On using splitter plates and flow guide-vanes for battery module cooling

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Abstract Thermal management of lithium-ion battery modules needs to be an integral part of the design process to guarantee that temperatures remain within a narrow optimum range. Also it is important to minimize uneven distribution of temperature throughout a battery module so as to enhance the battery life cycle, and, charge and discharge performances. This paper explores by simulation, the benefits of attaching thin surfaces extended into the near-wake of cylindrical lithium-ion cells, here termed integral wake splitters, and, of placing flow guide-vane in the vicinity of the near wake, regarding thermal management. When using the integral splitters it is found that the local Nusselt numbers in the very near wake of a single cylindrical cell are depressed and the temperature distribution within the cell was found to be reasonably constant. Similar results were found when the cells are in formation. Use of guide-vanes also show promise in maintaining constant temperature distributions throughout the module.

List of symbols

| Symbol | Description |
|--------|-------------|
| L      | Length of splitter plate |
| n      | Number of cells |
| m      | Cell mass |
| Mf     | Mass flow of air |
| Nu     | Nusselt number |
| p      | Pressure |
| Re     | Reynolds number |
| Sr     | Transverse pitch |
| t      | Time |
| T      | Temperature |
| Tf     | Average temperature |
| u      | Velocity field vector |
| U      | Velocity |
| U'     | Ratio velocity |
| V      | Volume |
| Wd     | Displacement of flow guide vanes downstream |
| Wg     | Gap between flow guide vanes |
| x, y, z| Cartesian coordinates |
| y      | Distance to wall |
| y+     | Dimensionless wall distance |

Greek symbols

| Symbol | Description |
|--------|-------------|
| α      | Angle of cell formation |
| μ      | Dynamic viscosity |
| ν      | Kinematic viscosity |
| ρ      | Density |
| τ      | Shear stress |

Subscripts

| Symbol | Description |
|--------|-------------|
| 1      | First cell |
| c      | Cell |
| f      | Fluid (air) |
| i      | Number, inlet |
| i, max| Last cell |
1 Introduction

Due to efficiency and environmental concerns, the importance of hybrid propulsion systems has been recognized for sustainable transportation [1–3]. Secondary lithium ion batteries (LIBs) are promising for applications in such systems because of their high energy density, high voltage, good cycle stability, and low self-discharge [4]. However, it is desirable to keep LIBs operating within a relatively narrow temperature range, say 20–40 °C to optimize the battery life cycle and enhance its safety, which is a relatively narrow range when compared to environmental temperatures [2, 5]. Overheating and an uneven cell temperature distribution, which commonly occurs in battery modules and packs, cause the degradation and failure of cells [6, 7]. It can be said that the thermal behavior of a battery module and its individual cells is of the utmost importance to the lifespan of a Li-ion battery and needs to be carefully addressed.

Cooling strategies have been widely researched using natural convection, forced convection (direct and indirect) both for air and water as well as innovative passive cooling using phase-change materials [8, 9]. However, it has been commented that air convection cooling methods (natural or forced) quite often are insufficient for effective heat dissipation from batteries under abuse conditions leading often to non-uniform temperature distributions within battery packs [9, 10]. Indirect liquid cooling of battery packs (both passive and active) can prove an efficient method for dissipation or addition of heat [11, 12]. However, it is desirable to keep the cooling fluid separate from the battery and for small battery packs cooling by liquid may not be possible. A review has been reported by Xu and He [13, 14] of forced air cooling methods and simulated heat dissipation performance of different airflow duct modes of a 55 Ah battery pack, while Wang et al. [15] have investigated different cell arrangements and forced air-cooling strategies by placing the cooling fan at different locations within the studied system.

The heat transfer characteristics of the separated boundary layer and recirculating wake downstream of a circular cylinder can be greatly influenced by the use of a splitter plate placed along the centre line of the near wake [16]. It has been shown for sub-critical Reynolds numbers (2700 ≤ Re ≤ 46,000) that a splitter plate appears to be responsible for the modification of the wake formation region without disrupting the usual von Karman vortex street [17]. Regarding heat transfer it was found [18] that longitudinally fined cross flow tube banks provided a high degree of heat transfer enhancement.

In an effort to enhance thermal management of a battery module the current work explores using numerical simulation, the use of thin surfaces extended into the near wake of cylindrical lithium-ion cells, here termed integral wake splitters, and of placing small flow guide-vanes also placed in the vicinity of the cylinder near-wakes. The numerical experiments are carried out in the Reynolds number range 102–103. Initially simulation of the flow and thermal characteristics in the vicinity of an isolated Li-ion cell, and, optimizing of the wake splitter plate length (L) to cell diameter (D) ratio, and, the gap between flow guide vanes (Wg) to distance of placement downstream of the cell centre-point (Wd) ratio was carried out (see Fig. 1). The plan view in the vicinity of the cylindrical battery cells is shown in Fig. 1. A three-dimensional view of a typical arrangement of battery cells, with splitter plates present, is also included, with a typical grid used for calculating conduction within the cells and splitter plates illustrated.

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**Fig. 1** Plane view of a splitter plates, and b flow guide-vanes, and, c three-dimensional view of a typical cell arrangement, with the grid used for conduction calculations included.
The study was then expanded to include examining the effect of using different cell arrangements on the thermal characteristics within a given module with the optimized splitter plates and flow guide-vanes in place.

2 Mathematical model

2.1 Governing equations

The fluid flow was considered as laminar, incompressible and with no body forces so the following equations were used for continuity and momentum transport equations,

\[ \nabla \cdot \mathbf{u} = 0 \]  
\[ \frac{\partial \mathbf{u}}{\partial t} + (\mathbf{u} \cdot \nabla) \mathbf{u} - \nabla \cdot \left[ \nu \left( \nabla \mathbf{u} + (\nabla \mathbf{u})^T \right) \right] = -\frac{1}{\rho} \nabla p \]

where, \( \nu \) is the kinematic viscosity, \( \rho \) is the density, \( \mathbf{u} \) is the velocity field and \( p \) is the pressure.

The heat equation used within the fluid computational domain, written in terms of absolute temperature \( T \), and ignoring viscous heating and pressure work was,

\[ C_p \frac{\partial T}{\partial t} + C_p \mathbf{u} \cdot \nabla T = \nabla \cdot \left( \frac{k}{\rho} \nabla T \right) \]

where, \( C_p \) is the specific heat capacity, and \( k \) is the thermal conductivity.

When the velocity is set to zero in Eq. (3), the equation governing pure conductive heat transfer is obtained for the solid computational domain, i.e.,

\[ C_p \frac{\partial T}{\partial t} = \nabla \cdot \left( \frac{k}{\rho} \nabla T \right) \]

Conjugate heat transfer was used between the solid domain and fluid domain.

2.2 Boundary conditions

The numerical experiments were carried out using a simple computational domain as illustrated in Fig. 2. Typically the length of the computational domain was set at 10\( D \) and the width was set at 4\( D \), with the cylinders placed close to the inlet to allow for the development of wakes.

The computational domain has a simple inlet and outlet arrangement, with the inlet flow considered as laminar and with various velocities ranging from 0.1 to 0.2 m/s. The temperature of the onset air was set at 293 K.

It has been stated that a cell can be considered as a homogeneous cylinder with an internal heat source since the detailed single cell structure has little effect on the temperature behavior of the battery module [1, 9, 19] and this approximation will be used here for each of the cylindrical cells, the diameter was set at 20 mm and the height 150 mm. Work has already been carried out using a lithium aqueous battery cell [20] to obtain realistic values for rises in temperature and thermal power generated during different discharge rates. In the following 1C, 2C and 3C are rates of discharge of a cell where 1C means that a cell fully discharges during 60 min, 2C that the cell fully discharges during 30 min and 3C that the cell fully discharges during 20 min, with the faster the discharge rate, the greater the increase in heat production. For the calculation of the thermal fields of sub-section “Thermal field results” later, the discharge rate is 3C. The results of Ref. [20] are summarized in Fig. 3 and Table 1 with the thermal power (\( P \)) calculated using,

\[ P = \frac{C_p m \Delta T}{t} \]

where \( C_p \) is the specific heat capacity, \( m \) is the cell mass, \( \Delta T \) is temperature difference, and \( t \) is time. The effective values of density, thermal conductivity and heat capacity.
needed for Eq. (5) are calculated using the suggestion of Chen et al. [21] where for example, the effective heat capacity can be calculated using

\[ \rho C_p = \sum_i \rho_i C_{p_i} V_i \]

Inside the battery cells, during the simulations, heat conduction was considered in all three directions, i.e. the radial, the azimuthal and the longitudinal directions as shown in Fig. 4.

![Fig. 3](Temperature profiles for different discharge rates)

**Table 1** Thermal power derived using Eq. (5)

| Discharge rate | \( \Delta T \) (K) | \( t \) (s) | \( P \) (W) |
|---------------|-------------------|------------|-----------|
| 1C            | 9.4               | 3600       | 0.1344    |
| 2C            | 18.8              | 1800       | 0.5347    |
| 3C            | 25.6              | 1200       | 1.0544    |

![Fig. 4](Directions of heat conduction inside a battery cell during calculations)

The grid used to discretize the computational domain had tetrahedral elements and was refined in the vicinity of the cylindrical elements and also in their near wakes. Grid independence was achieved when using \( 5 \times 10^5 \) nodes. Close to solid walls, very thin boundary layers can occur and should be captured by keeping the \( y^+ \) number defined by Eq. (7) at around a value of 1.0. Here this was achieved by fine-grid embedding, which refines the grid in the vicinity of solid surfaces.

\[ y^+ \equiv \frac{y \sqrt{\tau_w}}{\nu \rho} \]  

(7)

Here, \( \nu \) is the local kinematic viscosity, \( y \) the distance to the nearest wall (i.e., the thickness of the boundary layer grids), \( \tau_w \) the wall shear stress, and \( \rho \) the density of the fluid.

**2.3 Grid**

The grid used to discretize the computational domain had tetrahedral elements and was refined in the vicinity of the cylindrical elements and also in their near wakes. Grid independence was achieved when using \( 5 \times 10^5 \) nodes. Close to solid walls, very thin boundary layers can occur and should be captured by keeping the \( y^+ \) number defined by Eq. (7) at around a value of 1.0. Here this was achieved by fine-grid embedding, which refines the grid in the vicinity of solid surfaces.

**2.4 Validation of CFD code**

Before starting on the numerical experiments concerning battery modules, it was necessary to validate the CFD model used here using similar geometry, but for a different application [22]. Following the lead of Li et al. [23] a row of eight cells was set up in the simple computational domain shown in Fig. 2, and, different onset velocities ranging from 0.1 to 10 m/s were tested. This corresponded to Reynolds numbers of \((1130 \leq \text{Re} \leq 113,000)\) which covers the laminar and possible transition flow regimes of interest to the current work. The Reynolds number was defined for comparison with the experimental data [22] as,
Re = \frac{\rho U' D}{\mu} \quad \text{where} \quad U' = \frac{S_T}{S_T - D} U = 5U \quad (8)

where \( S_T \) is the transverse pitch between two cell centres.

Figure 5 shows the comparison of Nusselt number (Nu) versus Reynolds number (Re) obtained by the current calculations against experimental results [22] and previously published CFD results [23]. The Nusselt number was defined as,

\[ Nu = \frac{hD}{k} \quad (9) \]

where \( h \) is the heat transfer coefficient and \( k \) the heat conductivity of the air. The heat transfer coefficient \( h \) is defined using a log-mean temperature difference (LTMD) as shown in Eq. (10) [24],

\[ h = \frac{\dot{M}_f C_f (T_{f,0} - T_{f,i})}{n \pi D H \cdot T_{LTMD}} \quad \text{where} \]

\[ T_{LTMD} = \frac{(T_{f,max} - T_{f,o}) - (T_1 - T_{f,i})}{\ln \left( \frac{T_{f,max} - T_{f,o}}{T_1 - T_{f,i}} \right)} \quad (10) \]

\( \dot{M}_f \) is the mass flow of air, \( T_{f,o} \) is the average temperature of air at the outlet, \( T_{f,i} \) is the average temperature of air at the inlet (293 K in these studies), and \( T_1 \) and \( T_{f,max} \) are the average temperatures of the first and last cells in the row, respectively [1]. As can be seen from Fig. 4 the Nusselt number calculations performed with the current computer code agree reasonably well with those obtained by experiment [22] and calculations found in the literature [23] and in the region of \( \text{Re} = 10^3 \) the calculations were very close to the measurements.

Ref. [23] provides results of an experiment where a battery module with the arrangement shown in Fig. 6 is placed in an open jet wind tunnel and thermocouples are used for temperature measurement. As shown in Fig. 6 the battery module studied consisted of eight lithium ion cylindrical cells assembled in the configuration shown with on-set cooling air delivered as shown. The results of this experiment are used here to further validate the present CFD code.

Figure 7 shows calculations of the transient temperature obtained using the present CFD code compared with measurements reported by Ref. [23]. The temperature history is for the location T1 shown in Fig. 6, under 1C discharging, and, the results are for zero on-set velocity (natural cooling) and an on-set velocity of 5 m/s (forced cooling). The ambient air temperature was stable at 20.5 °C during the experiment and calculations.

As can be seen, the correct trend was obtained by the calculation method and the experimental and calculated results are in reasonable agreement. There was however a tendency to over-predict temperature during the first 35–40 min and under-predict temperature after that. However the errors were all less than 0.5 %, except initially. The comparatively large discrepancy initially may be due to experimental error [23].
3 Modelling results and discussion

The numerical experiments began with investigations of the velocity field and heat transfer characteristics of a single cell with an integral wake splitter attached or with flow guide vanes placed in the near wake.

3.1 Single-cell velocity field numerical experiments

The intention of using integral splitter plates is not primarily to help with the conduction of heat away from the cells using the splitter plate but to change the nature of the wake generally so as to enhance wake recovery both in the very near wake, the near wake and the far wake. The effect of using a splitter plate on the near wake flow pattern is demonstrated in Fig. 8. The onset velocity ($U_\infty$) was set at 0.2 m/s for this case, the diameter of the cell was $D = 20$ mm and the length of the splitter plate was $L = D$. Instead of a large region of very slow moving fluid in the near wake it can be seen that the presence of the splitter plate gives the flow an immediate downstream direction with very little re-circulation or separation present, and with very small boundary layers seen to form on the splitter plate itself.

Turning vanes are also used in this study, mainly to direct the mean flow over the downstream surface of the cylinder, and again to aid wake recovery, and hence help to cool cells farther downstream. The basic idea is shown in Fig. 9, where extensive turning vanes are employed to show the principle. As will be shown later the size of such turning vanes are rather excessive and not desirable in practice. However the principles of redirecting the flow are the same no matter what the size of the turning vanes. Here $U_\infty = 0.2$ m/s and $W_g/W_d$ (refer to Fig. 1) was set at 0.5.

3.2 Size and placement optimization

This section is concerned with determining an optimum length for the splitter plate ($L$), the effect of the gap size of the turning vanes ($W_g$), and also where to place the turning vanes to be most effective.

Many tests were carried out for different splitter plate lengths ranging from $L/D = 0.1$ to $L/D = 1.5$ with a sample of the results shown in Fig. 10. The optimum length
of splitter plate was thought of as when the wake recovers quickest, so providing an optimum mean flow to cool the next cell directly downstream. Figure 10 shows normalized mean flow velocity profiles across the wake at 2.5D downstream of the centre of the cylindrical cell. The spatial origin was set as the wake centre. It was found that the wake recovers most rapidly for \(L/D\) values around 0.5.

The wake recover can also be enhanced with the use of turning vanes placed appropriately in the vicinity of a cell near wake. The turning vanes need to be small so as not to disrupt the mean flow, but still remain effective and robust for manufacture and use. After testing it was decided to use the design shown in Fig. 1b with a size 0.1D for both \(x\)- and \(y\)-dimensions. Further numerical experiments were then conducted to optimize the values of \(W_g\) and \(W_d\) also shown in Fig. 1b.

The optimum placing of the turning vanes is to help with wake recovery as quickly as possible. A grid system was set up based on \(W_g\) and \(W_d\), and the simple criterion, ‘that wake recovery is enhanced’, if the wake centreline streamwise velocity increases was used as an indication of enhanced wake recovery. Care must also be taken that the turning vanes do not intrude and disrupt the recovering wake profile. This was indeed the case for the grid points (Fig. 11) \{0.75, 0.5\}, \{1.0, 0.5\}, \{1.25, 0.5\}, \{1.0, 1.0\}, \{1.25, 1.0\}. For these tests the turning vanes did increase the centerline streamwise velocity, but reduced other velocities across the wake profile, and, also created small undesirable recirculation within the wake flow structure. It was concluded therefore, on examining the rest of the results, that point \{0.75, 2.0\} was the most appropriate placement of the turning vanes.

### 3.3 Multiple-cell velocity field results

Following the preliminary results for the velocity field around single cells, some investigation was made of arrangements of cylinders. The example used here is shown in Fig. 12 for a formation of four cells. In keeping with the results for the single cell it was found that each cell without splitter plate or turning vanes had a relatively large region of recirculation in the near wake, with the velocities very small within this region and the far wake has noticeably small velocities.

However, with the inclusion of optimized splitter plates and turning vanes the flow patterns in both the near wakes and far wake are radically different as can be seen in Fig. 12. The far wake is seen to recover remarkably fast and...
there is a reasonable amount of flow over and between all four cells.

### 3.4 Thermal field results

Calculations of the thermal field are strongly dependent on the parameters thermo-physical properties of both the li-ion cells and the cooling air. The measured values of each are listed in Table 2. The values of density, thermal capacity and thermal conductivity for the cell were calculated using equations similar to Eq. (6).

The rate of heat release from each cell was estimated using Eq. 5 and set as a 1W source as typical of 3C discharge.

| Table 2 Parameters and thermo-physical properties used in this work |
|---------------------------------------------------------------|
| **Cell properties** |  | **Air properties** |
| Density (kg m$^{-3}$) | $\rho_c = 2130$ | Density (kg m$^{-3}$) | $\rho_f = 1.1614$ |
| Heat capacity (J/kg K$^{-1}$) | $C_c = 775$ | Heat capacity (J/kg K$^{-1}$) | $C_f = 1007$ |
| Heat conductivity (W m$^{-1}$ K$^{-1}$) | $k_c = 32.0$ | Heat conductivity (W m$^{-1}$ K$^{-1}$) | $k_f = 0.0263$ |
| Mass per cell (kg) | $m = 0.2$ | Dynamic viscosity (Pa s$^{-1}$) | $\mu_f = 1.846 \times 10^{-5}$ |

![Traverse 1](image1)

![Traverse 2](image2)

![Traverse 3](image3)

![Traverse 4](image4)

![Traverse 5](image5)

![Traverse 6](image6)

![Traverse 1](image7)

![Traverse 2](image8)

![Traverse 3](image9)

![Traverse 4](image10)

![Traverse 5](image11)

![Traverse 6](image12)

![Traverse 1](image13)

![Traverse 2](image14)

![Traverse 3](image15)

![Traverse 4](image16)

![Traverse 5](image17)

![Traverse 6](image18)
Calculations were made for a 4-cell formation and a 20-cell formation with temperature profiles reported for traverses across the computational domain with the 4-cell formation in place as summarized in Fig. 13. In each case the traverse was made through the appropriate centreline of the cells, and, the distance between streamwise placed cells was set at $3D$. The splitter plates had lengths of $D/2$, and the onset cooling air velocity was 0.1 m/s.

The corresponding temperature profiles for each traverse are shown in Fig. 14. Satisfactory results were found both for the overall range of temperature reached by each cell and for the almost constant temperature across each cell. The difference in temperature between the two streamwise cells was less than 0.1 %.

The cell arrangement for a larger battery module consisting of 20 cells also tested is shown in Fig. 15. Again temperature profiles across the computational domain were calculated to test for general evenness of temperature across the battery module and for temperature evenness across each individual cell. Selected profiles are reported here as described in Fig. 15. Again all profiles were calculated through the appropriate centerline of the cells. The splitter plates again had lengths of $D/2$, and the onset cooling air velocity was 0.1 m/s.

The corresponding temperature profiles for each traverse are shown in Fig. 16 with an interesting staircase profile obtained down the centre of the arrangement in the streamwise direction. The cells are seen to become a little hotter, but general could be considered as having the same temperature such is the small increase in temperature in adjacent cells. The temperature through the cells in the $z$-direction were also quite even and the discrepancy in temperature between front and back cells was small.

In summary it can be said that the calculations indicated very satisfactory results for general evenness of temperature across the module and for each individual cell.

4 Conclusions

The use of integral splitter plates and appropriately positioned turning vanes radically changes the flow patterns in the near wakes and far wake of a formation of cylindrical cells. This causes air to flow over the cells more uniformly and without relatively large regions of separation and recirculation, hence helping the dissipation of heat generated by the cells and reducing any uneven distribution of temperature across an individual cell as well as across a battery module.
As can be seen from the results for the thermal field, the temperatures of cells in formation remain both uniformly low due to the air cooling, and the temperature distributions across each cell and the battery module as a whole are very satisfactory.

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