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Lithium-Ion Thermal Management Structure Based on Phase Change, Liquid and Heat Sink Cooling

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Received: 04 April 2026; Accepted: 08 June 2026; Published: 27 October 2026

ABSTRACT: This paper focuses on the 21700 lithium-ion battery and addresses the heat dissipation requirements under a 3C high-rate discharge condition. Using numerical simulation methods, the thermal characteristics of the battery are systematically studied, and a multi-layer coupled thermal management system integrating honeycomb heat sink conduction, honeycomb channel liquid cooling, and phase change material (PCM) heat storage is designed and optimized. The research methodology adopts a stepwise multi-parameter optimization approach, progressing from a single cell to a complete system and from the inner layer to the outer layer. Based on the above experiments, the relevant thermophysical parameters of the test battery are obtained, and a high-precision thermal simulation model for a single cell is established. By comparing the simulated surface temperature rise of the battery under 1C–3C discharge with experimental data, it is found that the error between simulation and experiment is less than 4.55%, indicating that this simulation method is reliable and efficient. Subsequently, a battery–heat sink–channel–PCM multi-layer composite thermal management structure is proposed. Following a stepwise research approach from the inside out, the key parameters of the single-cell thermal management system are determined. The cooling effect is optimal when the honeycomb heat sink thickness is 2 mm, a counter-flow channel structure is adopted, and the channel inner diameter is 1.6 mm. The optimized composite system reduces the maximum battery temperature from 54.26°C to 39.85°C, and the temperature difference is reduced to 3.39°C.

KEYWORDS: 21700 lithium-ion battery; battery thermal management; coupled thermal management system; phase change cooling

1 Introduction

With the sustained growth of the global economy and the rapid development of the automotive industry, the global car parc has shown a continuously rising trend. Huang et al. [1] pointed out that as an important part of the national economy, the automotive industry occupies a pillar position in many developed and developing countries, and its scale is still expanding. Taking China as an example, according to information from the National Bureau of Statistics and the Ministry of Transport, the national passenger car parc reached 366 million in 2025, and this huge car base has become an important feature of modern society. The continuous development of the automotive industry has also put great pressure on energy and the environment.

According to Fig. 1, the sharp increase in car ownership has directly intensified the energy crisis and environmental pressure. Lin et al. [2] pointed out that the power source of traditional fuel vehicles comes

from non-renewable fossil fuels, and their continuous consumption is accelerating the depletion of global energy resources. During operation, fuel vehicles emit large amounts of harmful gases and particulate matter. Meanwhile, carbon dioxide (CO₂) and other greenhouse gases emitted by fuel vehicles have been scientifically identified as the primary anthropogenic drivers of global climate system disruptions and the enhanced greenhouse effect.

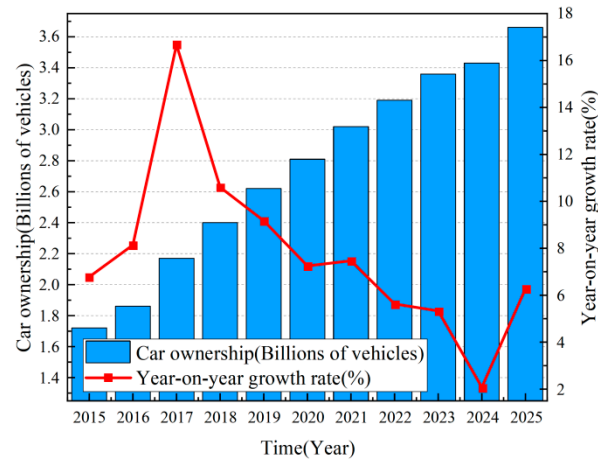


Figure 1: Car ownership and growth rate in recent years [1].

In response to the increasingly severe energy and environmental challenges, promoting the transformation of transportation energy structures and building a low-carbon, efficient transportation system has become a global strategic consensus. Ning et al. and Wang [3,4] pointed out that new energy vehicles (NEVs), with their significant advantages of low energy consumption and low emissions, have gradually become the core breakthrough for green transformation in the transportation sector. Driven by both policy incentives and market demand, China's NEV industry has experienced explosive growth. Data from the National Bureau of Statistics and the Ministry of Transport show that NEV sales have grown rapidly in recent years, with a year-on-year increase of 157.5% in 2021 and a total of 16.49 million vehicles sold in 2025. These data indicate that NEVs have entered a period of rapid development, with broad market prospects in the future.

According to Fig. 2, new energy vehicles (NEVs) are mainly classified into three types based on their energy sources: fuel cell vehicles, battery electric vehicles, and hybrid electric vehicles. Chen et al. [5] pointed out that among these types, fuel cell vehicles face challenges such as high hydrogen storage costs and technical difficulties, leading to slow progress in large-scale adoption. Currently, research and industrialization efforts remain focused on battery electric and hybrid electric vehicles. Huang et al. [6] pointed out that regardless of the technical route, the power battery serves as the energy hub of the vehicle, and its performance greatly affects the safety, power, and economy of the automobile. Lithium-ion batteries, with their advantages of high energy density and long cycle life, have become the most ideal power source for current NEVs. However, Tan et al. [7] indicated that temperature significantly affects the electrochemical performance and cycle life of lithium batteries. To ensure that lithium batteries operate in an optimal state, their temperature must be strictly controlled below 45°C, and the temperature difference between cells should not exceed 5 K [8,9]. This technical need has spurred research into battery thermal management systems (BTMS). By continuously monitoring the battery's real-time temperature, such systems can autonomously toggle between cooling and preheating modes, thereby ensuring that both the absolute temperature and the temperature differential stay within safe thresholds. Feng et al. and Alawi

et al. [10,11] pointed out that when electric vehicles are in operation, thermal management is essential to ensure power battery performance and to prevent safety accidents caused by high temperatures or thermal runaway.

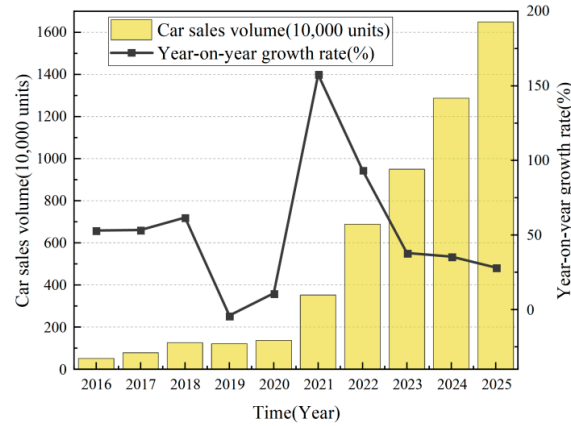


Figure 2: Sales volume and growth rate of new energy vehicles [4].

Asiri et al. [12] pointed out that as batteries continue to evolve toward higher energy density and higher power density, their thermal management issues become increasingly prominent. During rapid charging and discharging, batteries generate a large amount of heat. If this heat is not dissipated effectively and promptly, it can lead to performance degradation, shortened lifespan, and in severe cases, even trigger safety accidents such as thermal runaway [13]. This poses a realistic challenge to the overall reliability and environmental value of NEVs. Therefore, in the process of promoting the popularization of NEVs, the thermal safety design and intelligent thermal management of battery systems must be taken as the core direction of technical breakthroughs. Through integrated solutions that combine material innovation, system optimization, and intelligent control, a stable and efficient battery cooling system must be established. Such a system will enable NEVs to achieve low carbon emissions while offering safety and durability comparable to—or even exceeding—that of conventional fuel vehicles. In doing so, NEVs can truly become a reliable pillar supporting the global green transformation of transportation and sustainable development.

2 Thermal Model of Lithium-Ion Battery

2.1 Heat Generation Model of Lithium-Ion Battery

During operation, extremely complex electrochemical reactions occur inside a lithium-ion battery [14,15]. In the process of these electrochemical reactions, redox reactions take place between the positive/negative electrodes and the electrolyte, and ions move back and forth between the two electrodes, thereby generating heat [16,17]. From the structure and working principle of lithium-ion batteries, it can be understood that the heat generation of a lithium-ion battery mainly includes: chemical reaction heat Q_r , ohmic internal resistance Joule heat Q_j , battery polarization heat Q_p , and battery side reaction heat Q_s [18,19].

$$Q = Q_r + Q_j + Q_p + Q_s \quad (1)$$

(1) Chemical reaction heat

The electrochemical reaction of a lithium-ion battery is a thermodynamic process accompanied by entropy change. Heat of entropy change is generated when the entropy change of the electrochemical reaction of the electrode material is non-zero. The magnitude and direction of the entropy-related heat

depend on the specific battery reaction process and the thermodynamic properties of the substances involved. Consequently, this heat may appear as either endothermic or exothermic. Battery reaction heat refers to the reversible entropy change heat generated during the charging and discharging of a lithium-ion battery caused by the intercalation and de-intercalation of lithium ions in the crystal lattices of the positive and negative electrodes [20]. This part of the heat is reversible in an electrochemical sense; that is, the heat absorption and release effects can be converted into each other during the charging and discharging processes [21]. Its formula is as follows:

$$Q_r = T \frac{\Delta S}{\Delta t} \quad (2)$$

where T is the absolute temperature of the battery (K); ΔS is the entropy change of the electrochemical reaction (J/(mol·K)); Δt is the reaction time (s).

The entropy change formula is as follows:

$$\Delta S = -\frac{d(\Delta G)}{dT} = \frac{d(nFE)}{dT} = nF \frac{dE}{dT} \quad (3)$$

where G is Gibbs free enthalpy (J); n is the amount of substance of electrons (mol); F is the Faraday constant, 96,485 C/mol; E is the electromotive force (V).

Since the current is:

$$I = \frac{nF}{\Delta t} \quad (4)$$

Combining the above equations, the expression for battery reaction heat Q_r is obtained as:

$$Q_r = IT \frac{dE}{dT} \quad (5)$$

(2) Ohmic internal resistance Joule heat

There is a certain resistance inside the lithium-ion battery. The passage of current through a battery induces collisions between charge carriers and the lattice, resulting in energy dissipation in the form of heat. According to Joule's law, current generates heat on the ohmic internal resistance of the battery, and this heat is ohmic heat. Ohmic internal resistance heat is the most basic source of irreversible heat in batteries. Ohmic internal resistance includes the resistance of electrode materials, electrolyte, separators, and other components, as well as the contact resistance between them [22].

$$Q_j = I^2 R_e \quad (6)$$

where R_e is the ohmic internal resistance of the battery (Ω); I is the operating current (A).

This formula can be used for the calculation of ohmic internal resistance Joule heat during the discharge process of the 21700 lithium-ion battery.

(3) Battery polarization heat

Polarization heat is the additional energy loss caused by polarization phenomena during the charging and discharging of a lithium-ion battery, and it is an important component of irreversible heat generation. In lithium-ion battery charge/discharge processes, the sluggish kinetics of surface electrochemical reac-

tions lead to a deviation of the electrode potential from its equilibrium potential, a phenomenon known as polarization.

Polarization is divided into ohmic polarization, concentration polarization, and electrochemical polarization. Extra energy loss occurs during the polarization process, and these losses are released in the form of heat, forming polarization heat [23].

$$Q_p = I^2 R_p \quad (7)$$

where I is the operating current (A); R_p is the polarization internal resistance (Ω).

(4) Battery side reaction heat

Side reaction heat is heat generated by irreversible chemical side reactions of lithium-ion batteries under non-ideal operating conditions (such as high temperature, overcharging, aging), characterized by concentrated heat release and potential thermal runaway risks. During the use of lithium-ion batteries, aside from the normal charge–discharge reactions, several side reactions may take place, including decomposition of the electrolyte, chemical reactions between the electrode material and the electrolyte, and internal self-discharge. These side reactions are generally irreversible and accompanied by heat release; however, the heat generated is typically small and thus considered negligible [24].

In summary, the total heat generation during the battery charging and discharging process can be expressed as:

$$Q = IT \frac{dE}{dT} + I^2 R_e + I^2 R_p \quad (8)$$

Eq. (8) can be used for the subsequent calculation of the heat generation of the 21700 lithium-ion battery and serves as the core fundamental equation for the design of lithium-ion battery thermal management systems, numerical simulation of the thermal field of battery packs, optimization of charging and discharging strategies, and the design of thermal runaway warning and protection.

2.2 Thermal-Related Models for Lithium-Ion Batteries

Heat is generated during the charging and discharging process of a lithium-ion battery. Bernardi et al. [25] proposed a model regarding heat generation rate, which is currently the most commonly used calculation model for estimating battery heat generation. In this formula, the first term is the irreversible reaction heat of Joule heat and polarization heat, and the second term is the reversible heat of the electrochemical reaction. During the charging and discharging process, the sign must be considered for the reversible heat part.

$$q = \frac{I}{V_b} \left[(E - U) + T \frac{dE}{dT} \right] \quad (9)$$

where I is the battery charging and discharging current (A); V_b is the battery volume (m^3); E is the open-circuit voltage (V); U is the operating voltage (V); T is the battery temperature (K). Substituting in Eq. (9) $(E - U)$ as the product of current and internal resistance into the equation yields:

$$q = \frac{1}{V_b} \left[I^2 (R_j + R_p) + IT \frac{dE}{dT} \right] \quad (10)$$

This equation clearly indicates that the battery heat generation rate is composed of two parts: $I^2(R_j + R_p)$ is the sum of Joule heat and polarization heat, which is proportional to the square of the current; $ITdE/dT$ is the reversible reaction heat, which is proportional to the first power of the battery and temperature. This expression serves as the input for the volumetric heat generation rate in the subsequent thermal simulation. R_e and R_p can be obtained through HPPC internal resistance experiments, and dE/dT is the open-circuit voltage temperature coefficient. For ternary lithium-ion batteries, its typical value is between -0.4 and -0.2 mV/K, manifesting as endothermic (negative value) during discharge. In actual calculations, the relationship between battery internal resistance and temperature and SOC obtained from experiments can be used to process R_e and R_p as functions of temperature to consider the electrochemical-thermal two-way coupling effect.

The heat exchange between the battery outer surface and the ambient environment is defined by Newton's law of cooling:

$$-k \frac{\partial T}{\partial n} = h(T_s - T) \quad (11)$$

where n is the normal vector of the battery surface, h is the convective heat transfer coefficient ($\text{W}/(\text{m}^2 \cdot \text{K})$), T_s is the battery surface temperature (K), and T is the ambient temperature.

This equation provides the fundamental basis for key applications in lithium-ion battery technology, including pack-level thermal simulation, thermal management system design, cell thermal runaway risk assessment, and charge/discharge rate optimization.

2.3 Lithium-Ion Battery Temperature Rise Experiment

2.3.1 Experimental Equipment

The experimental equipment for the 21700 lithium-ion battery includes a high-precision battery test system, a constant temperature chamber, temperature sensors, and a data acquisition system. The test system is capable of completing charging and discharging operations of lithium-ion batteries under different powers. Based on experimental safety and research objectives, the focus is on conducting discharge tests at 1C, 2C, and 3C rates for the 21700 lithium-ion battery; collecting surface temperature data of the 21700 lithium-ion battery under various operating conditions; and recording the current and voltage variation patterns of the 21700 lithium-ion battery under different operating conditions.

The experiment focuses on the surface temperature distribution and internal impedance changes of a single cell during the charging and discharging process. The acquired data will provide a basis for heat source parameters for the numerical simulation of the battery pack.

According to Fig. 3, the discharge temperature rise characteristic experiment is a foundational experiment for obtaining the pattern of surface temperature changes of a battery during constant current discharge. By measuring the temperature-time curves of the battery at 1C, 2C, and 3C rates, the thermal behavior characteristics of the battery can be quantitatively mastered. It is possible to quantitatively determine the surface temperature variation patterns of the 21700 battery under 1C, 2C, and 3C discharge rates, and to verify the conclusion that the battery temperature exceeds the safety threshold under 3C discharge conditions. This underscores the need for thermal management system design and offers benchmark data on surface temperature evolution over time as well as discharge rate, which can be used for subsequent simulation model verification and validation.

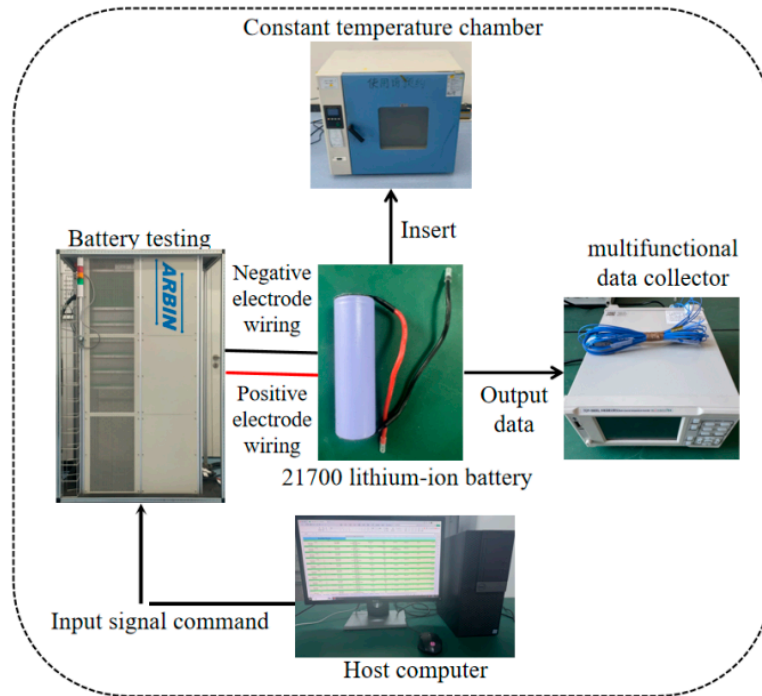


Figure 3: Battery thermal performance test bench.

To investigate the surface temperature changes of the 21700 lithium-ion battery at an initial temperature of 25°C under different discharge rates, this experiment uses the aforementioned testing and acquisition equipment to conduct temperature rise characteristic tests. Given the temperature inhomogeneity caused by the internal structure of the battery, the experiment measures temperatures at three points: upper, middle, and lower parts of the battery, using the average of the three to characterize the battery surface temperature. The battery connection and measurement point locations are shown in the Fig. 4 below.

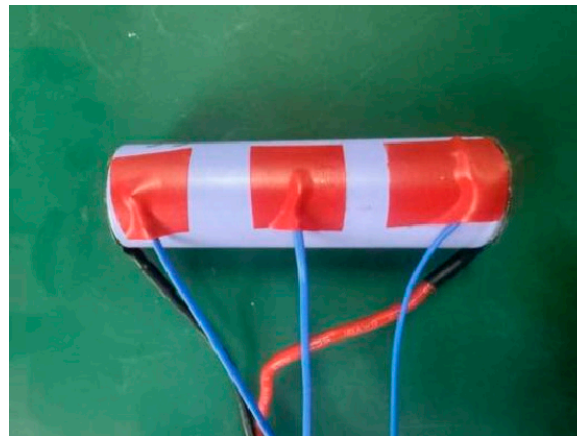


Figure 4: Battery cable connection position.

2.3.2 Experimental Steps

- (1) Fully discharge the 21700 lithium-ion battery;
- (2) Place the battery in the constant temperature chamber for 1 h to make it consistent with the ambient temperature;

- (3) Charge the battery at a 1C rate current (4 A) until the voltage reaches the upper cut-off voltage of 4.2 V, then perform constant voltage charging on the battery at 4.2 V, and stop charging when the current reaches 0.01C;
- (4) Let it stand for 1 h to make the battery temperature consistent with the constant temperature chamber;
- (5) Perform constant current discharge at a 1C rate until the lower cut-off voltage of 2.75 V is reached;
- (6) Place the battery in the constant temperature chamber for 1 h to make it consistent with the ambient temperature;
- (7) Organize the battery surface temperature data collected by the multi-function sensor;
- (8) Repeat steps 3–7, changing the discharge rate to 2C and 3C in turn to complete the discharge experiments at different rates.

2.3.3 Experimental Result Analysis

From the temperature curves of a 21700 lithium-ion battery during 1C–3C discharge, under the same ambient temperature, a higher discharge rate leads to a faster surface temperature rise and a higher peak temperature. This is because the discharge rate is proportional to the current, while the primary heat source—ohmic Joule heat—is proportional to the square of the current, so the heat generation power increases exponentially with rate. Moreover, a higher current intensifies polarization effects, accelerates electrochemical reactions, and sharply raises the Li^+ concentration gradient at the electrode interfaces. Consequently, concentration polarization, electrochemical polarization, and ohmic polarization are all enhanced simultaneously. The electrical energy lost in the polarization processes is converted almost entirely into heat, leading to considerable additional polarization heat generation during high-rate discharge. This heat accumulation ultimately appears as a rapid climb in the battery surface temperature. The constant temperature chamber has a temperature accuracy of $\pm 0.5^\circ\text{C}$. The multifunctional data acquisition system has a measurement accuracy of $\pm 0.2\%$ full scale (FS), and the battery charge-discharge tester has an accuracy of $\pm 0.05\%$ FS. The combined measurement uncertainty of all experimental data is within 5%.

According to Fig. 5, under the same ambient temperature and fixed discharge rate, the battery surface temperature rises continuously and peaks at discharge termination. This is because, during constant current discharge, internal heat generation from electrochemical reactions and polarization always exceeds heat dissipation via natural convection and thermal radiation, causing heat to accumulate progressively over time. Regarding the temperature rise rate pattern, the battery exhibits three distinct stages: a rapid increase in the early stage, a deceleration in growth during the middle stage, and a renewed rapid rise in the late stage. In the early discharge stage, the internal electrochemical reactions trigger quickly, causing the heat generation rate to momentarily exceed the heat dissipation rate, which leads to a sharp temperature rise. As discharge proceeds, the temperature difference increases, thereby enhancing the heat dissipation efficiency through conduction and convection. Consequently, the gap between heat generation and dissipation gradually narrows, resulting in a slowdown of the temperature rise rate. In the middle to late discharge stages, the battery's state of charge (SOC) continues to decline, leading to a sharp rise in electrode polarization and a significant increase in additional heat generation. The heat generation rate once again far surpasses the heat dissipation rate, ultimately causing the temperature rise rate to rebound rapidly, as reflected by a marked increase in the slope of the temperature curve.

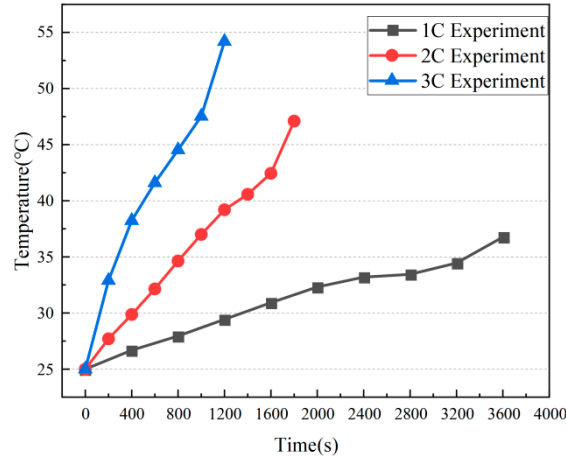


Figure 5: Lithium-ion battery 1C to 3C discharge.

This paper mainly investigates the thermal management problem of lithium-ion batteries under 3C discharge conditions. Under 3C discharge conditions, the temperature at the end of discharge approaches 55°C, which far exceeds the safe operating temperature of 45°C for the battery. Therefore, it is necessary to conduct thermal management on the battery to ensure the safe operation of the battery. This verification result is highly consistent with the research by Rezgui et al. [26]. The ohmic resistance, polarization resistance, open-circuit voltage and entropy coefficient in the model are strongly dependent on the state of charge (SOC) and defined as dynamic parameters instead of constant values. The open-circuit voltage varies nonlinearly with SOC, which dominates the fundamental characteristics of reversible heat generation of the battery. The ohmic resistance and polarization resistance increase gradually as SOC decreases, serving as the main sources of irreversible Joule heat and polarization heat. The entropy coefficient fluctuates dynamically with SOC, which directly affects the calculation accuracy of reversible heat generation under low-rate and ambient temperature conditions. All electrical and thermophysical parameters in this work are fitted based on the measured characteristics and published experimental data of 21700 lithium-ion batteries. The dynamic variation of parameters with SOC is fully taken into account, which can accurately reflect the actual heat generation characteristics of the battery during 1C–3C discharge and guarantee the computational reliability and simulation accuracy of the Bernardi heat generation model.

2.4 Lithium-Ion Battery Temperature Rise Simulation

To verify the accuracy of the numerical model, a simulation of the 21700 lithium-ion battery is built, and its predictions are compared with both the baseline and phase-change temperature rise experiments.

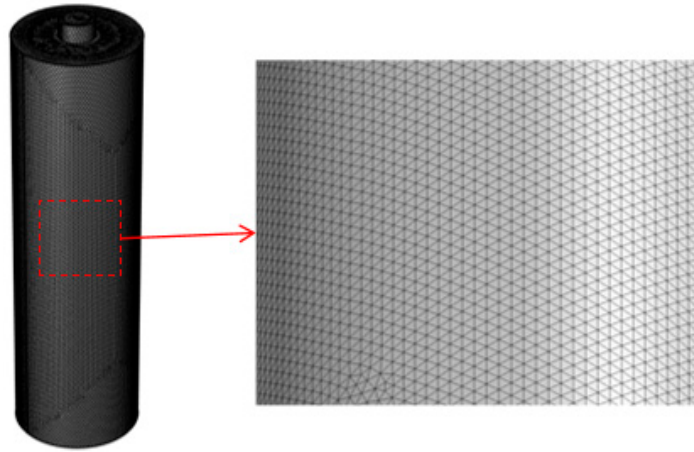
2.4.1 Simulation Model and Meshing

In terms of structure, a lithium-ion battery is composed of a positive electrode, negative electrode, separator, electrolyte, shell, etc. However, in the simulation process, restoring each part separately would consume a large amount of computing resources. Usually, its battery parameters are as follows:

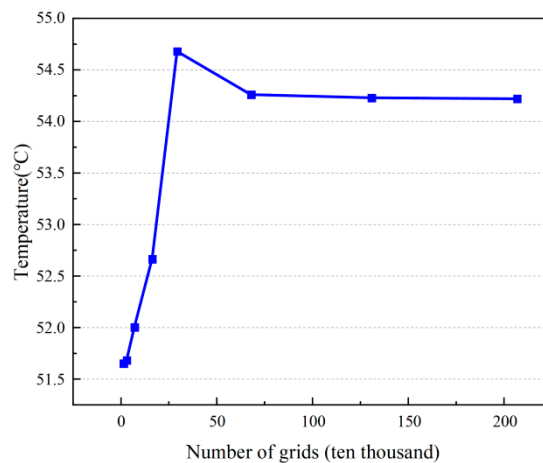
In the simulation process, the battery model is simplified as a cylinder with a diameter of 21 mm and a height of 70 mm. There is a cell core with a radius of 2 mm at the center of the battery, and a cylindrical terminal with a radius of 3 mm and a height of 3 mm above it, the remaining parameters are listed in Table 1. The simplified model and mesh division are shown in the Fig. 6 below:

Table 1: 21700 lithium-ion battery specifications.

Battery Parameter	Reference Value
Density (kg/m^3)	2850
Specific Heat Capacity ($\text{J/(kg}\cdot\text{K)}$)	1050
Transverse Thermal Conductivity ($\text{W/m}\cdot\text{K}$)	21.55
Radial Thermal Conductivity ($\text{W/m}\cdot\text{K}$)	1.55

**Figure 6:** 21700 Lithium-Ion Battery Simplified Model and Mesh Division.

Before simulation, mesh independence verification must be performed. In this study, the Comsol finite element simulation software is used to simulate the model. To ensure the accuracy of the simulation without wasting computing resources, mesh independence verification is performed first. Based on the battery model, 8 mesh schemes are proposed. The mesh numbers used are 13,250; 29,627; 70,364; 162,577; 292,466; 678,205; 1,304,876; and 2,063,349. The tests are conducted at an ambient temperature of 25°C , with the battery discharged at rates ranging from 1C to 3C. The relationship between the battery surface temperature and the mesh number is shown in the Fig. 7 below.

**Figure 7:** Grid Independence Verification.

Analyzing the figure above, the predicted battery temperature rises rapidly with mesh refinement at low grid counts and stabilizes after the mesh number reaches 678,205 (54.26°C). This trend is caused

by the insufficient resolution of boundary layer and near-wall temperature gradients in coarse grids, which underestimates convective heat transfer and leads to artificially low temperatures. With mesh refinement, these regions are adequately resolved, and further increasing the mesh number results in negligible temperature changes, indicating that the solution is mesh-independent. To balance simulation accuracy and computing resources, 678,205 meshes are finally selected as the calculation scheme. At this mesh element count, the global minimum orthogonal quality is 0.83, whereas the minimum orthogonal quality of the primary computational domain equals 0.87, meeting the requirements for boundary layer heat transfer calculations.

2.4.2 Simulation Result Analysis

The simulation result adopts the average surface temperature of the battery, and the final result is used as the battery surface temperature. The ambient temperature is set to remain unchanged at 25°C. The battery surface temperature rise characteristics under 1C–3C discharge rates are tested, respectively. The total heat generation refers to Eqs. (8) and (10). The comparison between simulation and experimental battery surface temperature rise curves is shown in Figs. 8 and 9 below.

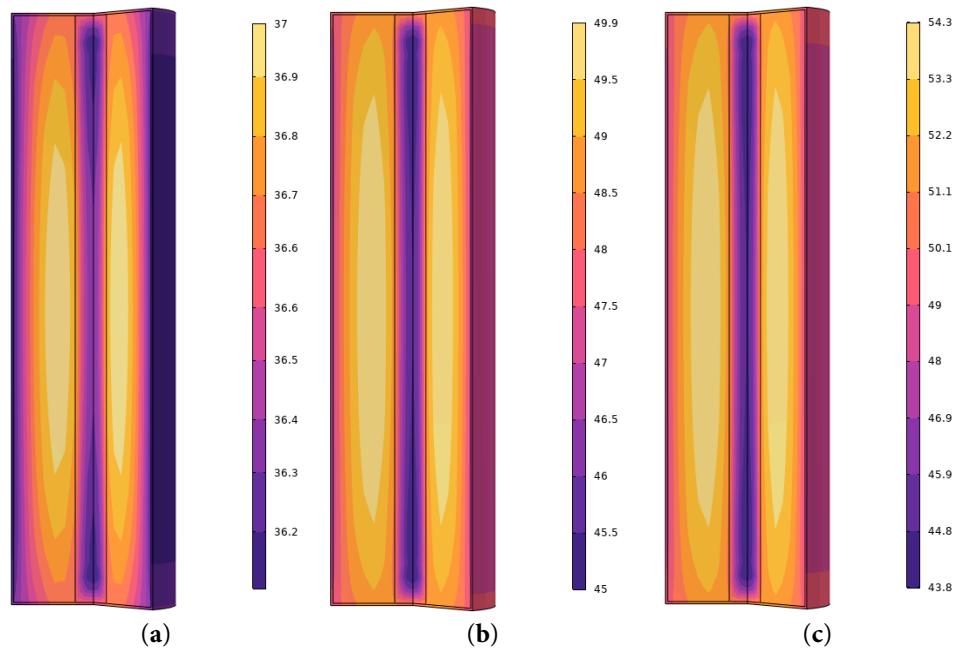


Figure 8: 21700 Lithium-Ion Battery 1C–3C Discharge Simulation. (a) 1C rate discharge; (b) 2C rate discharge; (c) 3C rate discharge.

As shown in Fig. 8 above, during the battery discharge process, due to thermal conductivity anisotropy and differences in external heat dissipation conditions, its temperature distribution presents obvious three-dimensional spatial distribution characteristics. Inside the battery, heat generation is primarily concentrated in the central region of the cell core and is conducted radially outward. Due to the low radial thermal conductivity of the wound structure, outward heat transfer is impeded, resulting in a high temperature at the radial center and lower temperatures in the surrounding areas. Along the battery axis, the heat conduction path to both ends is long and the dissipation conditions are poor, leading to a temperature distribution characterized by low temperatures at the ends and a high temperature in the middle.

The comparison between numerical simulation and experimental results is as follows:

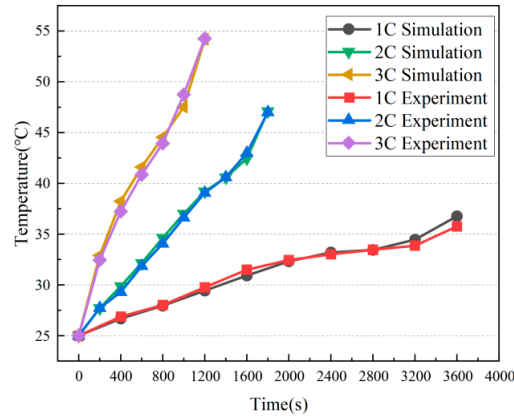


Figure 9: 21700 Lithium-Ion Battery Simulation and Experimental Comparison.

As shown in Fig. 9 above, in the early stage of simulation discharge, the temperature rises rapidly, and the magnitude of temperature rise is large. As the discharge time proceeds, the battery temperature rise slows down in the middle stage, and the growth rate of temperature rise begins to decrease. In the late stage of battery discharge, the temperature rise increases again, and the magnitude of the growth rate rises again. Under the same ambient temperature conditions, within the same discharge time, the higher the discharge rate, the faster the battery surface temperature rise, and the higher the surface temperature at the end of discharge.

The trend of the simulation results is generally the same as that of the experimental results. The maximum error between simulation and experiment at the 1C discharge rate is 2.78%, at the 2C discharge rate is 3.62%, and at the 3C discharge rate is 4.55%. Therefore, this simulation method is relatively accurate and has high simulation accuracy. This simulation method will be adopted in subsequent research.

3 Battery Cooling Structure Design and Simulation

The experimental research in Section 2 shows that the surface temperature of the 21700 lithium-ion battery can reach 55°C under 3C high-rate discharge conditions, far exceeding its safe operating temperature upper limit, creating an urgent need to design an efficient thermal management system. A single thermal management approach struggles to meet the dual requirements of suppressing transient thermal shocks and achieving steady-state heat dissipation. Active liquid cooling offers strong heat exchange capability but suffers from high energy consumption and system complexity. Passive phase-change thermal storage can effectively mitigate transient temperature rises; however, it is limited by low thermal conductivity and becomes ineffective once fully melted. Addressing the above problems, this chapter proposes a composite thermal management system coupling a regular hexagonal honeycomb heat sink, nested microchannels, and phase change material (PCM). On this basis, this chapter systematically establishes a complete coupled heat transfer theoretical model from a single cell to a battery pack, providing a theoretical basis for subsequent numerical simulation and structural optimization.

3.1 Battery Cooling Structure Design

The system leverages the high thermal conductivity of the aluminum alloy honeycomb skeleton for rapid heat diffusion, employs phase change material filled in the voids to absorb transient thermal shock via solid-liquid phase change, and uses the cooling liquid in the nested channels to achieve steady-state heat dissipation through forced convection. The specific structure is shown in the Fig. 10 below.

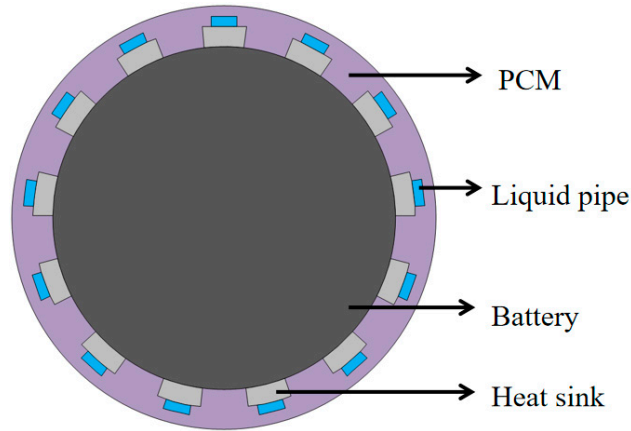


Figure 10: Simplified Model of Battery Thermal Management Structure.

The overall system is cylindrical and is divided radially from the inside out into four distinct computational domains. The core structure consists of a regular hexagonal honeycomb skeleton. Owing to its periodic arrangement, a single honeycomb unit can be selected as the computational domain for simplified analysis. The functional positioning and geometric characteristics of each computational domain are as follows:

(1) Battery solid domain

This domain is a cylindrical lithium-ion battery, the only internal heat source in the system. The battery adopts a wound structure, thus having significant thermal conductivity anisotropy.

(2) Aluminum alloy heat sink solid domain

This domain is a regular hexagonal honeycomb structure made of heat-dissipating metal, arranged close to the outer wall of the battery. The high thermal conductivity of the honeycomb skeleton makes it the main channel for heat transfer from the battery to the outside, playing the role of heat spreading and temperature equalization.

(3) Nested channel fluid domain

This domain is an independent cavity sharing the honeycomb structure of the heat sink, serving as a channel for the flow of cooling liquid. The cooling liquid performs forced convection inside the channel to carry heat out of the system.

(4) Phase change material domain

This domain is the phase change material, filled in all voids between the heat sink solid body and the channel wall. The phase change material absorbs a large amount of latent heat through solid-liquid phase change to suppress transient thermal shock.

3.2 Boundary Conditions for Individual Domains and Grid Independence

3.2.1 Basic Physical Model Assumptions

To strike a balance between the physical accuracy and mathematical tractability of the model, the following basic assumptions are proposed. These assumptions are based on the understanding of the thermophysical processes of lithium-ion batteries and reasonable simplifications for engineering applications, serving as the foundation for the subsequent derivation of control equations.

(1) Physical property constant assumptions

- (a) The thermal properties of the battery and aluminum alloy are constant and do not change with temperature.
- (b) The battery has anisotropic thermal conductivity in the radial and axial directions.
- (c) The thermal properties of solid/liquid paraffin phase change materials are constant in segments.
- (d) The coolant is an incompressible Newtonian fluid, and its thermal properties do not change with temperature.

(2) Flow assumptions

- (a) The coolant in the channel is axial laminar flow. The basis for judging laminar flow is the Reynolds number. For flow in a circular pipe:

$$Re = \frac{\rho_f u d_h}{\mu_f} \quad (12)$$

where ρ_f is the density of the coolant (kg/m^3); u is the flow velocity of the coolant (m/s); d_h is the hydraulic diameter of the channel (m); μ is the dynamic viscosity ($\text{Pa}\cdot\text{s}$). Taking one of the operating conditions in this study as an example, the coolant flow velocity is 0.015 m/s , the channel hydraulic diameter is 1.6 mm , the density of pure water is 997 kg/m^3 , and the dynamic viscosity is $8.9 \times 10^{-4} \text{ Pa}\cdot\text{s}$. Calculating gives $Re \approx 27$, which is far less than the critical value of 2300, confirming it as laminar flow.

- (b) Ignore radial and circumferential flow, and only consider the axial velocity component.

(3) Interface and boundary assumptions

- (a) The existence of contact thermal resistance will hinder heat transfer, but its exact value is difficult to determine and varies with factors such as pressure, temperature, and surface roughness. In engineering simplification, it is assumed that the interface is in ideal contact, i.e.,:

$$T_1 = T_2, \lambda_1 \frac{\partial T_1}{\partial n} = \lambda_2 \frac{\partial T_2}{\partial n} \quad (13)$$

- (b) The channel avoids the velocity no-slip boundary condition. According to basic fluid mechanics theory, the relative velocity of a viscous fluid on a solid wall is zero, i.e.,:

$$u_z|_{\text{wall}} = 0 \quad (14)$$

This condition forms a velocity boundary layer near the wall and is the basis for solving the momentum equation.

- (c) Ignore radiative heat transfer. The amount of radiative heat transfer is proportional to the fourth power of the temperature (Stefan-Boltzmann law).
- (d) The outer wall of the heat sink is set as an adiabatic boundary.

Assuming the system is well wrapped and no heat is dissipated to the environment, i.e.,:

$$\frac{\partial T}{\partial n} = 0 \quad (15)$$

This assumption closes the system boundary so that heat is discharged only through the two paths of coolant and phase change material. If considering actual heat dissipation, convective boundary conditions can be supplemented in the subsequent model.

(4) Heat generation assumption

The battery adopts the Bernardi uniform heat generation model, where the volumetric heat generation rate is uniformly distributed within the battery domain.

(5) Phase change assumption

The paraffin phase change occurs within a fixed temperature interval. The solidus temperature is 37°C, and the liquidus temperature is 40°C. The supercooling phenomenon is ignored.

3.2.2 Battery Domain Control Equation

(1) Derivation of heat conduction equation

Heat transfer inside the battery follows Fourier's Law. Taking a micro-element body, its radial range is from r to $r + dr$, and its axial range is from z to $z + dz$.

The radial heat flux density is:

$$q_r = -\lambda_{batt,r} \frac{\partial T}{\partial r} \quad (16)$$

The axial heat flux density is:

$$q_z = -\lambda_{batt,z} \frac{\partial T}{\partial z} \quad (17)$$

The net heat flow through the radial interface is:

$$q_r \cdot (rd\theta dz)|_r - q_r \cdot (rd\theta dz)|_{r+dr} = -\frac{\partial(rq_r)}{\partial r} dr d\theta dz \quad (18)$$

The net heat flow through the axial interface is:

$$q_z \cdot (rd\theta dz)|_z - q_z \cdot (rd\theta dz)|_{z+dz} = -\frac{\partial(rq_z)}{\partial z} dr d\theta dz \quad (19)$$

The rate of change of internal energy of the micro-element body is:

$$\rho_{batt} C_{p,batt} \frac{\partial T_b}{\partial t} \cdot r dr d\theta dz \quad (20)$$

The internal heat generation rate is:

$$q_{gen} \cdot r dr d\theta dz \quad (21)$$

According to energy conservation, the rate of change of internal energy equals the net heat inflow plus the internal heat source. Organizing yields:

$$\rho_{batt} C_{p,batt} \frac{\partial T_b}{\partial t} = \frac{1}{r} \frac{\partial}{\partial r} \left(\lambda_{batt,r} r \frac{\partial T_b}{\partial r} \right) + \frac{\partial}{\partial z} \left(\lambda_{batt,z} r \frac{\partial T_b}{\partial z} \right) + q_{gen} \quad (22)$$

where the term on the left is the rate of change of internal energy inside the battery. The first term on the right is the net inflow of radial heat conduction, the second term is the net inflow of axial heat conduction, and the third term is the internal heat source term. This equation describes the evolution law of the battery temperature field over time and is the foundation for subsequent numerical solution.

3.2.3 Heat Sink Domain Control Equation

The aluminum alloy heat sink serves as a high thermal conductivity channel. There is no internal heat source inside it, and heat transfer is dominated solely by thermal conduction. Using the same derivation method as for the battery domain, the control equation for the heat sink temperature field can be obtained:

$$\rho_{Al} C_{p,Al} \frac{\partial T_{Al}}{\partial t} = \frac{1}{r} \frac{\partial}{\partial r} \left(\lambda_{Al} r \frac{\partial T_{Al}}{\partial r} \right) + \frac{\partial}{\partial z} \left(\lambda_{Al} r \frac{\partial T_{Al}}{\partial z} \right) \quad (23)$$

This equation is similar in form to the battery heat conduction equation, with the difference being the absence of an internal heat source term.

3.2.4 Channel Domain Control Equation

The flow and heat transfer of the coolant inside the channel need to be described by simultaneously solving the three conservation equations of mass, momentum, and energy. These equations are derived based on the basic principles of fluid mechanics and heat transfer.

(1) Mass conservation equation

For an incompressible fluid, density is constant, and the mass conservation equation simplifies to the divergence of the velocity field being zero. In cylindrical coordinates, the continuity equation is:

$$\frac{1}{r} \frac{\partial(r u_r)}{\partial r} + \frac{\partial u_z}{\partial z} = 0 \quad (24)$$

For axial laminar flow, the radial velocity component is much smaller than the axial velocity and can be neglected, i.e., $u_r \approx 0$. At this time, the continuity equation simplifies to:

$$\frac{\partial u_z}{\partial z} = 0 \quad (25)$$

From this, it can be deduced that the axial velocity is constant along the flow direction and equal to the inlet velocity. This indicates that in the fully developed region, the velocity profile no longer changes along the flow direction.

(2) Momentum conservation equation

For steady incompressible fluid, ignoring gravity effects, the axial momentum equation in cylindrical coordinates is:

$$\rho_f \left(u_r \frac{\partial u_z}{\partial r} + u_z \frac{\partial u_z}{\partial z} \right) = -\frac{\partial p}{\partial z} + \mu_f \left[\frac{1}{r} \frac{\partial}{\partial r} \left(r \frac{\partial u_z}{\partial r} \right) + \frac{\partial^2 u_z}{\partial z^2} \right] \quad (26)$$

For fully developed axial laminar flow, the velocity profile is constant along the axial direction, so $\frac{\partial u_z}{\partial z} = 0$; meanwhile, ignoring radial flow, $u_r = 0$. Therefore, the inertial terms are zero. The momentum equation simplifies to:

$$0 = -\frac{dp}{dz} + \mu_f \frac{1}{r} \frac{\partial}{\partial r} \left(r \frac{\partial u_z}{\partial r} \right) \quad (27)$$

This equation describes the equilibrium relationship between the pressure gradient and viscous forces. Integrating yields the radial parabolic velocity distribution:

$$u_z(r) = \frac{1}{4\mu_f} \left(-\frac{dp}{dz} \right) (R_w^2 - r^2) \quad (28)$$

where R_w is the channel radius, and the maximum flow velocity occurs at the channel center $r = 0$. The average velocity is half of the maximum velocity. Eq. (28) is the analytical solution for the radial distribution of axial velocity for a fully developed laminar flow of an incompressible Newtonian fluid in a circular pipe. This equation accurately describes the velocity distribution law in the fully developed section of circular pipe laminar flow and is the core theoretical basis for the pressure drop calculation, flow distribution, and convective heat transfer capacity design of the liquid cooling channel.

(3) Energy conservation equation

The fluid energy equation describes the spatiotemporal evolution of the temperature field. According to energy conservation, the rate of change of internal energy of the micro-element body is equal to the sum of the convection term, conduction term, and viscous dissipation term. For an incompressible fluid, ignoring viscous dissipation, the fluid energy equation in cylindrical coordinates is:

$$\rho_f C_{p,f} \left(\frac{\partial T_f}{\partial t} + u_z \frac{\partial T_f}{\partial z} \right) = \lambda_f \left[\frac{1}{r} \frac{\partial}{\partial r} \left(r \frac{\partial T_f}{\partial r} \right) + \frac{\partial^2 T_f}{\partial z^2} \right] \quad (29)$$

where the first term on the left is the rate of change of internal energy of the fluid, and the second term is the heat convection term caused by axial flow. The terms on the right are the conduction terms, including radial and axial heat conduction. This equation reflects the influence of fluid flow on the temperature field, achieving the coupling of flow and heat transfer.

3.2.5 Phase Change Material Domain Control Equation

The heat transfer process of the phase change material involves solid-liquid phase change. The enthalpy-porosity model is used to uniformly describe the heat transfer in the solid state, phase change region, and liquid state, without the need to track the solid-liquid interface. Its core idea is to implicitly include the phase change latent heat in the enthalpy variable and obtain the temperature field by solving the enthalpy field.

(1) Enthalpy-temperature-liquid fraction relationship

The total enthalpy H of paraffin is composed of two parts: sensible enthalpy and latent heat of phase change:

$$H(T) = \int_{T_{ref}}^T C_{p,pcm}(T) dT + f_l(T) \cdot L \quad (30)$$

where the first term is the sensible enthalpy part, representing the energy change caused by temperature change; the second term is the latent heat part, representing the energy absorbed or released during phase change. $f_l(T)$ is the liquid fraction, describing the degree of melting of the phase change material, defined as:

$$f_l(T) = \begin{cases} 0 & T < T_s \\ \frac{T-T_s}{T_l-T_s} & T_s \leq T \leq T_l \\ 1 & T > T_l \end{cases} \quad (31)$$

This definition linearizes the phase change process, and within the phase change interval (T_s and T_l), the liquid fraction increases linearly with temperature. The solidus temperature and liquidus temperature, respectively.

(2) Phase change heat transfer control equation

The transient heat conduction equation based on the enthalpy method can be written as:

$$\rho_{pcm} = \frac{\partial H}{\partial t} = \frac{1}{r} \frac{\partial}{\partial r} \left(\lambda_{pcm} r \frac{\partial T_{pcm}}{\partial r} \right) + \frac{\partial}{\partial z} \left(\lambda_{pcm} \frac{\partial T_{pcm}}{\partial z} \right) \quad (32)$$

Substituting Eq. (29) into it and expanding using the chain rule gives:

$$\rho_{pcm} \left(C_{p,pcm} \frac{\partial T_{pcm}}{\partial t} + L \frac{\partial f_l}{\partial t} \right) = \nabla \cdot (\lambda_{pcm} \nabla T_{pcm}) \quad (33)$$

where the second term on the left is the contribution term of latent heat of phase change, which is significantly non-zero within the phase change interval ($T_s \leq T \leq T_l$) and is zero in the solid and liquid regions. This equation implicitly includes the phase change latent heat in the time derivative of enthalpy, avoiding the complexity of explicitly tracking the phase interface, and is particularly suitable for heat dissipation scenarios with multi-material coupling and complex geometric structures.

3.3 Coupled Boundary Conditions

The physical nature of coupling is captured through boundary conditions, which are key to connecting different physical domains. This section defines all boundary conditions in the order of contact interfaces, thereby ensuring the continuity of heat flow and temperature across domains. They couple the control equations of the battery domain, heat sink domain, channel domain, and phase change material domain into a complete mathematical model. Accurate setting of boundary conditions is crucial for the reliability of simulation results.

3.3.1 Battery and Heat Sink Coupling Boundary

The part where the outer wall of the battery is in direct contact with the aluminum alloy heat sink satisfies temperature continuity and heat flux density continuity:

$$\begin{cases} T_b = T_{Al} \\ \lambda_{batt,r} \frac{\partial T_b}{\partial r} = \lambda_{Al} \frac{\partial T_{Al}}{\partial r} \end{cases} \quad (34)$$

This boundary condition ensures that heat is transferred from the battery to the heat sink without energy loss and is the main entry point for system heat transfer. The first equation embodies the Zeroth

Law of Thermodynamics. When two objects reach thermal equilibrium, the temperatures on both sides of the contact surface are equal. The second equation embodies the conservation of energy. The rate of heat flow per unit area is equal on both sides of the interface; that is, no heat accumulates or disappears at the interface.

3.3.2 Battery and Phase Change Material Coupling Boundary

The part where the outer wall of the battery is in direct contact with the phase change material satisfies temperature continuity and heat flux density continuity:

$$\begin{cases} T_b = T_{pcm} \\ \lambda_{batt,r} \frac{\partial T_b}{\partial r} = \lambda_{pcm} \frac{\partial T_{pcm}}{\partial r} \end{cases} \quad (35)$$

This boundary condition establishes a direct heat transfer path between the battery and the phase change material, enabling the phase change material to absorb the heat generated by the battery more directly without passing through the heat sink.

3.3.3 Heat Sink and Phase Change Material Coupling Boundary

The outer surface of the heat sink is in contact with the phase change material. This interface is a solid-solid (or solid-liquid) coupling:

$$\begin{cases} T_{Al} = T_{pcm} \\ \lambda_{Al} \frac{\partial T_{Al}}{\partial n} = \lambda_{pcm} \frac{\partial T_{pcm}}{\partial n} \end{cases} \quad (36)$$

This boundary condition realizes the transfer of heat from the high thermal conductivity skeleton to the phase change material and is the main entry point for the phase change heat storage mechanism. The heat sink serves as a high thermal conductivity channel, rapidly diffusing the heat generated by the battery to the entire honeycomb structure, and then transferring the heat to the phase change material through this interface.

3.3.4 Phase Change Material and Channel Wall Coupling Boundary

The channel wall is in direct contact with the phase change material. This interface realizes the coupling of phase change heat storage and active cooling:

$$\begin{cases} T_{pcm} = T_{wall} \\ \lambda_{pcm} \frac{\partial T_{pcm}}{\partial n} = \lambda_{Al} \frac{\partial T_{wall}}{\partial n} \end{cases} \quad (37)$$

where T_{wall} is the channel wall temperature, i.e., the aluminum alloy temperature (K).

This boundary condition transfers the heat stored by the phase change material to the channel wall, providing a heat source for subsequent convective heat transfer, and realizing the coupling of passive heat storage and active cooling. This is a key link in the entire thermal management system, which can both receive heat from the phase change material and transfer heat to the coolant in the channel, carrying it out of the system through forced convection.

3.3.5 Channel Wall and Coolant Coupling Boundary

The inner wall of the channel is in contact with the coolant, which is the direct coupling interface between solid heat conduction and fluid convection:

$$\begin{cases} u_z = 0 \\ T_{wall} = T_f \\ \lambda_{Al} \frac{\partial T_{wall}}{\partial n} = \lambda_f \frac{\partial T_f}{\partial n} = h(T_{wall} - T_{f,bulk}) \end{cases} \quad (38)$$

where $T_{f,bulk}$ is the mainstream temperature of the fluid; h is the convective heat transfer coefficient. n is the coordinate in the direction normal to the solid wall.

This boundary condition realizes the two-way coupling of flow and heat transfer and is the core link of active cooling. The three equations correspond to the velocity no-slip condition, respectively. The relative velocity of a viscous fluid on a solid wall is zero, which is the basic boundary condition of fluid mechanics and determines the formation and development of the velocity boundary layer.

Determination of the convective heat transfer coefficient:

$$h = \frac{N_u \cdot \lambda_f}{d_h} \quad (39)$$

The Nusselt number is determined by the flow state. For the fully developed laminar section, $N_u \approx 4.36$ for constant heat flux boundary conditions and $N_u \approx 3.66$ for constant wall temperature boundary conditions.

In this system, due to the heat storage effect of the phase change material, the wall temperature may change along the flow direction, but it is between constant heat flux and constant wall temperature, so $N_u = 4.0$ can be taken as an engineering approximation.

3.4 Coupled Heat Transfer Characteristic Analysis

The previous sections have separately established the control equations for the battery domain, heat sink domain, channel domain, and phase change material domain, and given the boundary conditions for each coupled interface, constituting a complete mathematical physical model. However, solving a single system of equations makes it difficult to intuitively reveal the heat transfer enhancement mechanism under the synergistic effect of multiple domains. To gain a deeper understanding of the competitive relationships among heat transfer paths from the battery core to the outside, the thermal resistance bottlenecks at different stages, and the energy distribution law within the composite thermal management system, this section performs a coupled heat transfer characteristic analysis from the perspectives of heat transfer science and thermodynamics. By establishing an equivalent thermal resistance network, the thermal resistance of conduction and convection of three parallel heat transfer paths is quantitatively compared to identify the key links restricting heat dissipation efficiency. By constructing a transient energy balance equation, the heat absorption proportions of active liquid cooling and passive phase change heat storage in different discharge stages are quantified, elucidating the synergistic working mechanism of the composite system. This analysis provides a clear theoretical direction for the subsequent optimization design of heat sink parameters, channel structure, coolant selection, and phase change material modification.

3.4.1 Thermal Resistance Characteristic Analysis

The entire heat transfer process can be equivalent to a thermal resistance network combining series and parallel connections. According to Fourier's Law and Newton's Law of Cooling for convective heat

transfer, the thermal resistance for conduction is $R_{cond} = \frac{\delta}{\lambda A}$, and the thermal resistance for convection is $R_{conv} = \frac{1}{hA}$ where δ is the conduction distance, λ is the thermal conductivity, A is the heat transfer and heat exchange area, and h is the convective heat transfer coefficient.

The total thermal resistance R_{total} determines the battery temperature rise:

The equivalent thermal resistances of the three main heat transfer paths are in a parallel relationship because heat can be transferred simultaneously through three paths, and the total heat flow is the sum of the heat flows of each path:

$$R_{total} = \frac{1}{\frac{1}{R_{path1}} + \frac{1}{R_{path2}} + \frac{1}{R_{path3}}} \quad (40)$$

The thermal resistance composition of each path is as follows:

Path 1: Heat is transferred via Battery—Heat sink—Channel wall—Coolant. The thermal resistance is:

$$R_{path1} = R_{batt} + R_{Al} + R_{conv} \quad (41)$$

Path 2: Heat is transferred via Battery—Heat sink—PCM—Channel wall—Coolant. The thermal resistance is:

$$R_{path2} = R_{batt} + R_{Al} + R_{pcm} + R_{conv} \quad (42)$$

Path 3: Heat is transferred via Battery—PCM—Channel wall—Coolant. The thermal resistance is:

$$R_{path3} = R_{batt} + R_{pcm} + R_{conv} \quad (43)$$

Battery internal radial thermal conduction resistance:

$$R_{batt} = \frac{\ln\left(\frac{R_1}{R_{in}}\right)}{2\pi\lambda_{batt,r}L} \quad (44)$$

Heat sink thermal conduction resistance:

$$R_{Al} = \frac{\ln\left(\frac{R_2}{R_1}\right)}{2\pi\lambda_{Al}L} \quad (45)$$

Phase change material thermal conduction resistance:

$$R_{pcm} = \frac{\ln\left(\frac{R_{wall}}{R_2}\right)}{2\pi\lambda_{pcm}L} \quad (46)$$

Convective heat transfer resistance:

$$R_{conv} = \frac{1}{hA} \quad (47)$$

This parallel structure makes the total thermal resistance significantly lower than that of any single path. In particular, Path 3 bypasses the heat sink, shortens the heat transfer distance, and enables the phase change material to respond more rapidly to battery temperature changes.

3.4.2 Energy Balance Equation

Under transient conditions, the system satisfies energy conservation, which can quantify the heat absorption proportion of different cooling methods:

$$Q_{gen} = Q_{cool} + Q_{pcm} + Q_{loss} \quad (48)$$

where $Q_{cool} = \dot{m} C_{p,f} (T_{f,out} - T_{f,in})$ is the heat carried away by the coolant convection; $\dot{m} = \rho_f A_C u_{in}$ is the mass flow rate; $Q_{pcm} = m_{pcm} \left(\int_{T_{init}}^T C_{p,pcm} dT + f_l \cdot L \right)$ is the channel cross-sectional area. A_C is the total heat absorbed by the sensible heat and latent heat of the phase change material; m_{pcm} is the total mass of the phase change material. Q_{loss} is the heat dissipated by the system to the environment.

This equation reveals the distribution relationship of heat in the system. During discharge, the phase change material absorbs a large amount of heat, accounting for a high proportion. As the phase change material gradually melts, the proportion of heat carried away by the coolant increases. When the phase change material is completely melted, all heat is carried away by the coolant. Through the change in Q_{pcm} , the utilization rate of the phase change material can be evaluated.

The specific meanings of the parameters are shown in the Table 2 below.

Table 2: Nomenclature for Key Physical Quantities in the Thermal Model of the 21700 Lithium-Ion Battery.

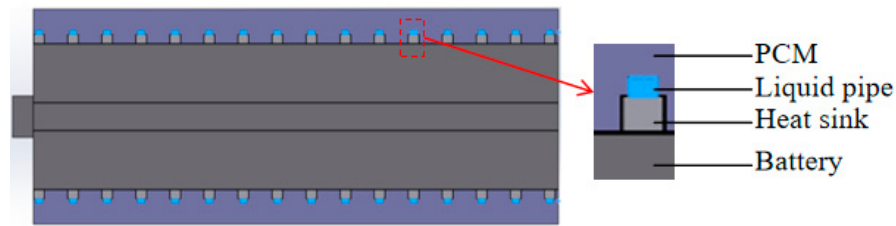
Symbol	Physical Quantity	Unit
Q	Total heat generation of the battery	W
Q_r	Reversible reaction heat (entropic heat)	W
Q_j	Joule heat	W
Q_p	Polarization heat	W
Q_s	Side reaction heat	W
T	Thermodynamic temperature	K
ΔS	Entropy change of the electrochemical reaction	J/(mol·K)
G	Gibbs free energy	J
n	Amount of substance of electrons	mol
F	Faraday's constant	C/mol
E	Electromotive force/Open-circuit voltage	V
I	Charge/discharge current	A
R_e	Ohmic internal resistance	Ω
R_p	Polarization internal resistance	Ω
q	Volumetric heat generation rate	W/m ³
V_b	Volume of the battery cell	m ³
U	Operating voltage of the battery	V
n	Boundary normal vector	—
h	Convective heat transfer coefficient	W/(m ² ·K)
T_s	Surface temperature of the battery	K
T_∞	Ambient temperature	K
$\lambda_{batt,r}$	Radial thermal conductivity of the battery	W/(m·K)
$\lambda_{batt,z}$	Axial thermal conductivity of the battery	W/(m·K)
ρ_{batt}	Average density of the battery	kg/m ³
$c_{p,batt}$	Specific heat capacity at constant pressure of the battery	J/(kg·K)
T_b	Bulk temperature of the battery	K
q_{gen}	Volumetric heat generation rate of the internal heat source	W/m ³
Re	Reynolds number	—
ρ_f	Density of the coolant	kg/m ³
u	Flow velocity of the coolant	m/s

Table 2: *Cont.*

Symbol	Physical Quantity	Unit
d_h	Hydraulic diameter of the channel	m
μ_f	Dynamic viscosity of the coolant	Pa·s
N_u	Nusselt number	—
F_l	Liquid fraction of the PCM	—
L	Latent heat of fusion of the PCM	J/kg
R_{total}	Total thermal resistance of the system	K/W
$\dot{m}$	Mass flow rate of the coolant	Kg/s

4 Selection of Cooling Structure and Heat Dissipation Analysis

In previous studies, Monika et al. and Sutheesh et al. [27,28] have proven that the hexagonal honeycomb heat sink has great advantages in heat transfer, so the heat sink type selected is a hexagonal honeycomb heat sink. The heat sink is the outer structure of the battery, accelerating the process of heat transfer from the battery surface to the outside. The channels absorb heat from the heat sink and PCM through the cooling liquid. The outermost layer is wrapped by PCM for phase change heat absorption. The final structural simplified model is shown in the Fig. 11 below:

**Figure 11:** Coupled thermal management system geometry.

This structure is a circumferentially fully wrapped composite thermal management section designed for cylindrical lithium-ion batteries, featuring an axisymmetric and coaxial annular layout. It integrates a dual temperature control mechanism combining active liquid cooling and passive phase change heat storage.

4.1 Cooling Structure Design

The modeling process of the composite cooling system from a single cell to a battery + heat sink + channel + PCM is shown in the Fig. 12 below in sequence:

The Fig. 12 above shows the hierarchical modeling process of the 21700 cylindrical lithium-ion battery composite thermal management system. Using the single cell as the core heat-generating body, a complete modeling sequence is constructed, ranging from a blank control group to a fully functional composite cooling system. For each structural group, both an axial 3D view and a radial cross-sectional top view are presented, clearly illustrating the structural topology, radial layout changes, and the progressive functional logic at each level.

Fig. 12c,d is the structural model of the single cell + honeycomb aluminum alloy heat sink. This structure achieves rapid circumferential homogenization and radial conduction of battery surface heat through the high thermal conductivity of aluminum alloy, greatly reducing the radial temperature gradient of the battery. At the same time, it provides structural support and efficient heat transfer interfaces for the subsequent channel and PCM structures, realizing the core heat spreading foundation for efficient thermal management.

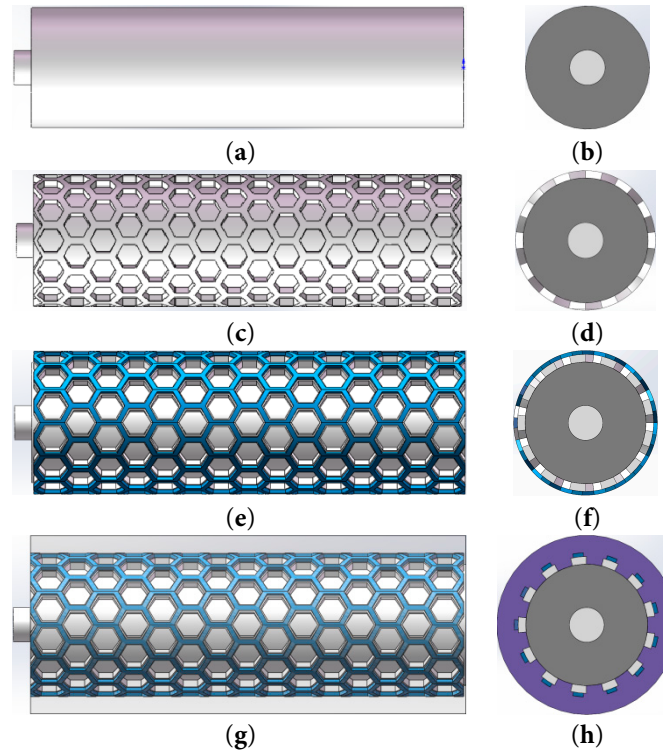


Figure 12: Battery Thermal Management System Modeling. (a) Battery; (b) Battery (From below); (c) Battery + Heat sink; (d) Battery + Heat sink (From below); (e) Battery + Heat sink + Liquid pipe; (f) Battery + Heat sink + Liquid pipe (From below); (g) Battery + Heat sink + Liquid pipe + PCM; (h) Battery + Heat sink + Liquid pipe + PCM (From below).

Fig. 12e,f is the active cooling model of the single cell + heat sink + nested liquid cooling channel. On the basis of honeycomb heat spreading, this structure adds an active heat discharge channel, realizing continuous temperature control in steady-state long-period operating conditions and solving the problem that single passive heat dissipation cannot effectively maintain battery operating temperature for a long time.

Fig. 12g,h is the final composite cooling model of the single cell + heat sink + channel + PCM. On the basis of the active liquid cooling steady-state temperature control, this structure achieves peak temperature suppression under transient thermal shock of high-rate discharge through the solid-liquid phase change latent heat effect of PCM. At the same time, through the continuous PCM medium, the circumferential temperature equalization effect is further enhanced. Three parallel heat transfer paths work synergistically to take into account the dual demands of transient heat buffering and steady-state long-term heat dissipation. This is the core composite thermal management scheme designed in this study.

4.2 Influence of Heat Sink on Heat Dissipation

The honeycomb heat sink is the core heat spreading component of the composite thermal management system. The thermal conductivity of its material directly determines the radial conduction efficiency of the battery surface heat generation, the circumferential temperature equalization effect, and is the core heat transfer bridge connecting the battery body, phase change heat storage layer, and liquid cooling channel. The simulation defaults the heat sink material to 3003 aluminum alloy, and the heat sink thickness d is 2 mm.

Fig. 13 is a schematic diagram of the honeycomb heat sink structure. Fig. 13a is the radial top view of the assembly of the battery and the heat sink. It can be seen that the regular hexagonal honeycomb

heat sink is uniformly distributed around the circumference of the 21700 cylindrical battery. The inner side of the heat sink is in complete contact with the outer wall of the battery, forming a circumferentially symmetric high thermal conductivity heat spreading network to achieve uniform radial conduction of heat generated by the battery. Fig. 13b is a structural unit of a single regular hexagonal honeycomb heat sink. It adopts a regular hexagonal annular topology, ensuring the structural isotropy and heat transfer uniformity, while possessing excellent structural stability. Fig. 13c is a side view of the heat sink. The mirror thickness of the fin is denoted as d , which is the core variable in this parametric analysis. Its value directly determines the cross-sectional area for heat conduction, heat spreading capability, and heat transfer area of the fin, making it a key structural parameter affecting the temperature control performance of the system.

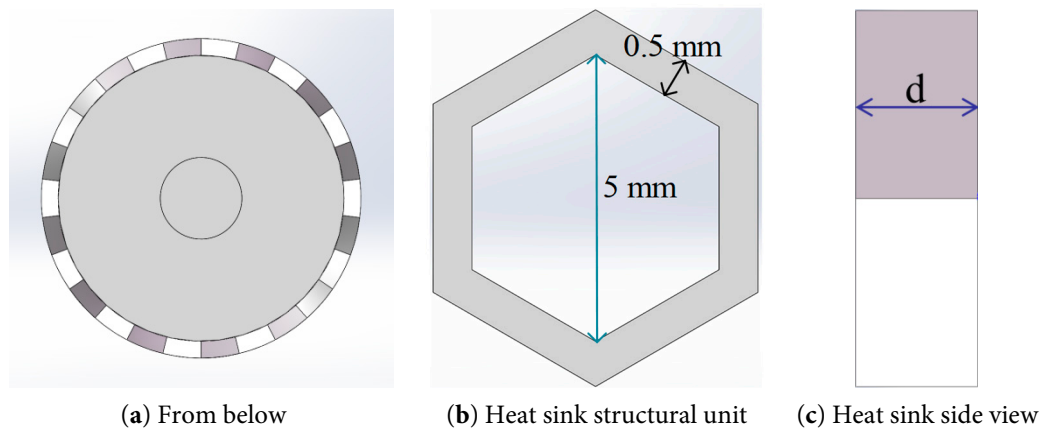


Figure 13: Heat sink structural unit. (a) From below; (b) Heat sink structural unit; (c) Heat sink side view.

4.2.1 Influence of Heat Sink Material on Heat Dissipation

The selection of heat sink material requires balancing temperature control performance, system weight, cost, and processing performance to achieve a multi-dimensional optimization. To clarify the influence of heat sink material on the temperature control performance of the battery pack, this section selects three metal materials commonly used in automotive power battery heat dissipation structures for simulation comparative analysis: metal copper, 3003-H18 aluminum alloy, and 6063-T83 aluminum alloy. The core thermophysical parameters of the three materials are shown in Table 3. The simulation conditions remain unchanged except for the heat sink material. The simulation results are shown in Fig. 14.

Table 3: Thermal conductivity parameters of metallic materials.

Material	Copper	3003 Aluminum Alloy	6063 Aluminum Alloy
Thermal Conductivity (W/(m·K))	400	155	201
Density (kg/m ³)	8960	2730	2700
Specific Heat Capacity (J/(kg·K))	385	893	900

Fig. 14 is the overall temperature field distribution clouds of the battery pack corresponding to the three heat sink materials at the end of 3C discharge. They clearly present the laws of influence of material thermal conductivity on the peak temperature and temperature uniformity of the battery pack. In terms of peak temperature, there are significant differences in the peak temperature of the battery pack corresponding to the three materials. The peak temperature of the battery pack under the metal copper heat sink condition is the lowest, followed by the aluminum alloy condition, and the peak temperature under the 3003 aluminum

alloy condition is the highest. In terms of temperature distribution uniformity, the battery pack temperature field distribution corresponding to the metal copper heat sink is the most uniform, with no obvious local high-temperature zones. The core mechanism of this law lies in the heat spreading effect of the heat sink. The thermal conductivity of metal copper is far higher than that of the two aluminum alloys. It can achieve faster radial uniform diffusion and radial efficient conduction of the heat generated on the battery surface, which not only eliminates local hotspots on the battery cell but also weakens the heat accumulation effect in the center of the array, finally achieving a reduction in peak temperature and an improvement in temperature uniformity. The thermal conductivity of 6063 aluminum alloy is slightly higher than that of 3003 aluminum alloy, so its temperature control effect also shows a small advantage.

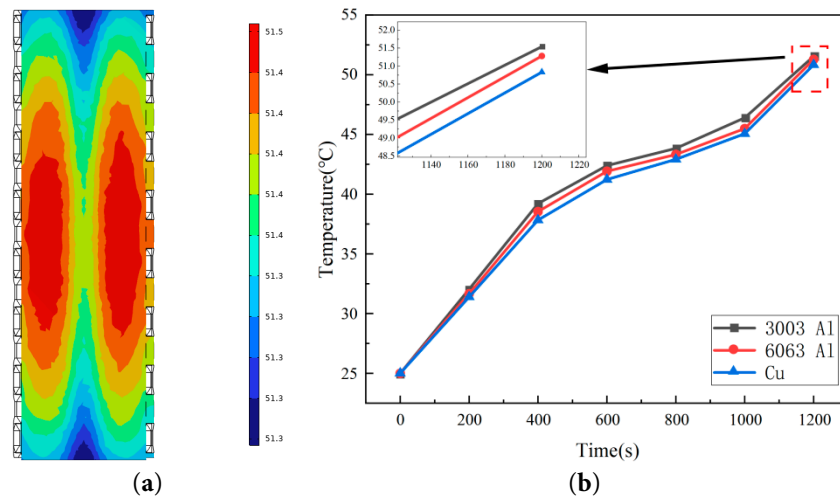


Figure 14: The impact of the heat sink on cooling performance. (a) Simulation results; (b) Comparison results.

Fig. 14 is the transient variation curve of the maximum temperature of the battery pack corresponding to the three heat sink materials during the 3C discharge cycle. It clearly presents the temperature rise laws and temperature control characteristic differences of different materials. In terms of the overall temperature rise trend, the temperature rise curves of the three materials basically coincide throughout the discharge process. In the early stage of discharge, the three curves are almost indistinguishable, and the battery temperature shows a nearly linear upward trend. This is because the battery heat generation rate is low in this stage, the system thermal load is small, and the thermal conductivity capabilities of the three materials can all meet the current heat transfer requirements, so the difference in thermal conductivity performance is not fully reflected. In the middle and late stages of discharge, as the battery heat generation continues to accumulate, the three temperature rise curves begin to separate. The higher the thermal conductivity of the material, the slower the temperature rise rate, and the temperature control advantage gradually appears. The local enlarged view can see a stepwise difference in peak temperatures. Metal copper has a more significant temperature control advantage in the early and middle stages of discharge. The core reason is that its high thermal conductivity can quickly transfer the heat generated by the battery to the surroundings, effectively suppressing the temperature rise rate. However, by the late stage of discharge, the high thermal conductivity advantage of metal copper can no longer be fully utilized, so the difference in peak temperature with the two aluminum alloys gradually narrows, presenting a characteristic of relatively weakened cooling effect in the later stage.

Metal copper and 6063 aluminum alloy show better thermal management performance than 3003 aluminum alloy, but the price of metal copper is far higher than that of 3003 aluminum alloy, and its density is

much greater than that of aluminum alloy, while the temperature reduction is only 0.71°C , a decrease of 1.38%. The price of 6063 aluminum alloy is 40%–50% higher than that of 3003 aluminum alloy, while the temperature is only reduced by 0.26°C , a decrease of 0.51%. In summary, considering temperature control performance, system lightweighting, cost, and processing performance, 3003 aluminum alloy is the optimal choice for the heat sink material in this study.

4.2.2 Influence of Heat Sink Thickness on Heat Dissipation

To investigate the influence of heat sink thickness on the thermal management system, the heat sink thickness d is now set to 1.6 mm, 1.8 mm, 2 mm, 2.2 mm, and 2.4 mm, respectively. The heat sink material is selected as 3003 aluminum alloy, and the remaining simulation conditions remain unchanged for simulation research. The simulation results are shown in the figure below.

As shown in the Fig. 15, the battery surface temperature decreases with the increase of heat sink thickness, but the magnitude of the decrease decreases as the heat sink thickness increases. The reduction is greatest when the heat sink thickness increases from 1.4 mm to 2 mm. After 2 mm, increasing the heat sink thickness results in a smaller temperature reduction. Considering that increasing the heat sink thickness also increases manufacturing cost and system weight, a 2 mm heat sink thickness is selected, with a final temperature of 51.54°C . In terms of the overall variation trend, the peak temperature of the battery shows a stepwise continuous decrease as the heat sink thickness increases, and the temperature control effect steadily improves. This directly verifies the heat spreading and temperature control effect of the honeycomb high thermal conductivity heat sink. It breaks the inherent bottleneck of weak radial thermal conductivity of the cylindrical wound battery through the high thermal conductivity aluminum alloy skeleton, greatly reducing the radial thermal spreading resistance of the battery and achieving rapid heat conduction and circumferential temperature equalization of the battery heat generation.

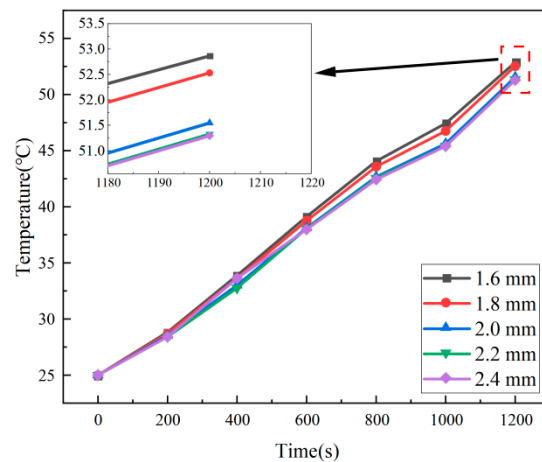


Figure 15: The effect of heat sink thickness on cooling performance.

4.3 Influence of Runner on Heat Dissipation

The cooling liquid channel plays an important role in liquid cooling. Modifications to the structure and inner diameter have a substantial impact on the flow rate and flow path of the cooling liquid, while also exerting a certain influence on the peak temperature and temperature uniformity. The channel unit structure is shown in Fig. 16.

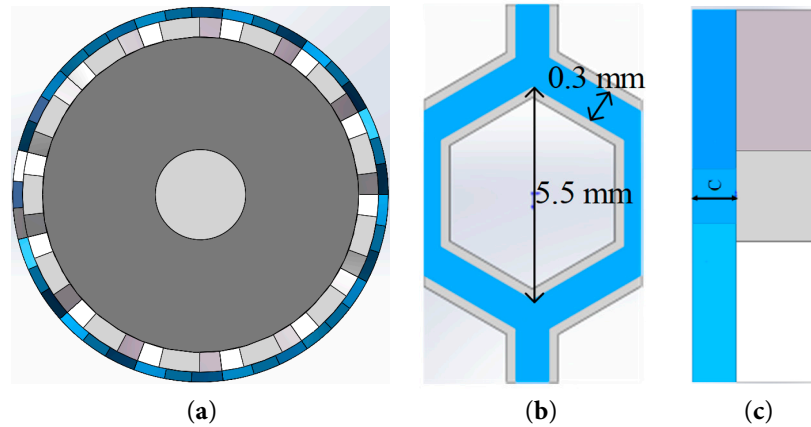


Figure 16: Runner unit structure. (a) From below; (b) Flow path structure unit; (c) Flow path side view.

Fig. 16 is the channel unit structure analyzed in this section, clearly presenting the integrated topology design of the channel with the honeycomb heat sink. Fig. 16a is the radial top view of the assembly of the battery and the channel. It can be seen that the cooling channels are uniformly distributed along the circumference of the 21700 cylindrical battery at equal intervals, corresponding one-to-one nested with the honeycomb aluminum alloy heat sink. The channel wall is seamlessly attached to the high thermal conductivity heat sink, realizing the efficient connection of heat sink spreading and channel convective heat transfer, and ensuring that the heat generated by the battery can be quickly transferred to the channel wall by the heat sink and continuously carried out by the coolant. Fig. 16b is a structural unit of a single channel. It adopts a regular hexagonal topology matching the honeycomb heat sink, which not only ensures the maximum contact heat transfer area with the heat sink but also maintains the structural circumferential symmetry, realizing uniform heat transfer in the full circumferential direction of the battery. Fig. 16c is a side view of the channel. c is the inner diameter of the channel and is the core variable of this analysis. Its value directly determines the channel flow area, coolant flow rate, total heat transfer area, and flow resistance, and is the core design parameter for the temperature control performance and operating energy consumption of the liquid cooling system.

4.3.1 Influence of Runner Structure on Heat Dissipation

To further improve the heat exchange efficiency of the honeycomb axial channel and the temperature uniformity of the battery, this section carries out an optimization design for the channel layout. The original single-inlet unidirectional serial channel is improved to a dual-inlet counter-flow, intermediate outlet symmetrical channel structure. The channel inner diameter is 1.4 mm. The coolant is water at a temperature of 25°C, and the flow velocity is 1 cm/s. The cooling effects before and after the flow channel improvement are compared and analyzed through simulation.

Fig. 17 is the topology comparison before and after the runner structure improvement, clearly presenting the core design idea of this optimization. The upper part is the original runner structure, adopting a single-inlet unidirectional serial layout. The coolant flows from the left inlet of the channel, sequentially passes through all honeycomb unit channels along the axial direction of the battery from left to right, and finally flows out from the right outlet. This belongs to a typical long-distance serial unidirectional flow. This structure has defects. The coolant continuously absorbs the heat generated by the battery along the flow direction, and its own temperature continuously rises along the path, resulting in a continuous expansion of the heat exchange temperature difference between the channel inlet and outlet, forming a significant

temperature gradient of low temperature at the inlet and high temperature at the outlet in the axial direction of the battery, and poor overall temperature uniformity. The lower part is the improved runner structure, adopting a dual-inlet counter-flow, intermediate outlet fully symmetrical layout. The coolant flows in simultaneously from the left and right ends of the channel and flows out uniformly from the middle outlet perpendicular to the axial direction. This design splits the original long-distance serial channel into two symmetric short channels by counter-flow. The single-pass flow distance of the coolant is shortened to half of the original structure, greatly reducing the temperature rise of the coolant along the path and improving the average heat exchange temperature difference and heat exchange efficiency along the entire channel. The simulation results are shown in Fig. 18.

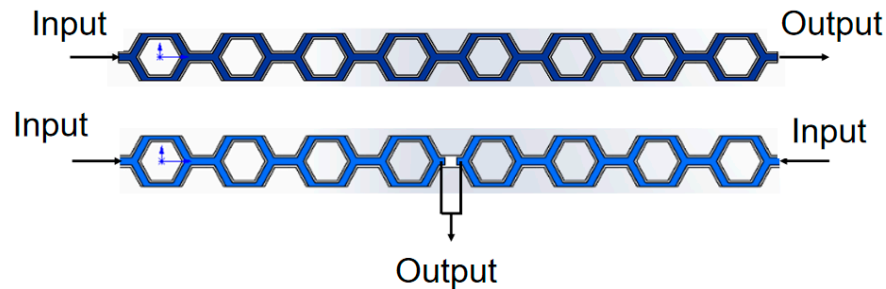


Figure 17: Runner structure improvement.

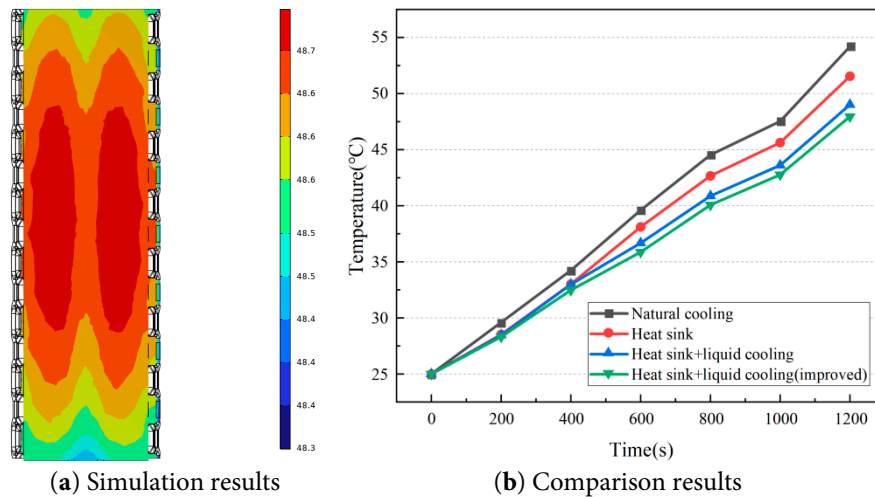


Figure 18: The effect of the improved flow channel on cooling performance. (a) Simulation results; (b) Comparison results.

From Fig. 18 above, it can be seen that after the flow improvement, the temperature variation law is the same as before, but the temperature has decreased compared to before the improvement. This is because after the improvement, the amount of incoming cooling liquid increases, carrying away more heat, leading to an increase in thermal conductivity and better cooling effect. Compared with battery natural cooling, heat sink cooling zone, and pre-improvement runner structure, the maximum temperature is reduced by 2.65°C, 5.18°C, and 6.24°C, respectively, lowering to 47.95°C. This improved runner has a good cooling effect. In terms of the overall temperature rise trend, the initial temperatures of the three operating conditions are all 25°C, and the temperature rise rates show significant differences during the discharge process. The temperature rise rate of battery natural cooling is the fastest, showing a nearly linear rapid

upward trend throughout the process. The temperature rise rate of the original runner + honeycomb heat sink structure slows down significantly, and the temperature control effect is preliminarily verified. The improved counter-flow channel structure has the slowest temperature rise rate and maintains the optimal temperature control effect throughout the discharge cycle.

In summary, this runner structure improvement, without changing the channel cross-sectional dimensions and total coolant flow, achieves a reduction in battery peak temperature and a substantial improvement in temperature uniformity through the layout optimization of symmetrical counter-flow. This fully verifies the rationality and effectiveness of this optimization scheme, providing a feasible technical idea for the design optimization of axial liquid cooling channels for cylindrical batteries.

4.3.2 Influence of Runner Inner Diameter on Heat Dissipation

In this parametric analysis, the axial cooling channel nested inside the honeycomb aluminum alloy heat sink is taken as the research object. The channel inner diameter c is taken as the core variable to investigate the influence laws of 6 groups of channel inner diameters: 1.0 mm, 1.2 mm, 1.4 mm, 1.6 mm, 1.8 mm, and 2.0 mm on the thermal management performance of the battery. The simulation results are shown in Fig. 19 below.

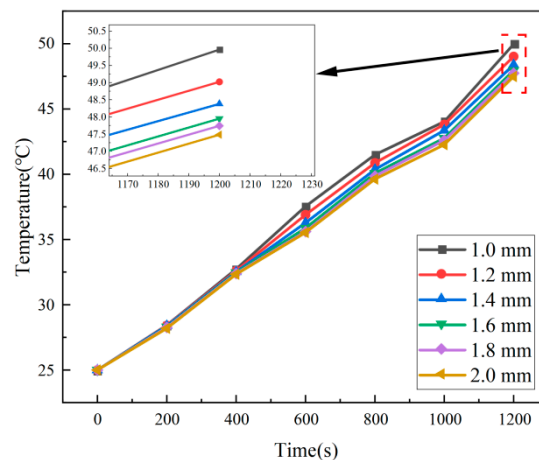


Figure 19: The effect of the runner inner diameter on cooling performance.

Fig. 19 is the transient temperature rise curve of the battery under different channel inner diameter conditions during the 3C discharge cycle, revealing the influence laws of channel inner diameter on the battery temperature rise rate and peak temperature. In terms of the overall temperature rise trend, the battery temperature rise rate and peak temperature at the end of discharge for all operating conditions configured with cooling channels are significantly lower than natural cooling, and the peak temperature of the battery shows a stepwise continuous decrease as the channel inner diameter increases, with the temperature control effect steadily improving. The influence of channel inner diameter on temperature control effect is essentially the comprehensive result of the combined action of flow area, coolant flow rate, heat transfer area, and flow resistance, finally achieving a continuous reduction in battery peak temperature. The battery surface temperature decreases with the increase of the channel inner diameter, but the magnitude of the decrease decreases as the heat sink thickness increases. The reduction is greatest when the channel inner diameter increases from 1 mm to 1.6 mm. After 1.6 mm, increasing the heat sink thickness results in a smaller temperature reduction. Considering that increasing the channel inner diameter also increases manufacturing costs, a 1.6 mm channel inner diameter is selected.

4.4 Influence of Coolant on Heat Dissipation

The coolant is the core heat exchange medium of the active liquid cooling system. Its four core parameters—type, composition ratio, inlet flow rate, and inlet temperature—directly determine the convective heat transfer efficiency inside the channel, the system operating energy consumption, the temperature control accuracy of the battery pack, and temperature uniformity. Based on the previously optimized battery pack model, this section conducts a systematic investigation—via multiple controlled-variable simulations—into the influence of key coolant parameters on temperature control performance, thereby achieving optimal matching between the liquid cooling system and the composite thermal management scheme and completing the parameter optimization of the liquid cooling system.

4.4.1 Influence of Coolant Type on Heat Dissipation

To investigate the influence of coolant type on heat dissipation, this section selects 5 cooling media commonly used in automotive thermal management systems for comparative analysis: water, ethanol, ethylene glycol, a 50% by volume aqueous ethylene glycol solution, and glycerol. The specific thermophysical parameters are shown in Table 4.

Table 4: Different coolant parameters.

Coolant	Density (kg/m ³)	Specific Heat Capacity (J/(kg·K))	Thermal Conductivity (W/(m·K))	Dynamic Viscosity (mPa·s)
Water	1000	4182	0.6	1.003
Ethanol	790	2470	0.182	1.2
Ethylene Glycol	1111	2415	0.252	157
50% Aqueous Ethylene Glycol Solution	1071	3300	0.384	3.39
Glycerol	1260	2427	0.286	799

The transient temperature rise laws and maximum temperature difference comparisons for different coolants are shown in Fig. 20.

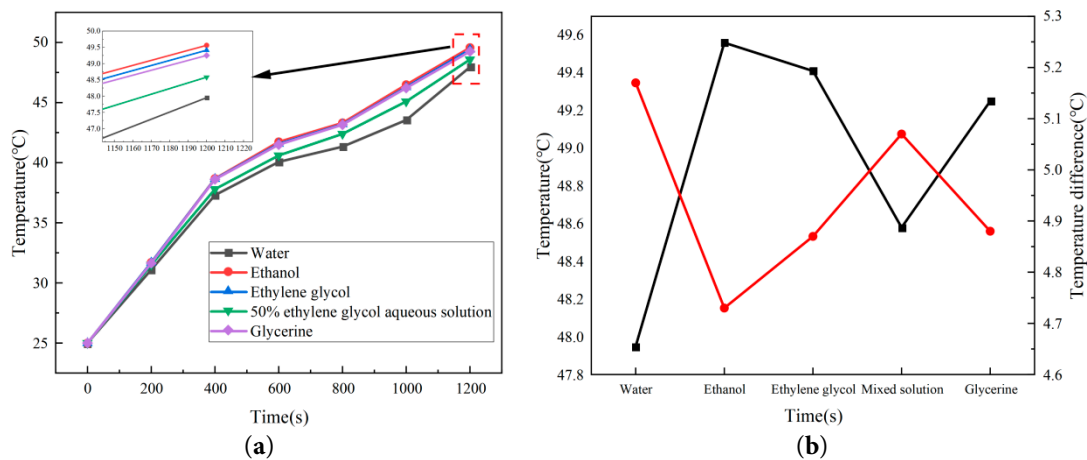


Figure 20: The effect of coolant type on cooling performance. (a) The effect of type on temperature; (b) The effect of type on temperature difference.

In terms of transient temperature rise laws, the temperature rise curves corresponding to the 5 coolants are completely consistent. In the early stage of discharge, all curves almost overlap, at which time the heat dissipation proportion of the liquid cooling system is high, but the influence of coolant type is not fully manifested. Entering the middle and late stages of discharge, the PCM gradually melts, the liquid cooling system becomes the core heat discharge path, and the difference in heat exchange capability of different coolants is fully reflected, showing a significant curve separation trend. The temperature rise rate of pure water is the slowest, and that of pure ethylene glycol and glycerol is the fastest, corresponding exactly to the peak temperature results in the temperature cloud map. However, analyzing from Fig. 20b, pure water has the highest temperature difference. Although pure water has strong cooling capacity, it has poor low-temperature adaptability and insufficient temperature uniformity. Overall, ethanol, pure ethylene glycol, and glycerol have fatal defects of insufficient cooling capacity or excessive system energy consumption. The 50% aqueous ethylene glycol solution balances temperature control effect and temperature uniformity and is the comprehensive optimal coolant type, but its composition ratio still needs further optimization analysis.

4.4.2 Influence of Coolant Composition Ratio on Heat Dissipation

The previous section verified the comprehensive performance advantage of the aqueous ethylene glycol solution. This section conducts a parametric analysis of the composition ratio based on 9 groups of volume fractions of ethylene glycol aqueous solution from 10% to 90%.

The core thermophysical parameters of aqueous solutions with different volume fractions are shown in Table 5. The remaining simulation conditions remain consistent with the previous section. The transient temperature rise laws and quantitative comparison of temperature control performance are shown in Fig. 21.

Table 5: Parameters of ethylene glycol aqueous solutions with different volume fractions.

Volume Fraction of Ethylene Glycol Aqueous Solution (%)	Density (kg/m ³)	Specific Heat Capacity (J/(kg·K))	Thermal Conductivity (W/(m·K))	Dynamic Viscosity (mPa·s)
10	1011.69	3981	0.552	1.08
20	1027.93	3826	0.503	1.46
30	1043.32	3660	0.459	1.92
40	1057.60	3485	0.419	2.57
50	1071.11	3300	0.384	3.39
60	1083.87	3106	0.352	4.52
70	1095.92	2903	0.324	6.24
80	1107.40	2690	0.299	8.29
90	1118.32	2467	0.278	10.59

The temperature rise curve in Fig. 21a shows that in the early and middle stages of discharge, the curves for different volume fractions almost completely overlap, as this stage is mainly dominated by coolant control, but the influence of coolant composition ratio is extremely small. Entering the middle and late stages of discharge, the curve separation trend becomes significant. The higher the volume fraction of ethylene glycol, the faster the temperature rise rate, and the higher the peak temperature of the battery. The reason is that as the volume fraction of ethylene glycol increases, the specific heat capacity and thermal conductivity of the aqueous solution decrease linearly, while the convective heat transfer capability is positively correlated with both, ultimately leading to a continuous weakening of the heat discharge capacity of the liquid cooling system and a steady rise in the battery peak temperature. Fig. 21b presents the reverse

variation law of temperature control effect and uniformity. As the volume fraction of ethylene glycol increases from 10% to 90%, the maximum temperature of the battery rises linearly from 47.76°C to 49.4°C, while the maximum temperature difference decreases continuously from 5.24°C to 4.99°C. As the volume fraction of ethylene glycol increases, the dynamic viscosity of the aqueous solution rises continuously. Comprehensively considering temperature control performance, temperature uniformity, system energy consumption, and low-temperature adaptability, the 50% by volume ethylene glycol aqueous solution is the optimal choice. At this time, the peak temperature of the battery pack is 48.58°C, approaching the optimal operating temperature range of the battery, and the maximum temperature difference is controlled at 5.07°C. Meanwhile, the freezing point is about -35°C , fully meeting the all-climate usage requirements of new energy vehicles. In summary, to ensure heat dissipation performance and temperature uniformity, an aqueous ethylene glycol solution with a volume fraction of about 50% is selected as the cooling medium.

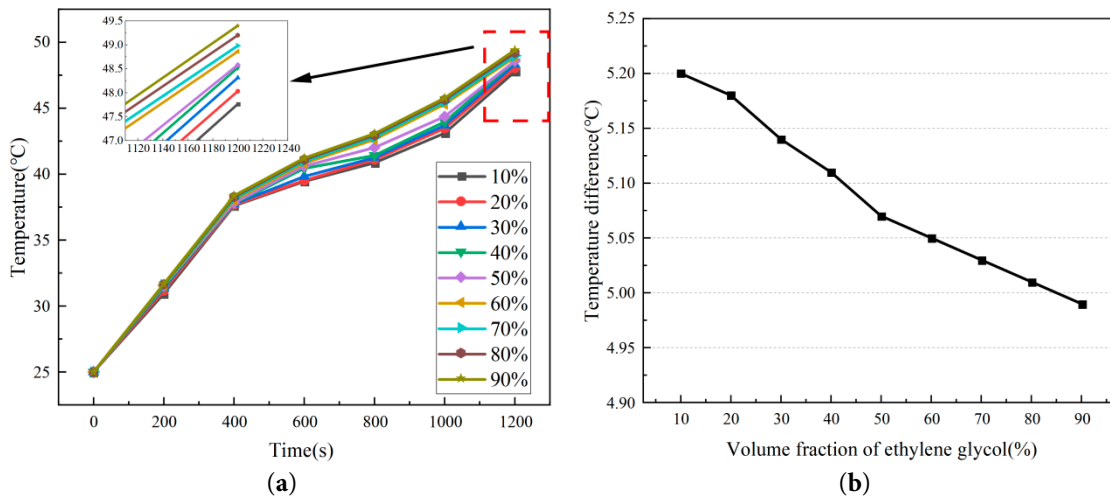


Figure 21: The effect of ethylene glycol solution concentration on cooling performance. (a) Effect of volume fraction on temperature; (b) Effect on temperature difference.

4.4.3 Influence of Coolant Flow Rate on Heat Dissipation

The inlet flow rate of the coolant directly determines the convective heat transfer coefficient inside the channel, the mass flow rate of the coolant, and the system flow resistance, and is the core parameter for the temperature control performance and energy consumption balance of the liquid cooling system. This section selects 5 groups of inlet flow rates: 0.5 m/s, 1 m/s, 1.5 m/s, 2 m/s, and 2.5 m/s. The cooling medium is a 50% aqueous ethylene glycol solution, and the remaining simulation conditions remain unchanged. The transient temperature rise laws and quantitative comparison of temperature control performance are shown in Fig. 22.

The temperature rise curve in Fig. 22a shows that in the early stage of discharge, the curves corresponding to different flow rates almost overlap, and the influence of flow rate is not fully reflected. Entering the middle and late stages of discharge, the influence of flow rate becomes significant. The higher the flow rate, the slower the temperature rise rate and the lower the peak temperature. In this system, the flow inside the channel is laminar, and the convective heat transfer coefficient increases with the increase of flow rate, while the mass flow rate of the coolant increases synchronously, significantly improving the thermal capacity and heat exchange capability of the system, enabling the heat generated by the battery to be carried out of the system more quickly, thus the peak temperature continues to decline. The quantitative

comparison in Fig. 22b shows that as the flow rate increases from 0.5 cm/s to 2.5 cm/s, the maximum temperature of the battery drops from 49.35°C to 47.38°C, a cumulative decrease of 1.97°C; however, the maximum temperature difference rises from 4.73°C to 5.76°C, a cumulative increase of 1.03°C. Comprehensively considering temperature control effect, temperature uniformity, and system operating energy consumption, 1.5 cm/s is the optimal inlet flow rate. The curve in Fig. 22c indicates that pressure drop increases with rising coolant flow velocity. Since this structure is designed with microchannels and the coolant operates at a low flow rate, the pressure drop reaches only 16.37 kPa at the maximum velocity of 0.025 m/s. Hence, pressure drop can be ignored in the design of this flow channel.

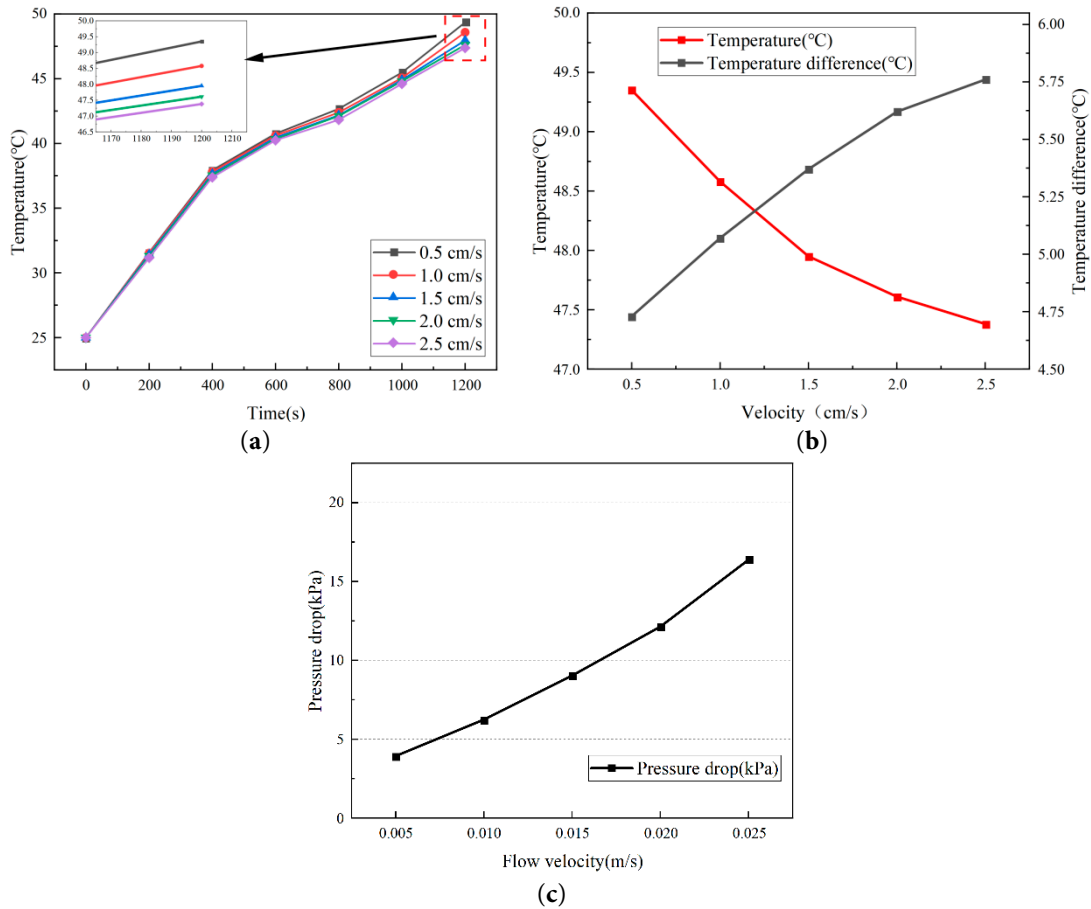


Figure 22: The effect of coolant flow rate on cooling performance. (a) The effect of flow rate on temperature; (b) The effect of flow rate on temperature difference; (c) Effect of coolant flow velocity on pressure drop.

4.5 Influence of Phase Change Material Type on Heat Dissipation

PCM is the core component of the composite thermal management system for suppressing transient thermal shocks. It absorbs heat generated by the battery through solid-liquid phase change latent heat, thereby flattening the temperature rise peaks. However, pure paraffin-based phase change materials suffer from inherent drawbacks: extremely low thermal conductivity and a slow heat transfer rate. These lead to a delayed phase change response, low latent heat utilization, and a tendency for local overheating and poor temperature uniformity within the battery pack, ultimately preventing full realization of the advantages of phase-change temperature control. To address this issue, this section presents a comparative simulation analysis of pure paraffin versus paraffin-expanded graphite composite phase change materials. The paraffin-

expanded graphite composite phase change material uses a ratio of 9:1, which not only maintains the high latent heat of paraffin materials but also continues the high thermal conductivity of graphite materials.

The core thermophysical parameters of the two phase change materials are shown in Table 6. The addition of expanded graphite significantly increases the thermal conductivity of the composite phase change material. Although the latent heat of phase change decreases slightly, it still possesses sufficient heat storage capacity, providing a foundation for the optimization of phase change temperature control performance. The influence law of thermal conductivity modification of battery pack thermal management performance is explored. The simulation operating conditions remain consistent with the optimized liquid cooling parameters in the previous section.

Table 6: Thermal conductivity parameters of different phase change materials.

Material Type	Density (kg/m ³)	Latent Heat of Fusion (J/g)	Thermal Conductivity (W/(m·K))
Paraffin	780	243.5	0.385
9:1Paraffin-Expanded Graphite	900	219.15	1.8

At the end of 3C discharge, the overall temperature field distribution clouds of the battery pack corresponding to the two phase change materials are shown in Fig. 23, intuitively presenting the significant optimization of battery pack temperature control effect and temperature uniformity by thermal conductivity. When using pure paraffin as the phase change material, the peak temperature of the battery reaches 42.39°C, which already meets the upper limit of 45°C for the optimal operating temperature of lithium-ion batteries. The maximum temperature difference of the battery reaches 4.32°C, meeting the safety threshold of a temperature difference $\leq 5^\circ\text{C}$. When using paraffin-expanded graphite composite phase change material (Fig. 23b), the peak temperature of the battery pack drops to 39.85°C, and the temperature difference drops to 3.39°C. The core reason for this difference is the full-region phase change heat absorption of the composite PCM, which greatly improves the thermal buffering capacity of the system, enabling it to more efficiently absorb the transient heat generation of the battery during high-rate discharge, strongly flattening the temperature rise peak.

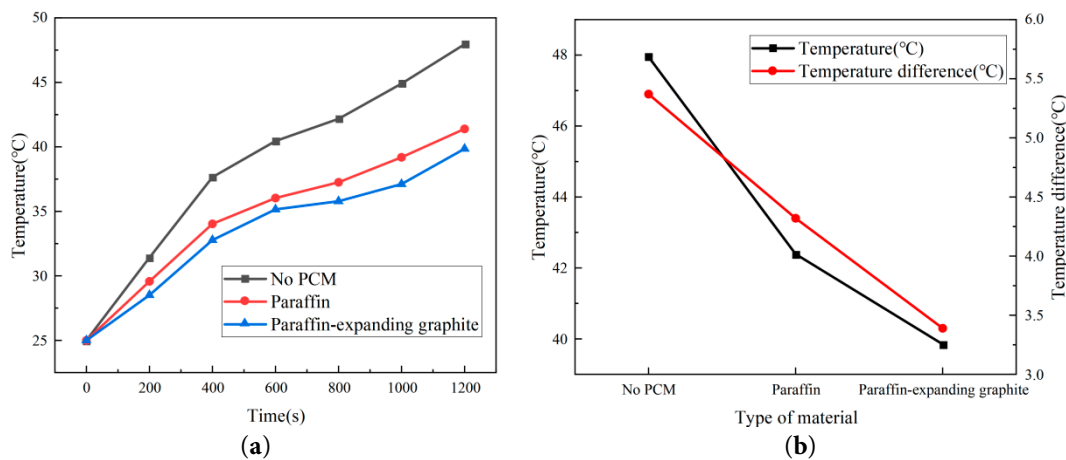


Figure 23: The impact of PCM on cooling performance. (a) The effect of PCM on temperature; (b) The effect of PCM on temperature difference.

Compared with pure paraffin, although the 9:1 mass fraction paraffin-expanded graphite composite material has decreased somewhat in latent heat of phase change, it has very strong thermal conductivity. This allows the composite material to conduct heat transfer faster, so that the vast majority of phase change material can exert its latent heat effect.

In summary, using this management method for battery thermal management, under a 3C discharge rate, the temperature can be reduced from 54.19°C to 39.85°C, a temperature reduction of 14.34°C, a decrease of 26.46%. The maximum temperature difference is reduced to 3.39°C, demonstrating that this thermal management system has very good temperature control effects. These optimization results are consistent with the laws of existing studies, verifying the rationality of the structural optimization settings in this paper.

5 Conclusion and Future Work

This paper takes the 21700 lithium-ion battery as the research object and addresses the thermal management problem under 3C high-rate discharge conditions.

5.1 Conclusion

Using a combination of numerical simulation and experimental verification, a multi-layer coupled thermal management system integrating honeycomb heat sink conduction, honeycomb channel liquid cooling, and phase change material heat storage is designed and optimized. The main conclusions are as follows:

- (1) Through 1C–3C discharge temperature rise experiments, it was found that the battery surface temperature rises significantly with the increase of the discharge rate. Under the 3C discharge condition, the battery surface peak temperature approaches 55°C, exceeding the safe operating temperature upper limit of 45°C, verifying the necessity of thermal management system design.
- (2) A thermal simulation model for the 21700 lithium-ion battery was established. The comparison between simulation and experimental data shows that the maximum errors at 1C, 2C, and 3C discharge rates are 2.78%, 3.62%, and 4.55%, respectively, verifying the accuracy and reliability of the simulation model.
- (3) A multi-layer coupled thermal management system was designed, consisting sequentially from the inside out of a battery, an aluminum alloy heat sink, a liquid cooling channel, and a phase change material, realizing the coordinated control of transient thermal shock suppression and steady-state heat discharge.
- (4) Through parameter optimization, the key parameters of the system were determined: the heat sink material is selected as 3003 aluminum alloy with a thickness of 2 mm; the channel adopts a dual-inlet counter-flow structure with an inner diameter of 1.6 mm; the coolant is a 50% by volume aqueous ethylene glycol solution with a flow velocity of 1.5 cm/s; and the phase change material adopts a 9:1 mass fraction paraffin-expanded graphite composite material.
- (5) Under the 3C discharge condition, the optimized composite thermal management system reduces the battery maximum temperature from 54.26°C to 39.85°C, a decrease of 14.41°C, and the maximum temperature difference is reduced to 3.39°C, meeting the safe operating temperature requirements of lithium-ion batteries.

5.2 Limitations

This study has several acknowledged limitations.

- (1) The composite thermal management system proposed in Section 4 is currently validated only by numerical simulations, with no physical experimental verification included in the present work. The accuracy of the predicted thermal performance remains to be further confirmed by experimental data.
- (2) Simplifications were adopted in the numerical model, such as treating the thermal conductivity of the phase change material (PCM) as a constant and idealizing the contact thermal resistance between the battery, PCM, and cooling components, which may introduce deviations from actual working conditions.
- (3) The system performance was only evaluated under specific conditions (1C–3C discharge rates at 25°C ambient temperature), and its behavior under extreme scenarios (e.g., high/low temperatures, fast charging) was not investigated.

5.3 Future Work

Corresponding to the above limitations, future research will focus on the following directions.

- (1) A physical experimental platform will be built to prepare a prototype of the composite system. Temperature monitoring tests will be conducted under different conditions to verify the simulation model and correct relevant parameters.
- (2) The system performance will be evaluated under extreme operating conditions, including high/low ambient temperatures, high-rate fast charging, and long-term cyclic discharge, to assess its stability and durability.
- (3) The numerical model will be improved by considering temperature-dependent PCM properties, interface contact thermal resistance, and battery aging effects to enhance simulation reliability.
- (4) Further structural optimization, such as adjusting the PCM composition ratio and optimizing the cooling channel layout, will be carried out to improve the thermal management performance under complex working conditions.

Acknowledgement: Not applicable.

Funding Statement: The authors received no specific funding for this study.

Author Contributions: Fei Wang and Jiaxin Liu drafted the main paper. Jiang Liu and Yongkang Niu provided technical support. Xilong Zhang made significant contributions during the revision of this paper. All authors reviewed and approved the final version of the manuscript.

Availability of Data and Materials: Data available on request from the authors.

Ethics Approval: Not applicable.

Conflicts of Interest: The authors declare no conflicts of interest.

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