Multi-objective optimization design of thermal management system for lithium-ion battery pack based on Non-dominated Sorting Genetic Algorithm II

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HIGHLIGHTS

- A novel liquid cooling system is proposed for lithium-ion battery pack.
- Multi-objective optimization of the cooling system is performed based on NSGA-II.
- Small hydraulic diameter enhances heat transfer coefficient at large friction factor.
- The cooling system achieves the desired thermal performance at a small pressure drop.

ARTICLE INFO

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- Battery thermal management system (BTMS)
- Multi-objective optimization
- Non-dominated Sorting Genetic Algorithm II (NSGA-II)
- Response surface approximation (RSA)

ABSTRACT

The thermal management of batteries was a significant issue considering the safety and efficiency. Optimal design of a novel liquid cooling system with symmetrical double-layer reverting bifurcation channel was performed by combining experimental, numerical simulation and multi-objective optimization techniques. The thermophysical parameters and heat production rate of the battery for numerical simulation were obtained by experiments. The convective heat transfer coefficient and the surface friction coefficient were chosen as objective functions to visually reflect the heat transfer process. Furthermore, batteries were confined to work at the optimal temperature (25–40 °C) and the optimal temperature difference between cells (less than 5 °C). The performance values of design points obtained by Latin hypercube sampling were calculated numerically. Response surface approximation was adopted to approximate the objective function and the constraint function to reduce computing time. The Pareto-optimal front between \( -h \) and \( f \) was obtained using Non-dominated Sorting Genetic Algorithm II. 17.19% change in heat transfer coefficient was accomplished by 85.53% change in skin friction coefficient. The results reported that the cooling system with optimized thermal performance can be obtained at low flow loss.

1. Introduction

Under the pressure of energy shortage and environmental pollution, automobile manufacturers have to turn their attention to green energy and clean cars [1,2]. The key task of developing clean energy vehicles is to find an energy storage system that can support high mileage and fast acceleration. Various batteries have been proposed, such as lead-acid, nickel-based, sodium-based and lithium-ion batteries [3]. In all these electrochemical systems, lithium-ion batteries are the most promising choice for electric vehicles (EVs) and hybrid electric vehicle (HEVs), because they have the advantages of high specific energy and power, long cycle life, no memory effect and fast charging and discharging speed [4,5]. However, batteries are very sensitive to temperature. Uneven temperature can result in partial deterioration of batteries. Excessive temperature can not only reduce the service life of batteries, but also threaten the safety of batteries, and even cause permanent damage [6–8]. The optimum operating temperature range of lithium-ion batteries is 25–40 °C, and the maximum temperature difference between cells is not more than 5 °C [9,10]. Therefore, to prolong the cycle life of batteries and maximize the performance of batteries, it is necessary to...
A large number of thermal management studies have focused on reducing the maximum temperature of lithium-ion batteries, pump power and the maximum temperature difference between cells. They mainly started from three aspects: numerical simulation scheme, experimental means and optimization scheme. According to the different cooling media, BTMS can be divided into air cooling, liquid cooling (liquid cooling plate and heat pipe cooling) and phase change material cooling [11].

Air cooling, including natural air cooling and forced air cooling, is widely applied because of its low cost and simple structure. Bernardo et al. [12] employed the optimization algorithm to optimize the air cooling system of the battery pack. It was found that the maximum temperature difference between battery modules was as high as 8.36 °C. Panchal et al. [26] found that if the temperature of the battery was higher than 66 °C, it would be difficult to cool it below 52 °C by air cooling. Therefore, air cooling is mainly applicable to the cooling of power batteries with less heat production.

Phase change materials (PCMs) are considered to be the best energy storage method because of their high fusion latent heat [14-16]. Akeiber et al. [16] found that PCMs may melt completely in hot summer or after several continuous charging and discharging cycles. Further, low thermal conductivity will hinder the heat transfer of PCM.

Chen et al. [11] investigated the thermal performance of indirect liquid cooling, and direct liquid cooling. The results showed indirect liquid cooling was more efficient than direct liquid cooling for vehicle cooling applications. Indirect cooling consists of liquid cooling based on cold plate and liquid cooling based on heat pipe. Heat pipe cooling provides a solid-state solution, but their cost, operating temperature range and cooling power limit its wide application in large battery packs [17-19]. Due to the strict space-limitation of battery pack, liquid cooling system based on cold plate is preferred [20]. Therefore, the liquid-cooling technique based on cold plate was adopted to cool the lithium-ion battery pack in our work.

Chen et al. [21] and Yu et al. [22] reported that the channel structure, namely, the route of coolant channel extension, was another parameter which had significant effect on the performance of cold plate besides the geometry of the channel. It can be generalized classified as parallel channel [23,24], serpentine channel [25,26] and multi-channel [27,28]. Patil et al. [24] numerically studied the inlet coolant mass flow rate, the inlet coolant temperature and the number of cooling channels in parallel channels. The results showed that the cooling efficiency was enhanced by the low coolant inlet mass flow, the low coolant inlet temperature and a high number of the channels. Panchal et al. [26] performed a comparative study of serpentine mini-channels for cooling
prismatic lithium-ion batteries combining experimental and numerical simulation methods. The results reported that the temperature of the cold plate increased with the increase of discharge rate and operating temperature. Stephen [29] evaluated the heat dissipation performance of cold plates with parallel channel, spiral channel and bifurcation channel. The results reported that the spiral channel design provided the best thermal performance at the expense of pump pressure (> 100 kPa). The bifurcation channel achieved good thermal performance and low pressure drop (< 10 kPa). There were great hot spots in parallel channel, and its comprehensive performance was worst.

It has been shown in many literatures that the bifurcation channel achieved better heat dissipation ability than parallel straight channel and serpentine channel for microelectronic devices. Furthermore, it offered the inherent advantage of low power consumption [30,31]. In the previous work [27], we designed a double-layered channel including four straight channels in the collection layer and bifurcation channels in the dispersed layer, which proved that the tree-shaped bifurcated cold plate design had good thermal performance and low pressure drop. Because the battery is always sandwiched by two cold plates, a novel symmetrical double-layered reverting bifurcation channel was proposed. In addition, the thermal performance and pressure drop of the liquid-cooled plate are always contradictory, and there are few literatures on multi-objective optimization of the liquid-cooled plate. Therefore, multi-objective optimization of the novel design was performed coupled with the surrogate model.

2. Design and analysis of battery pack

2.1. Battery pack design

In practical applications, the battery pack configuration is usually composed of many thin cell units. There is a cooling plate between each two cells. The heat generated by each cell is conducted into the cold plate through the contact surface and then transmitted by the coolant. Because the heat production at the electrode tab is very small, the electrode tab is omitted in this study to simplify the battery model. Fig. 1(a) illustrates a battery pack with cooling plates. In the battery pack, there are four square lithium-ion batteries, in turn named cell 1, cell 2, cell 3, cell 4, which are taken from commercial lithium-ion batteries. Fig. 1(b) and Fig. 1(c) display the structure of the cell and cooling plate, respectively. The cold plate in this study contains a symmetrical double-layer reverting bifurcation channel, which is improved on the basis of our previous work [27]. In addition to the channel thickness for the two layers, the cold plate thickness has three solid layer variables, namely, the thicknesses of the cover layers at the top and bottom, and the separating solid layer between the two channel layers. The coolant enters the collection layer channel from four inlets on the side and converges to the center. After flowing through the separating solid layer between the collection layer and the dispersion layer, it enters the dispersion layer channel, and finally leaves the cold plate through the four exits. Please note that the collection layer channel is exactly the same as the dispersed layer channel, only the flow direction is opposite.

2.2. Experimental setup and parameter extraction

The parameters related to the battery are listed in Table 1. In the numerical solution, the physical parameters of the battery include density (\(\rho_b\)), thermal conductivity (\(k_b\)) and specific heat capacity (\(c_{pb}\)). For simplicity, the battery is composed of a single material, so \(\rho_b\), \(k_b\) and \(c_{pb}\) are constants. According to the mass \(m_b\) and volume \(V_b\) of the cell, the density \(\rho_b\) of the battery is calculated to be 2049 kg/m\(^3\) (\(\rho_b = m_b/V_b\)). The thermal conductivity of the battery is generally taken as 3–5 W/(m °C) [23], \(k_b = 5\) W/(m °C) was considered in our work. The specific heat capacity and heat generation expression of the battery will be obtained by the following experiment. The test device is presented in Fig. 2.

The battery tester electronic load EBC-A40L can charge and...
discharge the battery according to the test procedure. Its voltage test
range is 0–5 V and current test range is 0.1–40 A. The thermostat can
provide a stable ambient temperature for battery testing. The tem-
perature range of the thermostat is 10–300 °C. The K-type thermocou-
ple is placed on the battery surface by Kapton tape, as shown in
Fig. 3. Point 2 and point 3 are placed near positive and negative elec-
trodes, respectively. Point 1 is placed on the center of the battery and
point 4 is located near the bottom of the battery.

The heat generated by the battery \( Q_{gen} \) is mainly composed of two
parts. One is the irreversible heat generated by the internal resistance
of the battery. The other is the reversible heat generated by the elec-
trochemical reaction inside the battery. The typical Bernadi heat gen-
eration model is usually used to describe the heat generation rate as fol-
lows [32]:

\[
Q_{gen} = I \left( U_{OCV} - U \right) + \frac{T_b}{\rho} \frac{dU_{OCV}}{dT_b}
\]  

(1)

where \( I \) \((U_{OCV} - U)\) is the Joule heat generated by the internal resistance
of the battery; \( I T_b \frac{dU_{OCV}}{dT_b} \) is the heat generated by the elec-
trochemical reaction inside the battery. Thus, the thermal generation of
batteries can also be written as follows [33]:

\[
Q_{gen} = I^2 R_j + I T_b \frac{dU_{OCV}}{dT_b}
\]  

(2)

As the temperature rises, the heat absorbed by the battery is as
below:

\[
Q_{abs} = m_c c_p \frac{dT_b}{dt}
\]  

(3)

In an insulated environment, the heat generated by the battery is
equal to the heat absorbed by the battery. After the above Eq. (2) and
Eq. (3) are transformed, the Eq. (4) can be obtained.

\[
\frac{1}{I} \frac{dT_b}{dt} = R_j \frac{1}{m_c c_p} I + \frac{1}{m_c c_p} T_b \frac{dU_{OCV}}{dT_b}
\]  

(4)

According the heat generation theory of the battery, as described in
Eq. (4), the finite element model of the battery is established. To make
an adiabatic environment, the battery was wrapped with the heat in-
sulated cotton. Discharging with 5 different rates (8 A, 16 A, 24 A, 32 A,
40 A) for 15 min, respectively. Further, five different \( dT_b/dt \) values are
obtained.

Chacko et al. [34] and Panchal et al. [35] suggested that the Joule
resistance \( R_j \) could be regarded as a constant when the battery operated
normally at 25–45 °C and SOC values of 0.2–0.9. The term \( T_b \frac{dU_{OCV}}{dT_b} \)
is related to the electrochemical reaction inside the battery. It
depends on SOC value and battery types, which can be determined as a
constant for same battery [36]. The term \( 1/\rho dU_{OCV} / dt \) is also a constant
for constant-current discharging within 15 min [37]. Therefore, the func-
tion \( 1/I(dT_b/dt) \) can be regarded as a linear relation about the current \( I \)
in Eq. (4). The linear relationship of \( 1/I(dT_b/dt) \) against \( I \) is obtained
by experiment, as shown in Fig. 4.

The linear equation is as follows:

\[
\frac{1}{I} \frac{dT_b}{dt} = 4.271 \times 10^{-8} I + 1.390 \times 10^{-4}
\]  

(5)

According to Eq. (4), the slope of liner line is \( R_j/\rho c_p \). The
equivalent specific heat of the battery \( (c_{pb}) \) is calculated to be 847 J/
(kg·°C). Therefore, combing the Eq. (3) with Eq. (4), the heat generation
\( Q_{gen} \) (W) of the battery is expressed as below:

\[
Q_{gen} = 2 \times 10^{-3} I^2 + 6.509 \times 10^{-3} I
\]  

(6)

As a result, the heat generation per unit volume of the battery \( (q_{gen}) \)
can be written as follows:

\[
q_{gen} = \frac{Q_{gen}}{V_b} = 7.409 I^2 + 241.139 I
\]  

(7)

<p>| Table 1 |</p>
<table>
<thead>
<tr>
<th>Parameters of lithium ion battery.</th>
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<td>Parameters</td>
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<td>Charging conditions</td>
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<td></td>
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<tr>
<td>Discharging conditions</td>
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</table>

Fig. 2. Test platform for the lithium ion battery.

Fig. 3. Thermocouple locations.

Fig. 4. Linear relationship of \( 1/I(dT_b/dt) \) against \( I \).
2.3.1. Initial and boundary condition

The initial conditions of the battery pack cooling system are given as follows:

- The initial temperature of battery is equal to 25 °C, and the inlet temperature of the coolant and the ambient temperature are equal to 25 °C;
- Apply a volume heat source to all batteries. Based on the experimental heat generation rate Eq. (7), when the battery is discharged at 3C (C-rate refers to the ratio of current to nominal voltage), the heat generation rate is calculated as 81715 W/m³.

The boundary conditions are provided in Table 3. The details are as bellow:

- Contact surface of battery pack with air: Free convection boundary condition with heat transfer coefficient of 5 W/(m² °C);
- Cold plate inlet: Mass flow rate \( q_m \).
- Cold plate outlet: The ambient pressure is used as the reference pressure of the fluid at the outlet and is equal to 0 Pa.
- Cold plate wall: If the outer wall is in contact with the battery surface, the solid-solid coupling is performed, and the inner channel wall and the fluid domain were fluid-solid coupled. In order to enhance the simulation of convective heat transfer, two boundary layers at the interface of fluid-solid interaction were added, which made the calculation of fluid-solid interaction more accurate.

2.3.2. Assumption

Because the actual heating generation process inside lithium-ion batteries is complex, the following assumptions are needed for some physical properties of batteries [5]:

① The physical properties of various media in the battery are uniform, and the density and specific heat capacity of batteries are constant.
② The thermal conductivity is anisotropic.

③ The radiation inside the battery has little effect on the heat dissipation and can be neglected.
④ The current density of the battery distributes uniformly during charging and discharging, and the heat generation of each part is uniform.

For cooling plate simulation, the following assumptions are put forward [38]:

1. The cold plate is homogeneous and isotropic.
2. The fluid is single phase, incompressible and steady.
3. The thermophysical properties of cooling water and aluminum are independent of temperature.
4. Ignore gravity and contact thermal resistance.
5. Ignore the influence of viscosity dissipation.

2.3.3. Governing equation

Based on the above assumptions, the governing equations of the numerical model can be written as follows [39]:

\[ \nabla \cdot (\rho l \vec{u} l) = 0 \]  
\[ \rho l (\vec{u} l \cdot \nabla \vec{u} l) = -\nabla p + \mu \nabla^2 \vec{u} l \]  
\[ \text{Energy conservation in the fluid domain:} \]
\[ \rho l c_p l (\vec{u} l \cdot \nabla T l) = k l \nabla^2 T l \]  
\[ \text{Energy conservation in the solid domain:} \]
\[ \nabla^2 T s = 0 \]

The Reynolds number at the inlet is defined as:
\[ Re = \frac{\rho l \mu D h}{\mu} \]  
The maximum inlet mass flow rate of coolant is 10 g/s and the corresponding Reynolds number is 2210.33. Thus, the laminar flow model is adopted.

The convective heat transfer coefficient and Nusselt number in the fluid domain are calculated by the following equations:
\[ h = \frac{Q_{con}}{n A s (T_w - T_l)} \]  
\[ N_u = \frac{h D h}{k l} \]

The skin friction coefficient of the fluid domain is calculated by the following equation:
\[ f = \frac{\Delta P}{\frac{1}{2} \rho l \frac{u^2}{2} \frac{1}{k l}} \]
To overcome the flow resistance, the pump power of driving coolant through all channels can be calculated as follows:

\[ P = \frac{\Delta P \cdot q_m}{\rho_1} \] (16)

In this work, the double-layered bifurcation channel contains many turning sections and variable cross-section channels. Therefore, the overall pressure drop in the channel consists of longitudinal pressure drop and local pressure drop.

### 2.3.4. Mesh independence test

Based on the above initial conditions and boundary conditions, a rigorous independent test of the volume mesh number was conducted to obtain accurate simulation results, as shown in Fig. 6.

It can be concluded that the volume grid number has little effect on the simulation results. The mesh model with number of 2,506,874 elements was selected for further study.

### 3. LiBTMS optimization methodology

#### 3.1. Design variables and objective functions

The double-layered bifurcation channel with variable cross-section contains many variables. According to Murray’s law [40], when the flow state of coolant is laminar, the width of all branches at the splitting point behaves the following relationship, which can reduce the pump power consumption.

\[ \sum D_{1\text{flow}}^3 = \sum D_{2\text{flow}}^3 \] (17)

Given the width value \( D_1 \) of the main channel and the width ratio \( \omega \), the width of all channels can be calculated. The length ratio \( \gamma \) was employed to determine the split point position of the internal channel along the X axes and Y axes, namely, to determine the relative length of \( L_1 \) and \( L_{0y} \) and that of \( L_1' \) and \( L_{0y}' \). When the bifurcation angle between channels (with widths of \( D_2 \) and \( D_3 \) is known, the channel length \( L_2 \) can be determined, and the values of length \( L_4 \) and \( L_5 \) are determined. To reduce the workload, the optimal bifurcation angle of heat transfer was obtained by numerical simulation. Therefore, the plane structure of the double-layered bifurcation channel can be determined by three variables: the channel width \( D_1 \), the width ratio \( \omega \), the length ratio \( \gamma \).

The width ratio \( \omega \) and the length ratio \( \gamma \) are defined as follows:

\[ \omega = \frac{D_1}{D_2} \] (18)

\[ \gamma = \frac{L_1}{L_1 + L_3} = \frac{L_1'}{L_1' + L_3'} \] (19)

For simplicity, the thickness of the cover layers at the top and bottom was designed to be constant at 0.5 mm. Consequently, the thickness of the cold plate (d) is determined by two variables, namely, the channel thickness (\( d_1 \)) and the separated solid layer thickness (\( d - d_r \)). One of the variables (channel thickness) was chosen for optimize. In addition, increasing mass flow rate can enhance heat transfer, but increase power consumption. Accordingly, the inlet mass flow rate \( q_m \) can be used as one of the optimization variables.

Therefore, the channel width \( D_1 \), the width ratio \( \omega \), the length ratio \( \gamma \), the channel thickness \( d_1 \) and the inlet mass flow rate \( q_m \) were selected as optimization design variables. The range of variation is given in Table 4.

In our work, convective heat transfer coefficient \( h \) and skin friction coefficient \( f \) were used as objective functions. The convective heat transfer coefficient represents the heat transfer rate. The skin friction coefficient reflects the flow of coolant in the channel. Increasing convective heat transfer coefficient usually causes an increase in skin friction coefficient. The convective heat transfer coefficient was maximized and the skin friction coefficient was minimized to obtain a cooling system with better heat transfer capacity and less flow loss. Besides, batteries were confined to work at the optimal temperature (25–40 °C) and the optimum temperature difference between cells (less than 5 °C).

#### 3.2. Response surface approximation (RSA)

Multi-objective optimization based on NSGA-II requires multiple evaluation of the objective function to find the optimal solution. In the absence of representative response functions, the evaluation becomes very expensive and time-consuming. Queipo et al. [41] suggested various surrogate models, including response surface approximation (RSA), Kriging (KRG) and radial basis neural network (RBNN). RSA surrogate model is the most widely adopted [42]. Therefore, RSA model was employed to approximate these performance functions.

RSA is a method of fitting polynomial function of discrete response. In our study, the second order polynomial with intercept term, linear term, square term and quadratic interaction term was used as the response function. As bellow:

\[ f(x) = \xi_0 + \sum_{i=1}^{N} \xi_i x_i + \sum_{i=1}^{N} \xi_i^2 x_i^2 + \sum_{i<j}^{N} \xi_i x_i x_j \] (20)

#### 3.3. Multi-objective optimization

NSGA-II evolutionary algorithm reduces the complexity of non-inferior sorting genetic algorithm, and provides the advantages of fast running speed and good convergence of solution set [43].

The above multi-objective problems can be formulated as follows: Minimize \( F(x) = [F_1(x), F_2(x), F_3(x), \ldots, F_m(x)] \)

Subject to \( g(x) = [g_1(x), g_2(x), g_3(x), \ldots, g_s(x)] \leq 0 \)

The work-flow of multi-objective optimization is illustrated in Fig. 7.
3.4. K-means Clustering

K-means clustering is based on the minimum distance method to classify samples. Cluster centers and objects allocated to them represent clusters. Therefore, K-means clustering was adopted to cluster Pareto optimal solutions, and representative solutions are obtained to analyze the relationship between objective function and design variables.

4. Result and discussion

4.1. Effect of bifurcation angle on battery pack cooling system

As mentioned above, the double-layered reverting bifurcation channel is determined by five design variables and the bifurcation angle between the channels with widths of $D_2$ and $D_3$. In order to reduce the workload, the influence of bifurcation angle on cooling system was studied by numerical simulation before multi-objective optimization. The maximum bifurcation angle ($\theta$) is calculated as 57°. Therefore, other variables remained unchanged and bifurcation angles of 0°, 25°, 35°, 45° and 55° were used for numerical simulation to obtain a $\theta$ value with better performance, respectively.

The variations of the maximum temperature of battery pack ($T_{\text{max}}$), the maximum temperature difference between cells ($\Delta T_{\text{max}}$), and the total pressure drop of cooling system ($\Delta P$) with bifurcation angle ($\theta$) are depicted in Fig. 8. It can be clearly noticed that when $\theta = 25^\circ$, $T_{\text{max}}$ is the smallest, and $\Delta T_{\text{max}}$ and $\Delta P$ are also lower. Thus, the bifurcation angle with better performance is equal to 25°.

4.2. Multi-objective optimization design

There is neither functional relationship between structural variables and objective functions, nor that between structural variables and constraint functions. Therefore, RSA surrogate model was employed to approximate the objective function and the constraint function. Before that, enough design points were selected in the design space using Latin Hypercube Sample (LHS), and their performance values were solved numerically based on STAR-CCM +, as shown in Table 5.

According to the discrete points and response values in Table 5, approximate fitting of $h$, $f$, $T_{\text{max}}$, and $\Delta T_{\text{max}}$ was obtained. As bellow:

$$h(D_1, \omega, \gamma, d_s, q_m) = 22.4913387 - 10.2257151x_1 + 13.8032977x_5 - 21.83124491x_2 + 48.01035174x_3 + 17.41384001x_4 - 0.0277601377x_5^2 + 8.053342345x_3^2 - 2.461636992x_2^2 + 1.921379319x_1^2 + 3.981526936x_5 + 0.1819920011x_4 - 0.09953x_3 - 0.09729628x_4 + 0.15490958x_5 - 0.24049958x_5^2 - 15.1479536x_3^2$$

$$f(D_1, \omega, \gamma, d_s, q_m) = 0.30995608 + 0.239938735x_1 + 1.064487442x_5 + 0.890910891x_2 + 3.71168x_3 - 0.1819920011x_4 - 1.765911966x_5^2 + 1.176740708x_3^2 + 2.470607888x_2^2 + 0.007706868x_2^2 + 0.081577525x_4 - 0.09953x_3 - 0.09729628x_4 + 0.005044x_5^2$$

$$T_{\text{max}}(D_1, \omega, \gamma, d_s, q_m) = 141.636654 + 5.93505637x_1 + 36.549x_4 - 16.6221x_1 + 0.38867563x_1^2 + 27.64085406x_5^2 + 18.20891082x_2^2 + 6.649952656x_3^2 + 1.058502118x_4^2 + 3.13399579x_2 - 0.33998x_3 + 3.429127712x_4^2 + 0.0473211x_4 + 59.91604969x_3 - 21.2755918x_4 - 2.17837537x_3 - 6.630932012x_4 + 0.289351024x_1 - 0.43196591x_4$$

Fig. 7. Work-flow of the multi-objective NSGA-II algorithm.

Table 4

<table>
<thead>
<tr>
<th>Limit</th>
<th>Variables</th>
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</thead>
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<td>$D_1$ (mm)</td>
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<tr>
<td>Lower</td>
<td>$D_1$ (mm)</td>
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</table>

Fig. 8. Effect of bifurcation angle on the performance of cooling system.
Table 5

<table>
<thead>
<tr>
<th>No.</th>
<th>Design variables</th>
<th>Objective functions</th>
<th>Constraint functions</th>
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<td>$D_i$, $\omega$, $\gamma$, $d_i$ (mm), $q_{pm}$</td>
<td>$h$ (W/°C)</td>
<td>$f$ ($m^2$)</td>
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The Pareto-optimal front using the NSGA-II and clustering points obtained through K-means clustering.

The pareto-optimal front using the NSGA-II evolutionary algorithm is plotted in Fig. 9. Because the convective heat transfer coefficient $h$ is maximized and the surface friction coefficient $f$ is minimized, a concave front can be seen in the Pareto front of $(h,f)$. From this figure, it is evident that the improvement of one objective always comes at the expense of the other, which can be attributed to the natural contradiction between convective heat transfer coefficient and skin friction coefficient. In addition, because each solution is Pareto’s global optimal solution, it can be found that none of the solutions in the Pareto front is completely superior to the others.

To validate the prediction accuracy of the surrogate model and to
Fig. 10. Velocity profile and temperature profile of representative Pareto-optimal solutions.
analyze the relationship between the design variables and the objective function, K-means clustering algorithm was used to select representative solutions (with red-marked) from Pareto-optimal frontier (POF), as shown in Fig. 9. The representative solutions and corresponding design variables are tabulated in Table 6.

<table>
<thead>
<tr>
<th>Design</th>
<th>A (m²)</th>
<th>B (m²)</th>
<th>C (m²)</th>
<th>D (m²)</th>
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<tbody>
<tr>
<td>f (m²)</td>
<td>0.329</td>
<td>0.842</td>
<td>1.426</td>
<td>2.177</td>
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<td>ΔP (Pa)</td>
<td>2283.174</td>
<td>4325.533</td>
<td>9451.370</td>
<td>29249.510</td>
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</table>

The predicted objective function values of representative solutions were compared with those calculated by CFD. The maximum relative errors in the predictions of convective heat transfer coefficient (h) and skin friction coefficient (f) are 3.17% and 7.28%, respectively, which demonstrates that the predicted values for the representative solutions have reached an agreement with the CFD results. It can be noticed from Table 6 that when the maximum temperature of the battery pack ranges from 25 °C to 40 °C and the maximum temperature difference between cells is less than 5 °C, the values of D1, ω, and qn tend to be near the upper bound, the γ value is near the middle and its lower bounds, but the d value fluctuates widely. In other words, the design variables D1, ω, and qn are less sensitive than that of γ and d. Besides, it can be observed that the convective heat transfer coefficient increases as the length ratio γ and channel thickness d decrease. This is because the increase in γ value makes the length of L2 channel larger. The surface integral values of convective heat transfer coefficient and skin friction coefficient enlarge with the expansion of heat transfer area. Decreasing the channel thickness can reduce the hydraulic diameter of the channel and enlarge the flow velocity of the coolant, thus increasing the convective heat transfer coefficient and the skin friction coefficient. 17.19% change in heat transfer coefficient is accomplished by 85.53% change in skin friction coefficient.

Fig. 10 illustrates the temperature profile and velocity profile of representative solutions. It can be seen that the flow velocity of cooling system corresponding to representing solutions A-D is increasing. In response the heat transfer is improved and the maximum temperature of the battery pack is decreasing. In addition, it can be observed that the flow velocity of coolant in inner channel is higher than that in peripheral channel, which results in heat concentration areas appear on the periphery of battery.

Behavior of pressure drop (ΔP) versus skin friction factor (f) is presented in Table 7. It can be concluded that the trend of pressure drop is consistent with that of skin friction coefficient, which is in accordance with Eq. (15). The flow loss of the cooling system was evaluated based on the transient indicator f and the result indicator ΔP. As expected, the pressure drop of the cooling system is much smaller than that of the cold system with serpentine channel [25], which reflects the inherent advantage of low power consumption in bifurcation channel.

5. Conclusion

Based on the evolutionary algorithm, the multi-objective optimization design of the novel battery pack liquid-cooling system was performed to maximize the convective heat transfer coefficient and minimize the skin friction coefficient. Furthermore, the maximum temperature of the battery pack and the maximum temperature difference between cells were considered as constraints. The design variables included the geometric variables of the channel cross section (the main channel width D1, the width ratio ω, the length ratio γ, and the channel thickness d), and the inlet flow rate qn. Latin hypercube sampling was used for the selection of enough design points. Further, response surface approximation models of the performance functions were constructed. The tradeoff Pareto front between the two objective functions indicated that the thermal performance can be improved at higher flow loss. The representative solutions illustrated that the channel thickness and length ratio had a great effect on the performance of the cooling system. As expected, the pressure drop of the studied channel was less than 1/2 of that of the traditional serpentine channel, which highlighted the inherent advantages of the bifurcation channel structure.

Declaration of Competing Interest

The authors declare that there is no conflict of interest.

Acknowledgements

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