Computational fluid dynamic and thermal analysis of Lithium-ion battery pack with air cooling

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HIGHLIGHTS

- We designed and analyzed the thermal behavior of the Li-ion battery pack.
- We analyzed the heat generation of 38,120 Li-ion cell using ARC.
- We validated the simulation results with experimental studies.
- We developed the correlations of Nu and Re for the air cooling battery pack.

ABSTRACT

A battery pack is produced by connecting the cells in series and/or in parallel to provide the necessary power for electric vehicles (EVs). Those parameters affecting cost and reliability of the EVs, including cycle life, capacity, durability and warranty are highly dependent on the thermal management system. In this work, computational fluid dynamic analysis is performed to investigate the air cooling system for a 38,120 cell battery pack. The battery pack contained 24 pieces of 38,120 cells, copper bus bars, intake and exhaust plenum and holding plates with venting holes. Heat generated by the cell during charging is measured using an accelerating rate calorimeter. Thermal performances of the battery pack were analyzed with various mass flow rates of cooling air using steady state simulation. The correlation between Nu number and Re number were deduced from the numerical modeling results and compared with literature. Additionally, an experimental testing of the battery pack at different charging rates is conducted to validate the correlation. This method provides a simple way to estimate thermal performance of the battery pack when the battery pack is large and full transient simulation is not viable.

1. Introduction

Dwindling of fossil fuel supplies and environmental concerns has spurred worldwide interest in the development of advanced energy storage system for electric vehicles (EVs). There are various types of battery used as an energy storage system in EVs and Hybrid Electric Vehicles (HEVs) such as lead acid, Nickel Metal Hydride (NiMH) and Lithium ion battery [1,2]. However, low energy density and depth of discharge (DOD%), sulfation, toxicity, short shelf life and high self discharge rate have deterred automotive manufacturers away from selecting lead acid battery as the primary energy storage system for EVs and HEVs [3]. On the other hand, Li-ion batteries have high energy density, no memory effect, no periodic deliberate full discharge requirement, low maintenance, fast charging capability and low self-discharge rate as compared to Nickel Cadmium (NiCd) and Nickel Metal Hydride (NiMH) batteries [4]. There is a variety of Li-ion batteries available in the market with different specific energy and voltage for diverse applications. The potential candidates of Li-ion batteries for EVs are Lithium Cobalt Oxide (LiCoO₂), Lithium Manganese Oxide (LiMn₂O₄), Lithium Iron Phosphate (LiFePO₄) and Lithium Nickel Manganese Cobalt Oxide (NMC) with different types of packaging such as spiral wound, elliptic and stacked plate make a good choice for the energy storage system [5–7].

Battery pack cycle life, capacity, charging duration, durability and warranty cost are the parameters that affecting the cost and
It will be reduced if the battery is operated at temperatures below 0°C depending on the battery chemistry. On the other hand, the reactant gases released from this reaction are dangerous and can cause an explosion or fire when the temperature difference of the battery system by 4°C and maximum cell temperature is reduced. As compared to uni-directional flow, reciprocating period of 120 s is effective to reduce the cells temperature difference of the battery system by 4°C and maximum temperature by 1.5°C. Ghosh et al. designed a battery pack cooling system for Ford Fusion Hybrid and Mercury Milan Hybrid [36,37]. CFD simulation was used to evaluate the performance of the cooling system. The configuration of battery pack comprised of 4 D-size NiMH cells arranged in series and 8 D-size NiMH cells arranged in parallel. A small temperature gradient of 1.2°C within the battery cell is achieved by controlling the wrap angle around the leading cells [36,37].

Liquid cooling is more complex, but offers a higher cooling capacity than air cooling system. Liquid cooling of the battery pack is realized by using a cold plate or heat spreader sandwich between the cells or submerging the cell in a dielectric fluid. Mineral oil, water, dielectric fluid and ethylene glycol mixture are normally used as a heat transfer medium. The disadvantages of liquid cooling system are the necessity of large space and the increase of vehicle total weight, higher cost, high pumping power, potential leakage of cooling fluid and poor thermal contact between the cold plate and cell [23,35]. The operating characteristics of the cold plate used in the battery pack are determined by the geometry of

reliability of the EVs and these parameters depend on the thermal management system of the battery pack. Battery temperature and uniformity have a strong influence on the battery pack power, cell balancing and charge acceptance during regenerative braking. Reaction heat, ohmic heat, reversible heat and external contact resistance heat are the heat sources of the battery during charging and discharging process. A rapid increase in the cell temperature during the end of state of charge (SOC) may cause the performance of battery deteriorate and reduce the cycle life of the cell [8]. Hence, this heat must be dissipated as fast as possible. The optimum operating temperature of Li-ion battery is between 25°C and 40°C and desirable temperature uniformity within a battery pack is less than 5°C [1,9]. The energy storage and cycle life of the battery pack are determined by the geometry of the cells or submerging the cell in a dielectric fluid. Mineral oil, water, dielectric fluid and ethylene glycol mixture are normally used as a heat transfer medium. The disadvantages of liquid cooling system are the necessity of large space and the increase of vehicle total weight, higher cost, high pumping power, potential leakage of cooling fluid and poor thermal contact between the cold plate and cell [23,35]. The operating characteristics of the cold plate used in the battery pack are determined by the geometry of

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the channel, route, width and length [11,28]. A serpentine-channel cooling plate was proposed for the liquid cooling system battery pack [11,28]. Numerical optimization was carried out to investigate the channel width and position to achieve the smallest pressure drop, average temperature and temperature uniformity. In order to avoid a local optimization of the cold plate design, Latin hypercube sampling was used to select the cold plate design. The cold plate needs to have a narrow inlet channel and widening toward the outlet to balance the effect of velocity, heat transfer area and fluid-solid temperature gradient to achieve the highest temperature uniformity [11,28]. However, the study was focused on low flow rate of the coolant and did not validate with any experimental results. Jin et al. introduce an oblique fin design for the liquid cold plate used in the EVs battery pack [38]. The main objective to introduce an oblique cut on the conventional straight channel is to improve the heat transfer and temperature uniformity along the axial direction by breaking the developed thermal boundary layer. Numerical simulation and experimental results show that the heat transfer coefficient of oblique mini-channel is higher than conventional straight fin at low flow rate [38].

Phase change materials (PCMs) utilized chemical bonds to store and release heat. Paraffin wax embedded in the graphite matrix is commonly used in the PCM enhanced battery pack. PCM enhanced battery pack also require auxiliary cooling systems to remove accumulated heat. The downsides of PCM enhanced battery pack such as heat accumulation at the center of the battery pack, additional weight and undesirable thermal mass [39]. Kim et al. used a lumped capacitance model to benchmark the performance of a PCM thermal management system with forced air cooling ($h = 15$ W m$^{-2}$ K$^{-1}$) and natural convection cooling ($h = 6$ W m$^{-2}$ K$^{-1}$) for a Li-ion battery pack under 40 A single discharge for 9 min [39]. The large thermal mass of the PCM results in a low battery pack peak temperature. Nevertheless, a large thermal mass and a smaller heat transfer at the surface give rise to a slower cooling rate for the PCM. At a high temperature of 40 $^{\circ}$C, the PCM reaches its melting point and prevents further rise in temperature by converting the heat generated from the battery into latent heat as the wax in the matrix melts. Therefore, the PCM module has the lowest temperature rise and slowest cool-down as the stored heat is slowly rejected to the environment [39]. Currently, the use of PCM in commercial EVs/HEVs/PHEVs is still being developed.

Wu et al. used numerical simulation to investigate the temperature distribution of the Li-ion battery (12 A h, cylindrical, 40 mm diameter and 110 mm height) by attaching two heat pipes with metallic fin on the battery surface [33]. From the study, the heat generated from the battery at high discharge rate is difficult to be removed using natural convection cooling. The temperature of the battery may reach 65 $^{\circ}$C for a discharge rate of 10 A [33]. However, this problem could be solved by forced convection with the metallic fins and heat pipe attached on the battery surface [33]. Pulsating heat pipe (PHP) can also be used for the battery thermal management system. Swanepoel designed the thermal management system for the Optima Spirocell (12 V, 65 A h) lead acid battery using PHP technology [34]. The simulation and experimental results show that PHP should be constructed with $d < 2.5$ mm pipe and using ammonia as working fluid [34].

In this study, parallel air flow architecture of the battery pack thermal management system was investigated. The battery heat generation was characterized using an accelerating rate calorimeter under constant current charging. Average heat generated in the cell at 3 $I_{rate}$-rate was used in the steady state simulation to investigate the temperature distribution of the cells in the battery pack under different cooling conditions. The heat transfer correlation is subsequently deduced from the simulation results and compared with open literature. Finally, experimental testing of the battery pack at different charging rates was carried out to validate the mathematical model.

2. Design and analysis of the battery pack

2.1. Battery pack design

A battery pack consists of 24 pieces of commercial Lithium Iron Phosphate (LFP) cells with an electric configuration of 12S2P (12 cells in series and 2 cells in parallel) was developed for the current study (Fig. 1). The nominal voltage and capacity of the battery pack were 38.4 V and 16 A h, respectively. Specifications of the LFP cell used in the modeling are summarized in Table 1. The cells were attached to copper bus bars with screws to form the battery array. The arrangement of the batteries in the pack was based on a close-pack structure. The spacing between the cells was 5 mm and 15.50 mm diameter venting holes were created on the holding plate in between four cells to allow cooling air to flow uniformly over the cell surfaces. The pack casing was made of aluminum and plexiglass. Anti-vibration rubber mounts were attached at the bottom of the battery pack to insulate the cells from possible vibration that will be harmful to the electrical connection of the battery pack. Tapered intake and exhaust plenums were used to direct the cooling air flow uniformly to each row of cells.

A constant flow rate of air was provided to the cells through the tangential blower. Cooling air was constricted when its flows through the venting holes and expanded to the cells surfaces. Similarly, constriction and expansion of cooling air occurred again when the cooling air flows out to the exhaust plenum through venting holes in the holding plate. The process of constriction and expansion will induce a pressure drop along the flow path.

The electric contact resistances between the bus bar and cell terminals, which will lead to energy loss, have always been overlooked in the thermal management of battery packs. The effect of electric contact resistance is localized and concentrated at the cell terminals. Poor contact resistance will cause the cell terminals to be hotter than the cell body. In this study, the above issue is taken into consideration and the heat generated due to electrical resistance solved by utilizing the effect of constriction and expansion of the air flow through the venting holes. Constriction and expansion of cooling air will result in significant cooling effects at both terminals of the cell and dissipate the heat generated [24].

2.2. Experimental setup and parameter extraction

Commercial 38,120 LFP cells with a capacity of 8 A h were used in the experimental study. Fig. 2 shows the air flow direction and location of thermocouples and the way they attached to batteries. The thermocouples are attached to battery body using aluminum tape and held firmly at their positions using Kapton tape. The charge and discharge processes of the cells were performed using a battery cycler (Maccor Instrument 4000). The specific heat capacity of a single cell was measured using an adiabatic accelerating rate calorimeter (THT ARC). The heat generation rate in a single cell during constant current charging at increasing SOC was measured using the ARC. The charging of the single cell and battery pack were carried out using three different charging rates, namely 1, 3 and 5 $I_{rate}$-rates. The charging current for 1, 3 and 5 $I_{rate}$-rates is 8, 24 and 40 A, respectively. The $I_{rate}$ rate as per the standard IEC61434 is defined as [40]

$$I_{rate} = \frac{C}{1h}$$  

where $I_{rate}$ represents the discharge current in amperes during one hour discharge and $C$ is the measured capacity of a battery pack or cell.
Table 1
LFP cell parameters.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Parameter</th>
<th>Value</th>
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</thead>
<tbody>
<tr>
<td>Nominal voltage, V</td>
<td>3.2</td>
<td>Internal resistance, Ω</td>
<td>0.0034</td>
</tr>
<tr>
<td>Nominal capacity, A h</td>
<td>8.0</td>
<td>Reference temperature, K</td>
<td>303.15</td>
</tr>
<tr>
<td>Weight, kg</td>
<td>0.335</td>
<td>Cell length, m</td>
<td>0.146</td>
</tr>
<tr>
<td>Cathode material</td>
<td>LiFePO₄</td>
<td>Cell diameter, m</td>
<td>0.038</td>
</tr>
<tr>
<td>Anode material</td>
<td>Graphite</td>
<td>Specific heat capacity, J kg⁻¹ K⁻¹</td>
<td>998</td>
</tr>
<tr>
<td>Thermal conductivity in radial direction, W m⁻¹ K⁻¹</td>
<td>0.2</td>
<td>Thermal conductivity in axial direction, W m⁻¹ K⁻¹</td>
<td>37.6</td>
</tr>
</tbody>
</table>

Fig. 1. (a) Overview of battery pack. (b) Top view of battery pack.

Fig. 2. Cross section view (left side view) of battery pack with thermocouple location on batteries and direction of air flow.

The tangential blower of the battery pack was set to operate at its maximum flow rate (ṁ = 30 g s⁻¹). The air flow rate was measured by a digital air velocity meter (TSI, velocIcalc 9565-P). A differential pressure transducer (Gems sensor, 5266 series) was connected to the pressure taps at the intake and exhaust plenum to measure the pressure drop across the battery pack. The cells were fully discharged to 2.0 V before the charging experiment started. The battery pack was placed in a temperature chamber (Weiss, T1500) and the chamber temperature was set to 30 °C. The experiment was initiated after the cells had achieved an equilibrium temperature of 30 °C. Twenty-seven T-type thermocouples were used in the experiment. Two thermocouples were attached to the intake and exhaust plenum of the battery pack to measure the intake and exhaust air temperatures. One thermocouple was placed outside the battery pack to measure the environment temperature in the chamber. While the temperature of the cell surfaces was measured using twenty-four thermocouples attached to the middle of the cell body. Measurements of battery surface temperature during different charging rates were done at an ambient temperature of 30 °C for an air cooling flow rate of 30 g s⁻¹. A HP 34,970A data acquisition system was used to record the temperature readings. All the tests were repeated three times and the average value was taken. The experimental results were compared with numerical results under similar cooling conditions.

2.3. Numerical procedures

Commercial Computation Fluid Dynamic (CFD) software-ANSYS-CFX was used to complement the experimental study and understand the flow field that is difficult to be observed in the experiment. The governing equations used to solve the time dependent three dimensional flow problems which involved heat transfer are the continuity equation, momentum equation, energy equation and equation of state given in Eqs. (2)–(8) [41].

Continuity equation

\[
\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \mathbf{u}) = 0
\]  

X-momentum

\[
\frac{\partial (\rho \mathbf{u})}{\partial t} + \nabla \cdot (\rho \mathbf{u} \mathbf{u}) = - \frac{\partial P}{\partial x} + \nabla \cdot (\mu \nabla \mathbf{u}) + S_x
\]  

Y-momentum

\[
\frac{\partial (\rho \mathbf{v})}{\partial t} + \nabla \cdot (\rho \mathbf{v} \mathbf{u}) = - \frac{\partial P}{\partial y} + \nabla \cdot (\mu \nabla \mathbf{v}) + S_y
\]  

Z-momentum

\[
\frac{\partial (\rho \mathbf{w})}{\partial t} + \nabla \cdot (\rho \mathbf{w} \mathbf{u}) = - \frac{\partial P}{\partial z} + \nabla \cdot (\mu \nabla \mathbf{w}) + S_z
\]

According to the first law of thermodynamics, the rate of change of energy of a fluid particle is equal to the rate of heat addition to the fluid particle plus the rate of work done on the particle [41]. This yield the following equation:

\[
\frac{\partial (\rho e)}{\partial t} + \nabla \cdot (\rho \mathbf{e} \mathbf{u}) = \rho \mathbf{u} \cdot \nabla \mathbf{U} + \mathbf{h} \cdot \nabla T + H + S_e
\]

Four unknown thermodynamic variables (ρ, P, i and T) from the five partial differential equations: mass conservation, x-, y- and z-momentum equations and energy equation can be obtained through thermodynamic equilibrium. Equations of state relate the other variables to the two state variables. For a perfect gas the following equations provide the link between the variables [41].

\[
P = \rho RT
\]

\[
h_f = \int_{T_{ref}}^{T_f} C_{pf} dT_f
\]

SST turbulence model
The SST turbulence model with automatic wall function is employed to predict the flow behavior in the present study. The SST model has proven to be stable and numerically robust and has a good predictive capability to give a good compromise between accuracy and robustness. Besides, the SST model has been designed to give accurate predictions of the onset and the amount of flow separation under adverse pressure gradients by the inclusion of transport effects into the formulation of the eddy viscosity. The superior performance of the SST model is validated by a large number of studies [42]. The SST model is also recommended for high accuracy boundary layer simulations. In free shear flows, the SST model is identical to the k-ε model. In addition, the SST model has been developed to overcome deficiencies in the k-ε and BSL k-ω model. One of the advantages is the near-wall treatment for low-Reynolds number computations. The SST model also incorporates a slight modification to the eddy viscosity for better prediction of the turbulent shear stress. The details of the SST model can be found in Sparrow et al. [43], Menter et al. [44] and Lee et al. [45]. The transport equations for the SST model are given below:

\[
\frac{\partial (\rho k)}{\partial t} + \frac{\partial (\rho U k)}{\partial x_i} = \frac{\partial}{\partial x_i} \left[ \nu + \sigma_k \mu_{\text{eff}} \right] \frac{\partial k}{\partial x_i} + \beta_{k} \rho \omega^2 + \frac{\partial}{\partial x_i} \left[ \frac{\rho}{\sigma_k} \frac{\partial k}{\partial x_i} \right] \left( \frac{\partial \sigma_k}{\partial x_i} \right) + 2(1 - F_1) \rho \sigma_{\omega^2} \frac{1}{\sigma_k} \frac{\partial k}{\partial x_i} \frac{\partial \sigma_{\omega^2}}{\partial x_i} - \frac{k}{C_{18/C_{19}}} q_{\text{wall}} \tag{9}
\]

\[
\frac{\partial (\rho \omega)}{\partial t} + \frac{\partial (\rho U \omega)}{\partial x_i} = \frac{\partial}{\partial x_i} \left[ \alpha \rho \omega^2 - \beta \rho \omega^2 \right] + \frac{\partial}{\partial x_i} \left[ \frac{(\mu + \sigma_k \mu_{\text{eff}})}{\alpha} \frac{\partial \omega}{\partial x_i} \right] + 2(1 - F_1) \rho \sigma_{\omega^2} \frac{1}{\sigma_k} \frac{\partial k}{\partial x_i} \frac{\partial \sigma_{\omega^2}}{\partial x_i} \tag{10}
\]

where the blending function \( F_1 \) is defined by:

\[
F_1 = \tanh \left\{ \min \left\{ \frac{y}{y^*} \left[ \frac{\sqrt{y^*/y}}{500} \left( \frac{500}{\gamma^*} \right) \frac{4}{\alpha \sigma_{\omega^2}} \frac{1}{\gamma^*} \right] \right\} \right\}^{1/4} \tag{11}
\]

where \( y \) is the distance to the nearest wall.

\( F_1 \) is equal to zero away from the surface (k-ε model) and switches over to one inside the boundary layer (k-ω model).

\[
C_{\text{D}_{\omega^2}} = \max \left( \frac{2 \rho \sigma_{\omega^2}}{10^{-16}} \right) \tag{12}
\]

The energy equation for the cell is given by:

\[
mC_p \frac{dT_c}{dt} = \nabla \cdot (\lambda_c \nabla T_c) + Q_{\text{gen}} \tag{12}
\]

A steady state conjugate heat transfer simulation was performed to predict the thermal performance of the battery pack with all the time derivative terms in Eqs. (2)–(6) equal to zero. Instead of layers of electrodes and separator, the cylindrical 38,120 LFP cell was modeled with a uniform solid structure and anisotropic thermal conductivity is assigned to the battery [25,46,47]. A heat generation rate of 4 W (corresponds to the average heat generation rate during 3 I_rate of charging) per cell was used for the steady state simulation. The heat generated in each cell in the battery pack was assessed to be uniform. Due to tight contact between bus bars and cell terminals, the contact resistance at the cell terminals is kept to minimum which is about 0.3 m Ω. Hence, it is neglected in this study.

The CAD model of the battery pack used for the simulation and cooling air flow path is shown in Fig. 3. A hybrid meshing was adopted to discretize the battery pack domain into 58,511,496 elements using ANSYS ICEM CFD 14.0 SP1. The coolant was air which was assumed to be an ideal gas. Since, the tangential blower of the battery pack operated in blowing mode, the intake of the battery pack was given a mass flow boundary condition while the pressure boundary condition was assigned to the outlet. The intake air temperature was kept at 30°C. The confining walls on the top, side and bottom of the battery pack were specified as no slip, adiabatic wall boundary conditions. Heat loss through the battery pack casing in the CFD analysis is assumed negligible. The Shear Stress Transport (SST) turbulence model with automatic wall function was selected for this study. This model will provide accurate prediction from laminar to turbulent flow and near-wall boundary conditions [48]. Besides, it also utilized automatic wall treatment for maximum accuracy in wall shear and heat transfer predictions as well as capturing the streamline curvature [48]. The computational domain was initialized with ambient conditions at 1 atmospheric pressure. CFX solver was used to solve the governing equations for the conservation of mass, momentum and energy. All simulations were executed with a high resolution scheme to achieve an accurate solution. A tight convergence criterion with an RMS of 1.0 × 10^-6 is applied to the continuity, momentum and energy equations (H-energy and T-energy) for all case studies. It was also ensured that there is no domain imbalance in momentum and energy. All simulations were computed on the 8 nodes-HP clusters. In addition, the grid independence test was carried out to refine the grid size until the simulation results were not affected by any further refinement of the mesh and the relative error of the results (cell temperature and pressure drop across the battery pack) is kept within 5%. Total computation time is about 52 h 20 min.

In actual charging process, a battery may not reach thermal steady state because of the short charging time, especially during fast charging when high current is involved. The common sense may lead to the idea that a comprehensive three dimensional transient modeling is more straightforward and powerful to capture the temperature distribution and evolution within the battery pack. 3D transient modeling can be conducted by importing the transient heat generation rate as a function of time into the CFD model. However, 3D transient modeling for a large battery pack is a tedious and time-consuming undertaking because of the complex air flow dynamics and the conjugate heat transfer between various batteries and the air flow in the battery pack. A steady state CFD simulation take about 52 h 20 min for each run on the 8 nodes-HP cluster and the computing time and results file size for the 3D transient simulation will build up into an unacceptable level, making it impractical for the current study. Therefore, in this study, the steady state modeling focuses on the investigation of temperature distribution and uniformity of the cells in the battery pack. Then, a correlation of Nu number to Re number was developed based on the steady state simulations to evaluate the cooling effectiveness of the battery pack thermal management system in transient state. The effective heat transfer coefficient derived from the steady state modeling was assigned to the surfaces of each cell.
to account for the convective heat transfer there. This simplified method is based on the fact that there is only a small change in
the air temperature and only a minor difference in temperature among the cells.

2.4. Data processing

The steady state of heat transfer from the cells to the air in the battery pack can be expressed as follows [49]:

\[ Q_{\text{total}} = Q_{\text{conv}} + Q_{\text{rad}} + Q_{\text{loss}} \]  

(13)

\[ Q_{\text{conv}} = \dot{m}C_P(T_{\text{out}} - T_{\text{in}}) \]  

(14)

Besides, the rate of convective heat transfer from the cells can also be expressed as

\[ Q_{\text{conv}} = hA_{\text{ext}}\left(T_s - \frac{T_{\text{out}} + T_{\text{in}}}{2}\right) \]  

(15)

The Reynolds number for the cooling air is calculated using Eq. (21) [50,51]:

\[ Re = \frac{\rho U D}{\mu} \]  

(18)

The steady state heat transfer rate through the air is equal to the heat loss of the cells, and the average convective heat transfer coefficient can be deduced via Eq. (14) [49].

\[ R = \frac{Q_{\text{conv}}}{A_{\text{ext}}\left(T_s - \frac{T_{\text{out}} + T_{\text{in}}}{2}\right)} \]  

(17)

The Nusselt number and the Reynolds number are calculated using Eqs. (21) and (22), respectively [52].

3. Results and discussion

3.1. Heat generation of cell

In order to show the accuracy of the measurements, a thorough uncertainty analysis was performed according to the method suggested by Moffat [53]. The uncertainties in this study were determined by the root-sum-square method [53]. The results are shown in Table 2. Average uncertainty measurement of the battery body temperature in the accelerating rate calorimeter and battery pack is about 1.03% and 2.21%, respectively.

Fig. 4 shows the heat generated in the cell measured in the ARC from 0% to 100% SOC at 1, 3 and 5 I-rate of constant current charging. A sharp increase in the heat generated in the cell was found at 90% SOC or greater. This is due to a sudden increase of the ohmic heat toward the end of charging process [47]. As shown in Fig. 4, average heat generated in the cell at 1, 3 and 5 I-rate are about 0.84 W, 4.03 W and 9.48 W respectively. As shown in Fig. 5, the temperature rise at the cell surface in the adiabatic calorimeter is about 7.3 °C, 14.4 °C and 18.8 °C for 1, 3 and 5 I-rate, respectively. The ambient temperature in Singapore is about 30 °C and at 1 I-rate of charging, the cell surface temperature may reach 37.3 °C under no cooling condition. Even though the surface temperature of the cell is within the optimum operating temperature range for the Li-ion battery, the internal temperature of the cell may exceed the optimum operating temperature limit. According to previous studies, large thermal resistance of the active material in the large cylindrical battery will cause about 5 °C of temperature difference between the center and the surface of the cell at the end of 3 I-rate of discharging with heat generation of 14 W and subject to natural convection cooling \((h = 5 \text{ W m}^{-2}\text{K}^{-1})\) [46]. Although the internal resistance of the cell is reduced at high temperatures, thermal aging of the cell is more severe and the cycle life span of the cell is also reduced [38,54]. Hence, an active thermal management system is needed to prolong the cycle life of the cell and optimize the cell performance by operating the battery within 25–40 °C.

3.2. Fluid flow analysis results

For the battery pack thermal management system design, steady state CFD simulations were performed using a mass flow rate from 5 to 75 g s\(^{-1}\) at 30 °C. High resolution of the mesh and fluid-thermal CFD model is important to capture the flow field and conjugate heat transfer in the battery pack. The typical velocity contour plot of the battery pack with the mass flow rate of 40 g s\(^{-1}\) is shown in Fig. 6. The cooling air is rapidly accelerated into the battery pack through the narrow intake plenum and venting holes, creating local high entrance velocities and a large entrance pressure drop. The cooling air is also rapidly accelerated to the battery compartment due to large contraction in the face area of the venting hole, resulting in high heat transfer coefficients on the battery surfaces.

In order to achieve uniform parallel air distribution, the air intake and exhaust plenum need to be designed carefully to obtain a uniform air flow to the battery compartment and minimize

<table>
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<tr>
<th>Table 2 Averahe uncertainties of the variables.</th>
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<tbody>
<tr>
<td>Properties</td>
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<tr>
<td>Specific heat capacity, %</td>
</tr>
<tr>
<td>(\Delta T_{\text{batt}}), °C (in ARC)</td>
</tr>
<tr>
<td>(\Delta T_{\text{batt}}), °C (in battery pack)</td>
</tr>
<tr>
<td>Heat generation, W</td>
</tr>
<tr>
<td>(A_{\text{ext}}, \text{in}^2)</td>
</tr>
<tr>
<td>Fan power, W</td>
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parasitic pressure drop. The recirculation flow and turbulence must be minimized. Moreover, the intake plenum design must be able to keep the inflow streamlined. A streamline plot of the cooling air, which could provide qualitative analysis of the cooling air distribution in the battery pack is shown in Fig. 7. The flow of cooling air has reasonable streamlines except at the front end of the intake plenum and corner of the battery compartment. Chamfers at the end of the intake plenum and taper design help to divert more flow to the last row of the cells and reduce creation of local turbulence that will cause an additional pressure drop. Cooling air is supplied by the tangential blower to the battery compartment through the venting holes. After picking up waste heat from all the cells, the warmed air is discharged to the exhaust manifold through venting holes and recombines in the exhaust plenum. Air exits the battery pack through tapered exhaust manifold and finally dumped out to the outer environment.

3.3. Temperature variation analysis results

Battery temperature and uniformity have a strong influence on the availability of the charging and discharging power, cell balancing and charge acceptance during regenerative braking. A large variation of temperature in a battery pack can lead to different cell charging and discharging at different rates and lead to electrically unbalanced cells and reduce the performance of the battery pack. A temperature difference of 5 °C would lead to about 10% degradation of power capability, and an increment of 25% of thermal aging kinetics [54]. The thermal aging kinetics can be calculated from the Arrhenius law as below [54]:

\[ k_{\text{aging}} = \exp\left(-\frac{E_a}{RT}\right) \]  

The degradation could escalate to 30–50% by increasing temperature approximately 10–15 K [53]. Hence, variation of temperature across the battery pack should be kept within 3–5 °C [54]. Variation of temperature within a battery cell should be kept within 5–10 °C with consideration of optimum cycle life [35].

![Fig. 4. Heat generated of the cell during various $I_\text{c}$-rate of constant current charging.](image)

![Fig. 5. Experimental cell surface temperature at various $I_\text{c}$-rates of constant current charging.](image)

![Fig. 6. Left side view of battery pack: Velocity contour of airflow through the intake plenum, battery compartment and exhaust plenum.](image)

![Fig. 7. Left side view of battery pack: Surface streamline plot of the air flow path into the battery pack.](image)

![Fig. 8. Average surface temperature and variation of cells temperature in the battery pack for mass flow rate of 5–75 g s$^{-1}$.](image)
The variation of temperature in a single cell is due to cell holder that covers some portion of the battery body and blocking the cooling fluid reaches the cell surface effectively and high thermal resistance across the cell.

Fig. 8 shows the average temperature of the cells in the battery pack for mass flow rate of 5–75 g s\(^{-1}\). The highest temperature was achieved at a mass flow rate of 5 g s\(^{-1}\) and gradually reduced to 33.2 °C at mass flow rate of 75 g s\(^{-1}\). At a flow rate of 40 g s\(^{-1}\), the average surface temperature of the cell is reduced to 35.7 °C. The variations of the cell temperature are about 6 °C for mass flow rate of 5 g s\(^{-1}\) and are gradually reduced to about 1.5 °C at 75 g s\(^{-1}\) as shown in Fig. 8. At 40 g s\(^{-1}\) the average variation of the cells surface temperature is about 1.7 °C and this is within the allowable limits. Overview of the surface temperature distribution of the cells in the battery pack is shown in Fig. 9, while the internal temperature distribution of the cells is shown in Fig. 10. Average difference of cells core temperature and surface temperature is about 2.6 °C. The maximum cell core temperature at 40 g s\(^{-1}\) of cooling air is about 39.2 °C. The cell in the center and front end of the battery pack is hotter than the cell on the side. This is due to lack of air flow to the end of battery pack and concentrating of heat at the center of the battery pack. Moreover, the highest temperature also occurs at the end of the cell body which is located in the slot on the holding plate and block the cooling air reached the cell surfaces. The simulation results confirm that the designed air cooling system is capable to maintain the battery temperature within the desired range.

Fig. 11 shows the correlation of the average Nusselt number for the current study. The average \(Nu\) along the cell surface is seen to increase with \(Re\) as expected. The average \(Nu\) for 30 g s\(^{-1}\) of cooling air is about 76.4 while for 75 g s\(^{-1}\) the \(Nu\) is about 169. Forced convection cooling requires parasitic power to overcome the flow resistance induced by narrow gaps between the cells in the battery pack. Fig. 12 shows the measured and simulated ideal fan power consumption for various flow rates. The measurement of the pressure drop in the battery pack is validated until the top limit of the tangential blower which is about 30 g s\(^{-1}\). At 30 g s\(^{-1}\), 0.53 W of fan power is needed to maintain the final temperature of the cells within the safety limit. Minor leakage occurring in the battery pack may have contributed to the slight deviation of experimental measurement of pressure drop and simulation results. The averaged relative error is about 5.03%.

Correlations of \(Nu\) with \(Re\) of the current study which determined via a least-mean-squares fit can be represented by Eq. (25) with \(R^2\)-square of 0.9985.

\[
Nu = 0.0374Re^{0.8014}
\]  

(25)
surface. The examined flows were in the Reynolds number range of $8.9 \times 10^4$ to $6.17 \times 10^5$. On the other hand, Sparrow and Geiger [50] derived the heat transfer coefficient for a circular disk facing a uniform air flow for a Reynolds number range from 5000 to 50,000 using the naphthalene sublimation technique. Ota and Kon [51] correlation is based on the heat transfer characteristic of reattached and redeveloped regions for longitudinal incompressible air flow along a blunt circular cylinder in a Reynolds number range from 24,900 to 53,600. The average Nusselt number for above mentioned studies are tabulated in Table 3. As shown in Fig. 13, Sparrow & Geiger, Ota & Kon correlations do not result in reliable predictions. Although, the examined Reynolds number range for Wiberg and Lior is higher than that for the current study, the results agreed well with the current study. The possible explanation could be because the flow of cooling air is through a circular disk which is similar to the current battery pack design with vent holes on the holding plate. On the other hand, other authors’ correlations are based on the direct impingement of cooling air on a circular body. From the above comparisons, it is found that the developed correlation is reasonable and would be applied to the following transient simulations of the battery pack.

Table 3

<table>
<thead>
<tr>
<th>Authors</th>
<th>$Re$</th>
<th>$C$</th>
<th>$e$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wiberg and Lior</td>
<td>89,000–61,700,000</td>
<td>0.070</td>
<td>0.734</td>
</tr>
<tr>
<td>Sparrow and Geiger</td>
<td>5000–500,000</td>
<td>0.927</td>
<td>0.5</td>
</tr>
<tr>
<td>Ota and Kon</td>
<td>24,900–53,600</td>
<td>0.109</td>
<td>0.701</td>
</tr>
</tbody>
</table>

3.4. Transient simulation and model validation

In this section, simplified transient simulations were performed to evaluate the temperature rise in the battery pack. In the numerical modeling, heat generation in the cell as a function of time of the cell obtained from the ARC measurement as shown in Fig. 4 was imported into the battery domains as transient heat sources. Heat conduction within the batteries is governed by the energy equation as in Eq. (12) and the heat transfer coefficient given by Eq. (25) was assigned to the battery surfaces as effective convective heat transfer boundaries. The simulations were performed for different charging rates and the average surface temperatures of the 24 pieces of batteries were plotted in Fig. 14. Additionally, an experimental testing of the battery pack was done to validate the developed mathematical model at various charging rates. Fig. 14 shows the comparison of experimental data and numerical prediction of the average rise of cell temperature in the battery pack at $m = 30 \text{ g s}^{-1}$. The figure shows good quantitative and qualitative agreement between experimental data and numerical prediction with average relative error of 13.1%, 13.7% and 13.6%, respectively for 1, 3 and 5 $I_t$-rates of constant current charging. Both experimental and numerical results showed a trend of increasing average cell surface temperature versus time. From the experiments, the variations of average cell temperature in the battery pack are about 1.6 °C, 2.6 °C and 3.4 °C for 1, 3 and 5 $I_t$-rates of constant current charging, respectively, confirming the findings from the simplified transient modeling that there is minor temperature difference among the cells. Although the internal temperature of the cell may exceed 40 °C during 3 $I_t$-rate of constant charging, the Li-ion cell can tolerate high temperature temporarily [54]. Sudden increase of the cell temperature occurred when the SOC of the cell reached 90%. Remaining charging time is less than 2 min. Hence, 30 g s$^{-1}$ of air flow rates and the current design of battery thermal management system is still capable of handling constant current charging till 3 $I_t$-rate. On the other hand, at 5 $I_t$-rate of charging, the maximum temperature rises of the cell is about 11.6 °C, which exceeds the top ideal operating temperature limit of Li-ion batteries. Therefore, a more powerful tangential blower or liquid cooling is recommended in this situation, especially for fast charging applications.

4. Conclusions

In this study, CFD analysis was utilized to analyze the air cooling of a battery pack comprising 38,120 cells. The simulation was able to predict the hot spots and cold spots within the battery pack. The simulation results demonstrate that an increasing of cooling air flow rate will result in the increase of heat transfer coefficient and pressure drop. A correlation of Nusselt numbers to Reynolds numbers was developed based on the steady state numerical simulations and compared with the correlations from the open literature. In general, the developed correlations show a similar trend with most of the correlations in the open literature. The developed correlation is then used to predict the transient thermal performance of the battery pack under different rate of charging. Finally, the numerical model was validated by a series of experiments done for active air cooling. The numerical results showed good agreement with the experimental results at various $I_t$-rates of constant current charging. For charging at 5 $I_t$-rate, a more powerful fan or liquid cooling was required to keep the cell temperature at optimum range. This method provides a simple way to estimate thermal performance of the battery pack thermal management system when the size of battery pack is large and full transient simulation is not viable.
Acknowledgement

This work was supported by a Grant No. EP/D009005/FRP(A) from Energy Market Authority Singapore and High Impact Research Grant No. HIR-D000006-16001 from University of Malaya.

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