TOPOLOGY OPTIMIZATION AND EXPERIMENTAL INVESTIGATION OF COLD PLATE HYDROTHERMAL PERFORMANCE

by

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> Original scientific paper https://doi.org/10.2298/TSCI240324194C

To enhance the hydrothermal performance of cold plates, this study constructs a topology optimization framework that integrates conjugate heat transfer principles, aiming to maximize heat generation and minimize fluid-flow dissipation. The robustness of the design is improved through the application of projection and density filtering techniques, with the projection intensity value incrementally increased via a parameter scanning method to bolster numerical convergence. The inlet and outlet distribution of the cold plate, aligned along its centerline, is determined through various target weight combinations, generating clear, continuous lay-outs. To further substantiate the effectiveness of the optimized configuration, a topology optimization design employing an objective function weighting factor of 0.7:0.3 is chosen to create the 3-D geometry of the cold plate. A parallel flow channel design, maintaining an identical fluid volume fraction and heat transfer boundary length, is introduced for comparative analysis. Moreover, performance indices such as surface temperature, pressure drop across the channel, average Nusselt number, and thermal resistance are assessed for two cold plates under varying inlet velocities. Both simulations and experiments indicate that the hydrothermal performance of the developed topological structure significantly surpasses that of the conventional parallel channel design, a disparity that amplifies with increasing inlet velocities.

Key words: cold plate, topology optimization, hydrothermal performance, thermal design

Introduction

Liquid cooling plates find extensive application across a variety of mechanical and electronic devices, including automobile engines, power battery plates, and communication base stations. An optimal geometric configuration of the flow channels plays a crucial role in enhancing heat transfer efficiency and, consequently, elevating the overall performance of the system [1, 2]. Recently, topology optimization methods have become recognized as highly flexible and efficient design tools for addressing conjugate heat transfer issues.

The topology optimization technique demonstrates the ability to determine the optimal path for heat dissipation autonomously [3, 4]. Matsumori *et al.* [5] addressed two types of design problems related and unrelated to heat sources and topology, respectively. They in-

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troduced an integral equation maintain constant input power at the inlet and solved for the optimal channel distribution based on the density method. Sato et al. [6] developed a Pareto front exploration algorithm for bi-objective topology optimization problems that adaptively determined weighting coefficients and explored the relationship between flow resistance and heat transfer. Zhao et al. [7] introduced a cost-effective strategy by substituting the Navier-Stokes equations with a Darcy flow model. By applying pressure drop and minimum size constraints, they solved the conjugate heat transfer problem under turbulent conditions. To tackle thermal dissipation challenges in power devices, Liu et al. [8] conducted topology optimization design five micro-channel structures with varying aspect ratios based on density to enhance the synergistic effect. Li et al. [9] optimized the distribution of liquid-cooled paths with various inlet and outlet arrangements using density-based topology optimization. The performance of liquid-cooled plates in thermal dissipation was investigated using simulations and experiments. Zhang et al. [10] applied topology optimization the cooling design of supersonic engines to improve system stability under non-uniform conditions. The 3-D topology optimization research faces challenges related to computational complexity, algorithm efficiency, manufacturability, and non-linear problems. To address this issue, Haertel et al. [11] optimized thermal resistance subject to constraints on heat generation and pressure drop using a pseudo-3-D model to reduce computational costs. Yaji et al. [12] effectively addressed the challenges associated with designing 2-D and 3-D heat flow systems using the level set method. Additionally, they managed the complexity of flow paths by adopting a regularization method. Pei et al. [13] developed a pseudo-3-D optimization framework composed of a flow design layer and a thermal conductive substrate layer to optimize forced air cooling heat sinks. In addition, topology optimization for transient thermal analysis has received increasing attention. Long et al. [14] proposed an efficient quadratic approximation method for multi-material transient thermal problems, which improves computational efficiency and reduces the number of iterations. Li et al. [15] verified the effectiveness of functional gradient constraints for solving transient topology optimization problems through numerical cases.

The design of liquid cooling channels faces the challenges of efficient thermal performance, design flexibility, and complex flow patterns. To address these challenges, this study introduces a density-based topology optimization for cold plates with inlet and outlet configurations distributed on the center line of the cold plate. The design approach aims to maximize heat transfer while minimizing pressure drop by coupling dual objective functions. The distinct topology structures of the flow channel are achieved, and the Pareto solution set for the dual-objective optimization problem is calculated. Furthermore, a parallel flow channel structure is designed for comparison with the optimized structure, maintaining identical heat transfer boundary lengths and fluid volume fractions. The hydraulic and thermal performance of the cold plates with these two types of flow channel structures is studied through numerical simulations. Finally, experiments on cold plate samples are conducted to verify the rationality of the proposed design.

Topology optimization design for cold plate

Concept of topology optimization for flow channel lay-out

When the flow channel structure is kept consistent in the direction normal to the horizontal plane, the cold plate design process can typically be simplified and treated as a 2-D problem. Figure 1 illustrates the topology optimization problem for the planar lay-out of the channel, encompassing the mixed design domain, fluid inlet and outlet, and physical boundary conditions. The characteristic dimension, L, of the inlet is 8 mm. To tackle this, a pseudo-densi-

ty value denoted as γ , which corresponds to the density method, is assigned to each unit within the design domain. The $\gamma = 0$ for solids and $\gamma = 1$ for fluids. The search for the optimized lay-out of the flow path evolves into resolving a topology optimization "0-1" problem to find the optimal solution by using γ as the design variable.

Mathematical model construction

Governing equation for fluid-flow

Considering the case of steady-state and incompressible Newtonian fluid-flow, the characteristic length at the inlet serves as the basis for calculating the Reynolds number, which is defined:

$$\operatorname{Re} = \frac{\rho UL}{\mu} \tag{1}$$



Fluid-flow behavior can be described by the continuity and momentum equations. The continuity equation can be expressed [16, 17]:

$$\nabla \cdot \mathbf{u} = 0 \tag{2}$$

The Navier-Stokes equation describes the movement of the viscous fluid, which is defined [16, 17]:

$$\rho(\mathbf{u} \cdot \nabla)\mathbf{u} = -\nabla p + \nabla \left\{ \mu \left[\nabla \mathbf{u} + (\nabla \mathbf{u})^{\mathrm{T}} \right] \right\} + F$$
(3)

where ρ is the fluid density, **u** – the velocity field, μ – the dynamic viscosity, and F and p are the body force and the pressure, respectively.

In thermal-fluid topology optimization, material distribution mimics porous media characteristics where fluid and solid phases coexist. Addressing fluid-flow within porous media necessitates considering the additional drag experienced by the fluid during permeation through the material. A resistance term associated with fluid velocity is introduced into the momentum equation using the Brinkman penalty model [18]. The assumption is made that the resistance is generated by the opposing force that scales with the fluid velocity, and this relationship is represented:

$$F = -\alpha \cdot \mathbf{u} \tag{4}$$

where α is the inverse permeability. When α is 0, it indicates that the porous media offers no resistance to fluid-flow, enabling smooth fluid-flow. Conversely, when α becomes infinitely large, the fluid is completely obstructed from passing through.

The rational approximation of material properties (RAMP) method is advantageous for its smooth interpolation of material properties, which enhances the efficiency and stability of the optimization process. Consequently, this study employs the RAMP method to achieve thermo-fluid topology optimization. The inverse permeability, α , is represented through a continuous function related to design variables, employing interpolation in the manner:

$$\alpha(\gamma) = \alpha_{\min} + (\alpha_{\max} - \alpha_{\min}) \frac{q(1-\gamma)}{q+\gamma}$$
(5)

where q is the interpolation parameter that regulates the trend of the function $\alpha(\gamma)$. The α_{\min} and α_{\max} are the inverse permeability for the exclusive presence of solid and liquid phases, respec-



Figure 1. Schematic representation of the design domain and the configuration of the inlet and outlet for the topology optimization problem

tively. In the Brinkman penalty model, a_{max} depends on the Darcy number, and the magnitude of the viscous force, which is expressed:

$$\alpha_{\max} = \frac{\mu}{\mathrm{Da}L^2} \tag{6}$$

Governing equation for conjugate heat transfer

The governing equations for heat transfer in the solid and fluid domains are given [3, 19, 20]:

$$k_{\rm s}\nabla^2 T + Q = 0 \tag{7}$$

$$\rho C_p \left(\mathbf{u} \cdot \nabla \right) T = k_f \nabla^2 T + Q \tag{8}$$

where the thermal conductivity of the solid is denoted by k_s , and that of the fluid by k_f . The C_p is the specific heat capacity and Q – the surface heat source, which can be defined by heat generation reliant on local temperature differences and is expressed by Newton's law of cooling:

$$Q = \eta \left(T_Q - T \right) \tag{9}$$

where η is the heat generation coefficient that depends on the temperature difference and T – the local temperature. The T_o denotes the prescribed temperature with a value of 80 °C.

During the initial stages of topology optimization, it is often challenging to distinctly allocate the material properties of the solid and fluid components. To address this issue, the design variable γ is employed to consolidate the aforementioned equations into a single equation that describes the conjugate heat transfer process:

$$\gamma \rho C_p(\mathbf{u} \cdot \nabla) T = \left[(1 - \gamma) k_s + \gamma k_f \right] \nabla^2 T + (1 - \gamma) Q$$
(10)

Treatment of numerical instability

To circumvent numerical instability phenomena during topology optimization, a combined approach of density filter and projection methods is employed. The density filter is implemented using a Helmholtz-type partial differential equation, the expression of which is [21]:

$$-r_{\text{filter}}^2 \nabla^2 \tilde{\gamma} + \tilde{\gamma} = \gamma \tag{11}$$

where $\tilde{\gamma}$ and r_{filter} are the filtered design variable and the filter radius, respectively.

The use of the density filter may introduce additional gray areas. To mitigate this, the Heaviside projection method is utilized in the optimization process to ensure a distinct channel structure [22]. The formulation is presented:

$$\hat{\gamma} = \frac{\tanh(\beta\gamma_{\beta}) + \tanh\left[\beta\left(\tilde{\gamma} - \gamma_{\beta}\right)\right]}{\tanh(\beta\gamma_{\beta}) + \tanh\left[\beta(1.0 - \gamma_{\beta})\right]}$$
(12)

where γ_{β} is the projection threshold, $\hat{\gamma}$ – the projection density, and β – the projection intensity. The γ_{β} and β are set to 0.5 and 12, respectively. To ensure the convergence of optimization results, the projection control parameter β is set through a parameter scanning method, starting from an initial value of one and experiencing a twofold increase every thirty iterations until $\beta = 32$.

Modelling of multi-objective optimization problem

To comprehensively evaluate the effectiveness of a cold plate, two factors should be carefully considered: flow and heat transfer. Evaluation of the heat transfer performance involves analyzing the total heat generation occurring within the designated domain. Thus, maximizing heat transfer efficiency is equivalent to maximizing internal heat generation, as expressed:

$$J_{\rm th} = \int_{\Omega} (1 - \gamma) \eta (T_Q - T) \mathrm{d}\Omega \tag{13}$$

where $1 - \gamma$ ensures that heat is generated independently by the solids. The fluid primarily acts as a heat transfer medium. By confining heat generation the solid components, the optimization can concentrate on channel structure design to improve heat transfer efficiency.

In addition, another objective involves minimizing fluid dissipation energy to enhance the flow characteristics of the cold plate and ensure smooth continuity of the flow path. This objective is defined by an integral-based objective function, expressed as:

$$J_{f} = \int_{\Omega} \left[\frac{1}{2} \mu \sum_{i,j} \left(\frac{\partial u_{i}}{\partial x_{j}} + \frac{\partial u_{j}}{\partial x_{i}} \right)^{2} + \sum_{i} \alpha(\gamma) u_{i}^{2} \right] d\Omega$$
(14)

After normalizing each objective, the weighting factors, denoted by w_1 and w_2 , are used to combine the two objectives into a unified objective function, which can be formulated:

$$J'_{\rm th} = \frac{J_{\rm th} - J_{\rm th,min}}{J_{\rm th,max} - J_{\rm th,min}}, \ J'_f = \frac{J_f - J_{\rm min}}{J_{f,max} - J_{f,min}}$$
(15)

The mathematical representation for topology optimization is presented below:

Find
$$\gamma$$

Minimize $J = -w_1 J'_{th} + w_2 J'_f$
Subject to (2) - (12)
 $\frac{1}{V} \int_{\Omega} \gamma d\Omega - V^* < 0$
 $0 \le \gamma \le 1$
(16)

where V is the total volume of the domain and V^* – the maximum allowable volume fraction of the fluid. The constraints ensure a balanced distribution between the fluid and solid regions within the cold plate. Moreover, the constraints contribute to the stability and convergence of the optimization algorithm.

Numerical implementation of topology optimization

Implementation method

The described problem is implemented in COMSOL Multiphysics, which employs a finite element analysis approach. The solution of the convection-diffusion equation and the heat diffusion problem in the substrate are carried out using the heat transfer module [23]. The flow field analysis is conducted using the CFD module, where velocity and pressure are solved through second-order and first-order discretization, respectively. The direct sparse solver PARDISO provided by COMSOL Multiphysics is available for solving discrete finite element equations [24]. Fluid, heat transfer, and optimization variables are solved by separate solution procedures. The sensitivity analysis is carried out utilizing the adjoint variable method, owing to the invisible relationships between the objective function and design variables. The iterative algorithm adopts the method of moving asymptotes (MMA) for updating the variables [25]. The MMA algorithm shows significant advantages in topology optimization due to its efficient handling of non-linear problems, rapid convergence, and applicability to large-scale problems. This approach can address various multi-physical topology optimization problems and is applicable across a wide range of scenarios. The convergence criterion is set:

$$\varepsilon = \left| J^{k+1} - J^k \right| \le 10^{-5}$$

Topology optimization results

Cold plates with consistent properties in the vertical direction are more advantageous for manufacturing. Additionally, research shows that at the height of the cold plate, the variations in the velocity field across different layers are relatively small [26]. Therefore, this study conducts topology optimization within a 2-D design domain. Subsequently, the 2-D result is extruded to create a 3-D structure for numerical analysis. The topology optimization is carried out using the given parameters of Re = 100, $\eta = 1 \cdot 10^6$ W/m²K. By adjusting the weight coefficients of the objective function, multiple optimized cold plate flow channel topologies are obtained, as depicted in fig. 2. Generally, the fluid enters the channel and divides into several main branches, which subsequently separate into smaller channels that resemble a *tree* distribution pattern. Finally, all branch channels converge into the main outlet channel, with a symmetric distribution of the final channel structure. Further analysis demonstrates that, when the objective of fluid pressure drop predominates in the optimization, there is a reduction in the number of independent solid domains and channel branches, alongside an increase in channel size. This effect arises because fewer branches and wider channels can significantly reduce the pressure drop loss inside the channels. As the weight of the heat transfer objective J_{th} gradually increases, the exclusive solid domains are further subdivided into smaller block domains, and the number of channels increases. Additionally, smaller branches emerge, and channels progressively envelop the entire design domain, ensuring sufficient heat transfer boundaries for effective heat exchange.



Figure 2. Topology optimization structures of the flow channel under different weighting factors for multiple objectives



Figure 3. Curves of pressure drop ΔP and heat transfer boundary length with changes in the weighting factor of the two objectives

The variations in pressure drop ΔP and the length of the heat transfer boundary with the weighting factor of the two objectives are shown in fig. 3. Observations reveal that the values of ΔP and the length of the heat transfer boundary are relatively small when the weighting factor is set to $w_1: w_2 = 0.2:0.8$. As the value of $w_{1802}: w_2$ starts to increase, both parameter values exhibit a gradual increase. The trends in changes for both parameters are consistent, aligning with the observed conclusions. Cai, Y., *et al.*: Topology Optimization and Experimental Investigation ... THERMAL SCIENCE: Year 2025, Vol. 29, No. 2A, pp. 797-810

Pareto front for multi-objective optimization

In multi-objective optimization problems, the Pareto front provides a comprehensive solution space, facilitating the selection of an optimal solution based on specific requirements. A suite of optimal solutions is obtained by solving multiple sets of multi-objective optimization problems with distinct weighting factors. This approach enables the exploration of various trade-offs between different objectives, thereby facilitating the identification of desirable solutions. Figure 4 shows the Pareto front for the objectives of flow and heat transfer. The Pareto



Figure 4. Pareto front for two objectives of flow and heat transfer

optimal points are fitted using a third-degree polynomial. Under the condition of satisfying the governing equations and constraints of the optimization problem, all the proposed optimal solutions are verified to converge. This verification process guarantees the reliability and accuracy of the optimization results.

Simulation analysis of cold plate thermal performance

Construction of simulation model

To validate the efficiency of the topology optimization design method, the optimization result for a channel with a weighting factor $w_1: w_2 = 0.7: 0.3$ is selected. Concurrently, a parallel flow channel cold plate is designed for comparison, ensuring that the fluid volume fraction and the length of the heat transfer boundary match those of the optimized design model. The two cold plate models are established, as illustrated in fig. 5.



Figure 5. The 3-D cold plate models of; (a) topology optimization flow channel and (b) straight flow channel

The following assumptions are made for the cold plate simulation model. Heat transfer is 3-D and steady-state. The fluid under consideration manifests as an incompressible laminar flow. The boundary condition attributed to the wall is that of a no-slip surface. All other wall

Relative error

boundaries are adiabatic except for a uniform heat source applied to the base plate. Simulations are conducted at different inlet velocities, ranging from 0.02 m/s $< u_{in} < 0.22$ m/s. The cold plate is fabricated using aluminum 6061, with water serving as the coolant. A reference ambient temperature is set at 20 °C, and the outlet pressure is maintained at 0 Pa. Furthermore, a uniform heat flux of $q_0 = 3.2 \cdot 10^4$ W/m² is supplied to the underside of the cold plate. According to the material database in COMSOL Multiphysics, the properties of both materials are detailed in tab. 1 [23].

Table 1. Property parameters of materials

Property parameters	Density [kgm ⁻³]	Specific heat [Jkg ⁻¹ K ⁻¹]	Thermal conductivity [Wm ⁻¹ K ⁻¹]	Dynamic viscosity [kgm ⁻¹ s ⁻¹]
Aluminum 6061	2700	896	167	_
Water	998	4180	0.6	0.001

Grid independence test

To avoid numerical errors attributed to suboptimal grid quality, a grid independence test is performed. As shown in tab. 2, four different grid partitioning schemes are used for numerical verification. Upon exceeding 864338 grid elements, the relative errors of ΔP at the cold plate inlet and outlet, and ΔT in the system, are all observed to be less than 1%. This indicates that the simulation results are relatively independent of grid quality.

Tuble 2. Results of grid independence test						
Number of grids	ΔT	Relative error	ΔP			
446139	32.7535	_	3.9832			

Table 2 Results of grid independence test

0				
446139	32.7535	_	3.9832	_
563705	33.3046	1.37%	4.0460	1.58%
634694	33.1609	0.43%	4.1566	2.73%
864338	33.1124	0.15%	4.1897	0.80%

Results and discussion

Analysis of numerical results

Figure 6 depicts the temperature distribution on the cold plate surface, as determined by simulations at an inlet velocity of 0.1 m/s. The topology-optimized model exhibits an average temperature of 44.2 °C, a maximum surface temperature of 54.3 °C, and a root mean square temperature of 6.2 °C. An elliptical low temperature zone is centered within the model. This configuration facilitates enhanced heat exchange with the low temperature coolant over a greater solid area, resulting in more efficient thermal performance. For the parallel channel model, the average temperature stands at 47.2 °C, with a maximum temperature of 65.0 °C and a root mean square temperature of 12.6 °C. The low temperature area presents a triangular distribution, predominantly near the inlet. Consequently, the optimized channel model significantly outperforms the parallel channel model in terms of surface temperature uniformity.

Figure 7 shows the numerical simulation curves for maximum temperature and pressure drop relative to the inlet velocity. As the inlet flow velocity increases, the heat exchange between the fluid and the wall is further strengthened, resulting in a decreased surface temperature of the heat-generating device. At an inlet flow velocity of $u_{in} = 0.18$ m/s, the topology optimization model exhibits a maximum temperature approximately 15% lower than that observed in the parallel channel model. Figure 7(b) shows the curve of pressure drop variation

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Figure 6. Temperature distributions on the cold plate surfaces with; (a) the topology-optimized channel and (b) the parallel channel configurations



Figure 7. Numerical simulation curves of; (a) the maximum temperature on the cold plate surface and (b) the pressure drop relative to the inlet velocity

with flow velocity. As the inlet flow velocity increases, there is a concurrent rise in the pressure drop across the channel. Nevertheless, it is essential to highlight that the pressure drop of the optimized channel consistently remains lower than that of the parallel channel structure. This underscores one of the advantages of topology optimization designs in reducing flow losses within the cold plate.

Thermal performance analysis

The hydrothermal capability of the cold plate is further evaluated by calculating the Nusselt number and thermal resistance. The formula for calculating the average Nusselt number, Nu_{avg} :

$$\mathrm{Nu}_{\mathrm{avg}} = \frac{h_{\mathrm{avg}} D_{\mathrm{h}}}{k_{f}} \tag{17}$$

where h_{avg} is the average heat transfer coefficient and D_{h} – the hydraulic diameter.

The thermal resistance $R_{\rm th}$ can be expressed: $R_{\rm th} = \frac{T_{\rm surf,max} - T_{\rm in}}{Q_{\rm wr}}$ (18)

where Q_w is the power of the heat source.

Figure 8(a) shows the curve of Nu_{avg} changing with the inlet velocity. The analysis reveals that the parallel channel configuration exhibits a lower Nusselt number compared to the topology optimization channel. Specifically, at an inlet velocity of 0.18 m/s, the topology optimization design demonstrates the Nusselt number increase of approximately thirty percent relative to the parallel channel configuration. The enhancement in heat transfer capacity is primarily attributed to the refined branch-shaped channel topology. In the optimized design, the channel gradually splits into smaller branches from the main channel. Following adequate heat exchange with the substrate, the flow converges back into the main channel, promoting fluid cross-mixing and heat transfer to the wall.



Figure 8. Numerical simulation curves of; (a) the Nu_{avg} and (b) the thermal resistance R_{th} vs. the inlet velocity

On the other hand, the presence of small branches and curved channels in the optimization process disrupts the flow boundary-layer and diminishes its thickness, increasing the heat transfer capacity. Figure 8(b) illustrates the curve of the total thermal resistance changing with the inlet velocity. With increasing velocity, there is a noticeable trend of the total thermal resistance decreasing. The optimized design exhibits a lower total thermal resistance compared to the parallel channel design. Specifically, when $u_{in} = 0.18$ m/s, the topology optimization design achieves a reduction in total thermal resistance of approximately 23%.

Experimental verification

Manufacture of cold plate and experimental system

The material selected for the cold plate during the topology optimization process is aluminum 6061. The cold plate is partitioned into two segments for the manufacturing procedure. Annular grooves, measuring 2.8 mm by 2.6 mm in cross-section, are designed around the flow channel and filled with a silicone seal. Sealing is accomplished by fastening the extruded sealant. The cold plate inlet and outlet are connected to the pipe-line using pneumatic quick connectors. An airtightness test confirmed the effectiveness of the seal. Figure 9 displays the cold plate sample.

Figure 10 displays the experimental set-up for the cold plate. In the experiment, a thermostatic water bath is used to regulate the inlet water temperature, offering a control range of -5 °C to 100 °C and a temperature control precision of ± 0.05 °C. A peristaltic pump, which can attain a maximum output flow rate of 1300 mL per minute with 1% accuracy, serves as the liquid driving force. A multiplex temperature tester is employed to measure the inlet water temperature and the temperature at nine specific monitoring points. Additionally, the pressure drop across the channel of the cold plate is measured using a pressure sensor, which has a range

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of 0-3 kPa and an accuracy of $\pm 0.075\%$. The uniform heat source is simulated by four high temperature ceramic heating plates with a heat flux of 3.2 W/cm².



Figure 9. The internal channel and external packaging structure of the cold plate sample



Figure 10. Experimental system for liquid cooling of; (a) the principle schematic diagram and (b) the photograph

Experimental results analysis

To study the hydrothermal performance at different flow velocities, the peristaltic pump is adjusted using the control panel to achieve the desired inlet velocities of $u_{in} = 0.06$ m/s, 0.1 m/s, 0.14 m/s, 0.18 m/s, 0.22 m/s, and 0.26 m/s. The nine temperature test points are evenly distributed on the surface of the cold plate. The aforementioned experimental scheme yields



Figure 11. Comparison curves between the experimental and simulation outcomes for; (a) the outlet temperature and (b) the average temperature of the measurement points across various inlet velocities

the average temperature of the measurement points and the outlet water temperature. Figure 11 shows the comparison curves of the experimental and simulation results. The graph unequivocally displays a consistent enhancement in the heat transfer capability between the fluid and the wall with increasing velocity. Consequently, the experimentally measured average temperature and the outlet water temperature exhibit a continuous decline, albeit at a diminishing rate. The experimental and simulated values show good agreement, with maximum errors of 3.07% and 3.57%, respectively. Moreover, as the flow velocity increases, the discrepancy between experimental and simulation data diminishes. The comparison of the simulation and experimental data verifies the effectiveness of the numerical analysis and proves the rationality of the topology optimization design method. These findings provide an effective reference for engineering applications.

Due to machining errors of the sample, experimental instrument errors, and other uncertainties, the final experimental results show some deviations. The absolute machining error of the inlet channel diameter is ± 0.01 mm. The relative errors of temperature mesaurement and pressure measurement are $\pm 0.5\%$ and $\pm 0.2\%$, respectively. The relative error of the heating element power loss is $\pm 4\%$, and the relative error of flow control is $\pm 3\%$. To analyze the uncertainty in this experiment, this method is used to calculate the uncertainty:

$$U_R = \sqrt{\sum_{i=1}^{i=N} \left(U_{x_i} \frac{\partial R}{\partial x_i} \right)^2}$$
(19)

where U_R is the absolute uncertainty of factor R, U_{x_i} – the uncertainty of each independent parameter, and N – the number of independent parameters.

It follows that the maximum relative uncertainty of the inlet velocity u_{in} can indeed be calculated to be 4.08%.

Conclusions

In this study, a complex mathematical model has been developed to optimize the topology of cooling channels, aiming to maximize heat transfer efficiency and minimize fluid dissipation. The density filter and projection combination method are employed to mitigate numerical instability. The efficacy of the model is demonstrated by solving the arrangement example of inlet and outlet distribution at the centerline of the cold plate, which yields a coherent and well-defined channel structure.

Optimization models under different weight combinations are solved. Through a comprehensive analysis of the pressure drop and the heat transfer boundary length, the impact of the objective function weight on the channel form is thoroughly investigated. The acquisition of the Pareto front in the topology optimization design provides designers with a range of feasible alternatives, thereby streamlining the decision-making process.

The topology optimization result, with a weighting factor of $w_1: w_2 = 0.7: 0.3$, is selected to establish a cold plate simulation model. For comparison, a conventional parallel flow channel cold plate is designed, maintaining the same fluid volume fraction and heat transfer boundary length. The cooling performance analysis demonstrated that the topology optimization channel exhibited significantly superior cooling performance compared to the parallel flow channel.

The cold plate sample is designed and manufactured, followed by a cooling performance experiment. The results reveal that the maximum error between the simulation and the experiment is within 4%, further proving the rationality of the topology optimization design method. This research not only advances the understanding of cooling channel optimization but also offers a robust framework for the design of more efficient thermal management systems. Cai, Y., et al.: Topology Optimization and Experimental Investigation ... THERMAL SCIENCE: Year 2025, Vol. 29, No. 2A, pp. 797-810

Acknowledgment

This work was supported by National Natural Science Foundation of China (No. 52275270), National Natural Science Foundation of China National Key R&D Plan of *Gravita-tional Wave Detection* Key Special Project No. (2021YFC2203501), and Natural Science Basic Research Program of Shaanxi, Project No. (2023-JC-JQ-38).

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