Manufacturing Rev. **13**, 3 (2026)
© H. Wang et al., Published by EDP Sciences 2026
https://doi.org/10.1051/mfreview/2025032

Manufacturing Review
Available online at:
https://mfr.edp-open.org

ORIGINAL ARTICLE OPEN ACCESS

Multi-objective intelligent optimization design of a bioinspired spider-web microchannel heat sink

Haichuan Wang[1], Jiayu Sun[2,3], Fei Qi[1,*], Xueliang Fan[4], Yuan Qiu[2,3], and Chunju Wang[1,*]

[1] School of Electrical and Mechanical Engineering, Soochow University, Suzhou 215137, PR China
[2] Shanghai Aerospace Electronic Communication Equipment Institute, Shanghai 201109, PR China
[3] Shanghai Key Laboratory of Collaborative Computing in Space Heterogeneous Networks (CCSN), Shanghai 201109, PR China
[4] School of Rail Transportation, Soochow University, Suzhou 215137, PR China

Received: 12 December 2025 / Accepted: 24 December 2025

Abstract. With the rapid increase in chip computing power, the heat flux density has sharply increased, becoming one of the bottlenecks restricting chip performance. This study presents a synergistic approach to thermal management by integrating a bioinspired spider-web microchannel heat sink with intelligent optimization. A multi-objective particle swarm optimization algorithm was employed to systematically optimize channel geometry, maximizing heat transfer while minimizing pressure loss. The TOPSIS-based decision-making identified an optimal solution that increases the convective heat transfer coefficient by 35.2% while reducing pressure loss by 8.3%. And, this advantage is particularly evident at 320 W/cm^2, with a maximum temperature reduction of 9.68 K. It means that the proposed structure significantly enhances both thermal and hydraulic performance compared to conventional designs, with improvements becoming more pronounced under increasing heat flux. Finally, the diamond/copper composite material was used to prepare the microchannel heat sink, and the heat dissipation performance of the designed biomimetic heat sink was verified by loop test. This work demonstrates the strong potential of combining bioinspired design with intelligent optimization for advanced thermal management in high-heat-flux applications.

Keywords: High heat flux / microchannel heat sink / bionic spider-web / diamond/copper composite material / multi-objective optimization / TOPSIS-based decision

Nomenclature

DOA	Domain of attraction
PEC	Performance evaluation criterion
PSO	Particle swarm optimization
MOPSO	Multi-objective particle swarm optimization
TPOSIS	Technique for order preference by similarity to ideal solution

1 Introduction

Driven by advances in artificial intelligence, the rapid increase in chip power density has posed unprecedented challenges to thermal management technologies. Traditional air cooling approach is increasingly inadequate for meeting the demands of high heat flux in modern electronic devices [1,2]. The high power consumption of high-performance chips inevitably leads to high heat flux density. Currently, NVIDIA's H100 GPU has reached a power consumption of 700 W [3], and the heat flux density for aerospace-grade high-performance chips is expected to reach 80 W/cm^2, and posing severe challenges for thermal management technologies. Relevant studies indicate that when the temperature of electronic equipment exceeds 338.15 K, its service life decreases by 50% for every 10 K increase [4]. Currently, mobile phone primarily relies on passive thermal management for processor cooling, where structural conduction can only dissipate up to several tens of watts. Conventional heat pipes have limited heat transfer capacity and are susceptible to gravity effects [5]. Existing thermal management technologies can no longer meet the cooling requirements of spaceborne high-performance chips [6], necessitating the exploration of new cooling methods. Compared with traditional cooling techniques, microchannel heat sink technology offers the advantages of a compact structure and high thermal performance, making it a vital solution for addressing the thermal challenges of high-heat-flux electronic devices [7–9].

* e-mails: cjwang@suda.edu.cn; qifei@suda.edu.cn

This is an Open Access article distributed under the terms of the Creative Commons Attribution License (https://creativecommons.org/licenses/by/4.0), which permits unrestricted use, distribution, and reproduction in any medium, provided the original work is properly cited.

Table 1. Simulation comparison of various microchannel topologies.

Topology	Heat transfer area (mm^2)	Pressure drop (KPa)	Maximum temperature of the chip (K)
River shape	50	148.1	368.6
Leaf vein shape	101	96.2	358.7
Honeycomb	85	90.8	358.1
Bionic spider-web	110	93.1	356

The concept of microchannel heat sinks was first proposed by Tuckerman and Pease in 1981 [10]. Their groundbreaking work involved etching microchannel grooves onto a silicon substrate bonded to the back of a chip and investigating the cooling effect of water forced through these microchannels. At a flow rate of 516 mL/min, their design achieved an exceptional heat dissipation capacity of 790 W/cm^2, establishing the potential of microchannel technology for thermal management. Following this foundational research, numerous scientists have conducted extensive studies to advance microchannel heat transfer. Chen et al. [11] developed a novel composite microchannel heat sink for phased array antenna thermal management. Their design demonstrated significant improvements in both flow performance and temperature uniformity compared to conventional microchannel structures. Shen et al. [12] designed and fabricated a staggered pin-fin microchannel structure on the backside of chip. This configuration exhibited superior heat transfer and hydrodynamic characteristics over traditional rectangular channels, resulting in an additional reduction of the chip surface temperature by 8 K. Building on these advancements, Chen et al. [13] proposed a double-layered microchannel array with cylindrical pin fins, achieving a remarkable heat dissipation capacity of up to 1200 W/cm^2.

Inspired by efficient natural structures, researchers have developed numerous bionic microchannel heat sinks to address thermal management challenges [14]. These designs leverage biological optimization principles to enhance both heat transfer efficiency and flow characteristics. Wu et al. [15] designed a tree-shaped microchannel liquid-cooled heat sink, which achieved an average surface temperature of only 339.78 K under a heat flux density of 8 W/cm^2. This performance demonstrates its suitability for cooling electronic chips. Han et al. [16] applied topology optimization methods to refine a bionic spider-web-inspired heat sink. The optimized topology exhibited lower thermal resistance compared to the original spider-web structure, highlighting the effectiveness of computational design in enhancing thermal performance. Wang et al. [17] drew inspiration from the placoid scales of shark skin to develop a bionic microchannel heat sink. By adjusting parameters such as the arrangement and density of the scale-like structures, they significantly improved the heat sink's overall performance. Wu et al. [18] compared the advantages and disadvantages of various biomimetic topologies under the same conditions, and the results are shown in Table 1. Bionic spider webs combine the advantages of large specific surface area of leaf vein heat dissipation and good honeycomb flow performance, and have good engineering application value.

In recent years, the integration of artificial intelligence algorithms with microchannel design has attracted significant attention from researchers [19]. Lu et al. [20] proposed an optimization method combining neural network prediction and multi-objective genetic algorithm for trapezoidal rib microchannel heat sinks. Compared to non-optimized heat sinks, the pressure drop was reduced by 87.9 Pa under the same Nusselt number. Liu et al. [21] used simulation results to construct an artificial neural network model, rapidly predicting the performance of heat sinks with different structural parameters, providing new ideas for optimizing heat sink design. Zhang et al. [22] employed a genetic algorithm to optimize the structural parameters of manifold microchannels; the optimized structure reduced the total thermal resistance by 12.51% under the same pressure drop.

In summary, designing microchannel heat sink structures using bionic topology and artificial intelligence algorithms can effectively improve the heat dissipation performance of heat sinks, optimize flow performance, and meet the thermal management requirements of high heat flux chips.

Based on the above research background and requirements, this study designed a microchannel spider-web structure microchannel heat sink, compared the heat dissipation and hydraulic performance of the channel structure with the traditional structure, and used an intelligent optimization algorithm to give the optimal size under different engineering objectives, and systematically analyzed the key evaluation indicators of the radiator such as pressure drop, convective heat transfer coefficient and thermal resistance of microchannel heat sink.

2 Numerical simulation

2.1 Bionic microchannel structure design

A comparative analysis of conventional microchannel heat sink structures was conducted, integrating bionics and manufacturing process considerations to identify an optimal bionic heat sink configuration that combines high thermal performance with feasible fabrication. Inspired by natural systems, a spider-web-like structure with large heat transfer area was selected. Subsequently, a bionic microchannel heat sink model was developed, and conjugate heat transfer simulations incorporating fluid-solid coupling were

Fig. 1. Schematic diagram of microchannel heat sink.

performed to numerically analyze the fluid dynamics and thermal characteristics. Based on these findings, a diamond/copper bionic microchannel heat sink was designed.

The microchannel heat sink model is shown in Figure 1. The structure includes a finned cover plate, a fluid chamber, and a base plate, with total dimensions of 70 mm in length, 8 mm in height, and 40 mm in width. The heat sink features square-shaped inlet and outlet ports, each measuring 4 mm × 5 mm. The wall thickness of the base plate is 1 mm.

The structure of the heat sink fin is shown in Figure 2, the total length of the heat sink cover plate is 46 mm, the thickness of the base is 3 mm, the cover plate is composed of 9 fins, the height of the fin is 4 mm, the thickness of the middle fin is 4 mm, the thickness of the remaining fins is 1.8 mm, and the fluid chamber is divided into 10 runners by fins, each with a runner width of 2 mm. At the same time, in order to reduce hydraulic loss, the corners and sharp corners of the fins are rounded.

2.2 Simulation boundary condition

In order to simplify the simulation calculations, the following simulation assumptions are made:

– The simulation model only has two heat transfer modes: heat conduction within solids components and thermal convection between solid walls and fluids, and the rest of the walls are set as adiabatic insulation, and thermal radiation is not considered.
– The thermophysical properties of fluids and solids are constant values that do not change with temperature, the fluid is incompressible, viscous dissipation and gravity are not considered, and the fluid flow state is laminar flow.
– The wall between the flow and solid is set as a no-slip boundary.

Based on the above assumptions, the governing equations of the simulation model are given as follows [23]:

Fig. 2. Structure of heat sink fins.

– Continuity Equation:

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}=0. \tag{1}$$

– Conservation Momentum Equation:

$$\rho(u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}+w\frac{\partial u}{\partial z})=-\frac{\partial P}{\partial x}+\nabla\cdot(\mu\nabla u) \tag{2}$$

$$\rho(u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}+w\frac{\partial u}{\partial z})=-\frac{\partial P}{\partial x}+\nabla\cdot(\mu\nabla v) \tag{3}$$

$$\rho(u\frac{\partial w}{\partial x}+v\frac{\partial w}{\partial y}+w\frac{\partial w}{\partial z})=-\frac{\partial P}{\partial x}+\nabla\cdot(\mu\nabla w). \tag{4}$$

– Energy Conservation Equations for Solid Domains:

$$\rho c_p(u\frac{\partial t}{\partial x}+v\frac{\partial t}{\partial y}+w\frac{\partial t}{\partial z})=\lambda(\frac{\partial^2 t}{\partial x^2}+\frac{\partial^2 t}{\partial y^2}+\frac{\partial^2 t}{\partial z^2}). \tag{5}$$

– Energy conservation equations for the solid domain:

$$k_s\nabla^2 T=0 \tag{6}$$

Fig. 3. Meshing of boundary layers.

Fig. 4. Grid independent verification.

where, u, v, w, is the velocity component of the fluid in the three directions of x, y, z; respectively, ρ is the density of the fluid; P is the pressure on the fluid microelements; μ is the dynamic viscosity of the fluid; λ, k_s, is the thermal conductivity of fluids and solids, respectively.

The boundary conditions of the microchannel heat sink heat-fluid coupled flow heat transfer simulation model are as follows:

- The working fluid is liquid water, the inlet flow rate is set to 100~600 mL/min, the inlet temperature is room temperature (293.15 K), the microchannel thermal sink outlet is set to the pressure outlet boundary condition, and the outlet pressure is 0 Pa;
- The material of the heat sink fin is diamond/copper composite [22], the thermal conductivity is 564.2 W/(m·K), the material of the heat sink base is aluminum alloy, the thermal conductivity is 140 W/(m·K), and the heat source material is ceramic, and the thermal conductivity is 1.39 W/(m·K).
- The heat source area is set to 0.9 mm×0.9 mm, and the heat flux density was 80 W/cm^2. All other external walls are set to adiabatic boundary.

2.3 Grid independence verification

To ensure the accuracy of the simulation, enhance computational efficiency and reduce the calculation time and costs, the grid independence test was conducted on the microchannel heat sink of the designed spider-web structure. For the test, the inlet flow velocity of the simulation model is 0.4 m/s, and two parameters the pressure drop ΔP between the inlet and outlet and the average temperature T_{ave} of the heat source coverage area are used as the evaluation indices.

Fig. 5. Comparison of simulation and theoretical calculations.

Figure 3 shows the meshing with 5 boundary layer meshes near the wall. As shown in Figure 4, the mesh counts selected for the test are 1688815,2642385, 3723501, 4537037, 5310105 and 6021814, which shows that with the increase of the number of meshes, the simulation values of the two evaluation indices gradually converge, and after increasing to the number of meshes of 4537037, the simulation values exhibit negligible changes, and continuing to increase the number of meshes will only increase the calculation time cost. Therefore, subsequent simulations are set to this number of meshes.

2.4 Model validation

To ensure the correctness of the CFD energy conservation and simulation parameter settings, the area-weighted average temperature of the fluid at the heat sink outlet fluid under different inlet Reynolds numbers Re is compared with the theoretically calculated outlet average temperature. The theoretical formula for calculating the Reynolds number and the average temperature of the fluid outlet is as follows [22]:

$$Re = \frac{\rho v_{in} D_h}{\mu} \tag{7}$$

$$Q = C_f \dot{m}(T_{out} - T_{in}) \tag{8}$$

where v_{in} is the average flow velocity of the fluid inlet, D_h is the hydraulic diameter of the thermal sink inlet, Q is the input heat of the heat source, C_f is the specific heat capacity of the fluid, m is the mass flow rate of the fluid, T_{out} and T_{in} are the average temperature of the fluid outlet and inlet respectively.

Fig. 6. Traditional flow channel structure: (a) Parallel-channel; (b) Straight-channel.

As shown in Figure 5, it is a comparison between the theoretical calculated and simulated values of the average temperature of the fluid outlet when the inlet Reynolds number changes from 370 to 2220 at 80 W/cm^2 heat flux density. It can be seen from the figure that the theoretical and simulated values are in good agreement, and the error of the two is the largest when inlet Reynolds number is 370, and the relative error is less than 0.06%. Moreover, the error decreases with the increase of the inlet Reynolds number, which verifies the effectiveness and reliability of the simulation model.

3 Result and discussion

3.1 Effects of fin structures on heat sink hydrothermal performance

In order to highlight the excellent characteristics of the thermal and hydraulic performance of the designed bionic spider web microchannel heat sink, it is compared with the traditional heat sinks (rectangular straight-channel and parallel-channel types) with the thermal-flow coupling simulation. The hydraulic diameter of the two traditional sinks of the microchannel channel is 2.48 mm, the inlet and outlet are both 5 mm x 4 mm rectangular structure is the fin thickness, the height is 4 mm, and the surface area of the three fin covers is the same, are depicted in Figure 6.

The heat sink of the three structures was simulated under the heat flux density of 80 W/cm^2 and the fluid inlet flow rate from 100~600 mL/min. As shown in Figure 7a, it is a simulated numerical comparison between the bionic spider-web heat sink and the straight-channel as well as parallel-channel heat sink at different fluid inlet velocities of the same heat flux density. Compared with the traditional heat sink structures, the maximum surface temperature of the spider-web heat sink is always the lowest, and with the increase of the inlet flow velocity, the heat dissipation advantage of the bionic spider-web heat sink becomes more and more obvious, when the inlet flow rate is 600 mL/min, the average temperature of the bionic spider-web heat sink surface decreases by 2.32 K and 1.58 K, respectively, compared with the straight-channel and parallel-channel heat sink structures. It can be seen from Figure 7b that the surface temperature difference of the bionic spider-web heat sink and the parallel-channel heat sink is always kept within 4.4 K, which has good temperature uniformity, while the temperature difference of the parallel-channel sink surface is large, and the temperature difference is more than 4.5 K under low flow rate. Figure 7c reflects the change of the inlet and outlet pressure drop of the heat sink with the inlet flow velocity, and it can be seen that the pressure drop of the three structures increases with the increase of the inlet flow velocity, and the pressure drop of the straight-channel heat sink and the bionic spider-web heat sink is basically the same, and the change is relatively gentle with the increase of the flow velocity, while the hydraulic performance of the parallel-channel heat sink is poor, and the pressure drop increases rapidly with the increase of the inlet flow velocity, reaching 425 Pa at the inlet flow rate of 600 mL/min.

Figure 8 shows the flow of fluids in the three structures under the inlet flow rate of 300 mL/min. It can be seen from Figure 8b that the fluid inside the straight-channel heat sink is mainly distributed in the middle four channels, while the fluid velocity of the upper and lower three channels decrease sharply, resulting in the heat in this part cannot be exported in time, and the heat dissipation area of the fins cannot be fully utilized. Since the heat source is located in the center of the heat sink, it can be covered by the middle four channels, so the temperature difference of the heat source surface is lower than that of parallel-channel heat sink. It can be seen from Figure 8c that the internal fluid of parallel-channel heat sink is mainly distributed in the channel near the outlet, while the channel flow velocity in the core area covered by the heat source is low, resulting in a large difference in surface temperature and high temperature. As can be seen from Figure 8a, compared with the previous two traditional heat sink structures, the fluid distribution inside the bionic spider-web heat sink is more uniform, and the fluid mixing degree increases due to the change of the fluid flow direction in the channel.

Table 2 reflects the maximum temperature of the heating surface of the three heat sink structures at the same inlet flow rate (300 mL/min) and different chip thermal power. It can be seen that with the increase of chip heating power, the influence of the bionic spider-web structure on the heat transfer ability of microchannel heat sink is also greater, and at the heat flux density of 320 W/cm^2, the maximum heating surface temperature of the bionic spider-web structure is nearly 9.68 K lower than that of the straight-channel heat sink. From the chip surface

Fig. 7. Changes of maximum temperature, temperature difference and inlet and outlet pressure drop with inlet flow velocity: (a) Maximum temperature change; (b) Temperature difference change; (c) Pressure drop changes.

Fig. 8. Fluid velocity contour plots of heat sink of three structures at inlet flow rate of 300 mL/min: (a) Bionic spider-web; (b) Straight-channel; (c) Parallel-channel.

Table 2. Maximum temperature of the chip at the same inlet flow.

Category	Maximum temperature of the chip (K)			
	80 W/cm^2	160 W/cm^2	240 W/cm^2	320 W/cm^2
Bionic spider-web	307.69	322.23	336.78	351.32
Straight-channel	310.27	327.40	344.52	361.64
Parallel-channel	309.36	325.57	341.78	358.00

Fig. 9. Changes of chip surface temperature cloud with heat flux density: (a) Bionic spider web; (b) Straight shape; (c) Parallel conjunction

temperature distribution in Figure 9, it can be intuitively seen that with the increase of heat flux density, the temperature of the bionic spider-web heat sink chip is always kept to the lowest, and the temperature uniformity is also better, which further reflects the enhanced heat transfer performance of the bionic microchannel heat sink. Therefore, comprehensive analysis shows that bionic spider-web microchannel heat sink has better heat dissipation capacity and good hydraulic performance than traditional heat sink structures.

3.2 Multi-objective intelligent optimization of flow channel structure

In order to further reveal the influence of microchannel heat sink structure on heat dissipation and hydraulic performance, and to determine the optimal structure size, a multi-objective particle swarm intelligent optimization (MOPSO) algorithm was used to optimize the pressure drop and convective heat transfer coefficient of microchannel heat sink by taking the flow channel width w_a and flow channel height h of microchannel heat sink as the optimization variables.

– Simulation of different heat sink structure parameters

The width of the microchannel heat sink channel varies from 1.2 mm to 2.4 mm (1.2 mm$\leq w_a \leq$2.4 mm), with design parameters set interval of 0.2 mm. The flow channel height varies from 3.4 mm to 4.6 mm (3.4 mm$\leq$h$\leq$4.6 mm), also with design parameters set at 0.2 mm, and there are a total of 49 simulated structures of microchannel heat sink, as shown in Figure 10. The design base width L is 40 mm, the middle fin plays the role of blocking and supporting, and the width w_c is 4 mm, which remains unchanged. w_b is the width of the remaining fins, calculated as follows:

$$w_b = \frac{L - w_c - n \cdot w_a}{n - 2} \tag{9}$$

where n is the number of flow channels.

The 49 sets of microchannel heat sink structures were simulated by CFD software. The simulation model sets the chip thermal power to 160 W/cm^2 and the inlet flow rate to 300 mL/min. At the same time, in order to ensure the convergence of the numerical results, the residuals of the continuity equation are set to 10^{-6}, and the residuals of energy and velocity are set to 10^{-6}. The average heat transfer coefficient of microchannel heat sink is calculated by the following formula:

$$h_{ave} = \frac{Q}{A_{cont}(\overline{T_w} - \overline{T_f})} \tag{10}$$

where A_{cont} is the convective heat transfer area between the sink fin and the fluid, $\overline{T_w}$ is the average temperature of the fin and the fluid contact wall, and $\overline{T_f}$ is the average temperature of the fluid.

Fig. 10. Dimensions of hot sink cross-section.

Fig. 11. Changes of pressure loss and convective heat transfer coefficient with channel width: (a) Pressure loss; (b) Convection heat transfer coefficient.

Fig. 12. Changes of pressure loss and convective heat transfer coefficient with flow channel height: (a) Pressure loss; (b) Convection heat transfer coefficient.

As shown in Figure 11, it can be observed that with the increase of flow channel width and channel height, the pressure loss and convective heat transfer coefficient show a downward trend. In the case of the same inlet flow, the smaller the flow channel width means that the flow speed of the fluid is larger, and a higher fluid velocity can take away the heat in time to avoid excessive temperature rise. In the same way, as shown in Figure 12, the increase of the flow channel height reduces the fluid flow velocity and reduces the pressure loss, and the increase of the height also increases the convective heat transfer area, which is conducive to the heat dissipation of the chip, but the convective heat transfer coefficient of microchannel heat sink also decreases, resulting in a decrease in the efficiency of the fin. On the other hand, the increase of the flow channel height will also lead to the

Fig. 13. Correlation between dimensionless pressure drop and dimensionless convective heat transfer coefficient.

increase of the heat deposition volume of the microchannel, which is not good for the lightweight of the heat dissipation system, and at the same time, the increase of the flow channel height will also increase the difficulty of fin manufacturing.

In order to intuitively compare the heat transfer and hydraulic performance of the designed bionic flow channel heat sink and the traditional heat sink, the pressure loss ΔP_0 and convective heat transfer coefficient h_{f0} calculated by traditional parallel-channel heat sink are used to normalized the pressure loss and convective heat transfer coefficient, as shown in Figure 13. The shaded area enclosed by the red dotted line represents the domain of attraction (DOA). In this region, the convective heat transfer coefficient ratio (h_f/h_{f0}) greater than 1 and a pressure loss ratio ($\Delta P/\Delta P_0$) less than 1, which means that the heat transfer efficiency and hydraulic performance are better than those of traditional heat sink structures. It can be seen that most of the biomimetic heat sink structures designed can reduce pressure loss while improving heat transfer performance compared with traditional heat sink structures.

– Establishment of multi-objective function model

In order to realize the multi-objective optimization of microchannel heat sink pressure loss and convective heat transfer coefficient, the dimensionless pressure loss and convective heat transfer coefficient objective function

$z = f(e_1, e_2)$ is constructed, the polynomial regression function is used for nonlinear fitting, and the three-dimensional response surface is constructed. In the equation, e_1, e_2 is the dimensionless flow channel width and dimensionless flow channel height, respectively, with

Table 3. Coefficient values of the objective functions.

$\Delta P/\Delta P_0$ coefficient	Value	h_f/h_{f0} coefficient	Value
a_0	6.213	b_0	3.904
a_1	−1.464	b_1	−0.744
a_2	−5.954	b_2	−2.575
a_3	0.248	b_3	0.035
a_4	0.384	b_4	0.350
a_5	1.747	b_5	0.534

Table 4. Objective function errors and coefficient of determination.

Objective function	*RMSE*	R^2
$\Delta P/\Delta P_0$	0.00977	0.9970
h_f/h_{f0}	0.00425	0.9989

their calculation formulas given as follows:

$$e_1 = \frac{w_a}{w_0}, e_2 = \frac{h}{h_0} \tag{11}$$

where $w_0 = 1.2$ mm, which is the minimum value of the runner width; $h_0 = 3.4$ mm, which is the minimum value of the runner height.

The fitting functions are as follows:

$$\frac{\Delta P}{\Delta P_0} = a_0 + a_1 e_1 + a_2 e_2 + a_3 {e_1}^2 + a_4 e_1 e_2 + a_5 {e_2}^2 \tag{12}$$

Fig. 14. Dimensionless response surface: (a) Pressure drop; (b) Heat transfer coefficient.

$$\frac{h_f}{h_{f0}} = b_0 + b_1 e_1 + b_2 e_2 + b_3 {e_1}^2 + b_4 e_1 e_2 + b_5 {e_2}^2 \quad (13)$$

In order to verify the accuracy of the fitting function, the simulation values are compared with the predicted values, and the error of the prediction model is measured by using root mean square error ($RMSE$) and coefficient of determination (R^2), and the fit of the model is evaluated:

$$RMSE = \sqrt{\frac{1}{n}\sum_{i}^{n}(Y_i - \hat{Y}_i)^2} \quad (14)$$

$$R^2 = 1 - \frac{\sum_{i}^{n}(Y_i - \hat{Y}_i)^2}{\sum_{i}^{n}(Y_i - \overline{Y})^2} \quad (15)$$

where n is the number of samples, Y_i is the simulation value, Y is the model prediction value, and d is the average value.

The specific coefficient values, root mean square error, and coefficient of determination values of the objective functions are shown in Tables 3 and 4 below.

The closer the $RMSE$ value is to 0 and the closer the R^2 value is to 1, the better the fitting performance of the objective functions. From the above table, it can be seen that the root mean square error of the two objective functions is lower than 0.05, and the R^2 value is higher than 0.99, indicating that the prediction accuracy of the model is high.

Figure 14 shows the response surfaces of dimension less pressure drop and dimensionless convective heat transfer coefficient with dimensionless channel width and height, and it can be intuitively seen that when the pressure drop decreases with the width and height of the flow channel, the convective heat transfer coefficient also decreases simultaneously, and the two show a contradictory trend.

– TOPSIS-Based Multi-objective particle swarm optimization (MOPSO) algorithm

The basic principle of the particle swarm optimization (PSO) algorithm comes from the simulation of the foraging behavior of birds in nature, each particle represents a solution to an optimization problem, the particles continue to fly in a given search space, and constantly update their position and speed through the information interaction between them, gradually moving closer to the global optimal value, and finally finding the optimal solution.

The formulas for calculating the velocity and position of each particle are as follows:

$$v_i^{t+1} = w \cdot v_i^t + c_1 \cdot r_1 \cdot (pBest_i - x_i^t) + c_2 \cdot r_2 \cdot (gBest - x_i^t) \quad (16)$$

$$x_i^{t+1} = x_i^t + v_i^{t+1} \quad (17)$$

where v_i^t and x_i^t represent the velocity and position vectors of the ith particle at moment t, w is the inertia weight of the particle, its magnitude reflects the global search ability of the particle, c_1 and c_2 are the individual learning factors and social learning factors of the particle, r_1 and r_2 are the random numbers between [0,1], which are used to increase the randomness of the search, and $pBest_i$ is the historical optimal position of the ith particle (individual optimal), $gBest$ is the optimal position for all particles (overall optimal).

The multi-objective particle swarm optimization algorithm adds multiple optimization targets to the traditional particle swarm algorithm, and finally obtains not a single optimal solution, but a series of non-dominated solution sets, that is, Pareto optimal solution sets, in which each solution is the optimal solution under the result. The optimization process of multi-objective particle swarm is shown in Figure 15. The number of iterations is set to 100, and the initial population size is 200.

Fig. 15. Flowchart of the multi-objective particle swarm optimization algorithm.

The multi-objective particle swarm optimization algorithm aims to maximize the convective heat transfer coefficient and minimize hydraulic loss, and its mathematical description model is as follows:

$$\begin{cases} Maximize: h_f/h_{f0} \\ Minimize: \Delta P/\Delta P_0 \\ Subjest\,to: 1 \le e_1 \le 2 \\ 1 \le e_2 \le 1.36 \end{cases}. \tag{18}$$

Since the solutions obtained after multi-objective optimization are non-dominated by each other, and the best solution needs to be selected from the solution set according to the actual situation of the project, the TOPSIS method is used to score and rank each solution, and the solution closest to the ideal goal is selected.

The TOPSIS method, also known as the technique for order preference by similarity to an ideal solution, is a comprehensive evaluation method that evaluates schemes by calculating the distance from the ideal solution and the negative ideal solution. It consists of four steps: benefit normalization, dimensionless normalization, relative closeness calculation, and score normalization. Benefit normalization ensures that all indicators are extremely large; Dimensionless normalization eliminates dimensional influences; Relative closeness is derived by comparing the distances to the positive and negative ideal solutions; Score normalization is then performed to yield the final evaluation scores. This approach is suitable for multi-metric decision problems.

Fig. 16. Pareto optimal solution set and three optimization schemes.

Three optimization schemes, namely Topsis1, Topsis2 and Topsis3, are designed to cope with different situations in the actual project. Among them, Topsis1 tended to have hydraulic performance, with a convective heat transfer coefficient weight of 30% and a pressure drop weight of 70%. Topsis2, which balances the importance of both, with a weight of 50%; Topsis3, on the other hand, pays more attention to heat transfer performance, with a convective heat transfer coefficient weight of 70% and a pressure drop weight of 30%. As shown in Figure 16, the distribution of the Pareto optimal solution set and the three solutions optimized for the multi-objective particle swarm optimization.

As shown in Table 5, the optimization results of the three schemes and the corresponding CFD numerical simulation results are summarized, of which the maximum error of the two is 2%, which indicates that the designed multi-objective intelligent optimization model can accurately find the optimal solution in the predefined parameter space, which can greatly reduce the amount of numerical simulation computation. Taking Topsis3 as an example, the corresponding flow channel width is 1.2 mm and the flow channel height is 3.87 mm, and the convective heat transfer coefficient is 6696.4 W/ (m^2· K), and the pressure loss was 121.3 Pa. Compared with the traditional parallel-channel heat sink, the convective heat transfer coefficient is increased by 35.2% while reducing the pressure loss by 8.3%, which greatly improves the heat transfer efficiency of heat sink.

The size of thermal resistance reflects the resistance during heat transfer, which is mainly related to the heat sink material, flow channel structure, and heat transfer area. The calculation formula is as follows:

$$R_{th} = \frac{T_{ave} - T_{in}}{Q} \tag{19}$$

Table 5. Optimization values and numerical simulation verification.

Devise scheme	Design value		Optimize values		Numerical simulation		Error	
	e_1	e_2	h_fW/(m^2·K)	ΔPPa	h_fW/(m^2·K)	ΔPPa	h_f%	ΔP%
Topsis1	1.28	1.26	5834.6	84.1	5856.5	82.5	0.37	1.95
Topsis2	1.29	1.14	6315.5	103.1	6378.3	99.5	0.98	3.62
Topsis3	1	1.14	6696.4	121.3	6565.4	121.4	2.00	0.08

Fig. 17. Microchannel thermal resistance changes with flow rate.

Fig. 18. PEC with flow.

where T_{ave} is the average temperature of the chip heating surface.

Figure 17 shows the relationship between the thermal resistance of the microchannel structure and flow rate of the three optimization schemes. It can be seen that with the increase of flow rate, the cooling water takes away the heat faster, and the thermal resistance of the microchannel gradually decreases. In addition, although Topsis3 has the smallest heat transfer area, the thermal resistance of Topsis3 is always kept to a minimum, which is mainly related to its slender flow channel structure. Under the same flow rate, the smaller the flow channel section, the larger the flow velocity, which can accelerate the heat transfer process, and the influence of the flow rate of the liquid on heat dissipation is greater than that of the heat transfer area.

The performance evaluation criterion (PEC) is defined and the designed microchannel hydrothermal performance is comprehensively evaluated from the aspects of fluid resistance and convective heat transfer capability, with its calculation formulas are given as follows:

$$PEC = \frac{(Nu/Nu_0)}{(f/f_0)^{1/3}} \tag{20}$$

$$f = \frac{\Delta P D_h}{2L\rho_f v_{in}} \tag{21}$$

where Nu is the average Nusselt number of the microchannel, f is the friction factor of the microchannel, Nu_0 and f_0 are the average Nusselt number and friction factor of the parallel conjunction structure at a flow rate of 100 mL/min, respectively, L is the length of the microchannel, and v_{in} is the average inlet flow rate.

As shown in Figure 18, the PEC of the optimized microchannel structure is higher than 1, and it continues to increase with the increase of flow, which indicates that the comprehensive performance of the optimized bionic microchannel hydrosettling exceeds that of the traditional runner structure, and the better the performance with the increase of flow, the better the improvement effect of heat transfer is better than that of pressure drop.

4 Experimental set and results

In order to verify the effectiveness of the designed biontic spider-web microchannel channel heat sink, Topsis3-optimized heat sink fins were fabricated by sintering diamond/copper composites, and the preparation method was referenced [24].

Mix 60% of the diamond particles by volume and copper powder with 40% of the volume fraction thoroughly, pour into the graphite mold prepared in advance, and then put the mold into the sintering furnace for hot pressing and sintering. The sintering temperature was 1173 K, the sintering pressure was 4 MPa, and the holding time was 30 min.

The fabricated heat sink is shown in Figure 19a. The maximum shrinkage rate of the fin height was 1.64% by super-depth of field microscopy, and the influence of this error on the experiment was negligible. The heat sink base, as shown in Figure 19b, was machined from aluminum alloy, the assembled structure is presented in Figure 19c.

Fig. 19. Sintered diamond/copper fins and base structure: (a) The physical fins; (b) Dimensional analysis of fin super-depth of field; (c) Heat sink base.

Fig. 20. Fluid Loop Test Platform.

Fig. 21. Comparison of experimental and simulated values of temperature and pressure drop of heat sink surface at different flow rates.

The loop test platform is shown in Figure 20, the fin and base are sealed by sealant, the cooling medium is deionized water, and the flow of cooling water is controlled by a peristaltic pump. A rubber block was placed under the microchannel heat sink to ensure it remained level. Ceramic heating plates were used as the simulated heat source, and the heat flux was set to 80 W/cm^2. In order to enhance heat transfer, thermal conductive silicone grease is applied to the contact surface between the heating plate and the heat sink, and heavy objects are used to pressurize to reduce the thickness of the silicone grease while removing the internal air to prevent excessive contact thermal resistance. The entire experiment was conducted in the constant temperature laboratory to ensure that the ambient temperature of the laboratory is 293.15 K, and when the system reached a steady state, The surface temperature of the heat sink around the heating plate is measured by thermocouples and the pressure difference between the heat sink inlet and outlet is measured using a differential pressure transmitter. The thermocouple model is T-type, the measurement accuracy is ±0.5 K. The thermocouple signal is collected by a multi-channel signal collector with model DAQ970A.

Change the peristaltic pump speed and adjust the flow rate of cooling water in the loop. Figure 21 compares the experimental values with the simulated values, it can be seen that the maximum temperature of the heat sink surface decreases with the increase of flow, and the decreasing trend slows down. On the other hand, the pressure drop increases, and the increasing trend becomes larger. This is in good agreement with the simulation results. This proves the validity of the simulation results.

At the same time, we also notice that there is a large error between the experimental value and the simulation value, and the analysis reasons are as follows:

– The simulation model does not consider the contact thermal resistance between the chip and the heat sink. In actual experiments, the heat sink surface has a certain roughness and cannot achieve perfect contact with the chip. Although thermal conductive silicone grease can enhance the bonding between the chip and the heat sink, its thermal conductivity is low (<20 W/(m·K)), which still exerts a significant impact on heat transfer.

– During the preparation of diamond/copper fins, defects are highly likely to form inside the fins due to the uneven distribution of diamond particles and copper powder. These defects reduce the actual thermal conductivity of the fins and further impair the heat dissipation performance of the heat sink.
– When machining the heat sink base, a circular extension was designed to facilitate connection with the hose. Cooling water passing through the junction of the circular extension and the heat sink's square inlets and outlets generates additional local resistance loss, resulting in a relatively large measured pressure loss.
– The inner surface of the channels prepared by hot-press sintering exhibits high roughness, which further aggravates the hydraulic loss.

5 Conclusions

This study introduced and systematically evaluated a novel spider-web inspired microchannel heat sink for high-heat-flux thermal management. The key findings and contributions are summarized as follows.

An innovative bio-inspired design is proposed to enhance the thermal-hydraulic performance of the heat sink: The proposed spider-web structure offers a novel paradigm for microchannel design by mimicking efficient natural topologies. Its core innovation lies in constructing interconnected flow pathways that promote highly uniform fluid distribution and enhance flow mixing. This mechanism effectively overcomes the common drawbacks of flow maldistribution and inefficient heat transfer area utilization in traditional straight and parallel designs.

Superior integrated performance balances high heat dissipation and low flow resistance effectively: Comprehensive numerical simulations confirm that the bionic design synergistically enhances thermal and hydraulic performance. It achieves a significant reduction in maximum surface temperature (up to 9.68 K under 320 W/cm^2) while maintaining excellent temperature uniformity and a relatively low pressure drop, comparable to that of a straight-channel heat sink.

Performance optimization via intelligent workflow effectively improves the heat sink's hydrothermal performance: Building upon the promising bionic architecture, a multi-objective optimization framework was successfully implemented. This process demonstrates that the inherent advantages of the spider-web design can be further amplified through geometric tuning, culminating in an optimal configuration (Topsis3) that boosts the convective heat transfer coefficient by 35.2% and reduces pressure loss by 8.3% compared to a conventional parallel-channel heat sink.

Excellent machinability ensures feasible fabrication of the flow channel via powder sintering: The designed bionic spider-web fin structure is relatively simple and can be fabricated using high thermal conductivity diamond/copper composites, which can further enhance the heat dissipation ability of microchannel heat sink.

In summary, this work validates the spider-web bioinspiration as a highly effective design paradigm for advanced thermal management. It provides not only a superior-performing hardware structure but also a systematic design methodology, offering significant potential for cooling next-generation high-power-density electronics in aerospace and other demanding applications.

Funding

This work was supported by the Shanghai Aerospace Science and Technology Innovation Fund (No. SAST2023-073).

Conflicts of interest

The authors have nothing to disclose.

Data availability statement

Data are available on request from the authors.

Author contribution statement

Haichuan Wang: Methodology; Writing-original draft. Jiayu Sun: Data curation (lead); Validation (lead). Fei Qi: Supervision (lead); Writing - review & editing (lead); Investigation (lead). Chunju Wang: Project administration (supporting); Funding acquisition (supporting); Formal analysis (lead). Xueliang Fan: Algorithm (supporting); Software (Supporting). Yuan Qiu: Conceptualization (lead); Validation (lead).

References

1. S.Y. Shi, S.Z. Hu, Y.K. Ding, S.F. Wang, J.Y. Du, J.W. Zhang, W. Wang, C. Zhang, Review of two-phase liquid cooling technology for aerospace processing chips, Aerosp. Shanghai. (2025) 1–22
2. H. Jiang, X. Wang, C. Ding, D. Shan, B.H. Guo, Qi, J. Xu, A review of emerging design and theoretical progress on vapor chamber for efficient thermal performance, Int. J. Heat Mass Transfer **231** (2024) 125814
3. A. Narayanan, Q. Wang, S. Ozguc, R.W. Bonner, Investigation of server level direct-to-chip two phase cooling solution for high power GPUS; ASME 2024 International Technical Conference and Exhibition on Packaging and Integration of Electronic and Photonic Microsystems. (2024) InterPACK2024-141261
4. Y.W. Wang, W.Z. Li, C. Qi, J. Yu, Thermal management of electronic components based on hierarchical microchannels and nanofluids, Therm. Sci. Eng. Prog **42** (2023) 101910
5. D.L. Xian, Development and application of microchannel-based chip heatdissipation technology, Southeast University, 2023
6. B. Lei, L.L. Zhang, K.M. Kang, Thermal design and simulation of high performance computer for spacecraft equipment, Electro-Mech. Eng. **37** (2021) 29–32
7. L. Du, W. Hu, An overview of heat transfer enhancement methods in microchannel heat sinks, Chem. Eng. Sci **280** (2023) 119081

8. J. Gao, Z. Hu, Q. Yang, X. Liang, H. Wu, Fluid flow and heat transfer in microchannel heat sinks: Modelling review and recent progress, Therm. Sci. Eng. Prog **29** (2022) 101203
9. Z.Q. Yu, M.T. Li, B.Y. Cao, A comprehensive review on microchannel heat sinks for electronics cooling, Int. J. Extreme Manuf. **6** (2024) 022005
10. D.B. Tuckerman, R.F.W. Pease, High-performance heat sink for VLSI, IEEE Electron. Device Lett. EDL-2 (1981) 126–129
11. Z.P. Chen, B. Yuan, J. Yang, Z. Zhang, Y.Q. Tang, Y. Yang, H.S. Zheng, Y. Chen, Numerical simulation of a novel composite microchannel for large-scale THz phased array antennas thermal management, Int. J. Therm. Sci **214** (2025) 109924
12. Y.T. Shen, Y.H. Pan, H. Chen, W.L. Cheng, Experimental study of embedded manifold staggered pin-fin microchannel heat sink, Int. J. Heat Mass Transfer **226** (2024) 125488
13. C. Ao, B. Xu, Z.Q. Chen, Dual-layer cylindrical array of pin–fin microchannels with heat dissipation capacity up to 1200 W/cm^2, Int. J. Heat Fluid Flow **116** (2025) 110003
14. L. Yin, J.Q. E, Y. Tu, W.H. Luo, Biomimetic microchannel structures and their topological optimization: a review, Int. Commun. Heat Mass Transfer **163** (2025) 108689
15. S.J. Wu, K. Zhang, G. Song, J.C. Zhu, B.R. Yao, Experimental study on the performance of a tree-shaped mini-channel liquid cooling heat sink, Case Stud. Therm. Eng **30** (2022) 101780
16. X.h. Han, H.l. Liu, G. Xie, L. Sang, J. Zhou, Topology optimization for spider web heat sinks for electronic cooling, Appl. Therm. Eng **195** (2021) 117154
17. K.C. Wang, Y. Shi, J.J. Chen, Y.C. Dai, A biomimetic microchannel heat sink for enhanced thermal performance in chip cooling, Biomimetic **10** (2025) 459
18. L.W. Wu, T. Lu, J.J. Chen, M.Y. Wang, P. An Du, A study of bionic micro-channel topology for chip cooling, Acta Electron. Sin **46** (2018) 1153–1157
19. G.G. Song, H.Q. Chen, Y.H. Zhang, J.J. Wei, X. Ma, Reviews: applications of optimization algorithm for micro-channel and microchannel heat sink on heat transfer, Int. J. Heat Fluid Flow **108** (2024) 109451
20. K.J. Lu, C.J. Wang, H.D. He, X.L. Fan, F. Chen, F. Qi, Artificial neural network prediction and multi-objective genetic algorithm optimization of the microchannel heat sink with trapezoidal ribs, Therm. Sci. Eng. Prog **50** (2024) 102546
21. Y. Liu, S.Q. Lv, Q.L. Cui, Y.J. Xia, M.X. Jiang, J. Lv, M.R. Huang, Y.X. Xue, Device packaging and integration optimization based on neural network method: effect of microchannel structure on heat sink performance, Mater. Sci. Semicond. Process **185** (2025) 108977
22. J.Z. Zhang, J.J. Xu, J. An, J.X. Chen, Y. Liu, Q. Zhou, H.F. Pan, L. Lei, G.M. Xin, Multi objective optimization of manifold microchannel heat sink with staggered microchannels, Int. Commun. Heat Mass Transfer **159** (2024) 108106
23. H. Tan, L.W. Wu, M.Y. Wang, Z.H. Yang, P.G. Du, Heat transfer improvement in microchannel heat sink by topology design and optimization for high heat flux chip cooling, Int. J. Heat Mass Transfer **129** (2019) 681–689
24. K.J. Lu, C.J. Wang, H.D. He, X.L. Fan, F. Chen, F. Qi, Study of the hot-pressing sintering process of diamond/copper composites and their thermal conductivity, J. Alloys Compd **960** (2023) 170608

Cite this article as: Haichuan Wang, Jiayu Sun, Fei Qi, Xueliang Fan, Yuan Qiu, Chunju Wang, Haichuan Wang, Multi-objective intelligent optimization design of a bioinspired spider-web microchannel heat sink, Manufacturing Rev. **13**, 3 (2026), https://doi.org/10.1051/mfreview/2025032