

Investigation on topology optimization of cold plate for battery thermal management based on phase change slurry

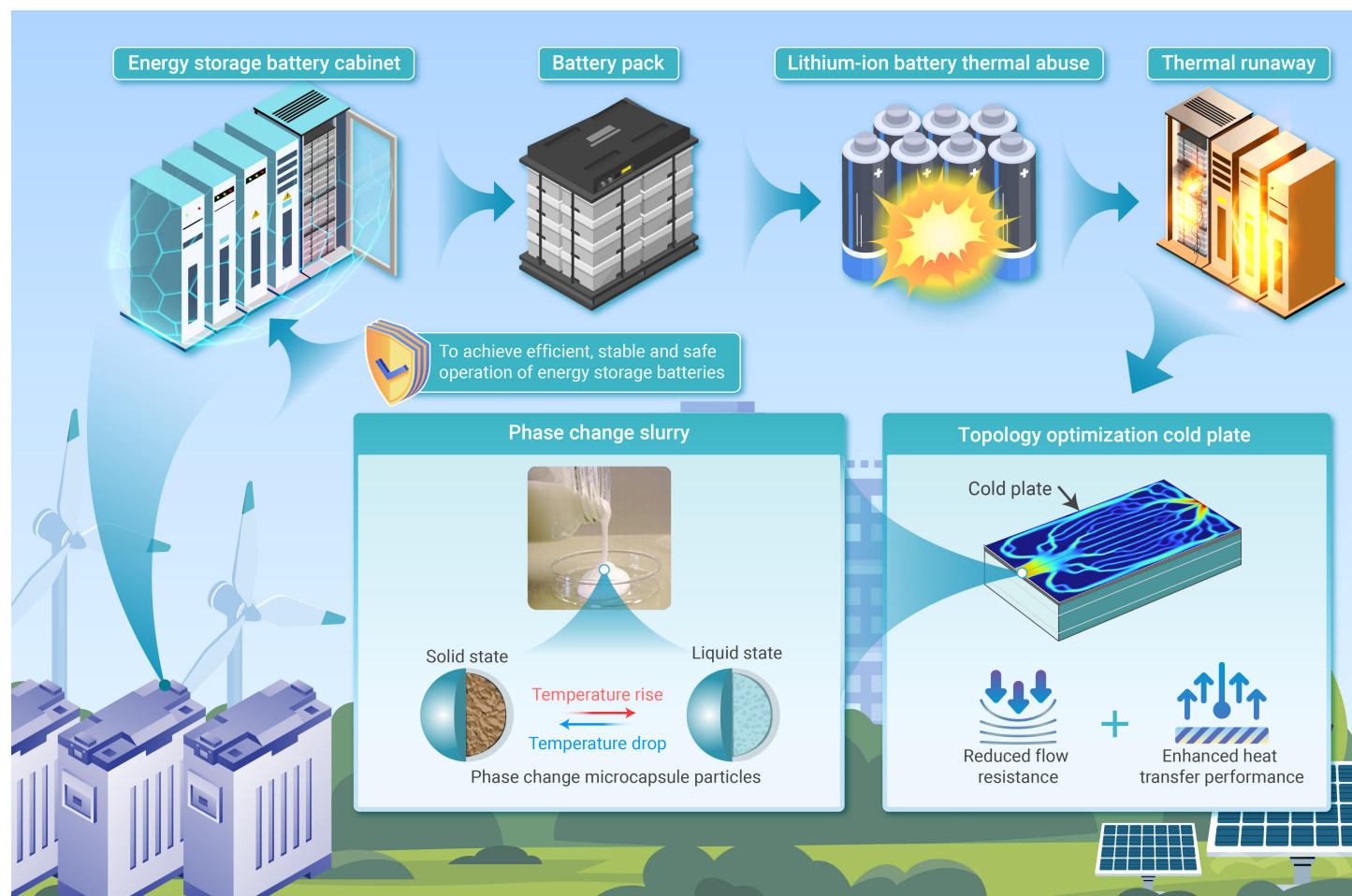
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GRAPHICAL ABSTRACT



PUBLIC SUMMARY

- Topology optimization of PCS-based cold plate for battery thermal management with multiple objectives is studied.
- TCP shows significant improvements in cooling performance and flow resistance compared to conventional RCP.
- PCS demonstrates superior cooling capabilities over pure water, enabling more uniform temperature distribution.

Investigation on topology optimization of cold plate for battery thermal management based on phase change slurry

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Addressing the issue that single liquid cooling/air cooling technology cannot meet the thermal management requirements of the battery under high power conditions, the topology optimization of the cold plate for battery thermal management based on phase change slurry (PCS) is numerically studied in this paper. The mathematical model of topology optimization is constructed with multi-objectives of minimizing both the average temperature of the cold plate surface and the power consumption of the fluid flow based on the variable density method. The numerical results of the topology optimization cold plate (TCP) are then compared with those of the conventional rectangular channel cold plate (RCP). Moreover, the effects of the PCS mass concentration, weight coefficient, volume fraction of the fluid domain, inlet velocity of coolant, heat source power on the cooling performance and flow characteristics of the cold plate are analyzed. The results indicate that the reduction of average temperature, root mean square temperature and flow resistance factor of TCP achieves 23.9%, 80.8%, 18.2%, respectively, compared with RCP by using PCS with 20% mass concentration. The PCS demonstrates superior cooling performance compared to pure water, resulting in a more uniform temperature distribution across the cold plate. Overall, this study offers a novel outlook on topology optimization of the cold plate based on PCS, providing an attainable approach for improving the cooling capabilities while reducing the pump power consumption of battery thermal management system (BTMS).

INTRODUCTION

The global energy transition is actively advancing. With the continuous increase of the scale of renewable energy generation, its characteristics of volatility and intermittency bring great challenges to the stable operation of the power system. Energy storage technology can effectively adjust the imbalance between the supply and demand of the power system. With flexible configuration and high energy density, battery energy storage technology can play different roles on the power supply side, the power grid side and the user side of the power system. However, with the continuous operation of battery energy storage power stations, their safety issues have become increasingly prominent. Lithium-ion battery (LIB) is a kind of energy storage battery with the best comprehensive performance, which has the advantages of high energy density, high cycle times, high conversion efficiency and fast response speed. The optimal operating temperature for LIB ranges between 20–40 °C, with a maximum temperature difference of 5 °C.¹ During the operation of the battery energy storage system, LIBs experience continuous charging and discharging cycles, resulting in the generation of heat from electrochemical reactions. The temperature of the LIB progressively rises with the accumulation of heat. Higher temperature significantly impacts the overall lifespan and performance of battery, and even causes thermal runaway and triggers explosions.^{2,3} Therefore, maintaining a reasonable temperature is vitally important for the operational efficiency and safety of the battery. The thermal management system is one of the prime methods to ensure efficient, stable, and safe battery operation.

Choudhari et al, Jaewan et al, and Li et al.^{4–6} have classified and elaborated on battery thermal management methods and systems in detail. Common battery thermal management technologies include air cooling,^{7,8} liquid cooling,^{9,10} phase change material (PCM) cooling,^{12–17} and heat pipe cooling.^{18,19} The cooling efficiency of air cooling is low and may not satisfy the cooling

requirements of batteries under high-power or extreme conditions.⁶ While the heat pipe cooling system is effective, it is expensive to manufacture and complex to install. PCMs could maintain a nearly constant temperature during phase change process, despite absorbing or releasing a significant amount of heat. It is beneficial for the thermal management of battery to control the temperature. However, it also presents challenges such as prolonged thermal response times and potential difficulties in timely heat transfer.

Liquid cooling using cold plates has been widely applied in battery pack due to their simple structure and high cooling efficiency.^{20–23} The conventional cold plates include the rectangular runner, S-shaped runner, and U-shaped types. Chen et al.²⁴ numerically studied the performance of the mini-channel cold plate with I-type, Z-type, and U-type flows. The maximum temperature difference ($\Delta T_{\max}$) of the heating surface and energy consumption (W_p) of the cold plate are, respectively, reduced by at least 43% and 61% after designing the edge width, convergence channel width and the symmetrical system. Fang et al.²⁵ investigated the effects of the coolant flow rate on the rectangular cold plates. The study demonstrated that the standard deviation of surface temperature decreased when the coolant flow rate increased from 0.065 kg/s to 0.115 kg/s. Zuo et al.²⁶ designed a double S-channel cold plate for prismatic LiFePO₄ batteries and found that the PEC is always higher than 1, which means the double S-channel cold plate is superior to the single S-channel structure in heat transfer efficiency and flow resistance. Patil et al.²⁷ conducted numerical studies on the U-turn type microchannel cold plate on both sides of lithium-ion pouch batteries. The results showed that optimized cooling parameters could maintain the maximum temperature and temperature non-uniformity of 50 V battery pack below 40 °C and 4 °C, respectively. The conventional cold plates design adopts the empirical design method, which relies on past experiences and repeated experiential design and verification to ensure that the structure meets the required design standards. However, it often leads to high costs and prolonged timelines.

Topology optimization, a type of structural optimization, is employed to determine the optimal material distribution within a given design domain subject to the specified constraints and objectives. It offers more freedom in structural shape than conventional optimization, resulting in unique flow configuration and excellent cooling effect. The application of topology optimization can also reduce the optimization time and cost caused by trial and error in the optimization process. Bendsøe and Kikuchi²⁸ first introduced topology optimization method to design mechanical structures capable of withstanding specified loads. Borrvall and Petersson²⁹ expanded its application to fluid flow, and later to convective heat transfer problems.³⁰ In recent years, topology optimization method has been widely applied to the design of cold plate for thermal management. Li et al.³¹ proposed a water-cooled heat sink design method based on topology optimization. The results showed that the surface maximum temperature with topology optimization channel is 11.7% lower than that of parallel design. Lee et al.³² investigated the performance of a new internal structure for water-cooled heat sink. The results showed that the varying-pitch cylindrical pin-fin structure inside the heat sink had advantages in higher heat transfer performance, lower pressure drop and the manufacturability. Zhang et al.³³ conducted topological optimization for a nanofluid heat radiator. The effects of energy dissipation constraints, temperature-dependent fluid properties, and nanofluid characteristics on the optimal configuration were examined. To improve the thermal performance of PCM-based heat sink, Ho et al.³⁴ used topology optimization to design and

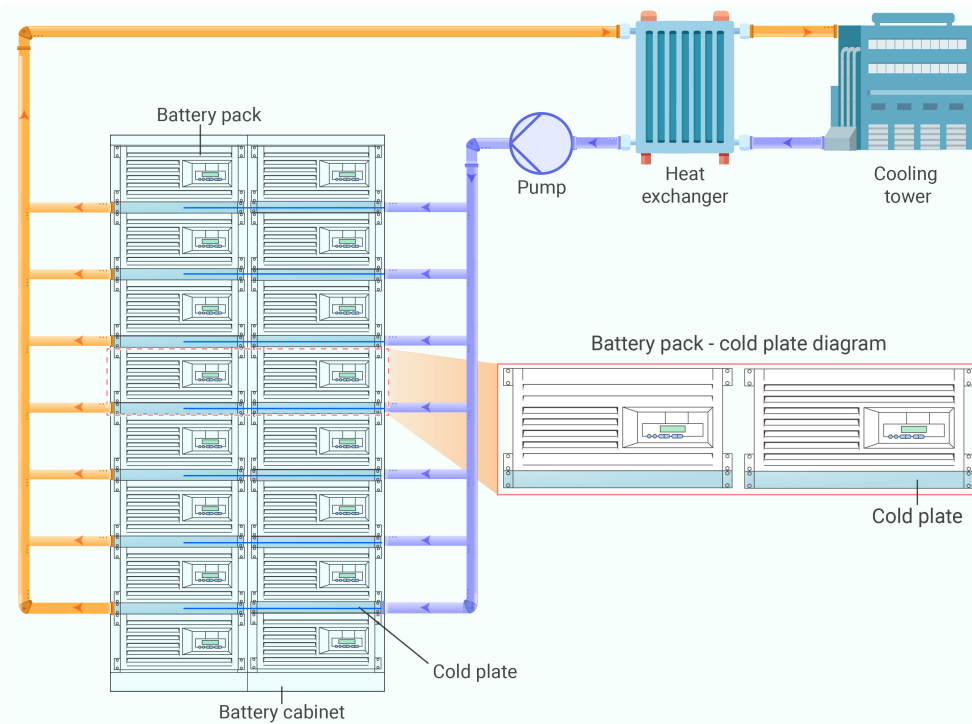


Figure 1. Schematic diagram of the BTMS.

MATERIAL AND METHODS

Physical model

Figure 1 presents the schematic diagram of the BTMS in this study. The system uses PCS as the cooling medium to transfer the heat generated by each layer of battery pack. The heat is transferred through the PCS in the cold plate to the heat exchanger, and then dissipated into the environment via the cooling tower. The battery packs and cold plates are arranged in a sandwich-type configuration, where two cold plates are provided for each layer. Considering that each layer of the battery packs in the battery cabinet are connected in series, the current remains constant, resulting in similar heating situations. Therefore, the research on the performance of the cold plate for a single battery pack can provide a basis for the subsequent analysis of the performance for the whole BTMS. The topology optimization of the cold plate for a single battery pack is conducted.

This study implements topology optimization

on a two-dimensional (2D) cold plate, the initial topology model is presented in Figure 2. The fluid inlet and outlet sections are located at the center of the left and right sides of the cold plate. The lengths and widths of inlet and outlet are all $L_1=80$ mm. The length and width of central rectangular section is $L=580$ mm and $W=470$ mm, respectively.

The boundary conditions for the TCP are also depicted in Figure 2. There is no gap between the cold plate and the battery, and the thermal contact resistance between them is negligible. The heat generated by the battery is transferred through the upper surface of the cold plate to the flow channel inside of the cold plate, and then removed by the coolant flowing through the cold plate. In this study, the heat flux Q transferred to the cold plate is presumed to be uniform based on the heat generation rate characteristics of the battery at high C-rate.³⁹ For the coolant, the inlet temperature T_{in} and inlet velocity u_{in} are constant. The outlet is set to atmospheric pressure p_{out} . The parameters calculated for topology optimization are shown in Table S1.

Mathematical model

Governing equation. The following are the assumptions about the flow and heat transfer in the design domain:

- (1) Heat dissipation is realized in the form of heat conduction and convection.
- (2) It is assumed that the mPCM particles are uniformly dispersed in the carrying fluid. Since the mass concentration of the PCS used in the study is less than 25%, the PCS is considered as a single-phase Newtonian fluid in the study.³⁶
- (3) The latent heat during the phase change process of mPCM particle is assumed to be equivalent to the specific heat capacity within the phase change temperature range. The physical parameters of the PCS (such as thermal conductivity, density, and dynamic viscosity) are assumed to be temperature independent.

The inlet velocity ranges from 0.005 m/s to 0.025 m/s to ensure that the Reynolds number of coolant flow in the channels of cold plate is generally between 80 and 2000, which is in the laminar flow state. Assuming that the coolant in the cold plate is an incompressible steady-state fluid, the governing equations of the PCS are as follows:

Continuity equation:

$$\nabla \cdot \mathbf{u} = 0 \quad (1)$$

Momentum equation:

fabricate a novel tree-shaped heat sink structure. The results shows that the TO tree-like structure heat sink has better performance, exhibiting up to 4 °C lower wall temperatures, than the conventional fin-structure heat sink. However, the cooling medium in most existing studies of TCP is water, which relies on sensible heat of temperature difference to transport heat and has low heat carrying density.

PCS is a kind of latent heat type, solid-liquid two-phase functional fluid. It consists of micro-encapsulated PCM (mPCM) and carrying fluid.³⁵ It combines the advantages of mPCM and liquids. Within a certain temperature range, microcapsules undergo a phase change, resulting in an increased apparent specific heat. This allows the PCS to absorb more heat at the same flow rate compared with other cooling media. PCS also exhibits excellent fluidity and heat transfer capability while not occupying additional design space, which can play an important role in battery cooling. Bai et al.³⁶ used PCS in RCP to achieve the thermal management of lithium-ion soft-pack batteries. The results showed that the cooling performance of 20 wt% PCS was better than that of pure water, glycol solution and mineral oil when the mass flow rate was less than 3×10^{-4} kg/s. To study the feasibility of applying PCS in BTMS, Li et al.³⁷ numerically established a BTMS model based on the U-channel liquid-cooled plate. They found that compared with natural convection condition, the liquid-cooling conditions at the end of 5C discharge can greatly reduce the maximum temperature of the tabs and on the battery body, but increase the maximum temperature difference on the battery body. Chen et al.³⁸ used PCS as the cooling medium for the cold plate constituted by a semi-elliptical serpentine tube. The results shows that compared with the blank module, GE-MPCMS-Si could reduce the battery module temperature and the temperature variance by 14.49 °C and 3.8 °C~2, respectively. At present, the research has mainly focused on the application of PCS in conventional cold plates, but there is rare investigation into the combination of PCS with TCP for BTMS.

In this study, PCS is utilized as the cooling medium in the cold plate designed by topology optimization method to effectively cool the battery. A two-dimensional cold plate model is established to design the flow channels of PCS using the topology optimization method. Initially, the cooling performance and flow characteristics of TCP are compared with those of RCP. Subsequently, the effects of the PCS mass concentration, weight coefficient, fluid domain volume fraction, inlet velocity, and heat source power on the performance of TCP are analyzed. The simulation results can be used as a reference for the arrangement of experimental scheme and the engineering application.

$$\rho(\mathbf{u} \cdot \nabla) \mathbf{u} = -\nabla p + \mu \nabla^2 \mathbf{u} + \mathbf{F} \quad (2)$$

where $\mathbf{u}$ is the velocity, ρ is the density, p is the pressure, μ is the viscosity, $\mathbf{F}$ represents the friction force in the solid domain, and can be evaluated as a Brinkman friction term in the porous medium.⁴⁰

$$\mathbf{F} = -a\mathbf{u} \quad (3)$$

where, the inverse permeability a is the Darcy interpolation function related to the design variable γ .⁴¹

$$a(\gamma) = a_f + (a_s - a_f) \frac{q(1-\gamma)}{q+\gamma} \quad (4)$$

where q is the penalty factor for the reverse permeability, which is 0.01. The subscript s and f represent solid and fluid material. a_f is 0 and a_s can be expressed as:⁴²

$$a_s = \frac{\mu}{Da \cdot L^2} \quad (5)$$

where Da is the Darcy number. Referring to the study on the effect of Da on the flow and topology optimization results in the literature,⁴³ Da is set to 10^{-5} in the study.

In fluid design domain with $\gamma = 1$, a equals 0, which means the friction force term is zero and the fluid velocity is not penalized. In solid design domain where $\gamma = 0$, a is infinite, which means the friction force term tends to infinity and the fluid velocity is penalized to zero.

Energy equation:

$$\rho C_p (\mathbf{u} \cdot \nabla T) = \nabla \cdot (k \nabla T) + Q \quad (6)$$

where C_p is the constant pressure specific heat capacity of the working fluid,

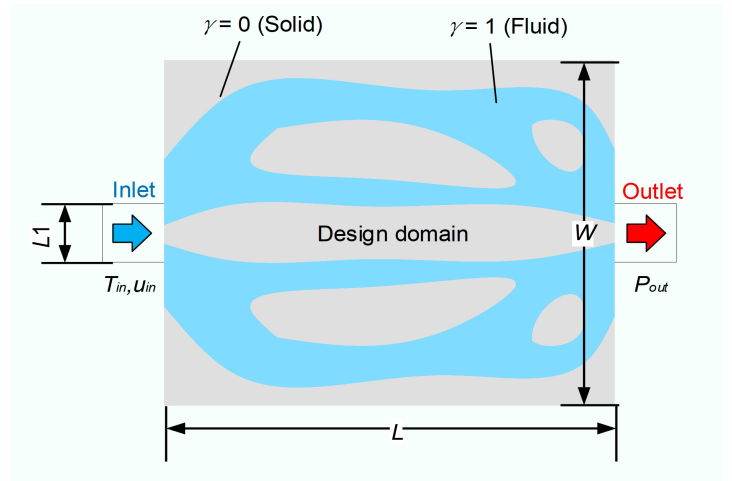


Figure 2. Two-dimensional model diagram of cold plate.

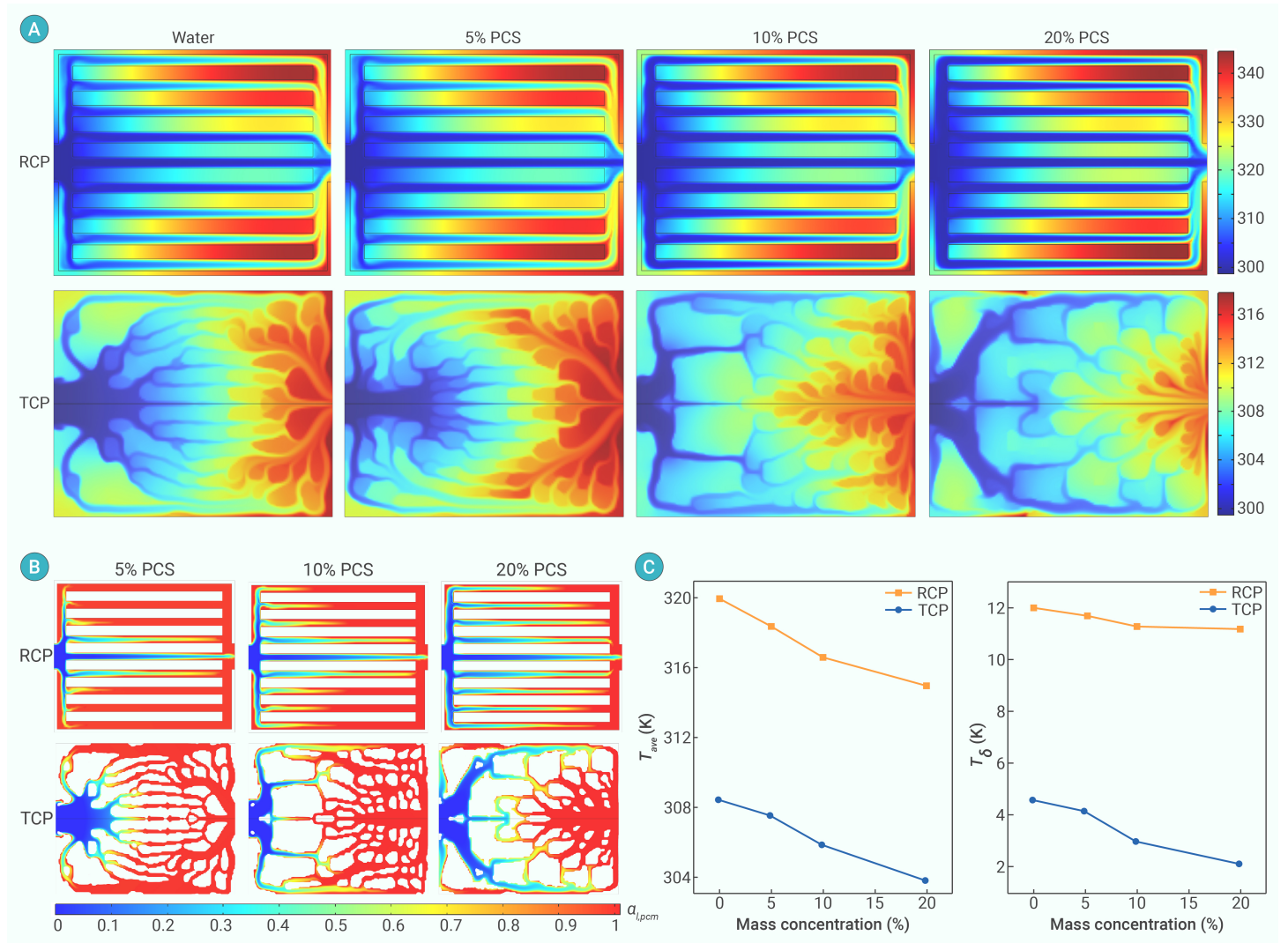


Figure 3. (A) Surface temperature distribution of RCP and TCP, (B) liquid phase volume fraction of PCS, (C) average temperature and root mean square temperature of TCP and RCP at different PCS concentrations.

T is the local temperature, k is the thermal conductivity of the fluid, Q is the cold plate heat source in the design domain.

The same interpolation function is also applicable to the density ρ , heat transfer coefficient k , and specific heat capacity C_p .⁴⁴

$$\rho(Y) = \rho_i + (\rho_s - \rho_i) \frac{q_p(1-Y)}{q_p + Y} \quad (7)$$

$$k(Y) = k_i + (k_s - k_i) \frac{q_k(1-Y)}{q_k + Y} \quad (8)$$

$$C_p(Y) = C_{p,f} + (C_{p,s} - C_{p,f}) \frac{q_{c_p}(1-Y)}{q_{c_p} + Y} \quad (9)$$

where q_k , q_p , and q_{c_p} represent penalty factors for thermal conductivity, density, and specific heat capacity with values of 0.01, 0.01, and 100, respectively.

Effective specific heat capacity of the PCS. In this study, the PCS can be considered as a single-phase Newtonian fluid due to the mass concentrations of the PCS are between 5-20% and the assumption of uniformly distributed mPCM particles.³⁶ The material of the cold plate and the thermo-physical properties of the PCS with different mass concentrations are shown in Table S2.

The latent heat of PCS can be modeled using the equivalent specific heat capacity method during the phase change temperature range.³⁷ In this study, the phase change temperature range of the PCS is defined from 299.7 K to 303.9 K. Eq. (10), which represents the equivalent specific heat capacity of the PCS, can be employed to calculate as follows:⁴⁶

$$C_{p,f} = \begin{cases} C_{p,ms}, & T < T_s \\ C_{p,ms} + \frac{h_f}{T_i - T_s}, & T_s \leq T \leq T_i \\ C_{p,ms}, & T > T_i \end{cases} \quad (10)$$

where $C_{p,ms}$ is the specific heat capacity of the PCS without phase change. h_f is the latent heat of phase change of PCS. The phase change process of a PCM progresses in the temperature range of T_s (solidus temperature, 299.7 K) to T_i (liquidus temperature, 303.9 K).

The liquid phase volume fraction of the PCM during the phase change process is critical for studying the melting process of the PCS. According to the equivalent specific heat model⁴⁷ utilized, the liquid phase volume fraction of the PCM could be estimated based on the temperature of the PCS as follows:

$$\alpha_{l,pcm} = \frac{\int_{T_i}^T c_{p,f} dT - (T - T_i) c_{p,ms}}{h_f}, T_s \leq T \leq T_i \quad (11)$$

When $T < T_s$ or $T > T_i$, $\alpha_{l,pcm} = 0$ or $\alpha_{l,pcm} = 1$.

Multi-objective optimization model. The purpose of the liquid cooling thermal management system for the cold plate in this study is to effectively cool the battery pack while reducing the pump power consumption of the system. It can be achieved by improving the cooling capacity of the cold plate and reducing the flow resistance of the fluid. Therefore, the objective of topology optimization for the cold plate in this study are twofold: maximizing the cooling capacity of the cold plate and minimizing the flow resistance. The greater the cooling capacity of the cold plate, the lower the average temperature of the design domain when subjected to a certain heat source. Consequently, this study adopts the minimum average temperature of the cold plate as the objective function for topology optimization aimed at enhancing cooling capacity. Regarding flow resistance, lower power consumption correlates with a reduced pressure drop between the inlet and outlet of the cold plate. Therefore, the optimization goal is to minimize the power consumption of the cold plate for fluid flow. The average temperature and flow power consumption are quantified by the following equation:⁴⁸

$$\Phi_T = \frac{\int_{\Omega} T d\Omega}{\int_{\Omega} 1 d\Omega} \quad (12)$$

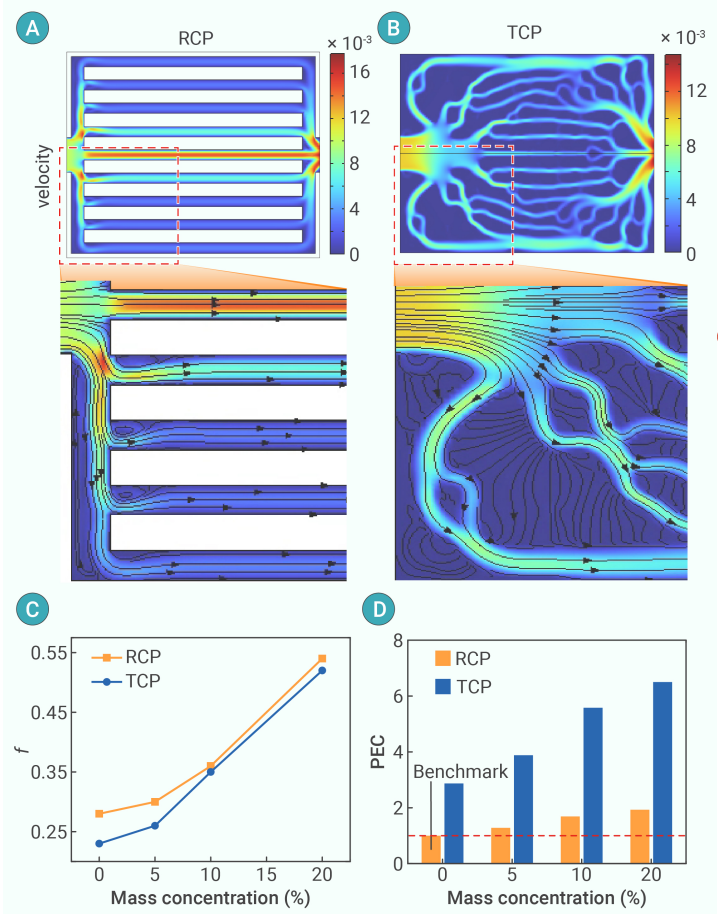


Figure 4. Numerical results of TCP and RCP at different PCS concentrations (A) Velocity distribution of RCP, (B) velocity distribution of TCP, (C) flow resistance factor, and (D) performance evaluation criterion.

$$\Phi_f = \int_{\Omega} \left[\frac{1}{2} \mu \sum_{ij} \left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right)^2 + \sum_i a(Y) u_i^2 \right] d\Omega \quad (13)$$

Considering the different magnitude of the two objective functions, the above objectives are normalized with $\Phi_{T,0}$ and $\Phi_{f,0}$ respectively, as shown below:

$$\overline{\Phi}_T = \frac{\Phi_T}{\Phi_{T,0}}, \overline{\Phi}_f = \frac{\Phi_f}{\Phi_{f,0}} \quad (14)$$

where subscript 0 represents the initial value of the objective function, which is calculated if the entire design domain is filled with fluid.

The multi-objective function Φ that combines two objectives $\overline{\Phi}_T$ and $\overline{\Phi}_f$ is constructed using weight coefficients:

$$\Phi = w_T \overline{\Phi}_T + w_f \overline{\Phi}_f \quad (15)$$

where w_T and w_f represent the weight coefficient of average temperature and the weight coefficient of power dissipation. The sum of the two weight coefficients is 1.

In addition, the volume fraction of the fluid domain (V_f) in design domain is limited, which is 0.5 except for section 3.3 in this study. The constraints are as follows:⁴⁹

$$\int_{\Omega} Y d\Omega / \int_{\Omega} 1 d\Omega \leq V_f \quad (16)$$

The mathematical model of topology optimization can be expressed as follows:

$$\text{Find } Y_i (i = 1, 2, \dots, N)$$

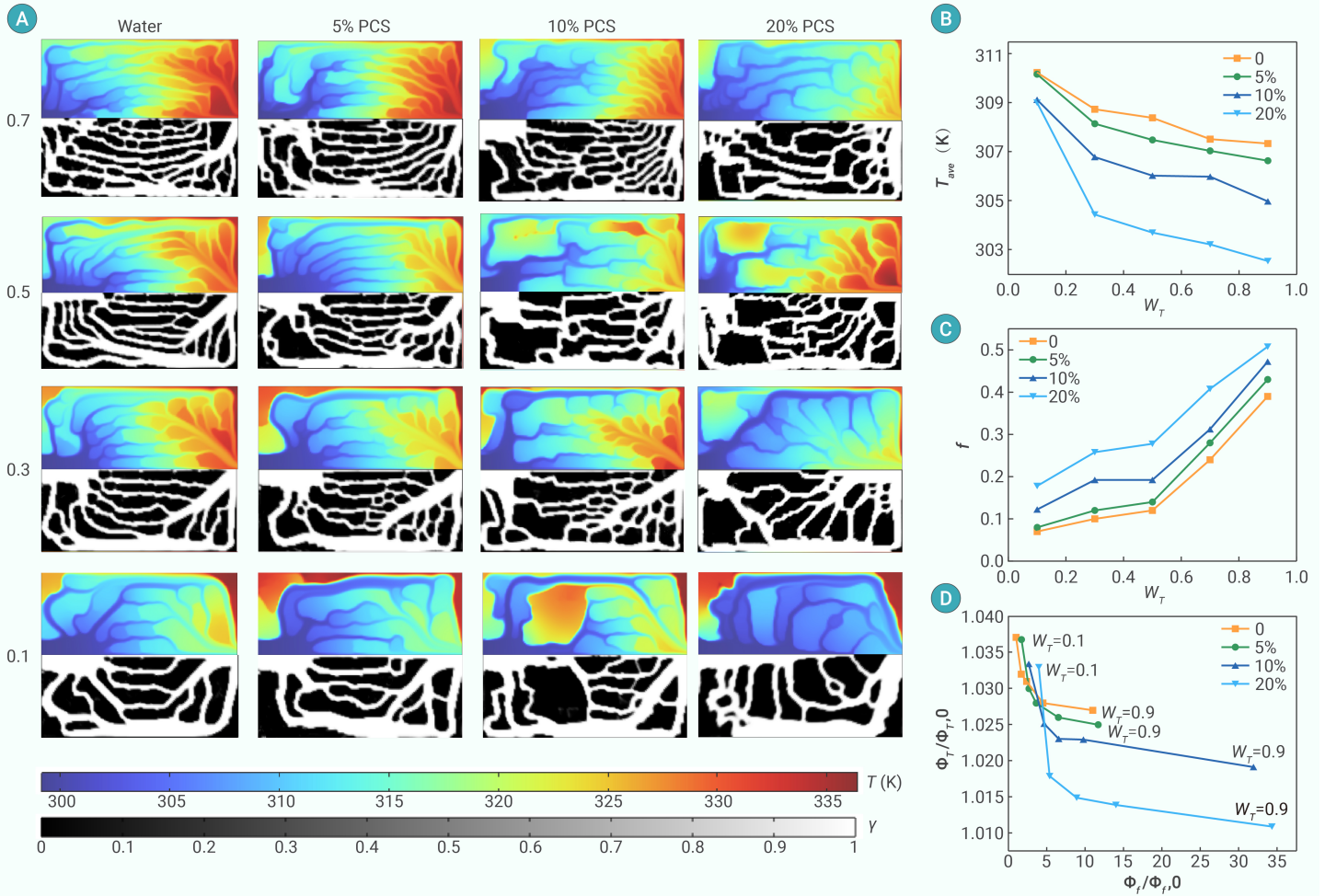


Figure 5. Numerical results of TCP with different weight coefficients (A) The temperature distribution and flow channel structure, (B) average temperature, (C) flow resistance coefficient, and (D) double objective Pareto boundary.

$$\text{Minimize } \Phi = w_T \frac{\Phi_T}{\Phi_{T,0}} + w_F \frac{\Phi_F}{\Phi_{F,0}}$$

$$\text{Subject to } \begin{cases} \nabla \cdot \mathbf{u} = 0 \\ \rho(\mathbf{u} \cdot \nabla \mathbf{u}) = -\nabla p + \mu \nabla^2 \mathbf{u} - \mathbf{F} \\ \rho C_p (\mathbf{u} \cdot \nabla T) = \nabla \cdot (k \nabla T) + Q \\ \int_{\Omega} \gamma d\Omega / \int_{\Omega} 1 d\Omega \leq V_F \\ 0 \leq \gamma_i \leq 1 \quad (i = 1, 2, \dots, N) \end{cases} \quad (17)$$

Filtration and projection. In the design of topology optimization, there are some issues such as grid dependence and checkerboard patterns. Thus, it is essential to filter the topology optimization results to mitigate these problems. This study employs the Helmholtz partial differential equation⁵⁰ to filter the topology optimization results. The expression for density filtering is as follow:

$$-r^2 \nabla^2 \tilde{\gamma} + \tilde{\gamma} = \gamma \quad (18)$$

where r is the filtration radius, γ is the design variable before filtration, and $\tilde{\gamma}$ is the design variable after filtration. In this study, the filter radius r is set as the maximum mesh size of the design domain.

After the filtering process, a significant number of gray units ($\tilde{\gamma} \in (0,1)$) are produced. The hyperbolic tangent projection is utilized to project the filtered design variable, thereby alleviating the extent of the grayscale region between the fluid and solid domains.⁵¹ The expression for the projection is as follow:

$$\gamma_i = \frac{\tanh(\beta(\tilde{\gamma}_i - \gamma_\beta)) + \tanh(\beta\gamma_\beta)}{\tanh(\beta(1 - \gamma_\beta)) + \tanh(\beta\gamma_\beta)} \quad (19)$$

where γ_i is the design variable after projection, γ_β is the projection point and β is the projection slope, which are set as 0.5 and 8 respectively.

Numerical process

The optimization process is implemented on the finite element software COMSOL Multiphysics 6.0. The numerical process of topology optimization is shown in Fig. S1 of Supplementary Information. First, an initial design variable field is given. Second, the design domain is divided into unstructured triangular meshes. Third, the governing equations are solved based on the finite element method. Fourth, the sensitivity analysis is conducted based on the adjoint method. Fifth, the design variable will be updated by the optimization solver based on the results of the sensitivity analysis. The moving asymptote method (MMA) is adopted. Finally, the topology optimization results are filtered and projected to obtain black and white clear results, which are two-dimensional design results. If the convergent criterion is satisfied when the max residual of design variable is less than 10^{-4} , the simulation will be terminated. If not, return to the third step and repeat the calculation process.

Performance index

The average surface temperature of cold plate (T_{ave}) is used to evaluate the heat dissipation performance, which can be expressed as:

$$T_{ave} = \frac{\int_{\Omega} T d\Omega}{\int_{\Omega} d\Omega} \quad (20)$$

The root mean square temperature (T_b) is used to represent the temperature uniformity of the battery.⁵² The smaller the T_b , the more uniform the temperature distribution over the cold plate. It can be calculated by:

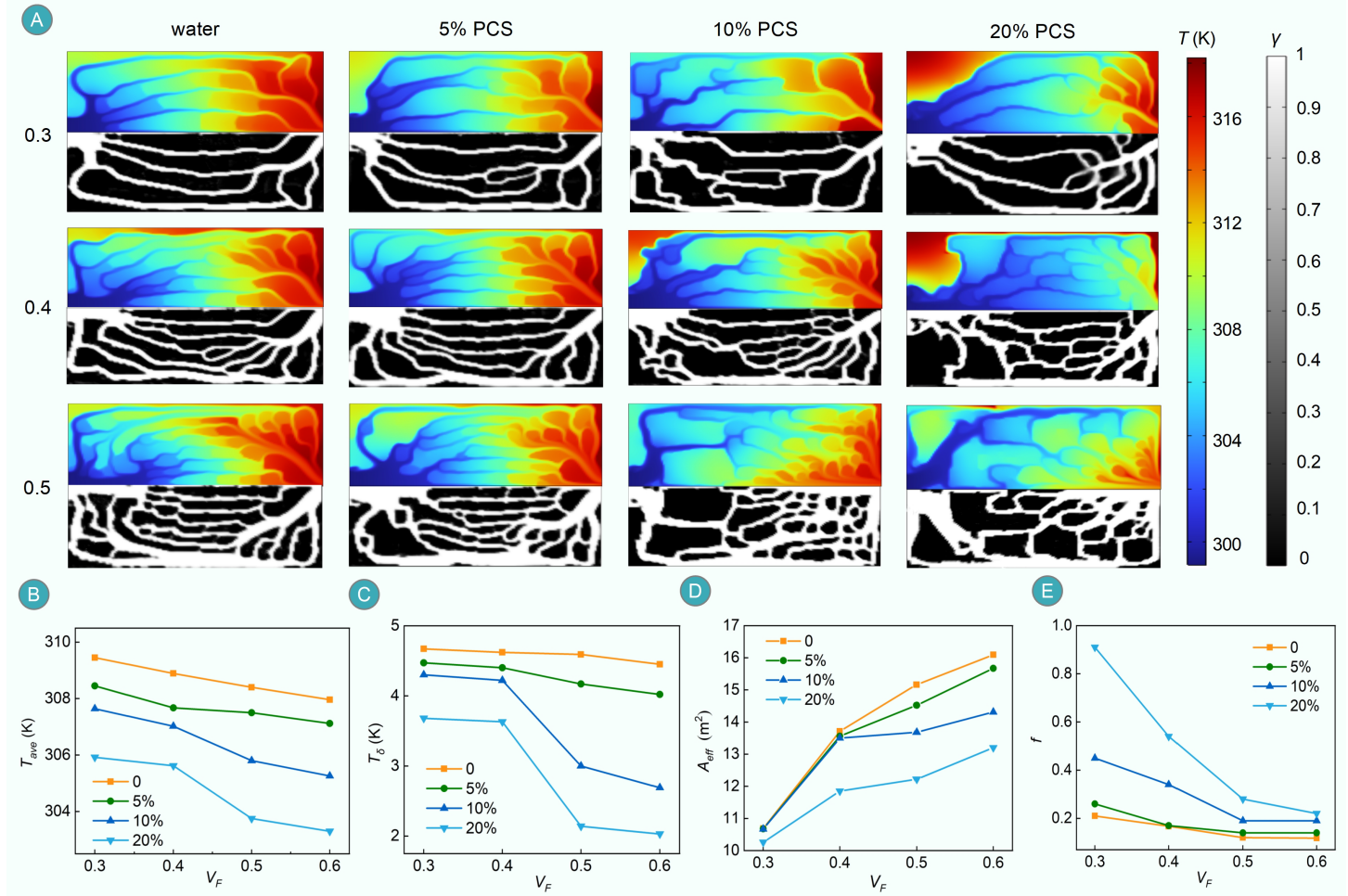


Figure 6. Numerical results of TCP at different volume fractions of fluid domain (A) The temperature distribution and flow channel structure, (B) average temperature, (C) root mean square temperature, (D) effective heat transfer area and (e) flow resistance factor.

$$T_\delta = \sqrt{\frac{\int_\Omega (T - T_{ave})^2 d\Omega}{\int_\Omega d\Omega}} \quad (21)$$

The Nusselt number (Nu) is used to analyze the heat transfer performance between the fluid and solid:

$$Nu = \frac{h_{avg} D_h}{k_f} \quad (22)$$

where D_h is the hydraulic diameter:

$$D_h = \frac{4A_c}{P_w} \quad (23)$$

where A_c is the cross-section area of the channels, P_w is the wetted perimeter of the channels. h_{ave} is the average heat transfer coefficient, which is defined as:

$$h_{ave} = \frac{Q}{A_{eff} (T_{ave} - T_{f,ave})} \quad (24)$$

where A_{eff} denotes the effective heat transfer areas between solid and fluid. $T_{f,ave}$ represents the average temperature of fluid.

The flow resistance factor (f) is used to evaluate the flow power consumption of the system.⁵³

$$f = \frac{D_h}{L_i} \frac{2\Delta p}{\rho_i u^2} \quad (25)$$

where L_i is the length of the flow channel from the inlet to the outlet.

The performance evaluation criterion (PEC) is used to evaluate both hydrodynamic and thermal performance of the designed flow channel structure,⁵⁴ which is expressed as follows:

$$PEC = \frac{Nu/Nu_{ref}}{(f/f_{ref})^{1/3}} \quad (26)$$

where Nu_{ref} and f_{ref} are reference values of Nu and f , respectively, defined by the performance of RCP under pure water conditions in this study.

Model validation

Grid independence validation. In order to balance the relationship between simulation accuracy and computing resources, the independence of the grid is tested. In this study, the grid is divided into free quadrilateral grids, which are faster in calculation and more accurate in results, as shown in Figure S2 of Supplementary Information. The verification result of grid independence is shown in Figure S3 of Supplementary Information. The grid numbers for verification are 1×10^5 , 3×10^5 , 5×10^5 , 7×10^5 , 9×10^5 and 1.2×10^6 , respectively. When the grid number is more than 7×10^5 , the relative errors of maximum temperature and root mean square temperature caused by continuing to increase the grid number are less than 0.14% and 0.52%, respectively. Therefore, in order to save computing resources, this study divides the grid number of 7×10^5 for the model.

Model method validation. In order to verify the accuracy of the numerical results of the topology optimization model, this study chooses the experiment of Zhou⁴⁴ for comparison. The experimental results of T2 are selected for comparison in this study. The results of comparison between this study and experiment at $Re=1715$ are shown in Figure S4. It can be seen that the numerical results are consistent with the experimental results and are thus highly reliable. The absolute difference of average temperature of cold plate does not exceed 1 K, and the relative deviation does not exceed 0.5%.

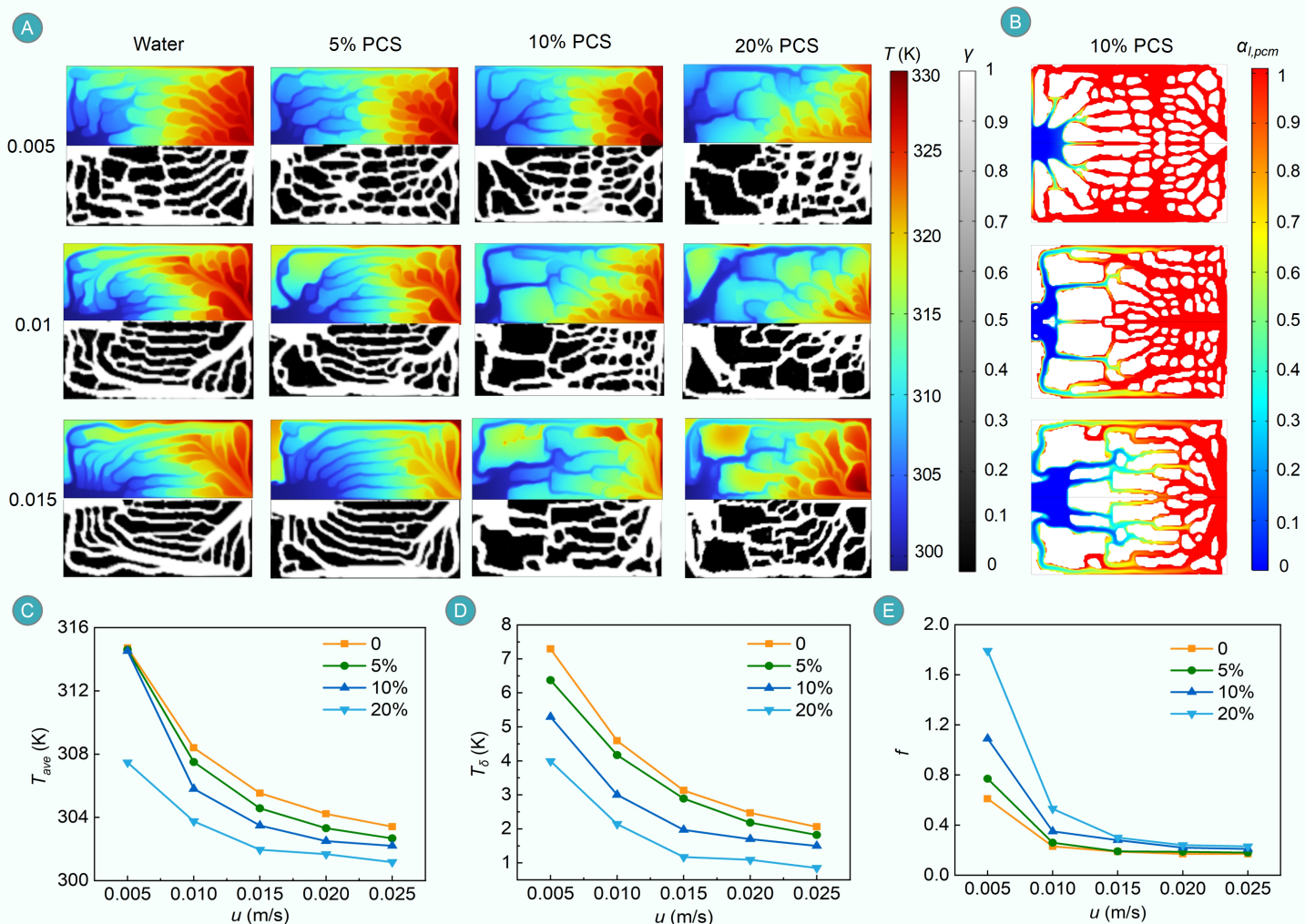


Figure 7. Numerical results of TCP at different inlet velocities (A) Temperature distribution and flow channel structure, (B) liquid phase volume fraction, (C) average temperature, (D) root mean square temperature and (E) flow resistance factor.

Results and discussion

Comparison of the performance of TCP and RCP

Cooling performance. To compare the performance differences between the TCP and RCP, an additional RCP model with the same fluid domain volume fraction of 50% as the TCP was established. The boundary conditions for both RCP and TCP are identical. Figure S5 in Supplementary Information shows the established RCP model, with specific dimensions detailed in Table S3.

Figure 3A shows the surface temperature distribution of the TCP and RCP at varying PCS concentrations. When water is used as the working fluid, the maximum temperatures of the TCP and RCP are 340 K and 316 K, respectively. The maximum temperature of the TCP is 24 K lower than that of the RCP, indicating superior cooling performance for the TCP. Additionally, as the PCS concentration increases, the maximum surface temperature of both the TCP and RCP decreases. Figure 3B shows the distribution of the liquid phase volume fraction of the melted PCM within the TCP and RCP. The liquid phase volume fraction gradually increases with the flow distance. Furthermore, for both TCP and RCP, an increase in PCS mass concentration results in a slower melting rate of the mPCM particles and a longer distance required for complete melting of particles.

Figure 3C shows the temperature values of TCP and RCP at different PCS concentrations. As shown in Figure 3C, T_{ave} of both RCP and TCP decreases with the increase of PCS concentration. Specifically, T_{ave} of RCP and TCP decreases by 10.7% and 13.2%, respectively, when 20% PCS is used as the coolant compared to water. This indicates that PCS exhibits superior cooling performance compared to pure water, and the cooling capacity of the cold plate can be improved by using PCS. Furthermore, T_{ave} of TCP consistently

remains lower than that of RCP at different PCS concentrations, with a maximum reduction of 23.9%. It demonstrates that topology optimization can further improve the cooling performance of the cold plate.

In addition, Figure 3C shows that T_b of both RCP and TCP decreases with the increase of PCS concentration. When 20% PCS is used as the coolant compared to water, T_b of RCP and TCP decreases by 6.8% and 53.5%, respectively. This suggests that PCS can improve the uniformity of the temperature distribution in the cold plate compared to pure water. Additionally, at different PCS concentrations, T_b of TCP is consistently lower than that of RCP, with a maximum reduction of 80.8%, indicating that the temperature distribution in TCP is more uniform. This result further confirms that topology optimization enhances the uniformity of the temperature distribution of cold plate.

Flow performance. Figure 4A-B shows the velocity distributions of different cold plate types. In the RCP, the central channel is directly connected to the entrance, resulting in significantly lower flow resistance in the middle flow channel compared with that in the side channels. Consequently, the coolant flows more rapidly through the middle channel, leading to a non-uniform velocity distribution. Conversely, the main channel of the TCP is subdivided into numerous smaller secondary channels. The similar flow resistance among these smaller channels ensures a uniform velocity distribution within the TCP. Additionally, as shown in the magnified view of the velocity in Figure 4B, the TCP channels exhibit a near-streamline shape, which minimizes the presence of sharp corners and, in turn, reduces the pressure drop.

Figure 4C-D show the flow resistance factor and performance evaluation criterion for RCP and TCP at different PCS concentrations, respectively. Figure 4C indicates that f of TCP is lower than that of RCP at different

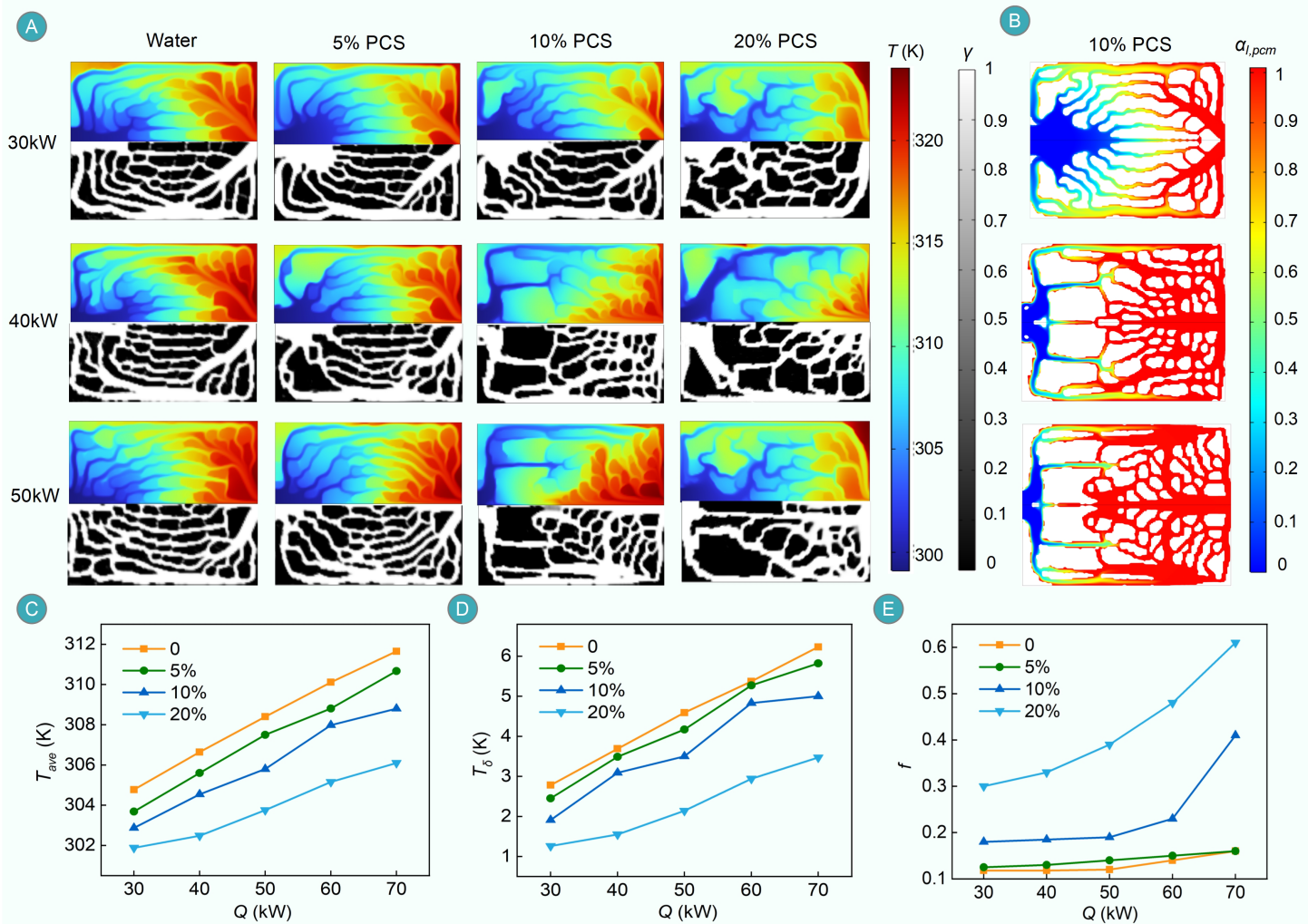


Figure 8. Numerical results of TCP at different heat source power (A) Temperature distribution and flow channel structure, (B) liquid phase volume fraction, (C) average temperature, (D) root mean square temperature and (E) flow resistance factor.

concentrations, with a maximum reduction of 18.2%. Therefore, TCP can achieve effective flow resistance reduction at different PCS concentrations. As seen from Figure 4D, for a given concentration of PCS, the performance of the RCP with pure water is used as the reference value. For the RCP, the cooling effect is enhanced when the coolant is PCS. The overall performance of TCP is superior to that of RCP at different PCS concentrations.

The influence of weight coefficients

For multi-objective topology optimization, it is necessary to consider the influence of the weight coefficient on the optimization results. Figure 5A shows the flow channel structure and temperature distribution at different weight factors w_T . The upper part of the figure represents the temperature distribution. The lower part represents the design variable distribution, where the black region indicates the solid structure of the cold plate and the white region delineates the flow channels.

From Figure 5A, it is observed that at the same PCS concentration, the number of flow channel branches decreases when w_T is reduced. There are no extra bent flow channels, resulting in low flow resistance. However, the temperature distribution in the cold plate is apparently nonuniform at smaller w_T . The temperature of areas away from the flow path is significantly higher than that of other areas due to the poor local heat transfer. The flow channel branches continue to increase with the increase of w_T , raising the complexity of the flow channel structure.

In addition, as shown in Figure 5A, at the same weight coefficient, the number of flow channel branches in the front region of the cold plate decreases with increasing PCS concentrations. This is because the higher concentration of PCS, the more mPCM particles contained in the unit volume

of fluid, and therefore the more heat absorbed by the mPCM particles during melting. The temperature in the front region of the cold plate is generally within the phase change temperature range of the mPCM particles, allowing mPCM particles to absorb latent heat. Consequently, fewer branches in the front area can satisfy the heat dissipation effect. However, in the rear region of the cold plate, the temperature exceeds the phase change temperature range, resulting in the complete melting of the mPCM particles. Here, heat is absorbed by the PCS as sensible heat, necessitating more branches to achieve the same heat dissipation effect.

Figure 5B-C show that the average temperature of the cold plate gradually decreases, while the flow resistance increases with the increase of w_T . Figure 5D shows the double objective Pareto boundary of the cold plate at different PCS concentrations. The Pareto boundary represents the influence of weight factor w_T on topology optimization objectives $\overline{\Phi}_T$ and $\overline{\Phi}_f$. It is observed that with increasing w_T , $\overline{\Phi}_T$ decreases, while $\overline{\Phi}_f$ shows an opposite trend at varying PCS concentrations. When w_T is less than 0.1 or greater than 0.9, $\overline{\Phi}_T$ and $\overline{\Phi}_f$ are approximately equal to extreme values, respectively, which makes the trade-off between the flow and heat transfer performance of the system inefficient. In this study, w_T is appropriately controlled within the range of 0.3 to 0.7 to ensure an effective trade-off between the two optimization objectives.

Influence of volume fraction of fluid domain

The volume fraction of fluid domain in the study represents the ratio of the fluid domain area to the entire cold plate. Figure 6A demonstrates that the flow channels are predominantly positioned in the central region of the cold plate at low volume fraction (V_f) of fluid domain. As V_f grows, the flow paths near the inlet and outlet extend towards the corners of the cold plate. The

accompanying increase in the number of flow channels leads to a more widespread distribution of channels, making the overall flow path layout resemble a rectangular. This channel structure helps to form a broader distribution of the PCS in the cold plate, thereby enhancing heat transfer capability and promoting a more uniform temperature distribution. Additionally, at a constant V_f , the number of branches in the flow channels in the front region of the cold plate tends to decrease with increasing PCS concentration.

Figure 6B-C exhibit the variations of the average temperature and the root mean square temperature of the cold plate. Both the two performance factors decrease as the V_f increases. The PCS with higher mass concentration shows better cooling behavior. Figure 6D displays the variation of heat transfer area between the fluid and solid domains. The larger the V_f , the greater the number of channel branches and the larger the heat transfer area, thus enhancing the heat dissipation performance of the cold plate. In addition, the heat transfer area for the high concentration PCS is the smallest, but the cooling effect is best, indicating that the PCS has a higher heat transfer efficiency due to the large heat capacity. Figure 6E shows that with the increase of V_f , the width of flow channels increases, so the flow resistance factor decreases. In addition, with the increase of PCS concentration, the flow resistance factor gradually increases. This is attributed to the fact that the apparent viscosity of PCS is greater than that of pure water, owing to the presence of mPCM particles.

Influence of inlet velocity

The effect of inlet velocity on the optimization results at different PCS concentrations is shown in Figure 7. The complexity of the flow channel structure decreases with increasing inlet velocity is shown in Figure 7A. The variation occurs in form of most of the narrower channels combining to a wider flow channel structure. The primary reason for this is that higher inlet velocities lead to a greater number of mPCM particles in the PCS over a given period, thereby increasing the melting heat absorption of the PCS. Consequently, a higher inlet velocity significantly enhances the cooling performance of the cold plate, making a simpler channel structure sufficient for effective cooling.

Figure 7B depicts the liquid phase volume fractions of the melting PCM in 10 wt% PCS at different inlet velocities. It can be seen that the distance of complete melting of mPCM particles increases with the increase of inlet velocity. It is observed that the distance required for complete melting of mPCM particles increases with higher inlet velocities. This occurs because an increase in inlet velocity results in a higher number of mPCM particles flowing through the channel per unit time, which in turn leads to a slower melting rate of the mPCM particles.

It can be observed from Figure 7C-D that the mean temperature and the root mean square temperature of the cold plate decrease as the inlet velocity increases for both pure water and PCS. At the same inlet flow rate, these temperatures also decrease with higher PCS concentration. However, when the inlet flow rate reaches 0.025 m/s, the mPCM particles in the PCS exit the cold plate without fully melting, thereby limiting latent heat absorption. Consequently, the decrease in the average and root mean square temperatures of the cold plate slows down. Figure 7E shows that as the inlet flow rate increases, the flow resistance factor for 20 wt% PCS surpasses that of pure water. Specifically, at an inlet velocity of 0.005 m/s, the flow resistance factor is 2.9 times greater than that of pure water. Additionally, the flow resistance factor increases with higher PCS concentration at the same inlet velocity. This increase in flow resistance factor is attributed to the enhanced apparent viscosity of the PCS at higher mass concentrations.

Influence of heat source power

The topology optimization results for varying heat source powers, ranging from 30 kW to 70 kW, are presented in Figure 8. It reveals that at lower heat source power and the same PCS concentration, the flow channel structure is simpler and the channel width is broader (Figure 8A).

Figure 8B demonstrates that at a lower heat source power (30 kW), the mPCM particles predominantly remain within the phase change process. Therefore, fewer branches are required to fulfill the heat transfer requirements. Conversely, as the heat source power increases, so does the average temperature of the cold plate. Once the temperature exceeds the phase

change range of the PCM, the cooling capacity of the cold plate diminishes due to the complete melting of the mPCM particles. This necessitates the formation of additional branches in the rear section of the cold plate to accommodate the increased heat dissipation demand, resulting in a finer flow path and a more complex structure.

As illustrated in Figure 8C-E, the average temperature (T_{ave}), root mean square temperature (T_{δ}), and flow resistance factor (f) decrease with the decline of heat source power. In addition, both T_{ave} and T_{δ} of the cold plate gradually decrease with an increase in the PCS concentration under constant heat source power. Therefore, the cooling performance and temperature distribution uniformity of the cold plate can be enhanced by increasing the PCS concentration, especially under high heat source power conditions.

CONCLUSION

This study presents work on battery thermal management based on the phase change slurry and topology optimization cold plate to improve the cooling capabilities while reducing the pump power consumption of BTMS. The main conclusions are summarized as follows:

(1) The cooling performance of the TCP is superior to that of RCP. Moreover, the flow resistance of the TCP across various PCS concentrations is lower than that of the RCP. The reductions in average temperature, root mean square temperature and flow resistance factor of TCP with 20wt% PCS are 23.9%, 80.8% and 18.2%, respectively, compared to RCP.

(2) The flow channel branches continue to increase with the increase in w_f . In addition, the number of flow channel branches in the front region of the cold plate decreases with the increase in PCS concentration at the same w_f .

(3) The number of flow channels increases with the increase in V_f , leading to a more widespread distribution of channels. This enhances the cooling capability and promotes a more uniform temperature distribution. The flow power consumption gradually increases with the increase in PCS concentration.

(4) The flow channel structure tends to simplify as the inlet velocity increases. The cooling performance and temperature uniformity of the cold plate progressively improve. The pressure drop increases with the increase in inlet flow rate and PCS concentration.

(5) When the heat source power increases, the mPCM particles rapidly reach complete melting, which diminishes the cooling ability of the cold plate. More branches appear in the rear section of the cold plate to accommodate the increased heat dissipation demands, resulting in a gradual increase in the flow resistance factor.

In the future, we will continue to conduct experiments to validate and refine the previous simulation studies. At the same time, the dimension constraint is applied to the topology optimization method to optimize the flow path structure of TCP, improve the manufacturability of the cold plate and reduce the processing cost.

Nomenclature

Symbols

A_c	heat transfer area (m^2)
A_{eff}	effective heat transfer area (m^2)
C_p	heat capacity ($J/(kg \cdot K)$)
Da	Darcy number
D_h	hydraulic diameter (m)
F	friction force (N/m^3)
f	flow resistance factor
h_f	latent heat of phase change of PCS (kJ/kg)
h_{avg}	average heat transfer coefficient ($W/(m^2 \cdot K)$)
k	convective heat transfer coefficient ($W/(m^2 \cdot K)$)
L	the length of cold plate (m)
$L1$	the length of inlet and outlet (m)

L_i	the length of the flow channel from the inlet to the outlet (m)
Nu	Nusselt number
p	pressure (Pa)
Δp	pressure drop between the inlet and outlet (Pa)
P_w	wetted perimeter of the channel (m)
∇	gradient operator
Q	heat generation (W)
q	penalty factor
q_a	heat flux per area (W/cm ²)
r	density filter radius (m)
Re	Reynolds number
T	temperature (K)
t	time (s)
$\mathbf{u}$	velocity vector (m/s)
V_F	volume fraction
W	the width of cold plate (m)
w_T	weight coefficient of average temperature
w_f	weight coefficient of power dissipation

Greek symbols

α	inverse permeability of the porous media
α_{pcm}	liquid phase volume fraction of phase change material
β	projection slope
γ	design variable before filtration
γ	design variable after filtration
γ_t	design variable after projection
γ_β	projection point
μ	dynamic viscosity (kg/(m·s))
ρ	density (kg/m ³)
Φ	multi-objective function
Φ_T	average temperature
Φ_f	power dissipation
$\Phi_{T,0}$	constant for normalization of average temperature
$\Phi_{f,0}$	constant for normalization of power dissipation
$\overline{\Phi_T}$	normalized average temperature
$\overline{\Phi_f}$	normalized power dissipation
Ω	design domain

Subscripts

i, j	Tensor notation
ave	average
δ	root mean square
in	inlet
out	outlet
h	hydraulic

ref	value of reference model
s	solid
f	fluid
l	liquid
ms	the PCS without phase change
pcm	phase change material
Acronyms	
PCS	phase change slurry
TCP	topology optimization cold plate
RCP	rectangular channel cold plate
BTMS	battery thermal management system
LIB	lithium-ion battery
PCM	phase change material
mPCM	micro-encapsulated PCM
C-rate	capacity-rate
MMA	moving asymptote method
PEC	performance evaluation criterion

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AUTHOR CONTRIBUTIONS

Peilin Hou: Conceptualization, Formal analysis, Methodology, Writing—original draft. Fei Ma: Writing—review & editing. Yang Li: Project administration. Lei Gu: Data curation, Validation. Guangliang Wang: Software. Rendong Shen: Visualization. Jing Wang: Resources. Jun Zhao: Supervision. All authors contributed to and approved the manuscript.

DECLARATION OF INTERESTS

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

DATA AND CODE AVAILABILITY

Data will be made available on request.

SUPPLEMENTAL INFORMATION

Supplementary material is available at: <https://doi.org/10.59717/j.xinn-energy.2024.100061>