



# Article Parametric Investigation on the Performance of a Battery Thermal Management System with Immersion Cooling

Yuxin Zhou <sup>1,2,3</sup>, Zhengkun Wang <sup>1,2,3</sup>, Zongfa Xie <sup>1,2,3,\*</sup> and Yanan Wang <sup>1,2,3,\*</sup>

- <sup>1</sup> School of Mechanical Engineering, Shandong University, Jinan 250061, China; zhouyuxin@mail.sdu.edu.cn (Y.Z.); wangzhengkun@mail.sdu.edu.cn (Z.W.)
- <sup>2</sup> Key Laboratory of High-Efficiency and Clean Mechanical Manufacture, Ministry of Education, Shandong University, Jinan 250061, China
- <sup>3</sup> National Demonstration Center for Experimental Mechanical Engineering Education, Shandong University, Jinan 250061, China
- \* Correspondence: zongfax@sdu.edu.cn (Z.X.); wyn@sdu.edu.cn (Y.W.)

Abstract: Lithium-ion batteries will generate a large amount of heat during high-rate charging and discharging. By transferring the heat to the environment in time, the batteries can be kept in a suitable temperature range. This allows them to work normally, prolongs their cycle life, and reduces the risk of thermal runaway. Immersion cooling is a simple and efficient thermal management method. In this paper, a battery thermal management system (BTMS) with immersion cooling was designed by immersing the lithium-ion cells in the non-conductive coolant—dimethyl silicone oil. The electric-thermal coupled model was adopted to obtain the heat production and temperature distribution of the cell during discharging, and the performance of the system was obtained by numerical calculation. It was found that, compared with natural cooling, immersion cooling could significantly reduce both the maximum temperature (MAT) of the cell and the temperature of the tabs during the 3C discharging process. However, the maximum temperature difference (MATD) of the cell was significantly increased. To solve this problem, the effects of the flow rate, viscosity, specific heat capacity, and thermal conductivity of the coolant on the performance of immersion cooling were further investigated and discussed, including the MAT and MATD of the cell, and the pressure drop of the coolant. The method and results could provide references for the design and application of the BTMS with immersion cooling in the future.

Keywords: immersion cooling; lithium-ion battery; thermal management; temperature; pressure drop

## 1. Introduction

Lithium-ion batteries are widely used in electric vehicles due to their great energy density, high voltage, and small self-discharge rate [1,2]. However, lithium-ion batteries should be working under certain temperature conditions. The best temperature range is usually between 20 °C and 40 °C, and the maximum temperature difference should be controlled within 5 °C [3,4]. Excessively higher or lower temperatures could affect the normal working of the battery, reduce its capacity, and shorten its life [5]. When the temperature is extremely high, thermal runaway may even occur, resulting in security incidents [6]. In addition, lithium-ion batteries will release a large amount of heat during long-term cycles and rapid charging and discharging processes. Insufficient and uneven heat dissipation can easily lead to local heat accumulations, which cause uneven temperature distributions in the batteries and in the battery pack, and affect their consistency and safety [7]. Therefore, it is essential to employ an effective battery thermal management system (BTMS) to regulate the temperatures and equalize the heat in the batteries.

According to the kinds of heat transfer mediums, the BTMSs commonly used in electric vehicles mainly include air cooling, liquid cooling, heat pipe cooling, PCM cooling, and hybrid cooling. The air-cooling type relies on the air as the heat transfer medium [8].



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**Copyright:** © 2022 by the authors. Licensee MDPI, Basel, Switzerland. This article is an open access article distributed under the terms and conditions of the Creative Commons Attribution (CC BY) license (https:// creativecommons.org/licenses/by/ 4.0/). However, due to the smaller specific heat capacity and lower thermal conductivity of the air, its temperature regulation and thermal equalization capabilities are usually not as good as those of the liquid cooling type. The liquid cooling type can be further classified into indirect cooling and direct cooling (also known as immersion cooling). Indirect cooling is to attach one or a few cooling plate(s) or pipe(s) on the cells, and the heat is taken away by the flowing coolant (such as a water–glycol solution) in the channel(s) of the cooling plate(s) or pipe(s) [9–12]. Direct cooling is usually to immerse the cells in the coolant, so it can directly contact the cells, and absorb the heat from them. According to the working state of the coolant, the BTMS with immersion cooling can be further classified into liquid-filled BTMS and liquid-circulated BTMS [13]. In the former, the coolant is stationary, and in the latter, the coolant is circulating. In addition, phase change materials (PCMs) and heat pipes are also commonly used in the research on BTMSs [14,15]. Moreover, hybrid cooling is used to combine two or more cooling methods to improve the cooling efficiency of the whole system. The characteristics of these BTMS technologies are summarized in Table 1, including their application levels, advantages, and limitations [16–20].

Table 1. Existing BTMS technologies.

Cooling Method	Application Level	Advantages	Limitations
Air cooling	Middle	The structure is relatively simple, the design is easy to implement, and the adaptability is good.	<ol> <li>The thermal conductivity of air is low, the specific heat capacity is small, and the temperature control ability under high-rate charge and discharge conditions is relatively weak. The temperature uniformity in the battery pack is relatively poor;</li> <li>When the active cooling method meets the thermal management requirements, the power consumption is greater, the space required for the system is larger, and the energy density of the battery pack is smaller;</li> <li>The potential for improvement is relatively small, the achievable temperature control limit is relatively low. It is mainly suitable for the battery pack with small energy density and low charge and discharge rates.</li> </ol>
Liquid cooling (indirect)	High	<ol> <li>Compared with those of air, the specific heat capacity and thermal conductivity of liquid are usually greater, and the cooling effect is usually better under the same power consumption;</li> <li>The cooling effect can be effectively improved through the flow flux, channel design and material properties of the coolant. The potential for improvement is relatively great;</li> <li>The temperature uniformity is usually better when the pipes or cooling plates are in contact with the side of the cells.</li> </ol>	<ol> <li>The structure of the system is more complex, the overall weight is greater, and the cost is higher;</li> <li>In order to prevent leakage and short circuits, the coolant should be in indirect contact with the cells, which increases the thermal resistance and suppresses the cooling effect;</li> <li>A pump is required to drive the coolant to circulate, and an additional cooling system is usually implemented to chill the coolant;</li> <li>The pipes or cooling plates usually have great thermal conductivity, which is not conducive to inhibiting the thermal runaway propagation in the battery pack.</li> </ol>

	Table 1. Cont.		
Cooling Method	Application Level	Advantages	Limitations
Liquid cooling (direct)	Low	<ol> <li>Simple and compact structure, light weight and low cost;</li> <li>The coolant is in direct contact with the cells, so the convective heat transfer is stronger, and the cooling effect is further improved;</li> <li>The coolant is dielectric, which could avoid short circuiting and inhibit the thermal runaway propagation;</li> <li>The pressure drop of the coolant is usually lower, and the power consumption is usually smaller.</li> </ol>	<ol> <li>The sealing requirements for the battery pack are higher, and conductive media are not allowed to enter the system;</li> <li>A pump and a cooling system are usually required to drive the coolant and reduce its temperature.</li> </ol>
Phase Change Material (PCM) cooling	Low	<ol> <li>Simple structure, light weight, low cost, and easy to implement;</li> <li>The PCM absorbs heat to reduce the temperature, and does not require additional cooling system. So, the power consumption is small;</li> <li>The shape of the PCM is easy to change, the layout of the system is simple, and the temperature uniformity is relatively good;</li> <li>PCMs usually have good insulation resistivity and can act as insulating materials to reduce the risk of short circuits.</li> </ol>	<ol> <li>After phase transition, the volume of the PCM usually changes obviously, increasing the possibility of leakage;</li> <li>Most PCMs have low thermal conductivity and poor sensitivity to temperature changes. Adding materials that can improve the thermal conductivity may reduce the insulation resistivity of PCMs;</li> <li>In the case of continuous cycles, the cooling effect declines. An additional cooling system is required to take away the heat absorbed by the PCM;</li> <li>The heat absorbed by a certain amount of PCM is limited. Increasing the PCM volume could improve the cooling efficiency, but it will also increase the mass and power consumption.</li> </ol>
Heat pipe cooling	Low	<ol> <li>Excellent thermal conductivity and wide application range. Sensitive to temperature changes, which can effectively control the temperature in real time;</li> <li>The working of the heat pipe alone does not require additional power consumption.</li> </ol>	<ol> <li>The structure of the system is complex, and is difficult to manufacture;</li> <li>Higher cost and risk of leakage. Small heat capacity;</li> <li>An additional cooling system is usually required to chill the condenser section of the heat pipe;</li> <li>The contact area with the cells is small, and additional cooling plates are usually needed to improve the temperature uniformity.</li> </ol>
Hybrid cooling	Middle	<ol> <li>Different kinds of thermal management methods are combined with complementary advantages and limitations. It is conducive to improving the overall performance of the system;</li> <li>The power consumption of the system can be reduced to a certain extent.</li> </ol>	<ol> <li>Compared with the single thermal management method, the volume and structure complexity of the hybrid cooling system increases, and the manufacture and maintenance costs increase accordingly;</li> <li>The control of the hybrid cooling systems is more difficult. The coupling between different kinds of cooling methods and the adaptability of various technologies need to be further studied.</li> </ol>

In immersion cooling, the coolant is in direct contact with the cells, eliminating the thermal resistance caused by the air gaps and thermal conductive materials. So, theoretically its heat transfer efficiency will be high. Chen et al. [21] compared the thermal management performance of several different cooling methods by experiment and calculation, including air cooling, plate cooling, and immersion cooling. It was found that when obtaining the same cooling effects, the air cooling consumed more energy, while the two liquid cooling methods consumed less. However, under small mass flow rates, the immersion cooling worked better in controlling the maximum temperature and maximum temperature difference of the cells than the plate cooling. Dubey et al. [22] compared the performance of immersion cooling and plate cooling applied on a 21,700 lithium-ion battery module by the CFD method. It was found that the performance of the two methods was basically the same at a lower discharge rate. However, at a higher discharge rate, the maximum temperature and maximum temperature difference in the battery module with immersion cooling were both remarkably lower than those in the battery module with plate cooling. Besides, with the increase in the flow rate of the coolant, the pressure drop in the cooling plate rose significantly, up to 15-25 times that of the immersion cooling.

However, since the cells are immersed in the coolant directly, the coolant must be a fluid with insulation resistivity. So, conductive coolants are no longer suitable for the BTMS with immersion cooling. For example, Sundin et al. [23] investigated the application of a non-conductive AmpCool AC-100 coolant developed by Engineered Fluids in the BTMS with immersion cooling. It was found that the average temperature of the cells was maintained at about 22.5 °C during rapid charging and discharging cycles, and the temperature fluctuation of each monitor was very small. In contrast, when forced air cooling was used, the average temperature was 28.7 °C, and the temperature fluctuation was much greater. Bhattacharjee et al. [24] designed an immersion cooling system for a lithium-ion battery stack at the scale of kWs. The system also used the AmpCool AC-100 coolant developed by Engineered Fluids. Simulation results showed that, compared with air cooling and plate cooling, the immersion cooling could significantly reduce the maximum temperature of the battery stack during discharging. Jithin et al. [25] applied deionized water, mineral oil, and AmpCool AC-100 to the BTMS with immersion cooling. It was found that the deionized water could control the average temperature rise of the cells within 5  $^{\circ}$ C under low mass flow rate (0.003 kg/s). Its cooling effect was the best, and its power consumption was the lowest. The cooling effects of mineral oil and AmpCool AC-100 were similar, but the power consumption of AmpCool AC-100 was less.

Tian et al. [26] applied hydrofluoroether (HFE-6120) to a BTMS with immersion cooling, and found that compared with those of air cooling, the maximum temperature, maximum temperature difference, and temperature standard deviation of the battery module with immersion cooling were significantly reduced under the same power consumption. Wang et al. [27] used a transformer oil with high insulation resistivity as the coolant, and designed an immersion cooling thermal management system for pouch-type lithium-ion cells. The effects of the immersion depth and the flow rate on the maximum temperature and the maximum temperature difference were investigated. It was found that when the cells were completely immersed in the coolant, the cooling performance was the best. By increasing the flow rate of the coolant, the cooling performance could be further improved. Patil et al. [28] immersed the pouch-type lithium-ion cells in a non-conductive coolant, and designed a BTMS with immersion cooling. It was found that when the coolant was flowing and tab cooling was assisted, the cells could obtain the lowest temperature rise and the smallest temperature difference. Meanwhile, the effects of the discharge rate, spacing between the cells, type and flow rate of the coolant, and inlet and outlet positions on the thermal management performance were investigated.

Typically, the coolant of the BTMS with immersion is in direct contact with the cells. However, Luo et al. [29] designed a special sealing structure for the battery module. Water was injected into the cooling channels as the coolant, which could directly contact the cells without causing a short circuit. It was found that the maximum temperature of the cells discharged at 2C could be controlled below 50  $^{\circ}$ C at the flow rate of 200 mL/min, and the maximum temperature difference could be controlled within 5  $^{\circ}$ C at the flow rate of 500 mL/min.

Thermal safety design for cells is also essential in BTMSs [30]. The immersion cooling with the application of the dielectric coolant can effectively inhibit the propagation if the thermal runaway of a single cell occurs [20]. For example, Wu et al. [31] used silicone oil as the coolant to design a BTMS based on immersion cooling. The temperatures of the battery modules with immersion cooling and with tubular liquid cooling were compared through numerical simulations. It was found that the maximum temperature rise and maximum temperature difference of the battery module with immersion cooling were significantly lower than those with tubular liquid cooling, and the immersion cooling could effectively suppress the spread of thermal runaway.

In summary, immersion cooling has a simpler design structure and lower pressure drop, and could achieve better performance in BTMSs. With the continuous increase in consumers' demands for the power of electric vehicles, as well as the rapid application of fast charging technology, higher requirements have been surfaced for BTMSs under high-rate charge and discharge conditions. It is necessary to investigate the performance and design method of immersion cooling in the BTMS more deeply. However, the current studies on BTMSs with immersion cooling did not investigate the effects of the viscosity, specific heat capacity, thermal conductivity, and other material parameters of the coolant on the thermal management performance of the system, nor did they discuss the pressure drop of the coolant. To fill the above knowledge gaps, this research designed a BTMS with immersion cooling based on the pouch-type lithium-ion cells and the coolant of dimethyl silicone oil. The coolant was in direct contact with the cells and absorbed the heat. The performance of the BTMS was obtained by a numerical method and compared with that of the natural cooling method. The effects of the flow rate, viscosity, specific heat capacity, and thermal conductivity of the coolant on the maximum temperature (MAT) and the maximum temperature difference (MATD) of the cell, as well as the pressure drop of the coolant, were further investigated, which could provide references for the design and application of BTMSs with immersion cooling in the future.

#### 2. Models and Governing Equations

#### 2.1. Battery

The battery used in this research was a pouch-type lithium-ion cell of 24 Ah. The basic parameters of the cell are shown in Table 2. The shape and dimensions of the cell are shown in Figure 1.

Parameters	Value
Capacity (Ah)	25
Rated voltage (V)	3.8
Cathode material	LiMn <sub>x</sub> Co <sub>y</sub> Ni <sub>z</sub> O <sub>2</sub> , LiMn <sub>2</sub> O <sub>4</sub>
Anode material	graphite
Charging cut-off voltage (V)	4.2
Discharging cut-off voltage (V)	3.0
Width (mm)	166
Height (mm)	205
Thickness (mm)	7.2

Table 2. Basic parameters of the cell [32].



Figure 1. (a) Shape of the cell; (b) dimensions of the cell (mm).

The electric and thermal behaviors of the cell were obtained by the electric–thermal coupled model. The electric model was the NTGK model in ANSYS/Fluent. This model has great advantages in the parameter setting and building process, and is widely used in the performance simulation of lithium-ion batteries [33]. Assuming that the current in the current collector is distributed on the spanwise plane of the cell, the potential in the current collector is:

$$\sigma_i \frac{\partial^2 \emptyset_i}{\partial x^2} + \sigma_i \frac{\partial^2 \emptyset_i}{\partial y^2} + \frac{J \cdot n}{\delta_i} = 0 \ (i = p, n) \tag{1}$$

where  $\sigma_i$  is the conductivity of current collector i (S·m<sup>-1</sup>),  $\emptyset_i$  is the potential in current collector i (V),  $\delta_i$  is the thickness of the current collector i (m), and p or n in the subscript indicates the positive or negative electrode. J is the current density across the separator (A·m<sup>-2</sup>), and  $\vec{n}$  is the normal unit vector perpendicular to the current collector.

The boundary condition at the junction of the current collector and the tab in the positive electrode is:

$$\overrightarrow{n} \cdot \left( -\sigma_p \nabla \varnothing_p \right) = \frac{I_{cell}}{L_{tab} \delta_p} \tag{2}$$

where  $I_{cell}$  is the current of a single pair of electrodes (A),  $L_{tab}$  is the width of the tab (m), and  $\delta_p$  is the thickness of the current collector in the positive electrode (m).

The boundary condition at the junction of the current collector and the tab in the negative electrode is:

Ø

$$n = 0 \tag{3}$$

According to the actual situation, other boundary conditions of the electric model are insulation conditions.

The current density passing through the separator can be calculated by the equivalent resistance:

$$U = \frac{U_{oc} - (\varnothing_p - \varnothing_n)}{R}$$
(4)

where *R* is the internal resistance per unit area in the cell ( $\Omega \cdot m^{-2}$ ), and  $U_{oc}$  is the opencircuit voltage of the cell (V).

The temperature distribution in the cell is:

$$\rho C_p \frac{\partial T}{\partial t} = \lambda_{in} \frac{\partial^2 T}{\partial X^2} + \lambda_{in} \frac{\partial^2 T}{\partial Y^2} + \lambda_{th} \frac{\partial^2 T}{\partial Z^2} + q$$
(5)

where  $\rho$  is the density (kg·m<sup>-3</sup>),  $C_p$  is the specific heat capacity at constant pressure (J·kg<sup>-1</sup>·K<sup>-1</sup>), and *T* is the thermodynamic temperature (K).  $\lambda_{in}$  is the thermal conductivity parallel with the spanwise plane of the cell (W·m<sup>-1</sup>·K<sup>-1</sup>),  $\lambda_{th}$  is the thermal conductivity perpendicular to the spanwise plane of the cell (W·m<sup>-1</sup>·K<sup>-1</sup>), and *q* is the heat production rate of the cell (W·m<sup>-3</sup>).

The heat production rate in the Bernardi's heat production model is expressed as:

$$q = q_{rev} + q_{irev} + q_{CC,p} + q_{CC,n}$$
(6)

$$q_{rev} = -\frac{J/2}{(d_{PE} + d_S + d_{NE})} T \frac{\partial U_{oc}}{\partial T}$$
(7)

$$q_{irev} = \frac{J/2}{(d_{PE} + d_S + d_{NE})} (U_{oc} - \varnothing_p + \varnothing_n) = \frac{J^2 R/2}{(d_{PE} + d_S + d_{NE})}$$
(8)

$$q_{CC,i} = \frac{\sigma_i (\nabla \varnothing_i)^2}{d_{CC,i}/2} \tag{9}$$

where  $q_{rev}$  is the reversible heat production rate (W·m<sup>-3</sup>),  $q_{irev}$  is the irreversible heat production rate (W·m<sup>-3</sup>), and  $q_{CC, i}$  is the ohmic heat production rate of the current collector (W·m<sup>-3</sup>).  $d_{PE}$ ,  $d_S$ , and  $d_{NE}$  are the thicknesses of the positive electrode, separator, and negative electrode (m).

The heat production at the tab area is originated from the impedance of the tab and the contact impedance between the tab and the wire:

$$q_{tab,i} = I_{tab}^2 \cdot R_{tab,i} \tag{10}$$

$$R_{tab,i} = H_{tab} \left( \frac{1}{\sigma_{tab,i}} + \frac{1}{\sigma_{c,i}} \right)$$
(11)

where  $q_{tab,i}$  is the heat production rate of the tab (W),  $I_{tab}$  is the current passing through the tab (A),  $R_{tab,i}$  is the impedance of the tab ( $\Omega$ ), and  $H_{tab}$  is the height of the tab (m).  $\sigma_{tab,i}$  is the conductivity of the tab (S·m<sup>-1</sup>),  $\sigma_{c,i}$  is the converted conductivity of the contact impedance between the wire and the tab (S·m<sup>-1</sup>).

The boundary condition of the thermal model is:

$$Q_c = hA_c \left( T_w - T_f \right) \tag{12}$$

where  $Q_c$  is the convective heat exchange (W), *h* is the convective heat transfer coefficient (W·m<sup>-2</sup>·K<sup>-1</sup>),  $A_c$  is the convective heat transfer area (m<sup>2</sup>),  $T_w$  is the temperature on the surface of the cell (K), and  $T_f$  is the temperature of the fluid (K).

We imported the three-dimensional cell model in Figure 1b into ANSYS and performed meshing. The result is shown in Figure 2a. The number of grids was 44,536 and the number of nodes was 57,704. Then, the electric–thermal coupled model of the cell was built in ANSYS/Fluent. The relevant material parameters are shown in Tables 3–5. The relationships of the open-circuit potential and internal resistance with the state of charge (SOC) are referenced from the experiment results in [32].



**Figure 2.** (a) Grids of the cell; (b) grids of the model used for calculation; (c) temperature distribution on the cell at the end of 1.5 C discharge.

Table 3. Material parameters of the cell [32].

Cell	Value	
Density $\rho_{cell}$ (kg·m <sup>-3</sup> )	2300	
Specific heat capacity $C_{p,cell}$ (J·kg <sup>-1</sup> ·K <sup>-1</sup> )	1243	
Convective heat transfer coefficient $h_{cell}$ (W·m <sup>-2</sup> ·K <sup>-1</sup> )	3.5	
Thermal conductivity $\lambda_{in}$ (W·m <sup>-1</sup> ·K <sup>-1</sup> )	21	
Thermal conductivity $\lambda_{th}$ (W·m <sup>-1</sup> ·K <sup>-1</sup> )	0.48	

Table 4. Material parameters of the current collectors [32].

Current Collector	Positive	Negative
Thickness $\delta_i$ (µm)	20	14
Electric conductivity $\sigma_i$ (S·m <sup>-1</sup> )	$37.8  imes 10^6$	$59.6 imes10^6$

Table 5. Material parameters of the tabs [32].

Tab	Positive	Negative
Width <i>L<sub>tab</sub></i> (mm)	50	50
Height $H_{tab}$ (mm)	50	50
Thickness $\delta_{tab}$ (mm)	0.3	0.3
Density $\rho_{tab}$ (kg·m <sup>-3</sup> )	8700	2700
Specific heat capacity $C_{v,tab}$ (J·kg <sup>-1</sup> ·K <sup>-1</sup> )	385	900
Electric conductivity $\sigma_{tab}(S \cdot m^{-1})$	$59.6  imes 10^6$	$37.8  imes 10^6$
Thermal conductivity $\lambda_{tab}$ (W·m <sup>-1</sup> ·K <sup>-1</sup> )	401	237
Convective heat transfer coefficient $h_{tab}$ (W·m <sup>-2</sup> ·K <sup>-1</sup> )	4	4
Converted conductivity $\sigma_{c,tab}(S \cdot m^{-1})$	$80 imes 10^6$	$70 imes10^6$

Based on the experiment conditions given in [32], the voltages of the cell at three discharge rates (0.5 C, 1 C, 1.5 C) were calculated and compared with the experiment results given in [32], as shown in Figure 3a. It could be seen that the calculated results were basically consistent with the experiment results. Meanwhile, the temperature distribution on the cell at the end of 1.5 C discharge was calculated, as shown in Figure 2c. The temperature gradually decreased from the top to the bottom, and the temperature at the junction of the positive tab and the cell was the highest. The maximum temperature,

minimum temperature, and maximum temperature difference were essentially consistent with the calculation results given in [32]. Due to the different software used (Comsol was used in [32]), the color distributions on the temperature scales are not completely identical, so the display styles of the temperature distributions on the cell are slightly different. As shown in Figure 1b, three temperature monitor points were set at the corresponding positions of the model. The temperatures of the three points during 1.5 C discharge were compared with the experiment results provided in [32], as shown in Figure 3b. It can be seen that the calculated results were in good consistency with the experiment results.



**Figure 3.** (a) Voltages of the cell during discharge at 0.5 C, 1 C and 1.5 C; (b) temperatures at point 1, point 8 and point 12 during discharge at 1.5 C.

When an object is placed in static air for natural convection, the variation range of the convective heat transfer coefficient is usually between 5 and 25. So, we set the convective heat transfer coefficient  $h_{cell}$  to 5, 10, 15, 20 and 25, respectively. The sensitivity analysis was performed with the maximum temperature and minimum temperature of the cell as the indexes. The results are shown in Figure 4. With the increase in  $h_{cell}$ , the maximum temperature and minimum temperature and significant downward trends. This indicates that the convective heat transfer between the cell and the air can significantly affect the temperature distribution in the cell. As shown in Figure 1, the thickness of the pouch cell was small, but its surface area was large.



Figure 4. Sensitivity analysis of convective heat transfer coefficient.

## 2.2. Thermal Management System

The structure of the BTMS with immersion cooling is shown in Figure 5. The coolant is pumped into the battery module, where it passes through the cells and takes the heat away. The temperature of the coolant rises after it absorbs the heat. Then, the coolant flows into the heat exchanger and exchanges heat with the cooling system, and the air before it flows into the battery module again. The coolant circulates in the system and keeps the cells working under suitable temperature conditions.



Figure 5. Structure of the BTMS with immersion cooling.

The structure of the battery module is shown in Figure 6a. The cells and the coolant channels were arranged vertically in the battery module, and separated from each other. The width of each coolant channel was 7.6 mm, and its length and height had the same dimensions with those of the cell, as shown in Table 6. Compared with the front or back surface area of the cell, the side surface area of the cell was very small. When designing the immersion cooling system, the inlets and outlets of the coolant channels are set on the planes of the side surfaces of the cells. So, the heat transfer between the side surfaces of the cells and the coolant can be ignored. Only the heat transfer between the front and back surfaces of the cells and the coolant was considered. Since the coolant was in direct contact with the cells, and the water-glycol solution commonly used in plate cooling was electrically conductive, it was necessary to use a non-conductive coolant. Silicone oil has excellent insulation resistivity, and its resistance and viscosity changes little in a wide temperature range, so it is often used as a heat carrier. At a suitable flow rate, silicone oil can provide good cooling effects for the cells [34]. Moreover, silicone oil has good shear stability. When the vehicle runs on a bumpy road, the impact of the oil on the cells is very small. So dimethyl silicone oil was selected as the coolant in this research. Its material parameters are shown in Table 7.



Figure 6. (a) Structure of the battery module; (b) model used for calculation.

Table 6. Dimensions of the coolant channel.

Dimension	Value
Length (mm)	166
Width (mm)	7.2
Height (mm)	205

Table 7. Material parameters of dimethyl silicone oil.

Material Parameter	Value	
Density $\rho_c$ (kg·m <sup>-3</sup> )	968	
Viscosity $\mu_c$ (N·s·m <sup>-2</sup> )	1.452	
Specific heat capacity $C_{p,c}$ (J·kg <sup>-1</sup> ·K <sup>-1</sup> )	1630	
Thermal conductivity $\lambda_c(W \cdot m^{-1} \cdot K^{-1})$	0.16	

In the model, the dimethyl silicone oil was assumed to be an ideal incompressible fluid, and no phase change occurred during the cooling process. The initial flow rate of the coolant at the inlet was 1 mm/s. According to the structure and material parameters, as well as the flow rates involved in the following investigation, the Reynolds numbers calculated were all less than 2300, so the flow pattern was a laminar flow.

The continuity equation of the coolant is:

$$\frac{\partial \rho_c}{\partial t} + \nabla(\rho_c v) = 0 \tag{13}$$

The energy conservation equation of the coolant is:

$$\rho_c \frac{\partial T_c}{\partial t} + \nabla \cdot (\rho_c v T_c) = \nabla \cdot \left(\frac{\lambda_c}{C_{p,c}} \nabla T_c\right)$$
(14)

The momentum conservation equation of the coolant is:

$$\rho_c \frac{dv}{dt} = -\nabla P + \mu_c \nabla^2 v \tag{15}$$

where  $\rho_c$ ,  $\mu_c$ ,  $\lambda_c$ ,  $C_{p,c}$  and  $T_c$  are the density, viscosity, thermal conductivity, specific heat capacity and temperature of the coolant, respectively. v and p are the velocity and pressure of the coolant, respectively.

The front and the back surfaces of each cell were both in contact with the coolant, that is, one cell was in contact with two fluid domains. To simplify the model, we took out one cell in the center of the battery module and the two fluid domains attached to it. We cut each fluid domain from its middle plane in the spanwise direction, and applied symmetrical boundary conditions on the middle planes. The final model used for calculation is shown in Figure 6b. The boundary conditions and initial conditions are listed in Table 8. Since the heat production rate of the positive tab was higher than that of the negative tab, the coolant flowed in from the positive tab side of the cell and flowed out from the other side.

Table 8. Boundary conditions and initial conditions.

Name	Condition
Inlet flow rate (mm/s)	1
Initial temperature of coolant (°C)	23
Flow pattern	Laminar
Outlet pressure	Atmosphere
Wall condition	Non-slip

Five different grid numbers were used for the grid independence verification, and the average grid qualities were all above 0.9. The maximum temperature (MAT) and minimum temperature (MIT) of the cell (excluding the tabs) at the end of 3 C discharge were used as indicators. As shown in Figure 7a, when the number of grids increased from 86,488 to 129,868, the MAT and MIT remained basically unchanged. Five different time steps were used for the time step independence verification. As shown in Figure 7b, when the time step decreased from 3 s to 1 s, the MAT and MIT remained basically unchanged. Considering both the amount of computation and the accuracy of results, the grid number of 86,488 and the time step of 3 s were finally adopted for the subsequent calculations. The final grids are shown in Figure 2b.



Figure 7. (a) Grid independence verification; (b) time step independence verification.

## 3. Performance of BTMS with Immersion Cooling

According to whether the coolant experiences phase transition during the convective heat transfer process [35], immersion cooling can be classified into single-phase immersion cooling and two-phase immersion cooling [20]. In this research, due to the thermal characteristics of the dimethyl silicone oil, the coolant did not experience phase transition, so it belongs to single-phase immersion cooling. Compared with two-phase immersion cooling, single-phase immersion cooling usually has a simpler structure and lower cost., The MAT and the maximum temperature difference (MATD) of the cell (excluding the tabs) with immersion cooling during the 3 C discharging process are shown in Figure 8. The temperature distribution on the cell at the end of the 3 C discharge is shown in Figure 9. There are two conditions considered in immersion cooling: stationary immersion cooling

(the flow rate of the coolant is 0 mm/s), and flowing immersion cooling (the flow rate of the coolant is 1 mm/s). For comparison, the temperature information on the cell with natural cooling is also provided in the figures. Natural cooling is to put one single cell in the air, cooled passively by natural convective heat transfer. The model is the same with the one in Section 2.1.



Figure 8. (a) MAT of the cell during 3 C discharging; (b) MATD of the cell during 3 C discharging.



**Figure 9.** Temperature distributions on the cell at the end of 3 C discharge: (**a**) natural cooling; (**b**) immersion cooling at 0 mm/s; (**c**) immersion cooling at 1 mm/s.

It can be seen from Figure 8a that the MAT of the cell with natural cooling demonstrated a rapid upward trend during discharging. At the end of discharge, the MAT of the cell reached 51.8 °C, which exceeded the suitable temperature range of lithium-ion batteries. However, with stationary immersion cooling, the MAT of the cell also demonstrated a rapid upward trend. At the end of discharge, the MAT reached 53.6 °C, which even exceeded that with natural cooling. This was caused by several reasons. First, the specific heat capacity of dimethyl silicone oil is relatively small (1630 J·kg<sup>-1</sup>·k<sup>-1</sup>), and is much smaller than that of water (4200 J·kg<sup>-1</sup>·k<sup>-1</sup>). So, its heat absorption and storage capability are relatively weak. Second, the model of the cell with immersion cooling is part of the battery module. The tight arrangement of the cells and the heat generated by the surrounding cells both affect the heat dissipation of the cell. Third, the stationary coolant only exchanges heat with the cell. Other thermal boundary conditions of the coolant are adiabatic. Therefore, during the discharging process, the temperature of the coolant increases continuously, and the heat exchange between the coolant and the cell decreases continuously. Eventually, the temperature of the cell rises rapidly. However, natural cooling involves only putting one cell in the air at 23 °C. The higher the temperature of the cell, the stronger the heat exchange between the air and the cell. So, at the end of discharge, the MAT of the cell with natural cooling is slightly lower than that with stationary immersion cooling. However, if the cell is put in a tightly packed battery module, the MAT at the end of discharge could be much higher.

With flowing immersion cooling, the MAT of the cell was significantly lower than those with natural cooling and stationery immersion cooling during the entire discharging process. At the end of discharge, the MAT was greatly reduced to 38.9 °C, within the suitable temperature range of lithium-ion batteries. This is because the flowing coolant can absorb and take the heat away from the cell continuously, thus greatly improving the cooling effect.

However, as shown in Figure 8b, the MATDs of the cells with stationary immersion cooling and flowing immersion cooling were both significantly greater than that with natural cooling during the entire discharging process. At the end of discharge, the MATD of the cell with stationary immersion cooling reached 11.6 °C, and that with flowing immersion cooling reached 12.6 °C. Both of them exceeded the limit of 5 °C. The reason is related to the temperature distribution of the cell. As shown in Figure 9a,b, with natural cooling and stationary immersion cooling, the MATS both appear at the junction between the positive tab and the cell. The MITs both appear at the lower part of the cell. Therefore, the MATDs of the cells were determined by the temperature near the positive tab and the temperature of the cell under these two conditions.

With natural cooling, the temperature near the tab is higher, where the convective heat exchange between the cell and the air is stronger. So, the increase in the MAT is inhibited. The temperature of the lower part of the cell is lower, where the convective heat exchange between the cell and the air is weaker. So, the increase in the MIT is accelerated. The combined action of the two factors results in a smaller MATD of the cell. With stationary immersion cooling, the temperature of the coolant close to the junction between the positive tab and the cell is higher after the coolant absorbs heat, so its cooling capability is weakened, and the increase in the MAT is accelerated. The temperature of the coolant far from the tab is lower, so its cooling capability is stronger, and the increase in the MIT is inhibited. The coolant with higher temperature is in the upper part, and coolant with lower temperature is in the lower part. It inhibits the convective heat transfer in the coolant, thereby further increasing the MATD of the cell.

As shown in Figure 9c, with flowing immersion cooling, the MAT appears at the left side of the cell, and the MIT appears at the right side of the cell. The temperature gradually increases from the right to the left. This is because the coolant with lower temperature was in direct contact with the cell at the inlet, so the cooling effect on the cell close to the inlet was very good. However, in the initial settings, the flow rate of the coolant was only 1 mm/s. During the flowing process, its temperature gradually increased, and the convective heat exchange between the coolant and the cell was weaker. So, the closer to the outlet, the greater the temperature rise of the cell, resulting in the increase in MATD. It is also noted that the MAT was transferred from the junction of the positive tab and the cell to the junction, and is not in direct contact with the coolant, a good cooling effect can also be achieved through heat conduction since it is closer to the inlet of the coolant. The negative tab is far away from the inlet, so its temperature is the highest, but still far below the corresponding temperatures with natural cooling and stationary immersion cooling.

Through the above analysis it can be seen that, compared with natural cooling and stationary immersion cooling, flowing immersion cooling demonstrates strong temperature suppression capability. The MAT of the cell can be controlled within 40 °C during the 3 C discharging process. The temperature of the tabs is also much lower. However, due to the

small flow rate of the coolant, the MATD of the cell cannot be controlled within 5 °C, which needs to be further improved.

#### 4. Effects of Different Parameters on the Performance of Immersion Cooling

The BTMS with immersion cooling has many parameters that could affect its performance, such as the flow rate, viscosity, specific heat capacity, and thermal conductivity of the coolant. By changing these parameters, their effects on the performance of immersion cooling were investigated, so as to provide references for the design and optimization of the BTMS. The method was to change one parameter at a time, while keeping the other parameters constant: the single variable method. The MAT and MATD of the cell (excluding the tabs) at the end of 3 C discharge, as well as the pressure drop of the coolant, were used as indicators. To facilitate analysis, the MIT of the cell was also supplemented as an indicator.

## 4.1. Flow Rate

The flow rate determines the mass flow of the coolant passing through the cell, which in turn determines whether it can quickly remove heat from the cell. However, the flow of the coolant requires power. The lower the flow rate of the coolant, the lower the energy consumption of the system. In the initial settings, the flow rate of the coolant was 1 mm/s. To investigate its effect on the performance of immersion cooling, the flow rate was set to 2 mm/s, 4 mm/s, 6 mm/s, 8 mm/s, 10 mm/s, and 12 mm/s, with the other parameters remaining unchanged. The MAT, MIT, and MATD of the cell at the end of discharge, as well as the pressure drop of the coolant, are shown in Figure 10.



**Figure 10.** Relationship between the flow rate of the coolant and the MAT, MIT, and MATD of the cell, as well as the pressure drop of the coolant.

It can be seen that, with the increase in the flow rate, the MAT of the cell decreased from  $38.9 \,^{\circ}$ C to  $29.9 \,^{\circ}$ C, a reduction of  $9.0 \,^{\circ}$ C. The MATD decreased from  $12.5 \,^{\circ}$ C to  $4.3 \,^{\circ}$ C, a reduction of  $8.2 \,^{\circ}$ C. Both of them demonstrate significant downward trends. This is because, as the flow rate increases, the coolant can take more and more heat away from the cell. However, as shown in Figure 9c, when the flow rate is  $1 \,\text{mm/s}$ , the temperature of the cell near the inlet of the coolant is already low, but the temperature of the cell near the outlet of the coolant and the cell near the flow rate increases, the convective heat exchange between the coolant and the cell near the inlet of the coolant will both increases. However, the increase in the latter is greater than that of the former. So, the decline in the MAT will be much greater than that of the MIT, which reduces the MATD of the cell. When the flow rate reaches 8 mm/s, the MATD decreases to  $5 \,^{\circ}$ C.

It can also be seen that the flow rate influences the pressure drop significantly. The higher the flow rate, the greater the pressure drop, and they are in direct proportion. This is because the flow pattern of the coolant is laminar, and the pressure drop is mainly composed of the frictional pressure drop in the channel:

$$\Delta P = f \frac{l}{D} \frac{\rho_c v^2}{2} \tag{16}$$

where  $f = \frac{64}{Re}$ , after substitution:

$$P = \frac{32lv\mu_c}{D^2} \tag{17}$$

where *v* is the flow rate of the coolant,  $\mu_c$  is the dynamic viscosity of the coolant, *l* is the length of the channel, and *D* is the hydraulic diameter of the channel.

Δ

The pressure drop of the coolant determines the pumping power of the circulating system. The pumping power required by the coolant in the flow channel is:

$$P = \Delta P \cdot A \cdot v \tag{18}$$

where *A* is the cross-sectional area of the flow channel. Therefore, the pumping power increases with the pressure drop. In practical applications, there is a trade-off between cooling performance and pumping cost.

Therefore, by increasing the flow rate of the coolant, the MAT and MATD of the cell can be effectively reduced. However, the pump providing the power will consume more energy. The flow rate should be adjusted in real time according to the actual needs of the BTMS.

#### 4.2. Viscosity

The viscosity of the coolant is one of the material properties. The viscosity of water is about  $0.001 \text{ N} \cdot \text{s} \cdot \text{m}^{-2}$ , while that of high-viscosity silicone oil can reach tens of thousands of  $\text{N} \cdot \text{s} \cdot \text{m}^{-2}$ . In the initial settings, the coolant is dimethyl silicone oil, and its viscosity is  $1.456 \text{ N} \cdot \text{s} \cdot \text{m}^{-2}$ . To investigate its effect on the performance of immersion cooling, the viscosity was set to  $0.1 \text{ N} \cdot \text{s} \cdot \text{m}^{-2}$ ,  $0.8 \text{ N} \cdot \text{s} \cdot \text{m}^{-2}$ ,  $1.5 \text{ N} \cdot \text{s} \cdot \text{m}^{-2}$ ,  $2.2 \text{ N} \cdot \text{s} \cdot \text{m}^{-2}$ , and  $2.9 \text{ N} \cdot \text{s} \cdot \text{m}^{-2}$ , with the other parameters remaining unchanged. The MAT, MIT, and MATD of the cell at the end of discharge, as well as the pressure drop of the coolant, are shown in Figure 11.



**Figure 11.** Relationship between the viscosity of the coolant and the MAT, MIT, MATD of the cell, as well as the pressure drop of the coolant.

It can be seen that the viscosity of the coolant basically does not influence the MAT and MATD of the cell. The MAT is maintained at 38.9 °C stably, and the MATD is maintained

at 12.5 °C stably. After calculating the Reynolds numbers of the coolant with different viscosities, it was found that they were all below 2100, so the flow patterns were all laminar. Therefore, the change in the viscosity has little effect on the flowing process of the coolant, and the convective heat transfer between the coolant and cell basically does not change.

However, the viscosity influences the pressure drop significantly. The greater the viscosity, the greater the pressure drop—in direct proportion—which is consistent with Equation (17). For the BTMS with immersion cooling in this research, shown in Figure 5, the coolant needs to circulate in the system and exchange heat in the heat exchanger. Higher viscosity is not conducive to the flow of coolant, and will also increase the energy consumption of the system. Therefore, the coolant with lower viscosity is better.

#### 4.3. Specific Heat Capacity

Specific heat capacity refers to the heat absorbed or released by a substance of unit mass when it rises or falls to unit temperature. The specific heat capacity of the coolant is also one of the material properties. At room temperature, the specific heat capacity of water is about 4200 J·kg<sup>-1</sup>·k<sup>-1</sup>. While in the initial settings, the specific heat capacity of dimethyl silicone oil is 1630 J·kg<sup>-1</sup>·k<sup>-1</sup>. To investigate its effect on the performance of immersion cooling, the specific heat capacity was set to 1700 J·kg<sup>-1</sup>·k<sup>-1</sup>, 2300 J·kg<sup>-1</sup>·k<sup>-1</sup>, 2900 J·kg<sup>-1</sup>·k<sup>-1</sup>, 3500 J·kg<sup>-1</sup>·k<sup>-1</sup>, 4100 J·kg<sup>-1</sup>·k<sup>-1</sup>, with the other parameters remaining unchanged. The MAT, MIT, MATD of the cell at the end of discharge, as well as the pressure drop of the coolant are shown in Figure 12.



**Figure 12.** Relationship between the specific heat capacity of the coolant and the MAT, MIT, MATD of the cell, as well as the pressure drop of the coolant.

It can be seen that with the increase in the specific heat capacity of the coolant, both the MAT and the MATD of the cell demonstrate downward trends. The MAT of the cell decreases from 38.6 °C to 33.8 °C, a reduction of 4.8 °C. The MATD decreased from 12.2 °C to 7.8 °C, a reduction of 4.4 °C. However, the MIT decreased from 26.3 °C to 25.9 °C, a reduction of only 0.4 °C. The reason for this phenomenon is that specific heat capacity reflects the heat capacity per unit mass of the coolant. In the convective heat transfer process between the coolant and the cell, the greater its value is, the more heat can be transferred through convection. So, the convective heat exchange is stronger, significantly reducing the temperature of the cell. However, since the temperature of the cell near the inlet of the coolant is already low, the change in the specific heat capacity has a greater effect on the MAT than on the MIT. So, the MATD also decreases with the MAT.

It can also be seen that the pressure drop of the coolant basically does not change with the specific heat capacity. This is because the specific heat capacity is a thermal parameter of the material. It mainly affects the heat exchange between the coolant and the cell, not the flow pattern of the coolant.

Therefore, when designing the BTMS with immersion cooling, the coolant with greater specific heat capacity can effectively enhance its performance. Adding phase change material (PCM) into the coolant is an effective way to increase its specific heat capacity, such as the PCM emulsion and encapsuled PCM [9,36].

## 4.4. Thermal Conductivity

Thermal conductivity refers to the heat transmitted directly by the material of unit section and unit length under unit temperature difference. The thermal conductivity of the coolant can directly affect the convective heat transfer between the coolant and cell. At room temperature, the thermal conductivity of water is 0.6009 W·m<sup>-1</sup>·k<sup>-1</sup>, and that of the nanofluid prepared by Huang et al. [37] increased by 4.3% compared with water. In the initial settings, the thermal conductivity of dimethyl silicone oil was 0.16 W·m<sup>-1</sup>·k<sup>-1</sup>. To investigate its effect on the performance of immersion cooling, the thermal conductivity was set to 0.1 W·m<sup>-1</sup>·k<sup>-1</sup>, 0.2 W·m<sup>-1</sup>·k<sup>-1</sup>, 0.3 W·m<sup>-1</sup>·k<sup>-1</sup>, 0.4 W·m<sup>-1</sup>·k<sup>-1</sup>, 0.5 W·m<sup>-1</sup>·k<sup>-1</sup>, 0.6 W·m<sup>-1</sup>·k<sup>-1</sup>, with the other parameters remaining unchanged. The MAT, MIT, and MATD of the cell at the end of discharge, as well as the pressure drop of the coolant, are shown in Figure 13.



**Figure 13.** Relationship between the thermal conductivity of the coolant and the MAT, MIT, and MATD of the cell, as well as the pressure drop of the coolant.

It can be seen that, with the increase in the thermal conductivity of the coolant, both the MAT and the MIT of the cell demonstrate downward trends. The MAT decreased from 40.4 °C to 37.0 °C, a reduction of 3.4 °C. The MIT decreased from 27.6 °C to 24.1 °C, a reduction of 3.5 °C. When the fluid with uniform temperature passes through a solid wall with another temperature, a fluid layer with a great temperature change will be formed near the solid wall, which is called the thermal boundary layer. It can be regarded as a fluid layer with a temperature gradient, which is also the main area where heat exchange occurs. In the thermal boundary layer of laminar flow, the heat transfer in the direction perpendicular to the wall mainly depends on thermal conduction. Therefore, the thermal conductivity plays an important role in the convection heat exchange between the coolant and the cell. Increasing thermal conductivity can better promote the convective heat exchange between the coolant and the entire cell is enhanced, the MAT and the MIT are reduced synchronously, resulting in insignificant changes in the MATD of the cell.

It was also noticed that, in the previous analysis, the MIT of the cell was always around 26 °C. However, after increasing the thermal conductivity, the MIT demonstrated

a significant downward trend. The reason is that the convective heat exchange between the coolant and the cell at the inlet determines the MIT. In the initial setting, the smaller thermal conductivity of dimethyl silicone oil functioned as a bottleneck, restricting the convective heat exchange between the coolant and the cell at the inlet.

It can also be seen that the pressure drop of the coolant basically did not change with thermal conductivity. This is also because the thermal conductivity was a thermal parameter of the material. It mainly affects the heat exchange between the coolant and the cell, not the flow pattern of the coolant.

Therefore, when designing the BTMS with immersion cooling, the coolant with greater thermal conductivity can effectively enhance its performance. Adding thermal conductive micro-particles into the coolant is an effective way to improve its thermal conductivity [13,38].

## 5. Conclusions

In this research, a BTMS with immersion cooling was designed based on the electricthermal coupled model of lithium-ion battery and the coolant of dimethyl silicone oil. The performance of the BTMS with immersion cooling was obtained through numerical calculations and compared with that with natural cooling. The effects of different parameters on the performance of immersion cooling were investigated. The following conclusions were obtained:

- 1. Compared with natural cooling and stationary immersion cooling, flowing immersion cooling can significantly reduce the MAT of the cell and the temperature of the tabs during the 3 C discharging process, even at a flow rate of 1 mm/s. However, both the stationary immersion cooling and the flowing immersion cooling at 1 mm/s increase the MATD of the cell significantly;
- 2. Within a certain range, the MAT and MATD of the cell can be greatly reduced by increasing the flow rate and specific heat capacity of the coolant. Increasing the thermal conductivity of the coolant can reduce the MAT and MIT of the cell, but has little effect on the MATD. The viscosity of the coolant basically does not affect the temperature of the cell;
- 3. The pressure drop of the coolant varies with the flow rate and viscosity of the coolant proportionally. Lower flow rate and smaller viscosity can reduce the energy consumption of the system. The specific heat capacity and thermal conductivity of the coolant basically do not affect the pressure drop;
- 4. Compared with other BTMS technologies, the structure design of the BTMS with immersion cooling is simple and compact. Its pressure drop is lower, and it can achieve a better cooling effect with smaller power consumption. The temperature uniformity can also be improved through the adjustments of design and working parameters. So, the BTMS with immersion cooling has good application prospects in electric vehicles.

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