**A Novel Approach for Active Cooling of a Battery: Air-Cooled Mini-Channel Heat Sink, Enhanced with Intermittent Metal Foam**

**M. Yang1, G. Mathew2, H. Nemati3, M. A. Moghimi2,\***

1School of Mechanical and Power Engineering, Nanjing Tech University, 30 Puzhu South Rd, Pukou, Nanjing 211816, China

2Department of Engineering, Staffordshire University, Stoke-On-Trent ST4 2DE, UK

3Department of Mechanics, Marvdasht Branch, Islamic Azad University, Marvdasht, Iran

\*Corresponding Author: Mohammad.Moghimi-Ardekani@staffs.ac.uk

**Abstract:**

Improving the performance and safety of electric vehicles (EVs) requires efficient thermal management of Li-ion batteries. Although a liquid cooling system with a mini-channel heat sink (MCHS) is an efficient and viable common approach, it makes the eventual battery pack complex, heavier and expensive. To address this problem, this study proposed the application of air as a coolant for an MCHS by adding intermittent porous zones inside the mini-channel. A numerical study was performed to understand the cooling performance of the mini-channel employed with intermittent porous zones. The results showed that the addition of intermittent porous zones in a mini-channel made the air efficient enough to maintain the battery temperature well below the operating limit. However, it was found that the position of porous zones inside the mini-channel significantly impacts the cooling performance. So, a novel approach was proposed for identifying the best locations of porous zones within a mini-channel, and its effectiveness is studied numerically. Results showed that the proposed method can give a very close to the optimum solution at the least expense.

*Keywords*: Hybrid electric vehicles, Lithium-ion battery pack, Air-cooled mini-channel, Porous metal foam, Thermal management.

**Nomenclature**

|  |  |
| --- | --- |
|  | Forchheimer’s constant |
|  | Specific heat capacity (J/kg.K) |
| d | Wall thickness between two channels (mm) |
| *F* | Faraday number (K/mol) |
| h | Height of each channel (mm) |
| *I* | Battery current (A) |
| *i* | Battery current per unit volume (A/m3) |
|  | Permeability (m2) |
|  | Thermal conductivity (W/m.K) |
|  | Effective thermal conductivity (W/m.K) |
|  | Fluid thermal conductivity (W/m.K) |
|  | Thermal conductivity of solid phase material (W/m.K) |
| L | Length of the channel (mm) |
|  | Pressure (Pa) |
|  | Rate of heat generation in the battery (W) |
|  | Rate of heat generation per unit volume of the battery (W/m3) |
|  | Battery internal electrical resistance per unit volume (Ω.m3) |
|  | Temperature (K) |
| *t* | Time (s) |
| *U* | Open-circuit voltage (V) |
| *V* | Cell voltage (V) |
|  | Velocity vector (m/s) |
| w | Width of each channel (mm) |
|  |  |
| **Greek characters and symbols** | |
|  | Operator |
|  | Entropy change (J/mol.K) |
|  | Density (kg/m3) |
|  | Porosity |
|  | Viscosity (kg/m.s) |
|  |  |
| **Subscripts** | |
| *b* | Battery |
|  | Effective |
|  | Fluid |
|  | Solid phase material |
|  |  |
| **Abbreviations** | |
| BTMS | Battery thermal management system |
| BTR | Battery thermal runaway |
| CFD | Computational fluid dynamics |
| EV | Electric vehicle |
| IC | Internal combustion |
| Li-ion | Lithium-ion |
| LIB | Lithium-ion batteries |
| MCHS | Mini-channel heat sink |
| PCM | Phase change materials |
| PPI | Pores per inch |
| SEI | Solid electrolyte interface |
| SOC | State of charge |

1. **Introduction**

Electric vehicles (EVs) are considered a sustainable alternative to traditional internal combustion (IC) engines [1] and a remedy for global warming in the transportation sector. Lithium-ion batteries (LIBs) have been widely used as the optimal storage for EVs because of their significant advantages, such as relatively low self-discharge rates, long cycle life, no memory effect, light weight, compactness, and high power and energy density [2, 3]. However, LIBs are prone to temperature variation in terms of performance, life, and safety [4-6]. It can operate reliably within a temperature range of 15 °C to 35 °C, with a maximum temperature deviation of less than 5 °C between the modules in the battery pack [4]. At high temperatures, lithium plating significantly reduces the battery's performance when it is being used at high rates of charging/discharging [7, 8]. According to Bandhauer et al. [9], the battery performance significantly drops at temperatures above 50 °C, and the cells are damaged irreversibly as the temperature exceeds 80 °C [10]. At higher temperatures (>120 °C), the solid electrolyte interface (SEI) may undergo decomposition, resulting in an internal short-circuit and battery thermal runaway (BTR) [11]. Eventually, the battery may fail, resulting in the release of toxic gases, fire, jet flames, and explosions. As the battery system costs about three-quarters of the total price of an EV power train, it is crucial to manage the battery thermally and consequently improve battery performance and its lifespan [12, 13].

Due to the exothermic nature of the electrochemical reaction during charging/discharging, the battery temperature rise is inevitable [14, 15]. However, it is possible to control the temperature of the battery within the operating range by utilizing an efficient battery thermal management system (BTMS). A common method in battery thermal management systems (BTMS) is using external devices such as fans, blowers, pumps, etc. to enhance the convective heat transfer in active cooling-based BTMSs (Air cooling and liquid cooling).

In commercial EVs, liquid cooling with mini-channel cold plates has been widely studied [16-27]. A mini-channel cold plate is a metal heat exchanger consisting of several small channels carrying a liquid coolant to exchange heat with a hot source [15]. Usually, low-viscosity fluids such as water and glycol are used as a coolant to not only transfer heat but also impose less pressure drop (less power consumption) in the system [28]. As the metallic plate of the cooling channel is in direct contact with the batteries, the MCHS offers a high surface area to volume ratio which significantly improves the heat absorption rate and temperature uniformity [29].

Despite the high heat transfer rate of liquids, the major drawback of a liquid active cooling system is its heavy weight, which is mainly due to the use of relatively high-density liquids such as water and the need for an additional cooling package to cool down liquid inside the channels [30]. Additionally, a closed loop is required to circulate the liquid between the min-channels and the cooling package to avoid leakage issues, making the system more complex [31]. On the other hand, air-based active cooling with cold plates can overcome the proposed shortcomings of liquid cooling cold plates. Furthermore, BTMS with air circuits is broadly considered in commercial applications due to its low cost, lightweight, simple structure, and lower parasitic power consumption rates [29].

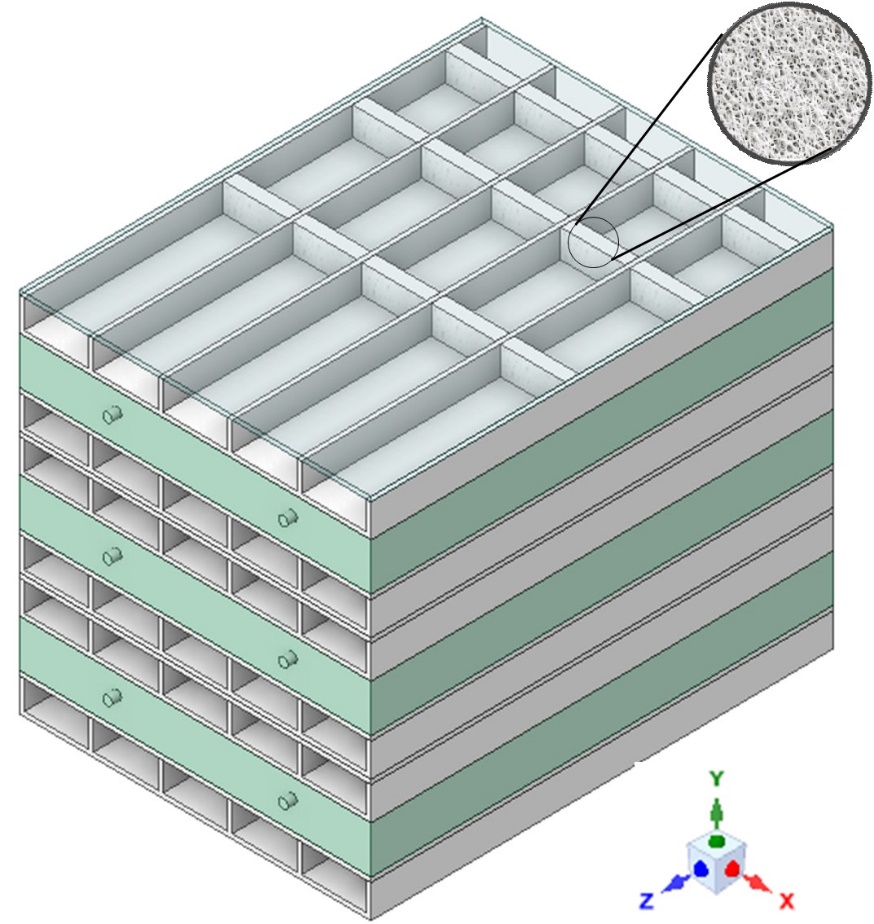
The main flaw of the air-cooling method is the low heat transfer of air which makes it suitable only for low-density batteries [31]. Although utilizing a mini-channel can overcome this weakness to some extent, embedding porous media in the mini-channel can enhance the heat transfer of a battery by improving the contact area with the cooling medium. Open-cell metal foams are a type of porous media with exciting properties such as high heat transfer area per volume, intensified fluid mixing, energy absorption, temperature tolerance, lightweight, low density, high mechanical strength, and high thermal conductivity [32]. A limited number of studies have used porous media to enhance the heat transfer rates in the BTMS. Recently, Heyhat et al. [33] investigated the thermal performance of a hybrid BTMS that couples the PCM with the metal foam, fins, and nanoparticles and found that the metal foam-PCM composition was the most efficient among them, and it could reduce the mean temperature by about 4-6 K than in the case with a pure PCM. Similarly, Liu et al. [34] developed a hybrid BTMS that couples the PCM-copper foam composition with helical liquid channels and reported a decrease of 30 K temperature drop than the natural convection case. Mohammadian et al. [35] studied the thermal behavior of a battery using an air-cooling method with metal foams embedded in the cooling channel. They compared the cases of cooling channels without the porous media with 30%, 70%, and 100% porous media within the channels. The results showed a significant decrease in the maximum temperature and the standard deviation of the temperature inside the battery using the porous media. Giuliano et al. [36] designed an air-cooled thermal management system with metal-foam-based heat exchanger plates to cool lithium-titanate batteries. They compared the obtained results with a conventional liquid cooling system and found that the proposed design was more efficient as it requires less parasitic power for the operation. In 2020, Bazkhane and Zahmatkesh [37] performed a sensitivity analysis of hydrodynamics and heat transfer of a nanofluid in a microchannel heat sink embedded with vertical/horizontal porous substrates. They used a 3-D solid-fluid conjugate model coupled with the two-phase mixture model for the nanofluids and Darcy–Brinkman–Forchheimer model for the porous substrate. By using the Taguchi method and analysis of variance (ANOVA), they found that adding vertical/horizontal porous substrates reduced the temperature, overall thermal resistance, and the required pumping power simultaneously.

The main challenge in adding porous metal to an MCHS is its high cost and weight. Even though the porous metals are light in weight, a channel fully filled with the porous metals makes the system heavier. Also, the addition of porous metals causes a significant pressure drop within the channels, leading to the additional power requirement [38]. So, it is proposed to use the intermittent porous zones within the channels to attain the benefits of enhancing heat transfer and reducing the weight and pressure drops of the cooling system. In this context, this study proposes mini-channel cold plates with intermittent porous zones within the channels for the thermal management of Li-ion batteries. The major challenge of this idea is to locate a proper place for the porous media. The main aim of this study is to propose a fast and straightforward method for identifying the proper locations of the porous zones within the cooling channel to reach the same cooling rate obtained for a fully porous channel, with a weight and pressure drop closer to the empty channel.

1. **Model Description**
   1. **Physical Problem**

The schematic of the batteries and MCHSs is shown in Fig. 1a. Each prismatic cell is sandwiched by two aluminum MCHSs with base plate thickness of 0.1 m. The lightweight and high thermal conductivity of aluminum make it a suitable material for the MCHS used for battery thermal management [39]. The geometric detail of the mini-channel is shown in Fig. 1b. The height of each channel is h = 3 mm, the width is w = 7 mm, the length is L = 50 mm, and the wall's thickness between the two channels is d = 0.1 mm. Due to the symmetry of channels, only half of one channel, as displayed in Fig. 1b, is considered for all the simulations to reduce the computational time and effort. This geometry will be used later for optimization as well as battery thermal management. In this study, it is considered to use sintered porous metal foam, inside each channel to increase the heat transfer and, consequently, decrease the maximum battery temperature. To reduce the cost and also the weight of used metal foam, only a limited volume of the mini-channel is to be filled with metal foam. Fig. 1b shows a schematic arrangement of metal foams inside the mini-channel. It is clear that the minimum length of each foam is limited by manufacturing restrictions. However, the main question is locating the proper position for each metal foam piece. To specify the possible positions for each metal foam, the channel is divided into 20 equal zones along its length to incorporate the porous metals into the mini-channel. The thickness of each zone is 2.5 mm, which will be considered as the thickness of the porous metals inside the channel. To focus on the heat sink design, battery has not been simulated in this stage and the generated heat inside the batteries is considered as heat flux boundary condition on the base of the mini-channel.

According to the experiments performed by Garrity et al. [38], which are used for validating the present model, aluminum foams with 40 PPI (Pores Per Inch) and with a porosity of 0.918 are used inside the mini-channel. Due to their large specific surface areas and high thermal conductivities, aluminum foams can transfer a large amount of heat from the battery, which makes them a perfect material for thermal management [38]. Air with a steady flow and constant properties is used as the coolant for the MCHSs. The detailed properties and parameters of the materials used in this study are listed in Table 1.



(a)



(b)

Fig. 1. a) Schematic of the batteries and MCHSs, b) Geometry of a mini-channel in MCHS and the schematic arrangement of metal foams inside it

Table 1: Thermo-physical properties of the materials

|  |  |  |
| --- | --- | --- |
| **Thermo-physical properties** | **Aluminum** | **Air** |
| Density, (kg/m3) | 2719 | 0.995 |
| Specific heat capacity, (J/kg.K) | 871 | 1009 |
| Thermal conductivity, (W/m.K) | 202.4 | 0.03 |
| Viscosity, (kg/m.s) | --- | 2.08E-05 |

* 1. **Governing Equations**

The energy conservation equation of the mini-channel cold plate is given as [26]:

|  |  |
| --- | --- |
|  | (1) |

The governing equations for the free channel are as follows [26]:

Continuity equation:

|  |  |
| --- | --- |
|  | (2) |

Momentum equation:

|  |  |
| --- | --- |
|  | (3) |

Energy equation:

|  |  |
| --- | --- |
|  | (4) |

where, and are fluid density and specific heat, respectively. and are fluid viscosity and thermal conductivity, respectively.

For the porous zone, the Forchheimer-Brinkman-Darcy model is employed [40]. It is assumed that the metallic foam used in this study is isotropic in nature. So, the governing equations will be:

Continuity equation:

|  |  |
| --- | --- |
|  | (5) |

Momentum equation:

|  |  |
| --- | --- |
|  | (6) |

where is the permeability and is Forchheimer's constant. Based on the experiment [38] used for the validation, and are 6.98e-10 m2 and 0.04,respectively.

Energy equation in the porous zone:

|  |  |
| --- | --- |
|  | (7) |

where and are the effective thermal conductivity and the heat capacity of the porous media, respectively. , , and are density, specific heat, and conductivity of solid phase material.

* 1. **Boundary Conditions**

The inlet flow velocity of air was taken as 1.5 m/s. It is assumed to have a uniform and constant velocity and temperature for the coolant at the flow inlet. Due to the symmetry, only half of one cooling channel, shown in Fig. 1b, was modeled. Based on the model, the top wall of the channel was considered adiabatic, vertical walls were considered symmetric, and the generated heat was dissipated into the air via the bottom of the mini-channel. Finally, at the outlet of the cooling channel, the gauge pressure was set to 0 Pa.

* 1. **Numerical Method**

The governing equations were numerically solved based on the finite volume method by using a commercial CFD solver, Ansys Fluent. A steady-state analysis with a laminar flow model was adopted to simulate the design model. The pressure-velocity coupling was done using a SIMPLE (Semi-Implicit Method for Pressure Linked Equations) scheme, and a second-order upwind method was applied for the space discretization of momentum and energy equations. An under-relaxation factor of 0.3, 0.7, and 1 were applied to the pressure, momentum, and energy equations, respectively, for a better convergence solution. Also, convergence criteria of 10-6 and 10-8 were applied for the flow and energy equations.

The hexahedral mesh was used to discretize the domain. The results of the mesh independence study for fully porous and non-porous channels are presented in Table 2, respectively. As clear in this table, the maximum temperature decreased only by 0.09 % and 0.01% for the fully porous channel and the non-porous one, respectively, as the number of mesh elements increased from 2.6E+5 to 4.7 E+5. This study shows that 2.6E+5 elements are generous enough to conduct the CFD simulation of this study.

Table 2: Grid study for two limiting cases, i.e. fully porous and non-porous channels

|  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- |
|  | **Max. Temp. in channel** | | | | | |
| Number of elements (105) | 1.0 | 1.7 | 2.2 | 2.6 | 3.4 | 4.7 |
| fully porous channel | 315.74 | 315.53 | 315.44 | 315.35 | 315.22 | 315.06 |
| Non-porous channel | 339.9 | 339.88 | 339.86 | 339.85 | 339.84 | 339.81 |

* 1. **Model Validation**

Before proceeding with the detailed simulations, it is necessary to validate the reliability of the present simulation model. Garrity et al. [38] conducted an experimental study to find the thermal performance of aluminum foams in a channel. The foam sample was 15.24 × 15.24 cm with a height of 2.54 cm. The lower wall of the foam sample in the channel was heated by an electric heater and the upper wall temperature was measured with an uncertainty of ±0.5 °C. They also located seven pressure taps, axially along the foam sample to measure the pressure variation along the channel. The duct was simulated and results were compared against experimental data. As shown in Fig. 2, at the different air velocities the simulated pressure drop perfectly agrees with the experimental results while the obtained wall temperature agrees acceptably with experimental data. The figure shows that at low velocities such as at 1.5 m/s, the maximum deviation between the two results is nearly 5%, and for velocities over 2.8 m/s, both are almost identical. This study shows the reliability of the simulated model for this investigation.

Fig. 2. Comparison of simulation and experimental results [38].

1. **Results and Discussion**
   1. **Impact of porous media and its positions on the temperature of cooling channels**

The properties such as high thermal conductivity and high heat transfer area per unit volume of open-cell metal foam can improve the heat transfer rate of the air flowing through the cooling channel. However, it is essential to know whether this addition of porous metals to the cooling channel is efficient enough to maintain the temperature of the batteries within the safe operating range. To understand the effectiveness of metal foam in a mini-channel, simulations were performed for the non-porous and fully porous channels. Dimensions of the channel were as shown in Fig. 1b and the inlet flow velocity of air was taken as 1.5 m/s. The used metal foam for validation (aluminum foams with 40 PPI and with a porosity of 0.918) was utilized in simulations. For this configuration, *Re*=301.4. Consequently, laminar flow regime could be assumed inside the channel.

From Table 2, it can be seen that the maximum temperature obtained for an empty channel was 339.85 K, which exceeds the maximum operating temperature limit (333 K) of the Li-ion batteries [41]. Therefore, it is clear that the simple channel cooled with air is inefficient for battery thermal management. However, for a fully porous channel, the maximum temperature was reduced to 315.35 K (Table 2), which is well below the maximum operating limit of the battery. So, from the above results, it is clear that the addition of porous metals can make the air an efficient coolant for the MCHS.

It is noteworthy that, the fully porous channels not only make the cooling system heavier and more expensive but also significantly increase pressure drop (from 4.3 Pa for the non-porous channel to 268.14 Pa for the fully porous channel, according to the conducted simulation) and consequently requires additional power for the operation. So, to overcome these challenges with the fully porous channel and make the air an efficient coolant for the MCHS, it is proposed to use the intermittent porous zones within the cooling channel. However, different arrangements and configurations of metal foam within the channel will impact the performance of intermittent porous zones (e.g. temperature distribution and uniformity of the batteries).

In that regard, a simulation study was conducted with 10% of the channel filled with aluminum foams to understand the importance of porous zone positions within the channel. To simulate, the channel was divided into twenty equal zones. So, for a 10% volume fraction of the porous zone, porous media (aluminum foams) can occupy two zones. It was found that 190 different arrangements are available to fill 10% of the cooling channel with aluminum foams. For higher volume fractions, there will be many more possibilities to meet. Therefore, it is crucial to have a robust and straightforward method to find the best arrangement. As the initial attempt in introducing the robust approach and to show the importance of metal foams arrangement, all 190 cases for 10% porosity were simulated and sorted based on the maximum recorded temperature. The results of this study are displayed in Fig. 3. As shown, with the same amount of metal foam inside the channel (10% metal foam inside the channel), the maximum temperature obtained within the channel varied from 317.45 K to 332.05 K. This significant difference in the temperature confirms the impact of metal foam arrangements within the cooling channel, as discussed earlier. To explain the reason for this significant difference in the temperature, one should note that there are two main factors contributing to the heat transfer enhancement in the porous area inside the mini-channel. The first factor is the considerable increase in heat transfer area and the second one which is mostly ignored is the considerable increase in thermal conductivity. The second factor is very important in a mini-channel with an intermittent porous zone.

Fig. 3. Maximum temperature corresponding to the different arrangements of a channel filled with 10% porous zones. The labeled points (1, 146, and 187) are selected arrangements that will be discussed later. For the sake of comparison, in this figure, the maximum temperature for non-porous and fully-porous cases is also presented.

To explain more, the temperature contour plot at several cross-sections along the mini-channel is shown in Fig. 4.a. Each cross-section belongs to one-half of the channel area. Critical examination of the contours shows that due to the low thermal conductivity of air, the fresh air in the core remains intact and cold. So, only the hot air flows in the vicinity of the channel wall. It is worth noting that the channel thicknesses were not shown in this figure. Also, the temperature variation along the centerline of the mini-channel is shown in Fig. 5. This figure shows that from the beginning of the channel up to its end, for the non-porous mini-channel, the temperature in the center of the channel does not change significantly. However, the presence of a porous zone with high thermal conductivity can change this stratified pattern and makes the temperature, uniform. So, the growth of the thermal boundary layer is intercepted and heat transfer increases. To compare, contour plots of three more cases are presented in Fig. 4. These cases were marked up in Fig. 3. For case no. 187, two consecutive porous zones are located at the beginning of the channel, while for case no. 146, consecutive porous zones are placed at the end of the channel. For case no. 1 which is the optimum case, zone 11 and zone 19 are filled with metal foam as shown in Fig. 4.

For case 187 (porous zones at the beginning of channels), as displayed in Fig. 4.b, there is a sharp change in the heat transfer from the solid wall into the fluid, because the porous zones increase the heat transfer area. However, the increase in the effective thermal conductivity is not beneficial, since the boundary layer thickness at the beginning of the channel is very small and the fluid is approximately at a uniform temperature close to ambient. For this reason, only one of those mentioned factors is effective and adding porous media is not associated with much superiority. Fig. 5 also shows that in this case, after a sharp change at the beginning of the channel, the temperature along the centerline remains approximately constant.

For case no. 146 (porous zones at the end of channels), as displayed in Fig. 4.c, the condition is slightly better. Because at the end of the channel, the thermal boundary layer is thickened. So, adding porous zones at the end of the channel not only increases the heat transfer area but also makes the average temperature uniform and colder. So, both factors are effective. However, porous zones at the end of the channel cannot affect heat transfer in the previous zones. So, as can be seen in Fig. 5, the centerline temperature for this case is much similar to the non-porous channel at the beginning of the channel.

|  |  |  |
| --- | --- | --- |
|  | |  |
| (a) |
|  | |  |
| (b) | | |
|  | |  |
| (c) | | |
| 000000000010000000103D.png |  | |
| (d) | | |

Fig. 4. Temperature contour:   
(a) at cross-sections along the mini-channel (non-porous channel);  
(b) at symmetry plane for case no. 187 (porous zones are at the beginning of the channel as highlighted);  
(c) at symmetry plane for case no. 146 (porous zones are at the end of the channel as highlighted);  
(d) at symmetry plane for case no. 1 (optimum porous zone arrangement as highlighted)

For the optimum case (case 1), the contour plot is basically different. Based on Fig. 4.d, the thermal boundary layer grows continuously up to the middle of the channel. The boundary layer thickness is large enough to cause a considerable reduction in heat transfer. At this point, adding the porous zone triggers both factors, and heat transfer increases considerably. Moreover, the thermal boundary layer is interrupted, and it starts again from the middle of the channel. So, the thermal boundary layer is thinner in the second half of the channel in comparison with other cases. In zone 19 (penultimate zone), the porous zone increases the heat transfer area and consequently the cooling effect. Based on Fig. 5, for this case, the temperature at the centerline is something between the non-porous and fully porous channels.

Fig. 5. Temperature variation along the centerline of the mini-channel for different cases

It is noteworthy that besides these two factors, conjugate heat transfer (conduction in the solid wall along the channel length) can affect the heat transfer and makes it difficult to explicitly predict the proper location for porous zones. In micro- or mini-channels, the effect of axial conduction cannot be ignored easily [42].

* 1. **Method to find the best locations of porous zones within a mini-channel**

To identify the best arrangement of porous zones within the channel, especially in a design with a higher number of possibilities (e.g. having more zones and or higher volume fractions of porous zones). Indeed, the simulations of all the possible arrangements are not practical as it is time-consuming. So, it is essential to find a novel approach that is fast and effective in identifying the best arrangement of porous zones within the channel, which can provide either an optimum or close to the optimum value for the temperature.

In this context, this paper proposed a new method to identify the proper locations of porous zones within the cooling channel. The main steps involved in this method are as follows:

**Initial Guess:**

1. As an initial guess, allocate all the porous zones at the beginning of the mini-channel and simulate to find the maximum temperature corresponding to each zone as well as the channel maximum temperature.

**Arrangement Modification Procedure:**

1. Rearrange the zones by moving the porous media from the coldest porous zone to the hottest non-porous zone (as captured in the previous step). Simulate for the new arrangement to find the maximum temperature of each zone as well as the channel maximum temperature.
2. Repeat the above step if the channel maximum temperature is decreasing. However:

* If the arrangement zones got repeated: stop the process and report the optimum solution.
* If the channel maximum temperature increases: record the previous channel maximum temperatureas *Tref*.Follow the **Correction Procedure**.
* If the hottest zone is a porous zone, follow the **Tuning Procedure**.

**Correction Procedure:**

1. If the channel maximum temperature got increased (which could be due to conjugated heat transfer attributions), put the porous zone in the middle of the captured positions in the last two steps.
2. Simulate the new arrangement and find the maximum temperature of each zone as well as the channel maximum temperature.
3. Do the above two steps until the channel maximum temperature decreases below the *Tref*.
4. Once the temperature gets reduced, continue the **Arrangement Modification Procedure**.

**Tuning Procedure:**

1. If the hottest zone is porous, start the tuning process by gradually moving the coldest porous zone towards the hottest porous zone.
2. Continue the above step until the maximum temperature of the hottest zone increases.

For further clarification on the proposed approach, its flowchart is presented in Fig. 6.



Fig. 6. The flowchart for finding the optimum locations of porous zones within a mini-channel

As a practical example, let’s consider the application of this method in a mini-channel filled with 10% porous zones. Table 3 shows the steps involved in finding the best locations of the porous zones. It is to be noted that in the table 0s indicate the non-porous zones, and 1s indicate the porous zones. The coldest porous zone is highlighted in blue and the hottest non-porous zone in yellow.

From Table 3, the arrangement modification procedure is continued up to the 3rd try and the maximum temperature of the channel decreases in each of these tries with the movement of the porous zones. However, after the 3rd try, when the coldest porous zone (zone 7) is moved to the hottest non-porous zone (zone 16), the maximum temperature increases from 319.36 K to 322.04 K. This shows the excess movement of the porous zone and therefore it needs to be corrected by bringing back towards the previous position. While analyzing the temperature distribution of the channel corresponding to the 3rd try, it is clear that to reduce the maximum temperature the porous zone must be placed somewhere between zone 7 and 16. Therefore as a corrective step, the porous zone is moved backward, to zone 12 which is halfway between zones 7 and 16 (closer to zone 16). This movement has reduced the maximum temperature to 317.79 K, which is even lower than the maximum temperature obtained on the 3rd try. Therefore, the arrangement modification procedure can be continued. However, the temperature increases again when the coldest porous zone (zone 12) moved to the hottest non-porous zone (zone 5). So, in the correction procedure, the porous zone needs to be moved halfway between the current zone and zone 12 as the 7th iteration, i.e. zone 8. This reduces the maximum temperature to 318.99 K, which shows that the porous zone movement is on the right way. However, the maximum temperature at this stage is still higher than the value corresponding to the 5th try; so, the correction process is continued by moving forward the porous zone towards zone 12. By the end of the correction procedure, the same arrangement of try 5 is repeated; therefore, the whole process can be stopped as no more possible movement is available for the porous zone using the proposed approach. So, the arrangement corresponding to the 10th try is obtained as the best arrangement with a maximum temperature of 317.79 K.

Table 4 lists the first 12 arrangements out of 190 cases sorted ascendingly based on the maximum temperature of the channel, captured in section 3.1. As listed, the maximum deviation of results from the optimum solution (the first solution in the list) is about 0.5 K. By comparing Table 4 and Table 3, one can see that the arrangement proposed by the approach discussed in section 3.2 (zones 12th and 20th as porous media) is very close to the optimum arrangement (zones 11th and 19th as porous ) reported in Table 4 based on simulating 190 cases.

Indeed the optimum locations of porous zones within the channel from the proposed approach are captured in 10 tries while without using this approach, it requires simulating all 190 cases. It means 94% saving in computational cost and effort (simulating 11 cases against 190 cases), with less than 0.4K deviation from the optimum goal. In summary, the proposed method can be considered an efficient and effective approach for identifying the best locations of the porous zones within the cooling channel.

|  |  |
| --- | --- |
| Table 3: Steps involved in identifying the best locations of porous zones for the mini-channel filled with 10% porous media |  |

Table 4: The first 12 arrangements (out of 190 cases sorted ascendingly based on the maximum temperature of the channel, captured in section 3.1) with 10% porous media.



* 1. **Further discussions**

This section discusses the effects of using higher percentage of intermittent porous zones in a mini channel. Fig. 7 shows the simulation study results on the mini-channel filled with different amounts of metal foams. The optimum locations of porous zones for various porous volume fractions are also presented in Appendix I.

Since the location of porous zones plays a crucial role in the cooling performance, all the results shown in the figure are obtained with the arrangement found using the proposed method in section 3.2. The figure shows that a fully porous channel reduces the maximum temperature by 25 K compared to an empty channel. However, the fully porous channel causes a significant pressure drop of 268 Pa, which in turn increases the parasitic power consumption of the cooling system. On the other hand, it can be seen that a maximum temperature comparable to a fully porous channel can be attained by using the intermittent porous zones with a much lower pressure drop. This shows the effectiveness of using intermittent porous zones in cooling the batteries. From the graph, the maximum temperature remains almost constant with the increase of porous zones from 10% to 80%. Thus, the mini-channel filled with 10-20% porous zones is the best solution for cooling the batteries as it is 80-90% lighter and cheaper than a fully porous channel and can reach a maximum temperature comparable to a fully porous channel while maintaining a pressure drop, comparable to a non-porous channel. For this reason in continuation of this investigation, the 10% porous channel is chosen for the battery thermal management at the cell level.

Fig. 7. Simulation study of intermittent porous zones employed in the mini-channel

1. **Battery Thermal Management at Cell Level**

To study the effect of the proposed arrangement of porous media in cooling a battery at the cell level, a LiPol pouch battery cell (model number: LPHDA034050) was added underneath the designed channel. Dimensions of the considered cell are 10 x 34 x 50 mm (LiPol Battery Company). Therefore, it was assumed that each battery cell is cooled by two heat sinks (one on each side see Fig. 8a). Each heat sink consists of five mini-channels and half the thickness of the battery cell was considered in the simulation. Dimensions of each channel were shown in Fig. 1b. It is the same geometry used for optimization in Sec. 3.2. To mimic the heat generation inside the battery cell, the following volumetric heat generation was imposed on the modeling [35]. The proposed equation is based on the thermodynamic energy balance inside the battery cell which takes into account heat generated by chemical reactions, the heat produced by resistive dissipation, the reversible entropic heat, and mixing heat due to the relaxation of concentration gradients in the cell. The simplified version of the equation is:

|  |  |
| --- | --- |
|  | (8) |

where is the rate of heat generation and is the electric current in the unit cell. and are unit cell open-circuit voltage and cell voltage, respectively. The first term on the right-hand side of Eq. (8) is the over-potential irreversible heat generation due to Ohmic losses in the cell, charge-transfer over-potentials at the interface, and mass transfer limitations. The second term on the right-hand side of the equation is the reversible entropic heat due to electrochemical reactions. Only one electrochemical reaction was assumed to occur in the battery in normal operation. Mixing and phase change effects are neglected in Eq. (8). By considering Faraday number (=96485 K/mol) and rewriting Eq. (8) per unit volume of the cell [43, 44]:

|  |  |
| --- | --- |
|  | (9) |

where , and are heat generation, internal resistance and discharge current all per unit volume of the cell.

Based on the experimental measurement [45], and can be presented as follows,:

|  |  |
| --- | --- |
|  | (10) |

for and K.

|  |  |
| --- | --- |
|  | (11) |

At the temperature between 293 K and 313 K, the entropy change is almost independent of temperature [45] while, depends on both state of charge (SOC) and temperature. SOC is also defined as:

|  |  |
| --- | --- |
|  | (12) |

=1.8 Ah is the electric capacity of the battery.

The governing equations are as presented in sub-section 2.2, but the energy equations (Eq. (1), Eq. (4) and Eq. (7)) are modified to:

Energy equation in the porous zone:

|  |  |
| --- | --- |
|  | (13) |

Energy equation in the fluid zone:

|  |  |
| --- | --- |
|  | (14) |

Energy equation in the heat sink solid zone:

|  |  |
| --- | --- |
|  | (15) |

Energy equation in the battery:

|  |  |
| --- | --- |
|  | (16) |

Boundary conditions and the simulated geometry are shown in Fig. 8b. Equations were solved numerically with the same strategy described in sub-section 2.4.

|  |  |
| --- | --- |
|  |  |
| (a) | (b) |

Fig. 8. a) The schematic of one battery cell with MCHSs, b) The simulated geometry and boundary conditions

For this study, the simulation was performed with an initial temperature of 300 K, a discharge rate of 3C and 20% cut-off state of charge. For a battery, C-rate is defined as the charge / discharge current divided by the nominally rated battery capacity. In other words, it takes 1/C hr for a battery to get fully charged or discharged. The effective total heat capacity of the battery under the experiment was 1.83 MJ/(m3 K) [45] and the thermal conductivities in the thickness direction and normal to the thickness direction were =1.09 W/(m K) and = 3.82 W/(m K), respectively [45]. Please note that three different cases were considered for the thermal management of the battery at the cell level:

- Case 1: The non-porous channel.

- Case 2: The 10% porous channel (located at zones No. 12 and 20 per Table 3)

- Case 3: The fully porous channel.

For these three cases, the battery average temperature and maximum recorded temperature versus SOC are presented in Fig. 9. According to these figures, the temperature rise for the channel without metal foam is about 24.6 K. While for the enhanced channel with metal foam, the temperature rise is controlled and about 12.1 K. The interesting result is that by using only 10% metal foam (but in proper positions) the average temperature is only 3.4 K more than a channel fully filled with metal foam.

(a)

(b)

Fig. 9. The battery average temperature for 3 proposed cases. Case 1: non-porous channel, Case 2: 10% porous channel (located at zones No. 12 and 20) and Case 3: fully porous channel.

Another interesting result is revealed in Fig. 9b. The maximum temperature for case 2 is very close to case 3. Because the optimization scenario in this study is based on reducing the maximum temperature. So, the maximum temperature in case 2 is very close to case 3. It shows the importance of selecting the correct position for metal foam in dealing with channels with intermittent metal foam.

The weight of a pair of MCHSs is approximately, 2.4 gr and the weights of 10% porous and fully porous MCHSs are 2.39 and 2.81. This shows that the weight of 10% porous MCHS is only 2% more than non-porous MCHS while the average temperature rise is less 9.1 K with a reasonable pressure drop. Table 5 shows a brief comparison between non-porous, 10% porous and fully porous MCHS. The number in parenthesis shows the ratio of the item value to the value of non-porous MCHS. For example by using 10% porous MCHS, the average temperature is 0.64 of average temperature for the non-porous MCHS.

Table 5: Comparison between non-porous, 10% porous and fully porous MCHS

|  |  |  |  |  |
| --- | --- | --- | --- | --- |
| **Item** | **Weight**  **(gr)** | **Max. temp. rise**  **(K)** | **Ave. temp. rise**  **(K)** | **Pressure drop**  **(Pa)** |
| Non-porous MCHS | 2.34 | 27.5 | 24.6 | 4.3 |
| 10% porous MCHS | 2.39 (1.02) | 16.7(0.61) | 15.5 (0.63) | 30.5 (7.1) |
| Fully porous MCHS | 2.81 (1.2) | 15.8 (0.57) | 12.1 (0.49) | 268.14 (62.4) |
| Note: number in parenthesis indicates the ratio to non-porous MCHS value | | | | |

1. **Conclusion**

Thermal management of Li-ion batteries is the main challenge for EV manufacturers. Even though many battery cooling systems have been introduced, it is still looking for an efficient and cost-effective cooling method for widespread application. From the literature study, it was found that liquid cooling with MCHS is a common method in commercial applications. In addition to the high heat transfer capacity of the liquids used, the MCHS will further improve the heat transfer rate with a reasonable pressure drop, which makes this method a preferable choice for manufacturers. However, this cooling system is heavier and more complex due to the high density and leakage issues of the used liquids. In this context, this study proposed using air as a coolant for MCHS by adding porous zones inside the channel. This significantly reduced the channel's temperature by about 25 K, which is efficient enough to maintain the battery within its operating temperature limit. However, it was found that a fully porous duct causes a pressure drop of 268 Pa, which considerably increases the parasitic power consumption of the system. So, it was proposed to use intermittent porous zones within the cooling channel. The simulation results indicate that the mini-channel with intermittent porous zones is as light as an empty duct and can provide a heat transfer similar to a fully porous duct, with a pressure drop closer to an empty duct. However, the challenge of this idea was to locate the porous zones correctly within the cooling channel. To overcome this challenge, a novel approach was proposed to identify the best locations of porous zones within the channel. It is an iterative method that works by moving the coldest porous zone towards the hottest non-porous zone. It was found that using the proposed method, a solution very close to the optimum value can be obtained within a few steps, which significantly reduces the computational time and effort. Finally, from the simulation study performed, it was found that the mini-channel filled with 10-20% porous zones was the best case for cooling the batteries. This can reduce the battery's temperature by 15.8 K with a reasonable pressure drop of 30.5 Pa (~11% of a fully porous channel) and 18% saving in weight in comparison with a fully porous channel.

**Appendix I**

The optimum location for porous zones for different porous volume fractions is shown in Table I. Based on this table, at a higher volume fraction, almost all the end of the duct shall be filled with porous metal foam.

Table I: Optimum location for porous zones for different porous volume fractions



**CRediT authorship contribution statement**

**M. Yang:** Methodology, Writing – Original Draft, Writing – Review & Editing

**G. Mathew:** Methodology, Simulation, Validation, Formal analysis, Investigation, Writing – Original Draft.

**H. Nemati:** Conceptualization, Methodology, Simulation, Validation, Formal analysis, Investigation, Writing – Original Draft, Writing – Review & Editing, Supervision

**M. A. Moghimi:** Simulation, Validation, Formal analysis, Investigation, Writing – Original Draft, Writing – Review & Editing, Supervision.

**Declaration of Competing Interest**

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

**References**

[1] W. Wu, S. Wang, W. Wu, K. Chen, S. Hong, Y. Lai, A critical review of battery thermal performance and liquid based battery thermal management, Energy Convers. Manag., 182 (2019) 262-281.

[2] V. Etacheri, R. Marom, R. Elazari, G. Salitra, D. Aurbach, Challenges in the development of advanced Li-ion batteries: a review, Energy Environ. Sci., 4 (2011) 3243-3262.

[3] J. Liu, Z. Huang, J. Sun, Q. Wang, Heat generation and thermal runaway of lithium-ion battery induced by slight overcharging cycling, J. Power Sources, 526 (2022) 231136.

[4] J. Kim, J. Oh, H. Lee, Review on battery thermal management system for electric vehicles, Appl. Therm. Eng., 149 (2019) 192-212.

[5] Y. Hua, S. Zhou, Y. Huang, X. Liu, H. Ling, X. Zhou, C. Zhang, S. Yang, Sustainable value chain of retired lithium-ion batteries for electric vehicles, J. Power Sources, 478 (2020) 228753.

[6] S. Abada, G. Marlair, A. Lecocq, M. Petit, V. Sauvant-Moynot, F. Huet, Safety focused modeling of lithium-ion batteries: A review, J. Power Sources, 306 (2016) 178-192.

[7] W.H. Zhu, H. Yang, K. Webb, T. Barron, P. Dimick, B.J. Tatarchuk, A novel cooling structure with a matrix block of microfibrous media/phase change materials for heat transfer enhancement in high power Li-ion battery packs, Journal of Cleaner Production, 210 (2019) 542-551.

[8] J. Lin, X. Liu, S. Li, C. Zhang, S. Yang, A review on recent progress, challenges and perspective of battery thermal management system, Int. J. Heat Mass Transf., 167 (2021) 120834.

[9] T.M. Bandhauer, S. Garimella, T.F. Fuller, A critical review of thermal issues in lithium-ion batteries, J. Electrochem. Soc., 158 (2011) R1.

[10] M. Al-Zareer, I. Dincer, M.A. Rosen, A novel approach for performance improvement of liquid to vapor based battery cooling systems, Energy Convers. Manag., 187 (2019) 191-204.

[11] D. Ren, X. Liu, X. Feng, L. Lu, M. Ouyang, J. Li, X. He, Model-based thermal runaway prediction of lithium-ion batteries from kinetics analysis of cell components, Appl. Energy, 228 (2018) 633-644.

[12] T. He, T. Zhang, S. Gadkari, Z. Wang, N. Mao, Q. Cai, An investigation on thermal runaway behaviour of a cylindrical lithium-ion battery under different states of charge based on thermal tests and a three-dimensional thermal runaway model, Journal of Cleaner Production, (2023) 135980.

[13] L. Sheng, H. Zhang, L. Su, Z. Zhang, H. Zhang, K. Li, Y. Fang, W. Ye, Effect analysis on thermal profile management of a cylindrical lithium-ion battery utilizing a cellular liquid cooling jacket, Energy, 220 (2021) 119725.

[14] K. Osmani, M. Alkhedher, M. Ramadan, D.S. Choi, L.K. Li, M.H. Doranehgard, A.-G. Olabi, Recent progress in the thermal management of lithium-ion batteries, Journal of Cleaner Production, (2023) 136024.

[15] T. Amalesh, N.L. Narasimhan, Introducing new designs of minichannel cold plates for the cooling of Lithium-ion batteries, J. Power Sources, 479 (2020) 228775.

[16] L.W. Jin, P.S. Lee, X.X. Kong, Y. Fan, S.K. Chou, Ultra-thin minichannel LCP for EV battery thermal management, Appl. Energy, 113 (2014) 1786-1794.

[17] Y. Huo, Z. Rao, X. Liu, J. Zhao, Investigation of power battery thermal management by using mini-channel cold plate, Energy Convers. Manag., 89 (2015) 387-395.

[18] Z. Qian, Y. Li, Z. Rao, Thermal performance of lithium-ion battery thermal management system by using mini-channel cooling, Energy Convers. Manag., 126 (2016) 622-631.

[19] H. Liu, E. Chika, J. Zhao, Investigation into the effectiveness of nanofluids on the mini-channel thermal management for high power lithium ion battery, Appl. Therm. Eng., 142 (2018) 511-523.

[20] E. Jiaqiang, D. Han, A. Qiu, H. Zhu, Y. Deng, J. Chen, X. Zhao, W. Zuo, H. Wang, J. Chen, Orthogonal experimental design of liquid-cooling structure on the cooling effect of a liquid-cooled battery thermal management system, Appl. Therm. Eng., 132 (2018) 508-520.

[21] Y. Huang, P. Mei, Y. Lu, R. Huang, X. Yu, Z. Chen, A.P. Roskilly, A novel approach for Lithium-ion battery thermal management with streamline shape mini channel cooling plates, Appl. Therm. Eng., 157 (2019) 113623.

[22] R. Jilte, R. Kumar, M.H. Ahmadi, Cooling performance of nanofluid submerged vs. nanofluid circulated battery thermal management systems, Journal of Cleaner Production, 240 (2019) 118131.

[23] Y. Yang, X. Xu, W. Li, G. Tong, Simulation analysis of the influence of internal surface morphology of mini‐channel on battery thermal management, Int. J. Energy Res., 44 (2020) 8854-8864.

[24] X. Xu, G. Tong, R. Li, Numerical study and optimizing on cold plate splitter for lithium battery thermal management system, Appl. Therm. Eng., 167 (2020) 114787.

[25] K. Monika, C. Chakraborty, S. Roy, S. Dinda, S.A. Singh, S.P. Datta, Parametric investigation to optimize the thermal management of pouch type lithium-ion batteries with mini-channel cold plates, Int. J. Heat Mass Transf., 164 (2021) 120568.

[26] H. Nemati, M.A. Moghimi, J.P. Meyer, Shape optimisation of wavy mini-channel heat sink, Int. Commun. Heat Mass Transf., 122 (2021) 105172.

[27] Z. Guo, Q. Xu, S. Zhao, S. Zhai, T. Zhao, M. Ni, A new battery thermal management system employing the mini-channel cold plate with pin fins, Sustain. Energy Technol. Assess., 51 (2022) 101993.

[28] J. Jaguemont, J. Van Mierlo, A comprehensive review of future thermal management systems for battery-electrified vehicles, J. Energy Storage, 31 (2020) 101551.

[29] J.R. Patel, M.K. Rathod, Recent developments in the passive and hybrid thermal management techniques of lithium-ion batteries, J. Power Sources, 480 (2020) 228820.

[30] C. Kannan, R. Vignesh, C. Karthick, B. Ashok, Critical review towards thermal management systems of lithium-ion batteries in electric vehicle with its electronic control unit and assessment tools, Proc. Inst. Mech. Eng. D: J. Automob. Eng., 235 (2021) 1783-1807.

[31] M. Shahjalal, T. Shams, M.E. Islam, W. Alam, M. Modak, S.B. Hossain, V. Ramadesigan, M.R. Ahmed, H. Ahmed, A. Iqbal, A review of thermal management for Li-ion batteries: Prospects, challenges, and issues, J. Energy Storage, 39 (2021) 102518.

[32] P.S. Liu, G.-F. Chen, Porous materials: processing and applications, (2014).

[33] M.M. Heyhat, S. Mousavi, M. Siavashi, Battery thermal management with thermal energy storage composites of PCM, metal foam, fin and nanoparticle, J. Energy Storage, 28 (2020) 101235.

[34] H. Liu, S. Ahmad, Y. Shi, J. Zhao, A parametric study of a hybrid battery thermal management system that couples PCM/copper foam composite with helical liquid channel cooling, Energy, 231 (2021) 120869.

[35] S.K. Mohammadian, S.M. Rassoulinejad-Mousavi, Y. Zhang, Thermal management improvement of an air-cooled high-power lithium-ion battery by embedding metal foam, J. Power Sources, 296 (2015) 305-313.

[36] M.R. Giuliano, A.K. Prasad, S.G. Advani, Experimental study of an air-cooled thermal management system for high capacity lithium–titanate batteries, J. Power Sources, 216 (2012) 345-352.

[37] S. Bazkhane, I. Zahmatkesh, Taguchi–based sensitivity analysis of hydrodynamics and heat transfer of nanofluids in a microchannel heat sink (MCHS) having porous substrates, Int. Commun. Heat Mass Transf., 118 (2020) 104885.

[38] P.T. Garrity, J.F. Klausner, R. Mei, Performance of aluminum and carbon foams for air side heat transfer augmentation, J. Heat Transfer, 132 (2010).

[39] S. Mousavi, M. Siavashi, A. Zadehkabir, A new design for hybrid cooling of Li-ion battery pack utilizing PCM and mini channel cold plates, Appl. Therm. Eng., 197 (2021) 117398.

[40] B. Alazmi, K. Vafai, Analysis of fluid flow and heat transfer interfacial conditions between a porous medium and a fluid layer, Int. J. Heat Mass Transf., 44 (2001) 1735-1749.

[41] R. Akula, C. Balaji, Thermal management of 18650 Li-ion battery using novel fins–PCM–EG composite heat sinks, Appl. Energy, 316 (2022) 119048.

[42] G. Maranzana, I. Perry, D. Maillet, Mini-and micro-channels: influence of axial conduction in the walls, Int. J. Heat Mass Transf., 47 (2004) 3993-4004.

[43] H. Fathabadi, A novel design including cooling media for Lithium-ion batteries pack used in hybrid and electric vehicles, J. Power Sources, 245 (2014) 495-500.

[44] H. Fathabadi, High thermal performance lithium-ion battery pack including hybrid active–passive thermal management system for using in hybrid/electric vehicles, Energy, 70 (2014) 529-538.

[45] Y. Inui, Y. Kobayashi, Y. Watanabe, Y. Watase, Y. Kitamura, Simulation of temperature distribution in cylindrical and prismatic lithium ion secondary batteries, Energy Convers. Manag., 48(7) (2007) 2103-2109.