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Numerical Optimization of Heat Dissipation Performance of a Parallel Cobweb-Channel Liquid Cooling Plate Using Genetic Algorithm

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ABSTRACT: This study proposes a novel parallel cobweb-structured gradient channel liquid cooling plate (optimized parallel web, OPW) aimed at enhancing the heat dissipation efficiency and energy consumption balance of lithium-ion battery thermal management systems. A numerical model of the liquid cooling plate with a non-uniform flow channel configuration is established. The effects of key structural parameters, including the main channel width (d), side channel width (L), inlet angle (α), and gradient ratio (η), on the thermal performance and pumping power are systematically investigated. Based on this, an efficient parameter optimization framework is developed by integrating Latin hypercube sampling (LHS), a response surface model (RSM), and the NSGA-II multi-objective genetic algorithm. The entropy-weighted TOPSIS method is employed to comprehensively evaluate the Pareto solution set, and the optimal structural parameters are finally determined as follows: main channel width $d = 8.54$ mm, side channel width $L = 6.52$ mm, inlet angle $\alpha = 89.8^\circ$, and gradient ratio $\eta = 7\%$. Under this configuration, CFD simulations yield a maximum system temperature $t_{\max} = 39.59^\circ\text{C}$, $\Delta p = 22.30$ Pa, and $\Delta t = 10.76^\circ\text{C}$. Compared with the IPW, the OPW exhibits negligible changes in $t_{\max}$ and Δt , while the pressure drop is reduced by approximately 7.8% and the cooling efficiency coefficient j_w is improved by approximately 9.2%, indicating that the proposed optimization scheme effectively reduces pumping power consumption while maintaining thermal performance, thereby achieving superior overall economic performance.

KEYWORDS: Lithium-ion battery thermal management; liquid cooling plate; bionic channel; multi-objective optimization

1 Introduction

Environmental pollution and the depletion of renewable energy sources have together made the transition to low-carbon energy systems a pressing global priority [1]. Electric vehicles (EVs) and hybrid electric vehicles (HEVs) are considered a key pathway to mitigating these issues [2,3]. Lithium-ion batteries have become the dominant power source for these vehicles, offering high energy density, large theoretical capacity, and long cycle life. However, battery performance, safety, and cycle life are heavily temperature-dependent, with an optimal operating range of approximately $20\text{--}45^\circ\text{C}$ [4,5]. Operation outside this window can cause losses in charge-discharge efficiency, capacity retention, and state of health (SOH), and may even pose safety risks. Therefore, a reliable battery thermal management system (BTMS) is indispensable for keeping the battery within its optimal temperature range and preserving both performance and durability.

Battery thermal management is commonly implemented through air cooling [6], liquid cooling [7], phase change materials [8], or heat pipes [9]. With the increase in energy density and power of traction batteries, the heat generation rate has risen significantly. Traditional air cooling, due to its limited heat dissipation capacity, can no longer meet the high-efficiency thermal management demands [10]. The main drawbacks of phase change materials (PCMs) for battery cooling are their low thermal conductivity and susceptibility to leakage. Moreover, as a passive cooling method, PCMs cannot actively remove heat, resulting in limited temperature control capability under high thermal loads [11]. Heat pipe cooling is highly efficient, passive, and reliable; however, due to structural and operational constraints, it suffers from drawbacks such as high cost and performance instability under extreme conditions [12]. In contrast, liquid cooling offers advantages including high temperature control accuracy, excellent temperature uniformity, flexible layout, and ease of manufacturing, assembly, and maintenance. These characteristics make it well-suited for new energy vehicles with high energy density and high-power requirements, and it has therefore been extensively studied and applied [13,14]. In recent years, inspired by natural biological structures, bionic design has been widely adopted in the channel design of liquid cooling plates. Chen et al. [15] proposed a bionic double-layer liquid cooling plate inspired by leaf-vein structures, in which the upper channels contact the battery module for heat exchange while the lower channels supplement the uncovered areas to improve thermal uniformity. The results demonstrated that this design effectively eliminated the large temperature difference between conventional dual-channel cooling systems. After parametric optimization, the optimal cooling efficiency at low pumping power was achieved with a branch angle of 80° , branch channel width of 5 mm, main channel width of 7.5 mm, and inlet mass flow rate of 0.03 kg/s. Xia et al. [16] developed a liquid cooling plate based on bionic leaf-vein and root-type flow channels. The distinctive feature of their design lies in configuring the front section of the plate as a conventional heat transfer zone and the middle-to-rear section as an enhanced heat transfer zone. After NSGA-II multi-objective optimization, under identical operating conditions, this specially shaped liquid cooling plate achieved a 14.2% reduction in pressure drop and a 0.7°C reduction in maximum temperature compared with a conventional liquid cooling plate. Li et al. [17] designed a bionic liquid cooling plate based on the structure of human blood vessels, which exhibited good heat dissipation performance under 1C–3C discharge rates. Notably, under the 3C rate, optimizing the outlet tube spacing, plate thickness, and turning radius significantly reduced the temperature rise, with the deviation between simulation and experimental results being less than 9.8%.

Despite the notable progress achieved in the application of bionic flow channels to liquid cooling plate design, the existing literature remains predominantly focused on various individual channel configurations, while systematic research on purely parallel bionic channel architectures has been comparatively limited [18]. Among the limited studies on parallel designs, most adopt uniform cross-sectional geometries, with little to no investigation into gradient channel configurations—namely, channels whose cross-sectional areas gradually decrease from the outer periphery toward the inner region—particularly within a cobweb-inspired parallel network. Furthermore, while parallel channels are effective in reducing pressure drop, they inherently suffer from a preferential flow path effect, wherein the coolant tends to travel along the shortest routes near the side walls. This leads to insufficient flow coverage in the central region, an unduly shortened effective heat exchange path, and consequent deterioration in both temperature uniformity and overall heat dissipation efficiency. This flow maldistribution issue has yet to be adequately addressed in existing studies on bionic parallel structures. In addition, the coupled influences of key geometric parameters—including main channel width, side channel width, inlet angle, and gradient ratio—on the thermal and hydraulic performance of such systems remain insufficiently understood.

To fill these identified research gaps, the present study proposes a novel parallel cobweb-structured gradient channel liquid cooling plate (optimized parallel web, OPW), in which a non-uniform gradient channel design is incorporated into a parallel cobweb architecture for the first time. By progressively tapering the channel width from the outer periphery toward the inner region, the proposed configuration actively counteracts the inherent shortest-path flow preference, thereby redirecting a larger proportion of the coolant toward the central region while simultaneously prolonging the effective heat exchange path. In terms of optimization, the present study integrates Latin hypercube sampling (LHS), response surface modeling (RSM), and the NSGA-II genetic algorithm to construct a multi-objective optimization framework, and further introduces the entropy-weighted TOPSIS method for quantitative comprehensive evaluation of the Pareto solution set to objectively determine the optimal compromise scheme. It should be noted that the aforementioned optimization methods themselves are relatively well-established techniques in the field of engineering optimization, and the core contribution of this study lies in their systematic application to parameter optimization and decision-making for the novel cobweb gradient channel structure. The specific objectives of this study are fourfold: (1) to propose and numerically validate the OPW structure for mitigating the flow maldistribution inherent in conventional uniform parallel cobweb designs; (2) to systematically examine the individual and coupled effects of the key structural parameters (d , L , α , η) on the thermal and hydraulic performance of the liquid cooling plate; (3) to construct a surrogate model based on LHS and RSM for accurate prediction of system responses; and (4) to combine NSGA-II with entropy-weighted TOPSIS to identify the optimal parameter set that best balances $t_{\max}$, Δt , and Δp , offering a practical reference for engineering the gradient parallel flow channels in BTMS.

2 Model Establishment and Computational Method

2.1 Cold Plate Structural Design

Drawing inspiration from the radial-and-loop architecture of a spider web, this study designs a liquid cooling plate with a cobweb-shaped channel network. Fig. 1 presents the geometric layout of the proposed parallel cobweb liquid cooling plate. During operation, the coolant flows in from the bottom inlet, then diffuses outward along the radial main channels. Assisted by the multi-layer concentric circumferential channels, the coolant circulates and achieves lateral connectivity among the main channels, eventually flowing out through the top outlet. While the coolant undergoes radial diffusion, the circumferential connections enable pressure equalization and flow redistribution, significantly enhancing the flow uniformity and heat dissipation performance across the entire liquid cooling plate. This design effectively overcomes the common issues in conventional liquid cooling plates, such as uneven coolant distribution, numerous flow dead zones, and high pressure drop. Meanwhile, after fully absorbing the heat from the battery, the coolant is discharged through the peripheral outlets, achieving efficient and uniform heat dissipation. The proposed cobweb channel design enhances battery thermal management, particularly under high discharge rates, and introduces a new bionic perspective for liquid cooling plate development.

The experimental setup parameters are as follows: the battery dimensions are 148 mm $\times$ 92 mm $\times$ 27 mm, and the liquid cooling plate dimensions are 283 mm $\times$ 152 mm $\times$ 8 mm. The angle between the side channels and the main channels is 55°. The main channel width, side channel width, inlet angle, and gradient ratio are denoted by d , L , α , and η , respectively. The settings of the factors d , L , and α are illustrated in Fig. 1b, where the gradient ratio η refers to the gradient rate of the channel width from the outer to the inner side for adjacent channels. The initial design parameters of the channel structure are as follows: main channel width $d = 8$ mm, side channel width $L = 6$ mm, inlet angle $\alpha = 90^\circ$, and gradient ratio $\eta = 0$. The liquid cooling plate is fabricated from aluminum alloy, whose high thermal conductivity and low

density help to enhance heat transfer and minimize system weight. The cooling medium is an ethylene glycol-water solution, and its key parameters are listed in Table 1.

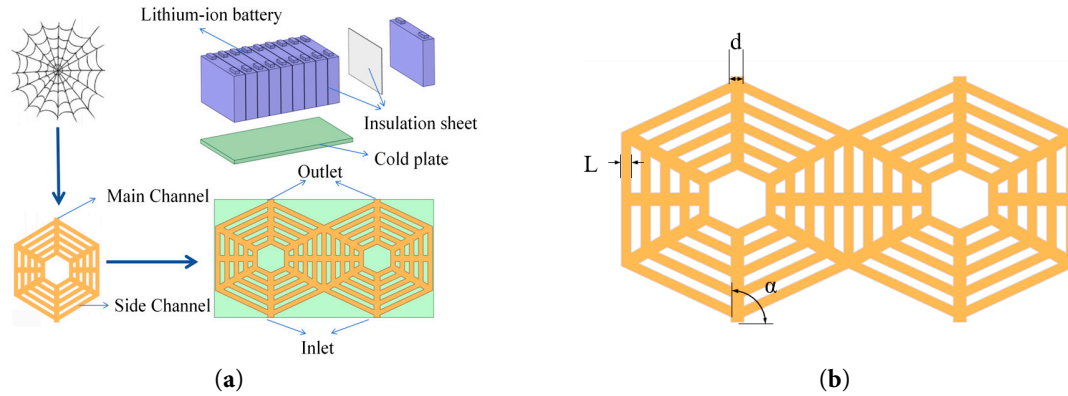


Figure 1: (a) Physical model of the battery cooling system; (b) channel geometry and design parameters.

Table 1: Material physical parameters.

Material	Density/(kg/m ³)	Thermal Conductivity/[W/(m·K)]	Specific Heat Capacity/[J/(kg·K)]
Lithium battery	2218	K ₁ = 23.4 K ₂ = 17.2 K ₃ = 5.3	1060
aluminum alloy	2690	201	900
thermal insulation sheet	3	0.021	1000
coolant	1070	0.384	3300

2.2 Boundary Conditions

Simulations are run in ANSYS Fluent using a pressure-based solver with the SIMPLE algorithm for pressure-velocity coupling and second-order upwind discretization. The model assumes temperature-independent material properties, incompressible coolant flow, and a constant heat source for the battery. The coolant inlet is set at 25°C (ambient: 26°C) with a 0.01 kg/s mass flow rate, and the outlet is a zero-gauge pressure outlet. No-slip and thermal continuity hold at solid-liquid interfaces. Conduction governs heat transfer between the plate and battery, while convection to air uses a fixed 5 W/(m²·K) coefficient. Table 2 gives the boundary condition summary.

Table 2: Boundary conditions for the numerical model.

Parameter	Value/Description
Solver type	Pressure-based
Solution algorithm	SIMPLE
Discretization scheme	Second-order upwind
Battery heat source	33,000 W/m ³
Coolant inlet temperature	25°C
Ambient temperature	26°C
Inlet boundary type	Mass-flow inlet, 0.01 kg/s
Outlet boundary type	Pressure outlet, gauge pressure 0 Pa
Flow regime	Laminar

2.3 Governing Equations

The present simulations employ a single-phase, incompressible coolant model under steady-state conditions with constant density. To characterize the flow behavior in the cold plate passages, the Reynolds number is introduced as:

$$Re = \frac{\rho_c V \cdot D_h}{\mu} \quad (1)$$

The transition between laminar and turbulent flow is governed by the Reynolds number. Laminar conditions persist below 2300, and turbulent behavior emerges above 4000. In this study, the coolant is an ethylene glycol aqueous solution. At room temperature, the kinematic viscosity of the ethylene glycol aqueous solution is $\mu = 3.9 \text{ mm}^2/\text{s}$, the mass flow rate is 0.01 kg/s , the inlet dimensions are $8 \text{ mm} \times 6 \text{ mm}$, and the density is 1070 kg/m^3 . The channel operates at a Reynolds number of about 342, well within the laminar regime. Given the calculated Re number, a laminar flow formulation is used. In the liquid cooling system, the coolant flow process satisfies the conservation of mass, momentum, and energy [19]. The governing equations are as follows:

$$\frac{\partial \rho_l}{\partial t} + \nabla \cdot (\rho_l v) = 0 \quad (2)$$

$$\rho_l (v \cdot \nabla) v = -\nabla P + \mu \nabla^2 v \quad (3)$$

$$\rho_l c_l \frac{\partial T_l}{\partial t} + \nabla \cdot (\rho_l c_l v T) = \lambda_l \nabla^2 T_l \quad (4)$$

The energy equation for the liquid cooling plate is as follows:

$$\nabla^2 T_s = 0 \quad (5)$$

A pump provides the driving force for coolant circulation. The corresponding pressure drop ΔP is adopted to quantify the pumping power consumption, and the formula is given as follows:

$$\Delta P = P_{\text{in}} - P_{\text{out}} \quad (6)$$

The temperature difference Δt can serve as an indicator for evaluating the cooling capacity and the uniformity of temperature distribution on the cold plate, and its formula is given as follows:

$$\Delta t = t_{\text{max}} - t_{\text{min}} \quad (7)$$

2.4 Battery Heat Generation Model

The thermal behavior of the battery during charge/discharge is represented by the classic Bernardi model [20]. This formulation splits the total heat into two parts: irreversible Joule heating from internal resistance and reversible heat from electrochemical reactions. The heat generation per unit volume is written as:

$$q = \frac{I}{V_b} \left[(U_{OCV} - U) - T \frac{\partial U_0}{\partial T} \right] \quad (8)$$

The simplified formula is as follows:

$$q = \frac{1}{V_b} \left[I^2 R_0 - IT \frac{\partial U_0}{\partial T} \right] \quad (9)$$

The battery under consideration is a ternary lithium-ion type with 50 Ah capacity, a nominal voltage of 3.65 V, and an internal resistance of 0.7 m Ω . The reversible reaction heat is positive for discharge and negative for charge, and the voltage difference is set to 0.0514 V. At a 2C discharge rate (100 A), the volumetric heat generation is calculated to be 33,000 W/m³.

To ensure the accuracy of the simulations, the battery thermal model was experimentally validated in this study. Zhu et al. [21] reported that the temperature rise of lithium-ion batteries at low discharge rates consists of three stages: a slow-rise stage, a plateau stage, and a rapid-rise stage. However, conventional numerical methods struggle to accurately capture these stages. As the discharge rate increases to 2C or above, the first two stages gradually disappear and the temperature rise becomes approximately linear. Accordingly, a constant heat generation power at a 2C rate was adopted as the thermal input in this study.

A validation case was performed by simulating the 2C discharge of a single battery at 25°C ambient temperature. Fig. 2 shows the numerical predictions against the experimental data. The predicted temperature curve agrees well with the measurements, deviating by less than 5% at all points, which confirms the adequacy of the thermal model adopted here.

3 Data Simulation and Result Analysis

3.1 Grid Independence Test

A mesh independence check was carried out to ensure that the numerical results are grid-independent. A hybrid meshing strategy was adopted: the main domain was discretized using unstructured polyhedral meshes, while prismatic boundary layer meshes were superimposed near the solid-liquid interface. The boundary layer expansion ratio was set to 1.2 to capture the heat flux gradient in the near-wall region more accurately. The overall mesh size was adjusted within the range of 1–6 mm, with the total number of elements gradually refined from 500,000 to 4,000,000. For critical regions with large geometric curvature, a second-order curvature-adaptive refinement technique was further introduced to enhance the resolution accuracy of local geometric features.

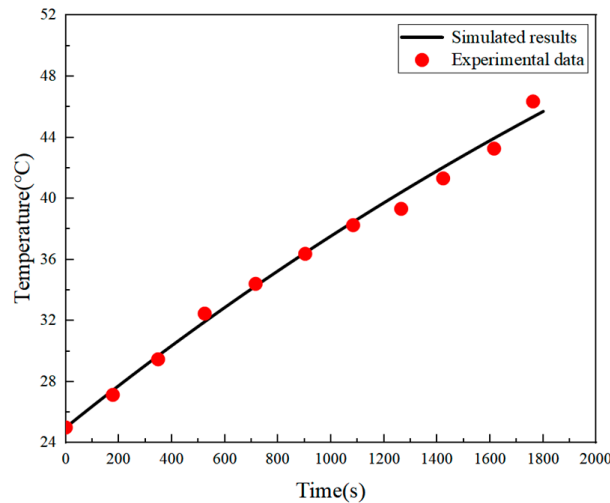


Figure 2: Comparison of simulation results and experimental data [20].

A mesh independence test was performed, with results given in Fig. 3. The monitored parameters—peak temperature, temperature non-uniformity on the battery surface, and pressure drop—level off once the element count reaches 2×10^6 . Increasing the mesh beyond this number changes the results by only a negligible margin. Therefore, a 2-million-element mesh is used for all subsequent calculations, offering a practical balance between accuracy and computational demand.

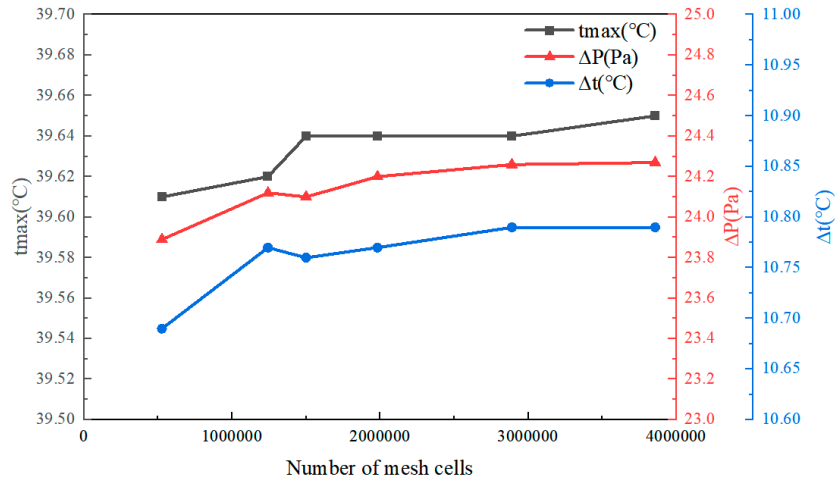


Figure 3: Grid independence test.

3.2 Performance Comparison of Different Liquid Cooling Plate Configurations

To deeply analyze the influence of the bionic dual-cobweb liquid cooling channel on the thermal management performance of the battery, this study constructed a conventional dual-serpentine flow channel layout with an area equivalent to that of the parallel cobweb channel, and compared the temperature contours under an equal pressure drop. Figs. 4 and 5 present the temperature distribution contours at a battery discharge rate of 2C. It can be observed from the figure that under the same pressure drop, the maximum temperature of the dual-serpentine liquid cooling plate is approximately 50.8°C, which is significantly higher than the 39.64°C of the parallel cobweb liquid cooling plate, representing a temperature reduction of about 22%. Moreover, the temperature field distribution of the latter is more uniform. This indicates that the parallel cobweb liquid cooling plate achieves more efficient heat dissipation and uniform temperature control without incurring additional pumping power penalties.

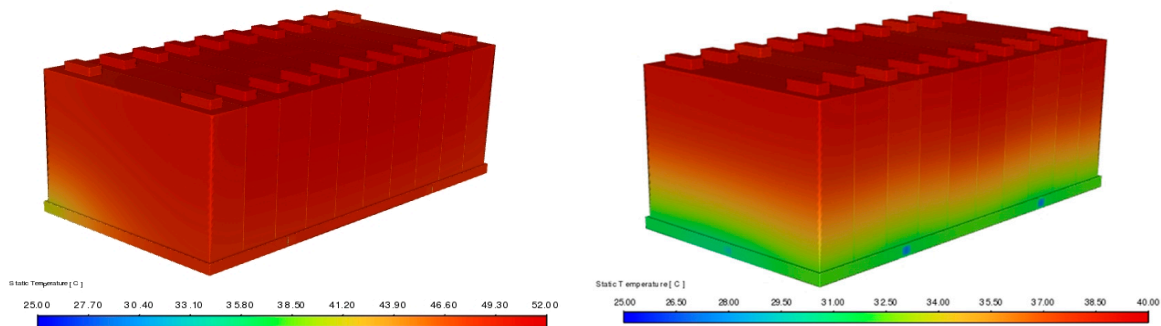


Figure 4: Temperature distribution of the battery pack with serpentine channel (left) and cobweb channel (right).

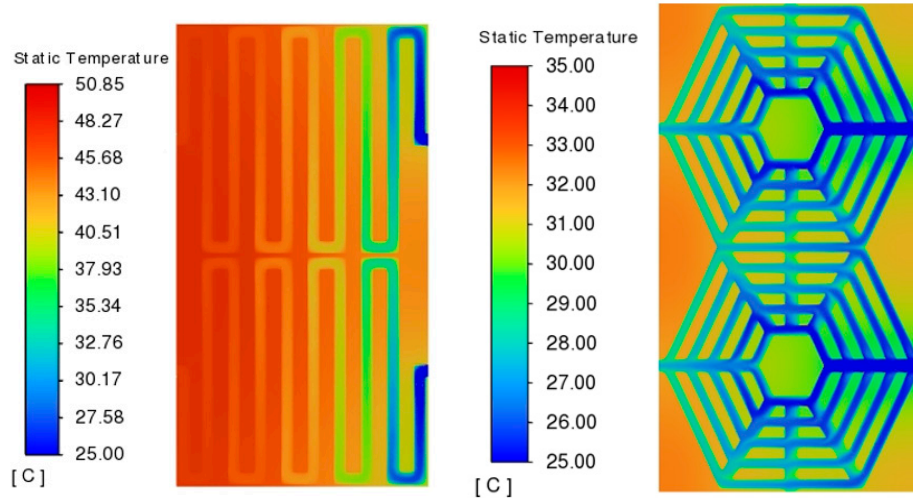


Figure 5: Cross-sectional temperature distribution of the liquid cooling plate: serpentine channel (left) vs. cobweb-type channel (right).

3.3 Single-Factor Experimental Analysis

Given that the performance of the liquid cooling system is influenced by multiple factors, and in light of the structural design, this study focuses on analyzing the effects of the main channel width (d), side channel width (L), inlet angle (α), and gradient ratio (η) on the maximum battery temperature ($t_{\max}$), battery temperature difference (Δt), and pressure drop (Δp) of the liquid cooling system. Each of these parameters- d , L , α , and η -is varied independently while the others are held fixed. The metrics $t_{\max}$ and Δt serve as evaluation indicators for the heat transfer performance of the liquid cooling plate, while Δp is adopted as the evaluation indicator for its power consumption.

3.3.1 Effect of Main Channel Width on Heat Dissipation Performance

The influence of the main channel width d on the cooling performance of the cold plate is shown in Fig. 6. When the main channel width increases from 6 mm to 10 mm, the variations in thermal and hydraulic performance are correlated with the change in d . During this process, the maximum battery temperature rises from 39.42°C to 39.84°C, while the pressure drop exhibits an opposite trend, decreasing from 35.98 Pa to 18.21 Pa. This indicates that as the inlet width increases, the battery temperature continuously rises, whereas the pressure drop gradually decreases. Fig. 7 illustrates the flow velocity distribution of the coolant. Under a constant mass flow rate boundary condition, increasing the channel cross-sectional area leads to a significant reduction in the average flow velocity of the fluid. Since the heat transfer efficiency is highly dependent on the frequency of fluid-wall contact and the velocity gradient, the reduction in flow velocity directly weakens the fluid's ability to transport heat. Concurrently, the increase in hydraulic diameter reduces the effective heat transfer area per unit volume of fluid, exacerbating local heat accumulation. The increase in channel width results in a thicker laminar boundary layer, making it difficult for heat in high-temperature regions to diffuse rapidly to low-temperature regions, thereby causing the temperature difference between the maximum and minimum temperatures to increase with width. Conversely, the significant reduction in pressure drop is primarily attributed to the widened flow cross-section resulting from the increased channel size, which lowers the fluid velocity and greatly reduces the fluid-wall contact area, thereby diminishing the frictional resistance along the flow path. This enables a substantial optimization of pumping power consumption in the system.

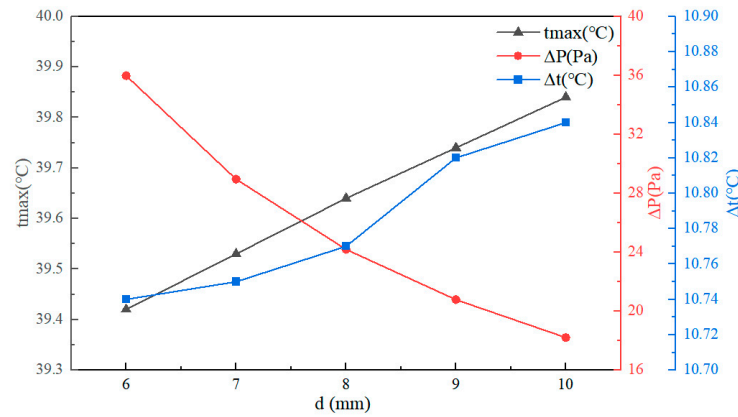


Figure 6: The influence of the main channel width d on the performance of the liquid cooling plate.

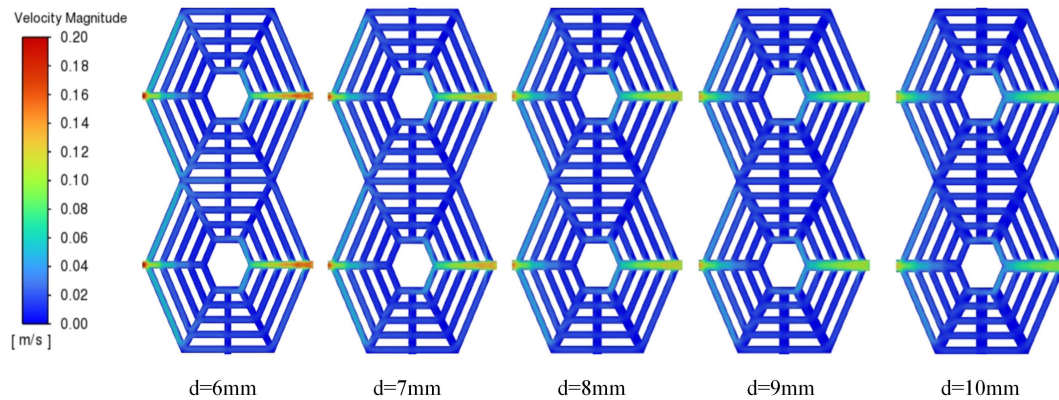


Figure 7: Flow velocity distribution under different main channel widths (d).

3.3.2 Effect of Side Channel Width on Heat Dissipation Performance

The influence of the side channel width L on the cooling performance of the cold plate is shown in Fig. 8. When the side channel width L increases from 4 mm to 8 mm, the battery surface temperature exhibits a monotonically increasing trend, rising from 39.54°C to 39.75°C. The system pressure drop shows a decreasing trend, dropping from 32.05 Pa to 21.23 Pa. Fig. 9 illustrates the flow velocity distribution of the coolant. The increase in the maximum temperature is attributed to the widening of the channels, which causes the majority of the liquid to flow through the central channels and only a small fraction to enter the outer channels. This directly reduces the contact area between the fluid and the heated wall, requiring heat to be conducted from more distant solid regions to the channel centers. Furthermore, the V-shaped variation in dt arises because excessively narrow channels result in insufficient fluid-wall contact area, impairing heat transfer, while excessively wide channels prolong the heat conduction distance due to uneven coverage. Both scenarios degrade temperature uniformity. Conversely, the significant reduction in pressure drop occurs because the side channels constitute the primary parallel flow network. Widening the channels not only greatly reduces the total contact area between the fluid and the tube walls, thereby lowering frictional resistance, but also enables fluid transport with lower energy dissipation, resulting in a decreased pressure drop.

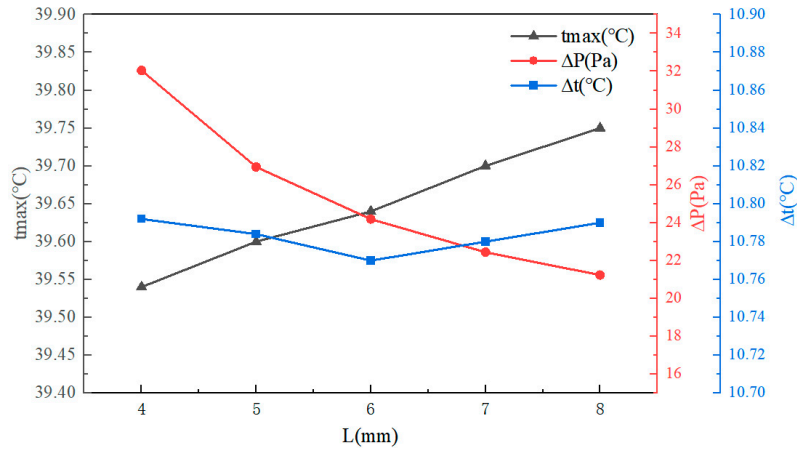


Figure 8: Effect of side channel width l on the performance of the liquid-cooled plate.

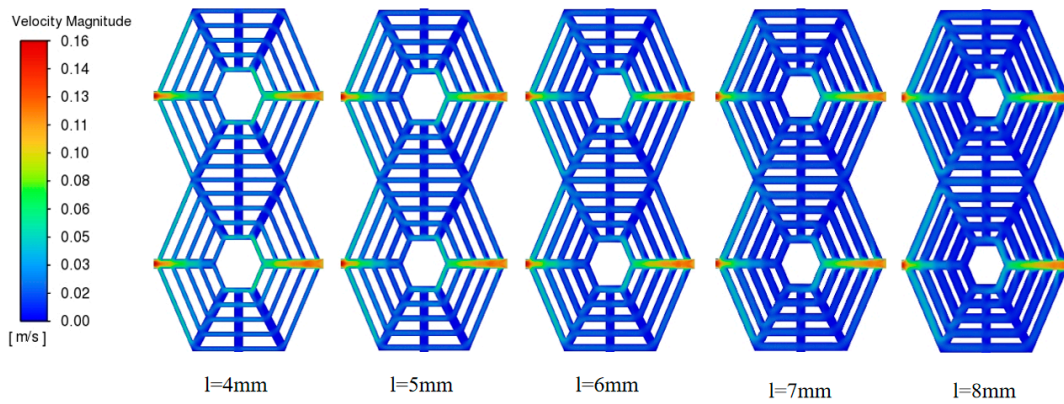


Figure 9: Flow velocity distribution diagrams under different side channel widths (L).

3.3.3 Effect of Inlet Angle on Heat Dissipation Performance

The influence of the inlet angle α on the cooling performance of the cold plate is shown in Fig. 10. When the inlet angle α increases from 86° to 90° , the battery surface temperature exhibits a monotonically increasing trend, rising from 39.39°C to 39.64°C . The system pressure drop shows a decreasing trend, dropping from 30.21 Pa to 24.19 Pa. Fig. 11 presents the velocity distribution for different inlet angles. A smaller inlet angle means that the fluid enters the main channel at a more oblique angle. This oblique impingement may generate stronger disturbances at the inlet, enabling the fluid to cover the peripheral regions earlier. As the angle approaches perpendicular, the fluid entry becomes gentler, causing a larger portion of the fluid to tend to flow through the central region. This results in a slight decrease in heat transfer efficiency in some areas, thereby causing a minor increase in the maximum temperature. A larger inlet angle generally facilitates a more symmetric and uniform radial flow distribution, reducing oblique impingement and allowing the fluid to be distributed more evenly among the side channels, thus diminishing the temperature difference between high-temperature and low-temperature regions. Furthermore, flow entry at a small angle causes a sharp flow direction change, leading to substantial impact energy dissipation. As the angle approaches perpendicular, the streamlines become progressively smoother, greatly reducing the local resistance coefficient and also lowering the frictional losses along the high-velocity flow path. This achieves a significant optimization of the total system pressure drop.

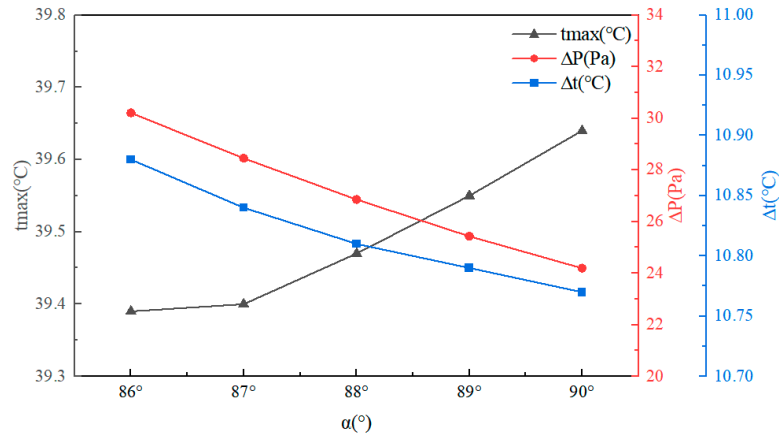


Figure 10: The effect of the import angle α on the performance of the liquid cooling plate.

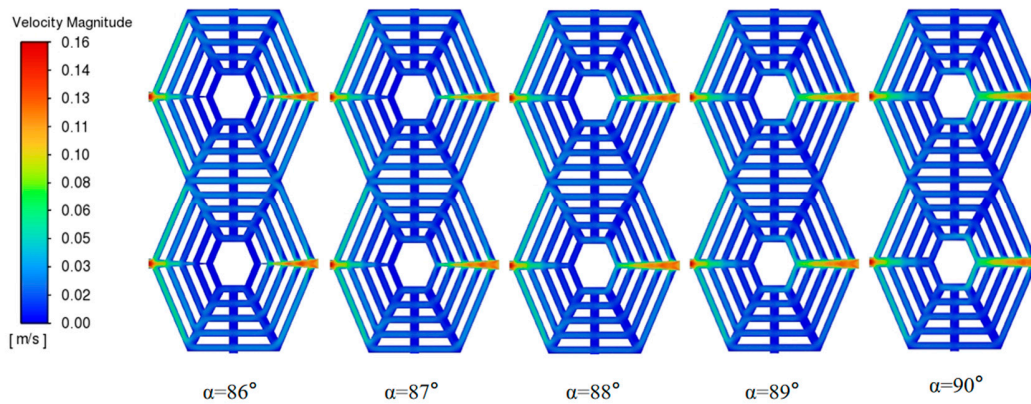


Figure 11: Flow velocity distribution diagrams under different inlet angles (α).

3.3.4 Effect of Gradient Ratio on Heat Dissipation Performance

The gradient ratio represents the gradual narrowing of channel width from the outer to the inner side; that is, a geometric gradient design introducing a “gradually narrowing from outside to inside” configuration into the cobweb liquid cooling channel. As the narrowing ratio of the channels from the outer to inner side increases (0% $\rightarrow$ 20%), its effect on cooling performance is shown in Fig. 12. With increasing ratio, the maximum temperature gradually decreases from 39.64°C to 39.32°C, and remains completely unchanged at 15% and 20%. This indicates that moderately narrowing the inner channels helps to improve heat transfer efficiency. Fig. 13 illustrates the flow rate distribution of the coolant under different gradient ratios. As the gradient ratio increases, the inner channels become narrower, leading to a higher flow velocity in the inner region and enhanced boundary layer disturbance. However, beyond a certain threshold (15%), further narrowing results in a saturated gain in the heat dissipation limit and may even offset the heat transfer advantage due to a sharp increase in flow resistance. The temperature difference (Δt) fluctuates slightly within the range of 10.70°C–10.77°C, showing no obvious trend. This suggests that the gradient variation in channel width has a limited effect on overall temperature uniformity and the system’s average thermal load; its primary role lies in suppressing local hot spots rather than restructuring the global temperature distribution. The pressure drop increases monotonically from 24.19 Pa to 31.66 Pa, a substantial increase of 30.9%, which is caused by the “narrowing” channel design. The reduction in cross-sectional area of

the inner channels forces an increase in flow velocity, leading to higher resistance. Furthermore, the narrowing of the internal flow passages forces more liquid to flow through the outer channels, resulting in a significant increase in frictional losses along the flow path. Consequently, although the system enhances heat dissipation capacity through high-velocity flow, it must consume pumping power to overcome the increased flow resistance.

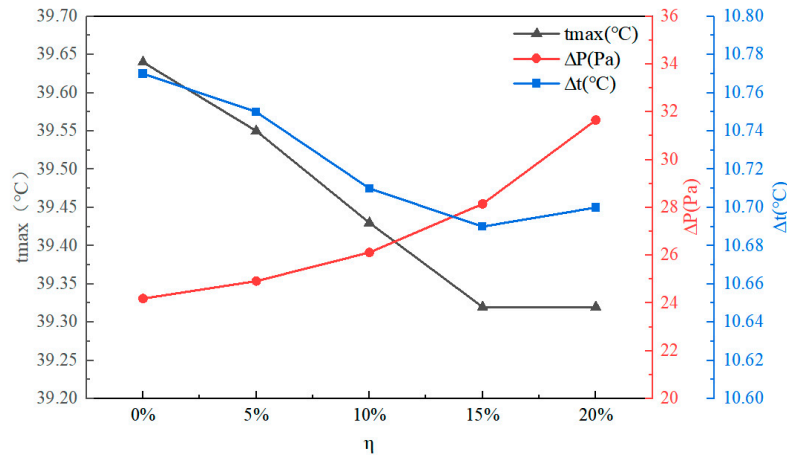


Figure 12: Effect of gradient ratio (η) on the performance of the liquid cooling plate.

4 Multi-Objective Optimization Based on NSGA-II

Based on the above experimental analysis, the thermal performance of the liquid cooling plate is significantly constrained by four key structural parameters: main channel width (d), side channel width (L), inlet angle (α), and gradient ratio (η), with each parameter exerting a different influence weight. More critically, the three core evaluation metrics—maximum temperature ($t_{\max}$), pressure drop (Δp), and temperature difference (Δt)—exhibit inherent trade-offs and competing relationships, making it difficult to achieve simultaneous optimization through the adjustment of a single variable. In view of this, overcoming the limitations of traditional single-factor optimization and introducing a multi-objective optimization algorithm to precisely determine the optimal combination of these four key parameters—thereby achieving the best balance among conflicting objectives—has become a central issue that urgently needs to be addressed to enhance the comprehensive thermal performance of the liquid cooling plate.

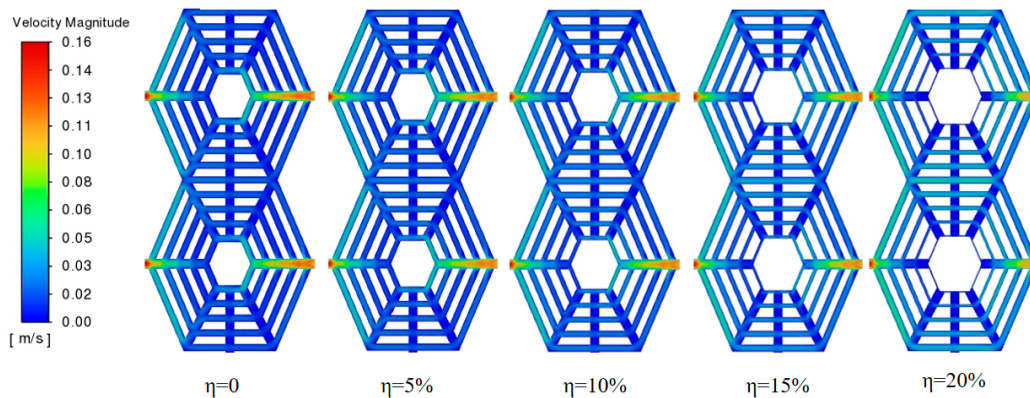


Figure 13: Flow velocity distribution under different gradient ratios (η).

The non-dominated sorting genetic algorithm II (NSGA-II), as an upgraded version of the classical NSGA, has become a mainstream evolutionary strategy for solving multi-objective optimization problems [22]. The core advantage of this algorithm lies in its ability to efficiently approximate the Pareto front when dealing with multiple competing objective functions, thereby providing a set of optimally balanced solutions rather than a single answer. Given its numerous successful applications and demonstrated excellent performance in the field of battery liquid cooling structure optimization [23], this study ultimately selects NSGA-II as the core tool for in-depth multi-objective optimization of the lithium-ion battery liquid cooling structure.

4.1 Surrogate Model

In this study, the maximum temperature ($t_{\max}$), pressure drop (Δp), and temperature difference (Δt) are selected as the multi-objective optimization functions, while the main channel width (d), side channel width (L), inlet angle (α), and gradient ratio (η) are defined as the design variables x_1 , x_2 , x_3 , and x_4 , respectively. Based on the Optimized Latin Hypercube Sampling (OLHS) method, an experimental design scheme containing 35 sample points is constructed within the design space. The sample points exhibit excellent uniform distribution characteristics and space-filling properties across the entire design domain, effectively avoiding sample clustering. Table 3 lists the complete input parameters and corresponding response values obtained from numerical simulations for these 35 sample sets. A preliminary inspection of the sample responses reveals that η and L exert the most significant influence on Δp , while $t_{\max}$ exhibits a saturation trend when η exceeds 15%, indicating diminishing thermal returns from further channel narrowing. The temperature difference Δt , however, remains relatively stable across all samples, suggesting that the cobweb topology inherently provides reasonable temperature uniformity. Based on the above dataset, a response surface model (RSM) is constructed. The coefficient of determination (R^2) and the root mean square error (RMSE) are adopted as key metrics to quantitatively evaluate the predictive accuracy and generalization capability of the surrogate model, both of which are calculated based on the same 35 sample points used for model fitting. R^2 should be greater than 0.9. The calculation formulas for R^2 and RMSE are as follows [24]:

$$R^2 = 1 - \sum_{i=1}^n \frac{(F_{i,pre} - F_{i,CFD})^2}{F_{i,CFD}^2} \quad (10)$$

$$RSME = \sqrt{\frac{1}{n} \sum_{i=1}^n (F_{i,pre} - F_{i,CFD})^2} \quad (11)$$

Table 3: Optimized latin hypercube sampling points and response values.

Number	L, mm	d, mm	α , °	η	$t_{\max}$, °C	Δp , Pa	Δt , °C
1	5	7	89	0	39.391	34.001	10.779
2	5	8	88	0.05	39.356	31.403	10.783
3	5	8	86	0.05	39.211	35.828	10.812
4	5	9	90	0.15	39.237	30.713	10.688
5	5	9	89	0.2	39.208	37.497	10.683
6	5	9	86	0.1	39.180	32.293	10.760
7	5	10	88	0.05	39.554	23.924	10.821
8	4	6	86	0	39.217	63.985	10.982

Table 3: *Cont.*

Number	L	d	α	η	$t_{\max}$	Δp	Δt
9	4	7	89	0.15	38.905	53.356	10.651
10	4	7	87	0.2	39.064	67.048	10.81
11	4	8	90	0	39.539	32.046	10.787
12	4	9	88	0.15	38.945	44.976	10.613
13	4	10	89	0.2	39.044	52.061	10.642
14	4	10	87	0.2	39.044	53.416	10.680
15	6	6	89	0.1	39.242	39.540	10.737
16	6	6	88	0.05	39.262	41.006	10.804
17	6	6	87	0.2	39.323	45.376	10.863
18	6	7	87	0	39.326	34.395	10.840
19	6	9	89	0.1	39.500	23.646	10.767
20	6	10	86	0.15	39.346	25.418	10.774
21	6	10	88	0	39.695	19.922	10.860
22	7	7	90	0	39.600	27.026	10.784
23	7	7	86	0.15	39.568	34.899	10.963
24	7	6	90	0.05	39.437	34.138	10.745
25	7	8	90	0.1	39.510	23.631	10.735
26	7	8	88	0.15	39.413	26.774	10.79
27	7	8	86	0.2	39.597	30.611	10.944
28	7	10	88	0.15	39.474	26.345	10.822
29	8	6	90	0.2	39.403	34.821	10.755
30	8	6	89	0.1	39.404	34.476	10.785
31	8	7	87	0.1	39.484	30.220	10.880
32	8	8	87	0	39.534	24.365	10.864
33	8	9	90	0.1	39.653	18.946	10.743
34	8	9	87	0.05	39.594	20.892	10.855
35	8	10	86	0.05	39.617	18.918	10.87

The prediction results of the response surface model are presented in Table 4. The relationship equations for the objective functions are as follows:

$$t_{\max} = 39.32 + 0.02215x_1 + 0.1104x_2 + 0.0303x_3 - 0.1069x_4 + 0.0014x_1x_2 - 0.016x_1x_3 + 0.0887x_1x_4 + 0.1040x_2x_3 - 0.0715x_2x_4 - 0.1225x_3x_4 - 0.0287x_1^2 + 0.0422x_2^2 + 0.0499x_3^2 + 0.0707x_4^2 \quad (12)$$

$$\Delta p = 27.66 - 10.3430x_1 - 9.4098x_2 - 2.8815x_3 + 4.9251x_4 + 2.1120x_1x_2 + 2.3350x_1x_3 - 3.9680x_1x_4 + 1.9188x_2x_3 + 2.5291x_2x_4 + 0.2190x_3x_4 + 7.0307x_1^2 + 4.8566x_2^2 - 0.7555x_3^2 + 2.3041x_4^2 \quad (13)$$

$$\Delta t = 10.7548 + 0.05472x_1 - 0.01392x_2 - 0.0678x_3 - 0.0298x_4 + 0.01472x_1x_2 - 0.0092x_1x_3 + 0.0396x_1x_4 + 0.0650x_2x_3 - 0.0180x_2x_4 - 0.0323x_3x_4 + 0.0014x_1^2 + 0.0352x_2^2 + 0.0162x_3^2 + 0.0365x_4^2 \quad (14)$$

Table 4: Response surface model evaluation index results.

	R^2	RMSE
$t_{\max}$	0.9617	0.0529
Δp	0.9876	1.75
Δt	0.9331	0.0289

4.2 Analysis of Multi-Objective Optimization Results

The optimization problem is formulated using the response surface models described above, with variable bounds taken from Table 5. NSGA-II is run with a population of 100 for 200 generations, using a crossover probability of 0.9, crossover index of 10, and mutation index of 20. Fig. 14 gives the Pareto front, which includes 50 viable designs. The trade-off relationship is clear: a reduction in pressure drop leads to higher maximum temperature and larger temperature difference. Since choosing from the Pareto front can be subjective, the entropy-weighted TOPSIS method is used to rank the solutions objectively. Here, entropy weighting determines the relative importance of each objective, and TOPSIS orders the alternatives by their distance from both the ideal and anti-ideal points. The computational steps are outlined as follows [25]:

Given m Pareto-optimal solutions and n evaluation criteria, the decision matrix X is constructed as follows:

$$X = \begin{pmatrix} x_{11} & x_{12} & \dots & x_{1n} \\ x_{21} & x_{22} & \dots & x_{2n} \\ \cdot & \cdot & \cdot & \cdot \\ \cdot & \cdot & \cdot & \cdot \\ \cdot & \cdot & \cdot & \cdot \\ x_{m1} & x_{m2} & \dots & x_{mn} \end{pmatrix} \quad (15)$$

where x_{ij} denotes the value of the i -th solution with respect to the j -th criterion. In this study, $m = 50$ and $n = 3$.

Table 5: Values corresponding to different parameters.

L (x1)	4	5	6	7	8
d (x2)	6	7	8	9	10
α (x3)	86°	87°	88°	89°	90°
η (x4)	0	5%	10%	15%	20%
Corresponding value	−1	−0.5	0	0.5	1

To eliminate the dimensional effects of different evaluation criteria, each data point in the decision matrix needs to be normalized. The normalization formula is as follows:

$$x'_{ij} = \frac{\max_j(x_j) - x_{ij}}{\max_j(x_j) - \min_j(x_j)} \quad (16)$$

where $\max_j(x_j)$ and $\min_j(x_j)$ denote the upper and lower bounds of the j -th criterion, and x'_{ij} is the normalized value of the i -th alternative under that criterion.

The entropy weighting scheme is adopted to determine the relative importance of each criterion. A larger entropy value suggests more uncertainty and therefore a lower weight. For each criterion j , the share z_{ij} of the i -th alternative under that criterion is calculated first. The formula for calculating z_{ij} is as follows:

$$Z_{ij} = \frac{x'_{ij}}{\sqrt{\sum_{i=1}^m x'^2_{ij}}} \quad (17)$$

The entropy value of each criterion is calculated as follows:

$$E_j = -k \sum_{i=1}^m z_{ij} \ln(z_{ij}) \quad (18)$$

here, k is a normalization constant that confines the entropy value to the interval $[0, 1]$. A smaller entropy value indicates a more concentrated information distribution for the j -th criterion.

The degree of redundancy d_j for the j -th criterion is then calculated from the entropy value as follows:

$$d_j = 1 - E_j \quad (19)$$

Finally, the weight w_j of the j -th criterion is calculated based on its redundancy degree d_j . The formula for calculating w_j is as follows:

$$w_j = \frac{d_j}{\sum_{j=1}^n d_j} \quad (20)$$

The similarity calculation requires two reference points: the positive ideal solution and the negative ideal solution. These are built from the normalized decision matrix and the criterion weights. The positive ideal A^+ takes the best attainable value for each criterion, and the negative ideal A^- takes the worst. They are defined as follows:

$$A^+ = \left(\max_j (x'_{1j} w_j), \max_j (x'_{2j} w_j), \dots \right) \quad (21)$$

where $\max_j (x'_{1j} w_j)$ denotes the weighted maximum value among all solutions with respect to the j -th criterion. Negative Ideal Solution A^- :

$$A^- = \left(\min_j (x'_{1j} w_j), \min_j (x'_{2j} w_j), \dots \right) \quad (22)$$

where $\min_j (x'_{1j} w_j)$ denotes the weighted minimum value among all solutions with respect to the j -th criterion.

For each alternative, its Euclidean distance to the positive ideal solution and to the negative ideal solution is subsequently determined. The formula for calculating the distance to the ideal solution is as follows:

$$D^+ = \sqrt{\sum_{j=1}^n (x'_{ij} - A_j^+)^2} \quad (23)$$

The distance to the negative-ideal solution is calculated as follows:

$$D^- = \sqrt{\sum_{j=1}^n (x'_{ij} - A_j^-)^2} \quad (24)$$

where D^- is the Euclidean distance from the i -th alternative to the negative ideal solution, and A_j^- is the negative ideal value under criterion j .

The closeness coefficient for each alternative is determined as follows:

$$C_i = \frac{D_i^-}{D_i^+ + D_i^-} \quad (25)$$

The value of C_i ranges between 0 and 1. In this paper, the cooling performance index is a negative indicator, where a smaller value indicates that the solution is closer to the ideal solution.

Solutions are ranked by relative closeness C_i , with the largest value indicating the best candidate—being closest to the positive ideal solution.

The weights obtained for maximum temperature, pressure drop, and temperature difference are 31.24%, 28.27%, and 40.49%, respectively. After applying the entropy-weighted TOPSIS method, the 19th candidate is found to have the largest relative closeness C_i , which means it is nearest the positive ideal and farthest from the negative ideal. This solution is therefore chosen as the optimal one. The corresponding optimized parameters d , L , α and η are 8.54 mm, 6.52 mm, 89.8° and 7%, respectively. Based on this parameter combination, the cooling performance of the BTMS is optimized, and the resulting performance indicators are presented in Table 6.

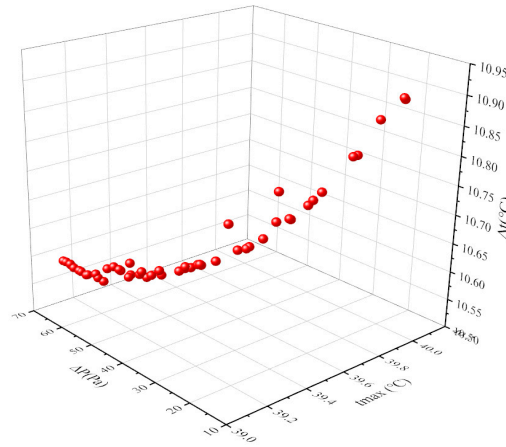


Figure 14: Pareto solution set.

Table 6: Optimization result error analysis.

	$t_{\max}$ (°C)	Δp (Pa)	Δt (°C)
Prediction	39.48	20.46	10.74
Simulation	39.59	22.30	10.76
Error	0.28%	8.25%	0.19%

4.3 Result Comparison

To validate the effectiveness of the structural optimization of the parallel cobweb liquid cooling plate, a three-dimensional simulation model of a conventional straight-channel cobweb liquid cooling plate with the same channel volume fraction as the optimized parallel cobweb plate is established for comparative analysis. Under a mass flow rate of $v = 0.01$ kg/s, thermo-fluid coupled numerical simulations are performed on the initial parallel web (IPW) liquid cooling plate ($d = 8$ mm, $L = 6$ mm, $\alpha = 90^\circ$, $\eta = 0$, channel cross-sectional area = $20,770 \text{ mm}^2$) and the optimized parallel web (OPW) liquid cooling plate ($d = 8.54$ mm, $L = 6.52$ mm, $\alpha = 89.8^\circ$, $\eta = 7\%$, channel cross-sectional area = $20,650 \text{ mm}^2$). Table 7 presents a comparative summary of

$t_{\max}$, Δp , and Δt between the IPW and OPW configurations. The OPW exhibits nearly unchanged $t_{\max}$ and Δt , but reduces Δp by approximately 8%.

Table 7: Comparison of $t_{\max}$, Δp , and Δt between IPW and OPW.

	$t_{\max}$ (°C)	Δp (Pa)	Δt (°C)
IPW	39.64	24.19	10.77
OPW	39.59	22.30	10.76

Fig. 15 compares the temperature field distributions of the original and optimized structures. The thermal performance improvement of the OPW is primarily attributed to the synergistic interplay between flow resistance matching and convective heat transfer enhancement. On the one hand, the 7% channel width gradient ratio produces progressively decreasing flow cross-sections along the radial direction among adjacent branches. Based on the principle of flow resistance balance, this graded design effectively compensates for the pressure drop imbalance caused by differences in flow path lengths, suppressing the preferential flow of coolant along the shortest central paths and thereby improving the uniformity of coolant distribution within the cobweb topology. On the other hand, the side channel width is increased from 6 mm to 6.54 mm, which enhances boundary layer perturbation and improves the convective heat transfer coefficient while maintaining the main transport capacity. Consequently, the peak temperature in the central heat source region is significantly reduced, and a more uniform overall temperature field is achieved.

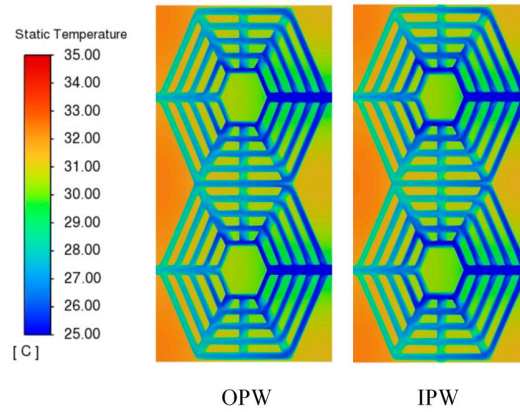


Figure 15: Temperature distributions of the original and optimized structures.

To comprehensively evaluate the thermal and economic performance of the liquid cooling plate, the cooling efficiency coefficient j_w is introduced as an evaluation metric, and its formula is as follows [26]:

$$j_w = \frac{Q_w}{P_w} \quad (26)$$

j_w can be simplified as follows:

$$j_w = \frac{\rho_l c_l (t_{out} - t_{in})}{\Delta P} \quad (27)$$

A larger value of this coefficient indicates that more heat is removed by the coolant per unit of pumping power, reflecting superior overall economic performance of the system. Fig. 16 presents the variation

of the cooling efficiency coefficient j_w for the OPW, IPW, and SLCP under different mass flow rates. It can be observed that as the mass flow rate increases from 0.01 kg/s to 0.04 kg/s, the j_w values of all three structures decrease significantly, indicating that increasing the flow rate substantially raises pumping power consumption, leading to rapidly diminishing cooling returns per unit energy input. The most pronounced decline occurs in the low-flow regime between 0.01 kg/s and 0.02 kg/s, where all three structures experience reductions exceeding 76.8%.

Under all flow rate conditions, the OPW consistently yields the highest j_w values among the three structures, followed by the IPW, while the SLCP performs substantially lower—by nearly two orders of magnitude. Specifically, the OPW outperforms the IPW with improvements ranging from 9.2% to 29.6%, with the most pronounced improvement occurring at the lowest flow rate of 0.01 kg/s. As the flow rate increases, the difference between these two structures gradually narrows, yet the OPW maintains a stable advantage throughout. In contrast, the SLCP yields j_w values less than 1% of those of the OPW across all tested conditions, indicating severely insufficient heat transfer capability for practical engineering applications. These results confirm that the optimized OPW structure effectively reduces flow resistance under various flow rate conditions, enabling greater heat removal per unit pumping power, and validating the effectiveness of its channel design in enhancing the overall economic performance of the liquid cooling plate.

The OPW structure features channel widths all exceeding 4 mm, allowing for low-cost manufacturing using conventional processes such as stamping or CNC milling. The gradual tapering configuration is readily achievable in mold machining, and the inlet/outlet arrangement on the same side facilitates integration into battery packs, demonstrating favorable engineering feasibility and economic viability.

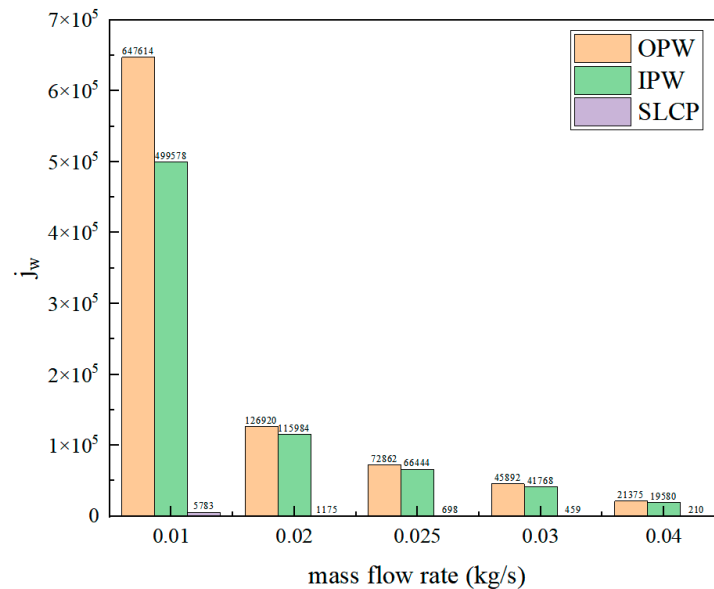


Figure 16: Variation of the cooling efficiency coefficient j_w for the IPW, OPW, and SLCP under different mass flow rate conditions.

5 Conclusions

This study introduces a cobweb-inspired liquid cooling plate with parallel channels, which provides a good trade-off between battery heat dissipation and pumping power consumption. The numerical model is built in Fluent. Single-factor analysis identifies four structural parameters— L , d , α , and η —as the key variables for optimization. With $t_{\max}$, Δp , and Δt taken as the objective functions, NSGA-II is employed to

perform multi-objective optimization, and the resulting Pareto solutions are ranked by entropy-weighted TOPSIS to identify the best overall design. The main conclusions are as follows:

- (1) The parallel cobweb flow channel configuration, featuring dual inlets at the bottom and dual outlets at the top, enables uniform coolant distribution and efficient heat transfer while shortening the flow path, thereby achieving favorable temperature uniformity and low pumping power consumption.
- (2) In the design, the uniform channels are replaced with non-uniform channels, where the channel width gradually narrows from the outer to the inner side according to ratios of 0, 5%, 10%, 15%, and 20%. The results indicate that moderately narrowing the inner channels helps to improve the heat dissipation performance of the liquid cooling plate. At a mass flow rate of 0.01 kg/s, compared with the uniform channel design, the maximum temperature is reduced by 0.32°C and the temperature difference is decreased by 0.08°C, while the system pressure drop increases by only 7.47 Pa, demonstrating a significant improvement in the heat dissipation efficiency of the cooling system.
- (3) This study establishes an optimization framework combining LHS, RSM, and NSGA-II, with the entropy-weighted TOPSIS method used to select the optimal solution from the Pareto set. The optimized parameters are $d = 8.54$ mm, $L = 6.52$ mm, $\alpha = 89.8$ and $\eta = 7\%$. CFD results give $t_{\max} = 39.59^\circ\text{C}$, $\Delta p = 22.30$ Pa, and $\Delta t = 10.76^\circ\text{C}$. Compared with IPW, OPW shows negligible changes in $t_{\max}$ and Δt , while Δp decreases by about 7.8% and j_w increases by about 9.2%. These results confirm that the proposed strategy effectively reduces pumping power without compromising thermal performance, offering a well-balanced design with superior economic efficiency.

A limitation of the current work is its exclusive reliance on CFD simulations. Experimental data for the cold plate—particularly in terms of temperature uniformity and flow resistance—are not yet available. Despite this, the comparison across design variants remains meaningful, since all cases share the same computational setup. To strengthen the findings, we intend to conduct dedicated experiments in our ongoing work, which will also help assess the manufacturability and operational reliability of the proposed cold plates.

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Nomenclature

Parameter definition		Unit
A^+	positive ideal solution	
A^-	negative ideal solution	
c_l	pecific heat capacity of coolant	$J \cdot kg^{-1} \cdot ^\circ C^{-1}$
C_i	relative closeness to the ideal solution	
d	Main channel width	mm
D_h	hydraulic diameter	mm
D^+	euclidean distance to the positive ideal solution	
D^-	euclidean distance to the negative ideal solution	
d_j	degree of redundancy of the j-th criterion	
E_j	the entropy value of the j-th criterion	
I	charge/discharge current	A
j_w	the cooling efficiency coefficient	
k	normalization constant	
L	Side channel width	mm
$\max_j(x_j)$	the maximum values of the j-th criterion	
$\min_j(x_j)$	the minimum values of the j-th criterion	
w_j	weight of the j-th criterion	
x'_{ij}	the normalized value of the i-th solution on the j-th criterion	
P_{in}	inlet pressure	Pa
P_{out}	outlet pressure	Pa
T_{min}	minimum battery temperature	$^\circ C$
T_{max}	maximum battery temperature	$^\circ C$
U	terminal voltage	V
U_{OCV}	open-circuit voltage	V
V_b	battery volume	m^3
v	coolant flow velocity	$m \cdot s^{-1}$
x_{ij}	the value of the i-th solution with respect to the j-th criterion	
Z_{ij}	Normalized proportion value	
Greek Symbols		
α	inlet angle	$^\circ$
η	Gradient ratio	
λ_l	thermal conductivity of coolant	$W \cdot m^{-2} \cdot K^{-1}$
ρ_l	Coolant Density	$kg \cdot m^{-3}$
Abbreviations		
IPW	Initial parallel spider-web liquid cooling plate	
OPW	Optimized parallel spider web liquid cooling plate	

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