    | The Brinkman model: \( \mu_{nf} = \frac{\mu_{bf}}{(1 - \phi)^{2/3}} \)       |
| Nanofluid Effective LHV          | Fitted model from experimental data of Bhuiyan et al. [18] \( h_{fg,nf} = (-130.4 \cdot \ln(100\phi)) + 1446 \) |

The nanofluid effective properties were determined at five different nanoparticle volume concentrations; 0.05%, 0.1%, 0.5%, 1.0% and 1.5%. These properties were then substituted inside the equations (2) and (3), replacing the properties for liquid, in order to calculate the thermal resistance of the evaporation and condensation for the heat pipes.

3. Results & Discussion

3.1 Numerical Model Validation

For the purpose of validating the numerical model, the simulation results obtained when the working fluid inside the heat pipes is water (or 0% vol Al₂O₃) is compared against the experimental data acquired from the authors’ previous work [14]. The maximum battery surface temperature for different heat inputs is shown in Figure 4 and 5 for both simulation and experimental results (with 10% error bar). The figure shows that the battery surface temperature predicted by the simulation varies less than 10% from the experimental results. The temperature difference of the battery surface predicted by the simulation varies within 12% from the experimental data, indicating reasonable agreement with each other (Figure 5). The temperature difference is defined by the difference between the maximum and the minimum battery surface temperature. Thus, it is concluded that the simulation technique employing solid continuum body to represent the heat pipes can produce reasonably accurate results.
3.2 Effect of $\text{Al}_2\text{O}_3$ volume concentration

Figure 6 and 7 show the simulation results for the HPTMS at different heat inputs for varying $\text{Al}_2\text{O}_3$ volume concentration. In general, $\text{Al}_2\text{O}_3$ nanofluid-filled heat pipes are able to reduce the temperature of the battery surface. At 30 W and 0.05% volume loading of $\text{Al}_2\text{O}_3$, the battery surface temperature reduces from 60.98°C to 58.62°C (3.87%). At the highest volume loading of $\text{Al}_2\text{O}_3$, the temperature reduces by 4.44°C (7.28%).

As the $\text{Al}_2\text{O}_3$ volume concentration increases from 0.05% to 1.5%, the battery temperature reduces by 2.08°C at 30 W heat input and 1.59°C at 20 W heat input (Figure 7). The temperature of the battery reduced by as much as 3.54%, indicating insignificance of the effect of the nanoparticle volume concentration.

Figure 4. The maximum battery surface temperature at different heat inputs ($\phi = 0$ %vol)

Figure 5. The temperature difference within the battery at different heat inputs ($\phi = 0$ %vol)

Figure 6. Maximum battery surface temperature at different heat inputs.

Figure 7. Maximum battery surface temperature at different $\text{Al}_2\text{O}_3$ volume concentration
The results of the total thermal resistance for the HPTMS are shown in Figure 8 and 9. At 30 W, Al$_2$O$_3$ nanofluid is capable of decreasing the thermal resistance of the heat pipes from 0.4677 °C/W to 0.394 °C/W (15% reduction) indicating superior performance by the nanofluid as compared to water alone. At 20 W, similar performance was also observed for the nanofluid-filled heat pipes. In Figure 9, as the nanoparticle volume concentration increases, the thermal resistance decreasing by as much as 8%. This result shows that Al$_2$O$_3$ volume concentration has a significant effect on the total thermal resistance of the HPTMS.

4. Conclusion

In this work, the usage of Al$_2$O$_3$ nanofluid-filled heat pipes as the lithium-ion battery TMS for EV application were numerically investigated. The heat pipes were modelled as the solid continuum body with high equivalent thermal conductivity. This thermal conductivity was determined using the thermal resistance network analysis.

Reasonable agreement between the numerical simulation results and that from experimental works indicates the thermal resistance network model can be used to determine the equivalent thermal conductivity of the heat pipes. The results also indicate that by using 1.5 vol% Al$_2$O$_3$ as the working fluid in the heat pipes can effectively lower the temperature of the battery surface further by as much as 4.4°C (7.28%), and reduces the total thermal resistance by as much as 15% when compared with water-filled heat pipes. It was also observed that although there is insignificant effect of the nanoparticle volume concentration on the battery temperature, it plays an important role in reducing the total thermal resistance of the HPTMS.

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