INTRODUCTION

The development of new energy vehicles (especially electric vehicles) has become one of the most important ways to solve such problems as nonrenewable energy shortage, environmental pollution, and climate change in the world.\(^1,2\) The power battery has become a limitation to the development of new energy vehicles.\(^3\) Compared with other power batteries, Li-ion batteries have become the leader in power batteries due to their high energy density, long service life, low self-discharge rate, and good adaptability.\(^4-6\) The pouch Li-ion battery has the advantage of higher energy density, which makes it more and more equipped in electric vehicles. However, in case of high rate discharge, the Li-ion battery generates a large amount of heat,\(^7,8\) and Panchal et al studied the thermal behavior of Li-ion battery through experiments and established thermal models.\(^9-11\) If the heat accumulation cannot be dissipated in time, the pouch Li-ion battery expands due to internal gas generation. And burning and explosion may occur in more serious cases.\(^12\) Therefore, the power Li-ion battery needs to be controlled at a reasonable operating temperature range of 25-40°C.\(^13\) In addition, when the temperature difference in the Li-ion battery pack exceeds

Thermal management performance of cavity cold plates for pouch Li-ion batteries using in electric vehicles

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Abstract
The cold plate cooling system has become one of the most practical and promising Li-ion battery thermal management systems for electric vehicles. The existing cold plate has complex structure with high production cost, and the energy consumption is relatively high. In this paper, a detailed optimization study of cavity cold plate is carried out. The heat generation data of the pouch Li-ion batteries were obtained through experiments, and the data were applied to the finite element method simulation of the cavity cold plate thermal management system. The effects of cavity cold plate thickness \((d_1)\), cold plate inlet and outlet width \((d_2)\), and inlet coolant mass flow rate on battery temperature and cold plate pressure drop were discussed. The results showed that the optimization allowed the cavity cold plate to control the battery maximum temperature within 40°C and temperature difference within 5°C. Under the same volume of cold plates, the average pressure drop reduction rates of models where \(\Delta d\) (the difference between \(d_1\) and \(d_2\)) is 1 mm are 65% higher than that of the models where \(d_2\) is 1 mm. This study has practical significance for improving the design efficiency of cavity cold plate.

KEYWORDS
cavity cold plate, liquid cooling, pouch Li-ion battery, pressure drop, thermal management system
the consistency of the Li-ion battery is deteriorated and the service life is lowered. Therefore, the battery thermal management system (BTMS) of the pouch power Li-ion battery is very important. As an integral part of the power Li-ion battery pack, the thermal management system should have the advantages of high heat dissipation efficiency, low energy consumption, low economic cost, safety and stability. Therefore, the structure and boundary conditions need to be designed and optimized.

Thanks to the research by scholars all over the world, and there are now a variety of battery thermal management solutions. At present, among many solutions, the liquid-cooled heat management system based on cold plate has a good application prospects because of its compact structure, strong heat dissipation capability, and low cost. The current research method is mainly using the finite element model to simulate the cooling effect of BTMS on the basis of experimental data. Although the results of the experiment are more realistic and reliable, the variables that can be tested are less and the results are more limited. The numerical simulation results are also of great significance by establishing more accurate models and setting reasonable boundary conditions.

In recent years, scholars from various countries have done a lot of innovative research on cold plate cooling schemes. In terms of the material, the common cold plate material is aluminum. In order to enhance heat transfer, scholars have designed BTMS based on silicon cold plates. Although the silicon cold plate cooling system can meet the heat dissipation requirements of the Li-ion battery pack, it has higher economic cost and lower structural strength compared with aluminum cold plate. On types of coolants, the classic coolants are water, oil, and aqueous solution of ethylene glycol (EGW). In order to improve the thermal properties of the coolants and enhance the efficiency of the system, many new coolants have been studied. These new coolants have better thermal properties, but they need higher technical requirements than the classical coolants. The aqueous solution of ethylene glycol has a low freezing point, which is more practical.

In order to improve the heat dissipation capacity of the BTMS, multistage heat dissipation systems combining liquid cooling with other cooling methods were studied. Although the cooling effect of these combined cooling methods is higher than that of individual cooling methods, the economic cost of the BTMS for the large-capacity battery pack is greatly improved, and the addition of the phase change material increases the quality of the battery pack and the battery additional power consumption. By optimizing the structure of the cold plate, the cooling efficiency of the BTMS can also be improved, and the energy consumption can be reduced. The improvement of the above microchannel structure can better improve the cooling effect of the cold plate, but it increases the processing difficulty of the cold plate and therefore reduces the possibility of large-scale commercial use.

In summary, scholars from various countries have conducted various researches on the materials, coolants, combination of cold plate and other cooling methods, and optimization of microchannels inside the cold plate. Although the cooling effects of the BTMS have been improved, the proportion of coolant in most of the cold plates is small, and the heat dissipation of the cold plate has a great limitation. They have also increased production cost and technical requirements, which were not conducive to practical applications and widespread promotion. In this paper, large heat dissipation, low energy consumption, and easy-to-manufacture cavity cooling plates were analyzed in detail, and the effects of the thickness of the cooling plate, the inlet width of the cooling plate and the mass flow rate of the coolant on the maximum temperature, temperature difference, and pressure drop at 4 C-rate (the charge and discharge current with respect to battery's nominal capacity) discharge were analyzed in detail. It has reference value for the design of the cooling system of the Li-ion battery cavity cooling plate.

The rest of this article is organized as follows. Section 2 introduces the physical model in detail. The mesh model is determined, and the specific settings of the numerical calculation software FLUENT 17.0 are given. At the same time, the battery heat generation model is determined. Section 3 analyzes the results of numerical calculation: When the cold plate thickness and the cold plate inlet width is 1 mm, the effects of the cold plate size on the maximum temperature and temperature difference of the cells are analyzed. The effects of the mass flow rate of coolant on the maximum temperature and temperature difference of cells are analyzed. And the analysis of the pressure drop of the cold plate in different cooling systems is done. Section 4 is the conclusions.

## 2 | MODEL AND METHODOLOGY

### 2.1 | Physical model of cavity cold plate cooling system

The concept is shown in Figure 1A.B. The cavity cold plate cooling system presents a classic laminate structure in which a cavity cold plate is placed between two battery cells. The inlet and outlet are on the same side of the cold plate. The coolant flows from inlet to outlet and fills the entire cavity of the cold plate. The cells are commercial Li-ion batteries made by Shandong RealForce Enterprises Co., Ltd, a battery manufacturer in China. The specifications of cells are shown
in Table 1. In the next numerical calculation, the cells were reasonably simplified because of the main research on heat dissipation capacity of cavity cold plate, and the main part of the pouch battery was retained.

Figure 1C,D show the sizes of the cold plate model. The length and height of the cold plate are the same as those of cells while being symmetrical in the length direction of battery. As can be seen from these figures, \(d_1\) is the thickness of the cold plate, \(d_2\) is the width of the inlet and outlet of the cold plate, and \(\Delta d\) is the difference between the thickness of cold plate and the width of cold plate inlet and outlet. The shape of inlet and outlet is rectangular with the length of 25 mm. The distance between the cold plate inlet and the outer surface is 25 mm. The shell thickness of the cold plate is uniform, which is 0.5 mm.

| Property         | Parameter          | Property         | Parameter          |
|------------------|--------------------|------------------|--------------------|
| Type             | F22-87155235       | Cathode material | LiFePO₄            |
| Size (mm)        | 155 × 8.7 × 235    | Anode material   | Graphite           |
| Weight (g)       | 610                | Electrolyte      | Carbonate based    |
| Capacity (Ah)    | 22                 | End/Charging voltage (V) | 2.0/3.65 |
| Nominal voltage (V) | 3.2               | Maximum discharge current (A) | 100 |

Note: All data are provided by the manufacturer.

2.2 Governing equations

The coolant is 50% aqueous solution of ethylene glycol (50%EGW). The continuity equation for 50%EGW is described by the following equation:

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

(1)

where \(\nabla\) is gradient operator, \(\rho_{\text{EGW}}\) is density of coolant (kg m\(^{-3}\)), \(t\) is time (s), and \(\mathbf{u}\) is velocity vector of coolant.

The momentum conservation equation is given:

\[
\frac{\partial (\rho_{\text{EGW}} \mathbf{u})}{\partial t} + \nabla \cdot (\rho_{\text{EGW}} \mathbf{uu}) = -\nabla \cdot \mathbf{p} + \nabla \cdot (\mu_{\text{EGW}} \nabla \mathbf{u})
\]  

(2)
where $p$ presents static pressure (Pa) and $\mu_{E_{GW}}$ is dynamic viscosity of coolant (Pa s).

The energy conservation equation for coolant is as follows.\textsuperscript{38}

$$\frac{\partial(p_{E_{GW}}c_{E_{GW}}T)}{\partial t} + \nabla \cdot (p_{E_{GW}}c_{E_{GW}}Tu) = \nabla \cdot (\lambda_{E_{GW}} \nabla T)$$ (3)

where $c_{E_{GW}}$ is heat capacity (J kg$^{-1}$ K$^{-1}$), and $\lambda_{E_{GW}}$ is thermal conductive coefficient of coolant (W m$^{-1}$ K$^{-1}$).

## 2.3 | Numerical solution

By controlling the values of $d_1$ and $d_2$, a total of seven cavity cold plate cooling system models were established. And the specific values are shown in Table 2. The cavity cold plate is made with aluminum, and the outer surface of the cells is made of Al-plastic film. Table 3 is the thermal property parameters of each material measured by experiment or simulation.

Integrated computer engineering and manufacturing code computational fluid dynamics (ICEM CFD) 17.0 was used to divide the models into hexahedral meshes. Considering the impacts of the number of grids on the accuracy of the simulation results and to prevent the waste of computing resources, the grid independence test was performed on the cooling system model. Since the seven cooling systems are similar, Model 4 was divided into hexahedral meshes with different grid cell numbers (ranging from 889 248 to 6 702 200). The inlet mass flow rate was 0.001 kg/s, and the cell heat generation was 65 kW/m$^3$. The results show that the grid with 5 563 360 hexahedral cells in this model is accurate enough, as can be seen in Figure 2. The minimum volume is $1.29 \times 10^{-10}$ m$^3$, and the max volume is $1.42 \times 10^{-10}$ m$^3$.

Therefore, the grids with about 5.5 million hexahedral cells of all cooling system models in this numerical calculation were chosen. The boundary layers of coolant grid following that $y^+$ are chosen to be 10. This value is consistent with the recommended value for near-wall flows while being closer to acceptable value for the Laminar Viscous Model. All grids qualities are greater than 0.9.

Numerical calculations were performed on all cooling systems using FLUENT 17.0. Prior to this, the heat generation test of battery at 4 C-rate had been performed to obtain the maximum temperature, and then, User Define Function (UDF) in FLUENT was used to define the heat generation characteristics of cells. The heat generation density of cell can be estimated as Equation (4). The battery was fully charged at 100% state of charge (SOC) by constant current (0.5 C-rate)–constant voltage (cutoff current is 0.03 C-rate) (CC-CV) before discharge operation. The discharge time was 900 seconds. The calculation results were recorded every 30 seconds. And the comparison between the simulation results and the experimental data is shown in Figure 3.

As can be seen, the relative error between the simulation result and the experimental data is less than 2%.

$$\bar{q} = -63t + 93957.8$$ (4)

where $\bar{q}$ is average heat generation density of cell (W/m$^3$).

Before the simulation, the Reynolds numbers of the seven cooling system models had been calculated. The results show that the Reynolds numbers ($101.72 < Re < 567.28$) are less than 2300 in this paper. So, the Laminar Viscous Model was chosen in FLUENT. The slip of coolant relatively to the inner surface of cavity cold plate was not taken into consideration, and thermal contact resistance in cooling systems was ignored. The inlet boundary type was mass flow inlet, and the mass flow rate ($q_m$) was set to 0.005 kg/s, 0.01 kg/s, 0.015 kg/s, 0.02 kg/s, and 0.025 kg/s, respectively. Initial temperature of coolant was 25°C. The outlet boundary type was outflow. Cells and cold plate to air contact surfaces were set to Convection Thermal Condition, and the Heat Transfer Coefficient and Free Stream Temperature were set to 5 W/(m$^2$ K) and 25°C, respectively.

## 3 | RESULTS AND DISCUSSION

In this study, the maximum and minimum temperatures were obtained from the whole cell, and the points where the values were got are not the same.

### 3.1 | Effects of plate size $d_1$ when $d_2$ is 1 mm

#### 3.1.1 | Effects on the maximum temperature

Figure 4 shows the maximum temperature of cells in Model 1, Model 2, Model 3, and Model 4. It can be seen that the maximum temperature of the cells firstly increases and then

| TABLE 2 | $d_1$ and $d_2$ values of different models |
|---------|---------------------------------|
|         | Model 1 | Model 2 | Model 3 | Model 4 | Model 5 | Model 6 | Model 7 |
| $d_1$ (mm) | 2 | 3 | 4 | 5 | 3 | 4 | 5 |
| $d_2$ (mm) | 1 | 1 | 1 | 1 | 2 | 3 | 4 |
decreases with the discharge time increasing. This is related to the heating power of batteries. At \( q_m \) of 0.015 kg/s, the highest maximum temperature in Model 3 is 34.8°C, which is 1.5°C higher than that of 33.3°C in Model 4. But the highest maximum temperature in Model 2 is 0.7°C higher than that in Model 1. Because the fluid volume of them is small, flow rate of coolant is slow. When \( q_m \) is 0.025 kg/s, the highest maximum temperature in Model 4 is 30.3°C, which is 4.2°C lower than that of 34.5°C in Model 1. Temperature gradient between battery and cold plate is greater due to the increase of coolant volume. More heat in the battery is transferred to coolant. When the mass flow rate is 0.015 kg/s and 0.025 kg/s, the maximum temperature of the above models is less than 40°C, which meets the thermal management requirement of the Li-ion batteries.

### 3.1.2 Effects on the temperature difference

Temperature difference of cells in Model 1, Model 2, Model 3, and Model 4 is shown in Figure 5. When \( q_m \) is 0.015 kg/s, the maximum temperature difference of Model 3 is 8.9°C, which is 14.42% lower than that of 10.4°C in Model 2. The maximum temperature difference in Model 4 is 7.0°C, which is 19.10% lower than that in Model 3, but the temperature difference is more than 5°C. When \( q_m \) is 0.025 kg/s, the maximum temperature difference in Model 3 is 7.5°C; the maximum temperature difference in Model 4 is 4.1°C, which decreases by 45.33% and meets the thermal requirement.

### 3.2 Effects of plate size when \( \Delta d \) is 1 mm

#### 3.2.1 Effects on the maximum temperature

Figure 6 is the comparison of maximum temperatures in Model 1, Model 5, Model 6, and Model 7. When \( q_m \) is 0.005 kg/s, the maximum temperatures of cells in all cooling system models go up with time. The highest maximum temperature in Model 5 is 40.3°C and that in Model 1 is 40.9°C. The highest maximum temperature in Model 5 is 0.6°C lower than that in Model 1. When \( q_m \) is 0.015 kg/s and 0.025 kg/s, the highest maximum temperatures in Model 5, Model 6, and Model 7 have the same growth trend that the maximum temperature goes up with the increase of \( d_i \) as discharge time passes by. The highest maximum temperature is 35.9°C in Model 7 when \( q_m \) is 0.015 kg/s, but it is only 0.1°C lower than that in Model 6. The highest maximum temperature in Model 7 is 35.3°C at 660 seconds when \( q_m \) is 0.025 kg/s and that in Model 6 is 34.4°C at 570 seconds. Velocity contours of coolant with \( q_m \) of 0.025 kg/s in different models are shown in Figure 7. Velocity of coolant reduces in
cold plate with increase of $d_1$, so coolant flowing into the bottom of the cold plate decreases in the same time. Heat generated by cells accumulates and, therefore, the largest maximum temperature increases with increase of $d_1$ at the same flow rate. When $q_m$ are 0.015 kg/s and 0.025 kg/s, the maximum temperatures of all models are within 40°C at all times. With the increase of $q_m$, Model 1 has the best effects on the maximum temperature.

3.2.2 Effects on the temperature difference

The temperature difference of cells in Model 1, Model 5, Model 6, and Model 7 is shown in Figure 8. It can be seen that the regulations of temperature difference in Model 1, Model 5, Model 6, and Model 7 are similar to that of maximum temperature. The highest temperature difference of cells gradually decreases with the increase of $d_1$. When $q_m$ is 0.005 kg/s, the maximum temperature difference is 13.6°C in Model 7, 12.26% lower than that in Model 1 of 15.5°C, so they are all greater than 5°C. When $q_m$ are 0.015 kg/s and 0.025 kg/s, the temperature differences are also greater than 5°C. And there is a different trend that the temperature difference goes up with the increase of $d_1$. When $q_m$ is 0.025 kg/s, the highest temperature difference in Model 5 of 8.70°C at 570 seconds is 0.02°C lower than that of 8.72°C at 600 seconds in Model 6. And the highest temperature difference in Model 7 is 9.5°C at 690 seconds. So, the time of highest temperature difference delays. This regulation is related to that of maximum temperature, and the reason is the same as maximum temperature.
3.3 | Effects of coolant mass flow rate

3.3.1 | Effects on the maximum temperature

Figure 9 is the diagrams of the effects of different mass flow rates on the maximum temperature in the cavity cold plate cooling systems when \( d_2 = 1 \) mm.

According to Figure 9, in Model 1, the maximum temperature of cells gradually falls with the increase of mass flow rate. When \( q_m = 0.01 \) kg/s, the highest maximum temperature of cells is 38.3°C at 720 seconds, and it decreases by 2.6°C when \( q_m = 0.005 \) kg/s, with the reduction rate of 6.36%. When \( q_m = 0.015 \) kg/s, the highest maximum temperature of the cells appears at 630 seconds, and it decreases by 1.6°C when the \( q_m = 0.01 \) kg/s, with the reduction rate of 4.18%. When \( q_m = 0.02 \) kg/s, the highest maximum temperature is 35.4°C at 570 seconds. And it decreases by 1.3°C when \( q_m = 0.015 \) kg/s, with the reduction rate of 2.99%. When \( q_m = 0.025 \) kg/s, the highest maximum temperature appears at 540 seconds, which is 0.9°C lower, and the reduction rate is 2.54%. It can be seen from the above that in Model 1, with the increase of mass flow rate, the time point at which the highest maximum temperature of cells appears gradually advances and the reduction rate gradually decreases. When \( q_m \) is greater than 0.01 kg/s, the maximum temperature first increases and then decreases with discharge time, and both of them are less than 40°C.

In Model 4, the regulations of cells’ maximum temperature are similar to that in Model 1. When \( q_m = 0.01 \) kg/s, the highest maximum temperature of cells is 35.6°C at 690 seconds, which is 8.25% lower than that of 38.8°C when \( q_m = 0.005 \) kg/s at 900 seconds. When \( q_m = 0.015 \) kg/s, the highest maximum temperature is 6.46% lower than that when \( q_m = 0.01 \) kg/s. When \( q_m = 0.02 \) kg/s, the highest maximum temperature is 31.1°C, which is 2.2°C lower than that when \( q_m = 0.015 \) kg/s at 420 seconds. And when \( q_m = 0.025 \) kg/s, the highest maximum temperature at 360 seconds is 0.8°C lower than that when \( q_m = 0.02 \) kg/s. It can be seen that the time to reach the highest maximum temperature gradually advances. And the maximum temperature of cells is below 40°C at different mass flow rates in Model 4.

Figure 10 is the diagrams showing the effects of different mass flow rates on the maximum temperature of cells in the cavity cold plate cooling systems when \( \Delta d = 1 \) mm.

It can be seen from Figure 10 that as the mass flow rates increase, the maximum temperature of cells shows a downward

![FIGURE 7](image_url) Velocity contours of coolant when \( q_m = 0.025 \) kg/s; (A) Model 1 at 540 s; (B) Model 5 at 540 s; (C) Model 6 at 570 s; (D) Model 7 at 660 s

![FIGURE 8](image_url) Temperature difference of cells in Model 1, Model 5, Model 6, and Model 7; (A) \( q_m = 0.005 \) kg/s; (B) \( q_m = 0.015 \) kg/s; (C) \( q_m = 0.025 \) kg/s
trend, too. In Model 5, when $q_m$ is 0.01 kg/s, the highest maximum temperature is 37.9°C, which is 2.4°C lower than that when $q_m$ is 0.005 kg/s, with the reduction rate of 5.96%. When $q_m$ is 0.015 kg/s, the highest maximum temperature is 1.6°C lower than that when $q_m$ is 0.01 kg/s, with the reduction rate of 4.22%. When $q_m$ is 0.02 kg/s, the highest maximum temperature is 35.1°C, which is 1.2°C lower than that when $q_m$ is 0.015 kg/s, with the reduction rate of 3.31%. When $q_m$ is 0.025 kg/s, the highest maximum temperature is 0.8°C lower than that when $q_m$ is 0.02 kg/s, with the reduction rate of 2.28%. It can be seen that in Model 5, as the mass flow rate of coolant increases, the highest maximum temperature reduction rate of cells reduces slowly.

In Model 6, when $q_m$ is 0.01 kg/s, 0.015 kg/s, and 0.02 kg/s, respectively, the highest maximum temperature of cells reduces by 5.54%, 4.00%, and 2.78%. When $q_m$ is 0.025 kg/s, the highest maximum temperature reduces by 1.71% compared with 35.0°C when $q_m$ is 0.02 kg/s. Similarly, in Model 7, when $q_m$ is 0.01 kg/s, 0.015 kg/s, and 0.02 kg/s, the highest maximum temperature reduces by 5.10%, 3.49%, and 1.95%, respectively. But when $q_m$ is 0.025 kg/s, the highest maximum temperature rises by 0.28% compared with 35.2°C when $q_m$ is 0.02 kg/s.

Although the increase of mass flow rate has different specific effects on temperature of two models’ types, the trend is consistent. The increase of $q_m$ leads to inlet velocity increase in the same models, which subsequently results in the increase of heat dissipation at the bottom of cavity cold plate, and ultimately the gradual decrease of maximum temperature. In the case of adjacent mass flow rates differing by 0.005 kg/s, the growth proportion of $q_m$ decreases as the mass flow rate increases, resulting in a decrease in reduction rate of the highest maximum temperature.

### 3.3.2 Effects on the temperature difference

Figure 11 shows the effects of different mass flow rates on the temperature difference of cells in the cavity cold plate cooling system when $d_2$ is 1 mm. As the discharge time increases, the temperature difference increases first and then decreases. In Model 1, as the mass flow rate increases, the temperature difference gradually drops. However, when $q_m$ increases to 0.025 kg/s, the maximum temperature difference is still greater than 5°C. In addition, as the mass flow rate increases, the reduction of maximum temperature difference decreases gradually.

In Model 4, when $q_m$ is 0.005 kg/s, the maximum temperature difference is 12.9°C at 900 seconds. When the $q_m$ is 0.01 kg/s, the maximum temperature difference is 9.5°C at 720 seconds. When $q_m$ is 0.015 kg/s, the maximum temperature difference is 7.0°C at 600 seconds. When $q_m$ is 0.02 kg/s,
the cells reach the maximum temperature difference of 4.8°C at 450 seconds. When $q_m$ is 0.025 kg/s, the maximum temperature difference is 4.1°C at 390 seconds. In summary, in Model 4, as the $q_m$ increases, the time points at which cells reach the maximum temperature difference advance and they arrive later than the time to reach the highest maximum temperature. In addition, the maximum temperature difference gradually decreases and those of $q_m$ for 0.02 kg/s and 0.025 kg/s are less than 5°C.

Figure 12 shows the effects of different mass flow rates on the temperature difference of cells with $\Delta d$ of 1 mm. According to the figures, the trend of temperature difference with mass flow rate is similar to that of maximum temperature. As $q_m$ increases, the maximum temperature difference of cells gradually drops. But in Model 7, the maximum temperature difference with $q_m$ of 0.025 kg/s is 0.07°C higher than that with $q_m$ of 0.02 kg/s. In addition, the reduction rates of maximum temperature difference reduce with the increase of $q_m$. And all of the maximum temperature differences are more than 5°C.

3.4 | Effects on pressure drop of cavity cold plate in different cooling systems

Cold plate pressure drop is an important indicator to evaluate the power consumption of the cold plate cooling system, so it is necessary to compare and analyze the pressure drop of different cooling models. Figure 13 shows the pressure drop of the cavity cold plate under different mass flow rates and sizes of plate. It can be seen that both the mass flow rates and sizes can affect the cold plate pressure drop. When $q_m$ is 0.005 kg/s, the minimum average pressure drop is 18.8 Pa in Model 7.
When $q_m$ is 0.025 kg/s, the maximum average pressure drop is 5693.2 Pa in Model 1. Therefore, in order to fulfill the requirement of heat dissipation, the cold plate pressure drop can be reasonably controlled by optimizing the structure of cavity cold plate and mass flow rates of coolant, which will reduce the power consumption of the cold plate cooling system.

In order to accurately analyze the pressure drop on different models, the models are divided into two categories: those with $d_2$ of 1 mm and those with $\Delta d$ of 1 mm. Taking Model 1 as the reference, two diagrams in Figure 14 are obtained with mass flow rate of coolant.

Under the premise of meeting heat dissipation requirements of cells, it can be concluded from Figure 14A that when $q_m$ is 0.02 kg/s, the average pressure drop of the cavity cold plate in Model 4 is 807.8 Pa. When $q_m$ increases to 0.025 kg/s, the average pressure drop increases to 1244.6 Pa, 54.07% higher than that with $q_m$ of 0.02 kg/s.

Although the heat dissipation of cavity cold plate with $\Delta d$ of 1 mm is worse, the average pressure drop is smaller. From Figure 14B, when $q_m$ is 0.02 kg/s, the average pressure drop of the cavity cold plate in Model 5 is 901.5 Pa. And when $q_m$ increases to 0.025 kg/s, the average pressure drop is 166.7 Pa, 81.51% lower than that in Model 5. When $q_m$ is 0.025 kg/s, the average pressure drop in Model 6 is 465.4 Pa, 63.63% lower than that in Model 5, and the volume increases by 33.33%. The average pressure drop in Model 7 is 266.7 Pa, with a decrease of 42.69% compared with that in Model 6 and an increase of 59.99% compared with that with $q_m$ of 0.02 kg/s. In the models with $\Delta d$ of 1 mm, inlet velocity significantly decreases with the increase of inlet sizes, so the pressure drop decreases obviously.

The volumes of the cavity cold plate in Model 3 and Model 6 are the same. When $q_m$ is 0.025 kg/s, the average pressure drop in Model 6 is 67.49% lower than that in Model 3. Similarly, in Model 4 and Model 7, when $q_m$ is 0.02 kg/s, the average pressure drop in Model 7 is 79.36% lower than that in Model 4; when $q_m$ is 0.025 kg/s, the average pressure drop in Model 7 is 78.57% lower than that in Model 4. At the same mass flow rate, the inlet flow velocity in the models with $d_2$ of 1 mm is higher than that in the models with $\Delta d$ of 1 mm, and therefore, the energy loss of coolant is greater with the same $d_1$.

4 CONCLUSIONS

In this paper, a number of power Li-ion battery thermal management systems of cavity cold plate are established. The effects of mass flow rate ($q_m$), the thickness of the cavity cold plate ($d_1$) and the width of cold plate inlet ($d_2$) on the maximum temperature, the temperature difference and the pressure drop were studied by using computational fluid dynamics and finite element method at 4 C-rate discharge of power Li-ion battery. The conclusions obtained are as follows:

1. In the models with $d_2$ of 1 mm, the heat dissipation of cavity cold plate is better with the increase of $d_1$, and the reduction of temperature difference increases gradually. In the models with $\Delta d$ of 1 mm, when $q_m$ is greater than 0.015 kg/s, the maximum temperature is lower than 40°C, but the temperature difference does not meet the requirements of thermal management with all kinds of conditions.

2. In the two kinds of models, the reduction of maximum temperature and temperature difference decreases gradually with the increase of $q_m$.

3. Although temperature difference is more than 5°C in the models with $\Delta d$ of 1 mm, the pressure drop obviously falls. So, the structure of cavity cold plate will be optimized to reduce its energy consumption and improve heat dissipation capacity.

To sum up, the Li-ion battery cavity cold plate thermal management system has good heat dissipation capacity after optimizing. At the same time, the pressure drop of the cooling system can be greatly reduced by changing the sizes of cold plate under the same volume, so as to reduce the energy consumption of BTMS. This study provides some guidance for the design of cavity cold plate thermal management systems.

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