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Recent Research Progress in Microchannel Heat Sink Technology

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Abstract: As electronic devices continue to miniaturize and experience higher power densities, thermal management has become a critical factor limiting their performance and reliability. Microchannel heat sinks (MCHS) have emerged as a promising heat dissipation solution due to their excellent thermal performance and compact structure. Thermal resistance and pump power are two key performance indicators of MCHS. In recent years, significant efforts have been made to enhance the heat dissipation capacity of MCHS and reduce pump power, leading to numerous research achievements. This paper reviews the recent progress in microchannel heat sink research, including its working principles, design optimization, manufacturing processes, performance evaluation, and applications in various fields. Additionally, the challenges faced by MCHS and future development directions are discussed.

Keywords: Microchannel heat sink, Heat dissipation, Thermal resistance, Thermal-hydraulic performance.

1. Introduction

Electronic products are developing towards miniaturization, multifunctionality, and high power, with increasing integration levels. To ensure reliable operation and prevent problems such as concentrated heat and rising heat flux density, efficient heat dissipation for high-density integrated microsystems must be achieved. Compared to traditional heat dissipation technologies, microchannel heat sinks (MCHS) are particularly suitable for thermal management of high-density integrated microsystems due to their advantages such as good compatibility with manufacturing processes, short heat dissipation paths, and strong heat dissipation capabilities.

An MCHS utilizes a high-conductivity substrate with a large number of parallel small-diameter channels for heat dissipation. These heat sinks are very lightweight and compact, and the coolant can undergo phase change (boiling) along the channels. Compared to single-phase heat exchangers, they provide a much higher heat transfer coefficient (HTC) than that of single-phase mode. This technology is crucial for addressing the heat dissipation challenges of future ultra-high heat flux density chips.

The concept of MCHS was first proposed by D. B. Tuckerman et al. in 1981 [1]. They used an etching solution to create rectangular microchannels on one side of a thick silicon wafer and fabricated a thin-film thermistor on the other side for heating. Experimental studies were conducted with water as the cooling medium, and it was found that, under certain pressure losses, the heat dissipation capability exceeded the levels achievable by known cooling methods at the time.

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Subsequently, scholars conducted extensive studies on the flow and heat transfer characteristics of microchannels. M. I. Hasan et al. [2] investigated the effect of different channel cross-sectional shapes (such as circular, square, rectangular, equilateral triangular, and trapezoidal) on performance. The results showed that increasing the number of channels improved the HTC, but it also led to a sharp increase in pressure drop and a deterioration in temperature uniformity. Scholars also researched the flow boiling characteristics of MCHS, such as Zong et al. [3], who found that the phase-change boiling process in MCHS exhibited unstable phenomena. B. Agostini et al. [4] showed that uneven heat distribution in microchannels could cause unstable backflow during the flow boiling heat transfer process. These studies indicate that traditional MCHS suffer from high pressure drops and large temperature differences. To ensure stable system operation, researchers have attempted to address these issues through structural optimization.

2. Working Principles

2.1. Heat Transfer Mechanism

Microchannel heat sinks primarily dissipate heat through forced convective heat transfer. Heat convection is the process of heat transfer caused by the relative displacement of particles in a fluid. Common types of convective heat transfer are natural convection and forced convection. Natural convection occurs due to density differences caused by temperature differences, while forced convection involves using mechanical means to drive fluid flow for heat transfer. When a fluid comes into contact with a solid surface at a different temperature, heat will be transferred between them. This process is called convective heat transfer, which can be expressed as:

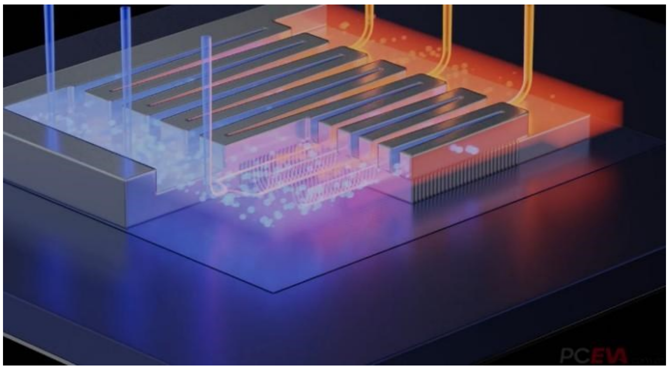

Figure 1: Effect Diagram of Convective Heat Transfer.

$$\varphi = h_{\mathrm{A}}(\mathrm{t_w} - \mathrm{t_f}) \tag{1}$$

where Φ is the convective heat transfer rate (W), h is the convective HTC (W/m²·K), A is the effective heat transfer area of the surface (m²), t_w is the temperature of the solid surface (°C), and tf is the temperature of the coolant. MCHS greatly enhance the heat exchange efficiency between the fluid and solid due to their high specific surface area.

2.2. Flow Characteristics

Flow within the microchannels is usually laminar. Because the channel dimensions are extremely small, the flow characteristics differ from those at macroscopic scales, requiring consideration of micro-scale effects. Laminar flow refers to a flow state in which the fluid flows in layers. When a fluid flows slowly within a pipe, it exhibits laminar flow, with particles moving smoothly in straight lines along the direction parallel to the pipe axis. The fluid velocity is maximum at the center of the pipe and minimum near the walls. The ratio of the average flow velocity to the maximum flow velocity within the pipe is 0.5.

2.3. Trade-off Between Pressure Drop and Thermal Resistance

Fluid pressure drop is an energy loss caused by the internal friction the fluid overcomes during flow and the momentum exchange resulting from particle collisions in turbulent flow. This is reflected in the pressure difference between the inlet and outlet of the fluid flow, known as pressure drop. The magnitude of the pressure drop changes with the flow velocity in the pipe. During the operation of air conditioning systems, the smoothness of the pipes and the connection method, including whether constrictions or throttling occur, also affect the pressure drop. Increasing the flow velocity enhances heat transfer but also increases the pressure drop. Therefore, the design of MCHS requires a trade-off between thermal resistance and pressure drop. In practical engineering applications, the heat dissipation capability of a substrate with embedded MCHS is often assessed by the maximum heat flux q of the substrate and the overall heat dissipation performance coefficient (COP), which is is defined as the ratio of the fluid's heat dissipation power to the pump power required to drive the fluid flow within the microchannels:

$$\mathrm{COP}\frac{P_{\mathrm{extract}}}{P_{\mathrm{pump}}} \tag{2}$$

For a substrate with embedded MCHS, the main factor influencing its heat dissipation power is the thermal resistance R of the substrate:

$$P_{\mathrm{extract}=\frac{T_t-T_b}{R}} \tag{3}$$

where T_t and T_b are the temperatures of the top and bottom surfaces of the substrate, respectively, along the heat transfer path. The pump power is primarily determined by the flow resistance of the microchannels. Microfluidic heat sinks with both low thermal and flow resistances exhibit high overall heat dissipation performance, and both flow and thermal resistances depend on the structure and geometric dimensions of the microchannels [5].

3. Improving the Performance of Microchannel Heat Sinks through Structural Design and Optimization

3.1. Optimization of Geometrical Parameters

The geometrical parameters of MCHS have a decisive impact on their thermal and hydraulic performance. The main geometrical parameters include the cross-sectional shape, size, aspect ratio, and depth-to-width ratio of the channels. The optimization of these parameters is a key issue in the design of MCHS. The rectangular cross-section is a common shape. In terms of convective heat transfer capacity, the higher the aspect ratio of a microchannel, the stronger the axial heat transfer capacity of the working fluid inside the microchannel. Based on rectangular microchannels, when the Reynolds number (Re) of the cooling medium flow is between 100 and 1,000 and the hydraulic diameter is close, the HTC of the rectangular microchannel is highest, while the HTC of the triangular microchannel is the lowest. Studies show that, under the same hydraulic diameter, circular channels typically have the minimum pressure drop, while triangular channels exhibit the highest heat transfer efficiency. Du et al. [5] studied silicon MCHS with trapezoidal cross-sections and found that when the hydraulic diameter of the channels ranges from 51 to 169 μm, the pressure gradient and friction coefficient inside the channels are both higher than the predicted values under classical laminar flow theory. Kim et al. [7] compared the flow distribution performance of triangular and trapezoidal inlet manifolds at low Re (Re = 50–300), and the results showed that triangular inlet manifolds provide

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better flow distribution. Kumaran et al. [8] investigated the flow characteristics of three manifold shapes—rectangular, trapezoidal, and triangular—and recommended using triangular inlet manifolds to control uneven flow distribution.

By establishing a thermal resistance model for rectangular microchannels, studies on the thermal resistance characteristics of rectangular microchannels under different flow conditions show that the aspect ratio of the microchannel is inversely proportional to its overall thermal resistance. Luo et al. [9] proposed that reducing the channel width and fin width could lower the average wall temperature of the heated surface, but the pressure drop at the inlet and outlet increases sharply. Xia et al. [10] suggested that the channel width and fin thickness determine the total number of microchannels that can be fabricated on a limited surface. The smaller the fin thickness, the larger the heat transfer surface area per unit volume. However, when the microchannel width is fixed, thinner fins are not always better, and more channels are not always beneficial. The following are the research results [11].

(Gradually contracted and expanded rectangular cross-sectional MCHS with parallel channel layouts) [11]

1) Reducing the small-to-large end width ratio increases the pressure drop.

2) Microchannels with large-end inlet structures have significantly better thermal performance than those with small-end inlet structures.

3) Staggered channels can improve thermal performance, but this comes at the cost of increased pressure drop.

4) Microchannel heat sinks with counterflow structures perform better than those with parallel flow structures.

(Gradually expanded and contracted rectangular cross-sectional MCHS with parallel channel layouts) [11]

1) Increasing the inlet-to-middle cross-sectional width ratio leads to an increase in the maximum temperature and middle cross-sectional width, a reduction in the pressure drop, and an increase in the overall performance coefficient.

2) Increasing the middle cross-sectional width reduces the maximum temperature and thermal resistance but increases the pressure drop, resulting in a decrease in the overall performance coefficient.

3) Increasing the channel height reduces the maximum temperature and thermal resistance but increases the pressure drop, leading to a decrease in the overall performance coefficient.

(Tree-shaped gradually contracted rectangular cross-sectional MCHS) [11]

1) Increasing the outlet-to-inlet width ratio increases the maximum temperature and total thermal resistance. The pressure drop initially decreases and then increases, while the overall performance coefficient first increases and then decreases.

2) Increasing the sub-to-main pipe width ratio increases the maximum temperature and total thermal resistance. Small outlet-to-inlet width ratio: pressure drop decreases, overall performance coefficient increases. Large outlet-to-inlet width ratio: pressure drop increases, overall performance coefficient decreases.

3) Increasing the length ratio reduces the maximum temperature and total thermal resistance, but the pressure drop increases, leading to a decrease in the overall performance coefficient.

4) Increasing the number of branching layers reduces the maximum temperature and total thermal resistance but increases the pressure drop, resulting in a decrease in the overall performance coefficient.

3.2. Layout Optimization

In addition to the geometric parameters of individual channels, the overall layout of the channels is also an important consideration in the design of MCHS. Layout optimization mainly involves the

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arrangement and distribution density of the channels. The following discussion will address this from two perspectives: topological structure and biomimetic design.

3.2.1. Topological Structure

Numerical simulations have shown that topologically optimized fins can effectively enhance heat transfer. These microchannels can have different extension patterns, including straight, serpentine, and helical configurations. For example, Figure 2 shows a comparison of the performance of a new structure versus a traditional straight fin heat sink [12].

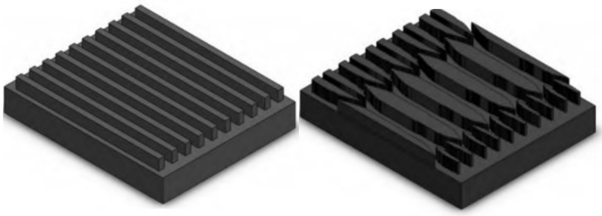

Figure 2: 3D Model of a Straight Fin Structure (Left) and Optimized Structure (Right). (Note: The base dimensions of the straight fin model: 100 mm × 120 mm × 17 mm, fin part: 10 mm × 108 mm × 8 mm).

Comparison of the average temperature of the base of the topological structure heat sink and the straight fin heat sink at different inlet velocities:

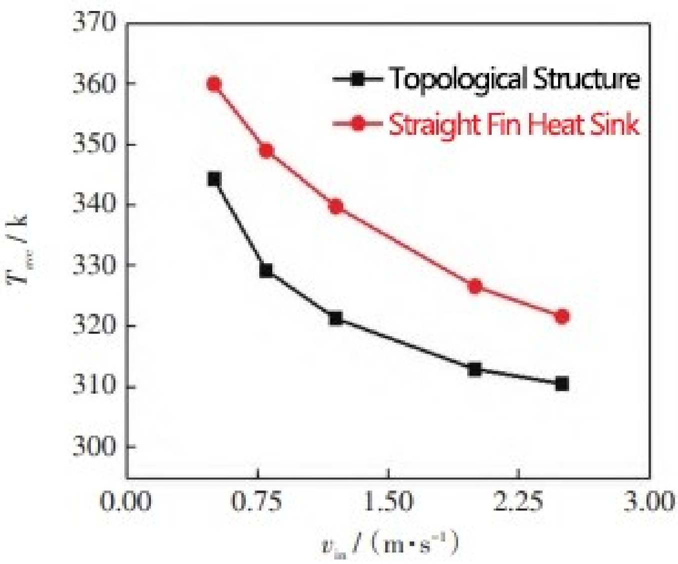

Figure 3: Comparison of the Average Temperature of the Heat Sink Base between the Topological Structure and the Straight Fin Heat Sink (Note: The black lines represent the topology structure, while the red lines represent the straight fin structure).

At an inlet velocity of 1.2 m/s, the temperature and velocity fields of the topological structure heat sink and straight fin heat sink with a channel height of 4 mm are shown below:

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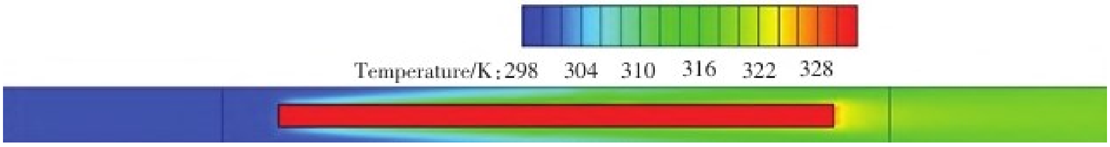

Figure 4: Temperature Contour Map of the Straight Fin Structure.

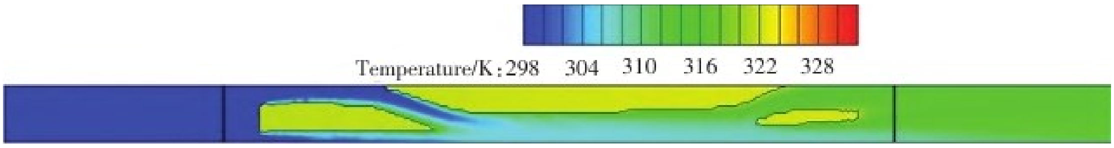

Figure 5: Temperature Contour Map of the Topological Structure.

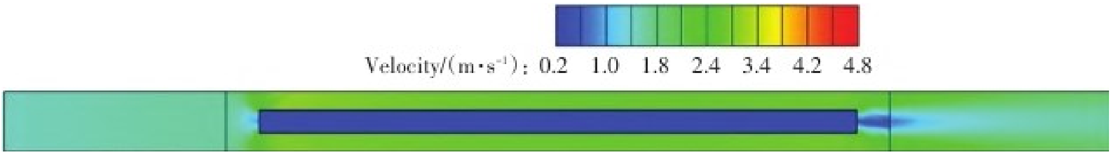

Figure 6: Velocity Contour Map of the Straight Fin Structure.

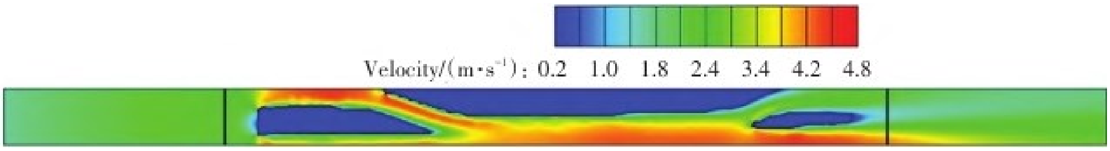

Figure 7: Velocity Contour Map of the Topological Structure.

At an inlet temperature of 293 K, the effect of different inlet velocities on the topological structure heat sink and the straight fin heat sink is shown below:

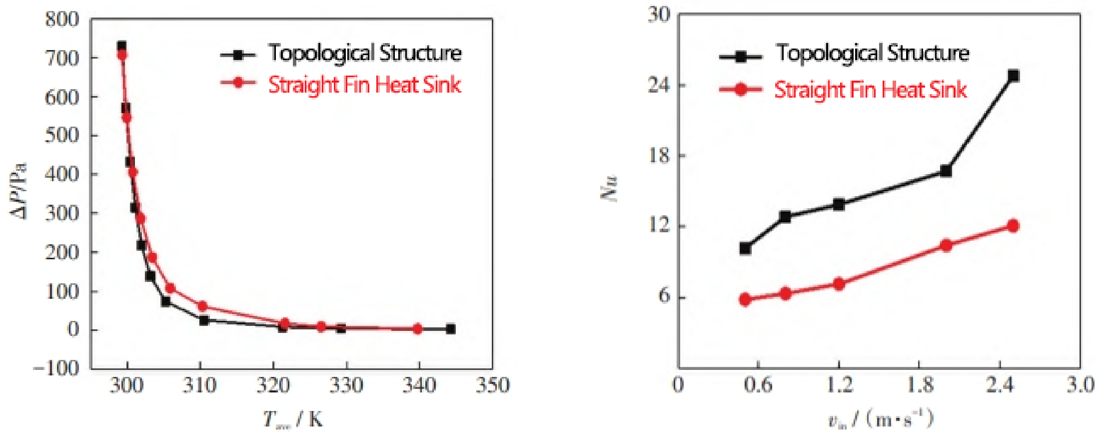

Figure 8: Performance Comparison of Topological Structure and Straight Fin Structure (Note: The black lines represent the topology structure, while the red lines represent the straight fin structure).

The horizontal axis represents the average temperature of the heat sink base, and the vertical axis represents the pressure difference at the inlet and outlet.

The horizontal axis represents the inlet velocity, and the vertical axis represents the Nusselt number. When comparing the heat transfer and pressure drop performance of the optimized topological structure and the straight fin structure with the same volume ratio, it was found that the fins of the topological structure heat sink have localized high-speed regions at both the inlet and outlet, which

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can rapidly carry away heat and facilitate heat dissipation. At an inlet velocity of 1.2 m/s, the average temperature of the topological structure heat sink was reduced by approximately 5.4% compared to the straight fin heat sink, while the Nusselt number increased by approximately 94.3%. This indicates that the topological structure heat sink effectively enhances heat transfer performance.

3.2.2. Biomimetic Design

To enhance the heat dissipation capacity of high heat flux density chips, various biomimetic microchannel topologies inspired by natural networks with excellent mass transfer and heat transfer properties have been designed.

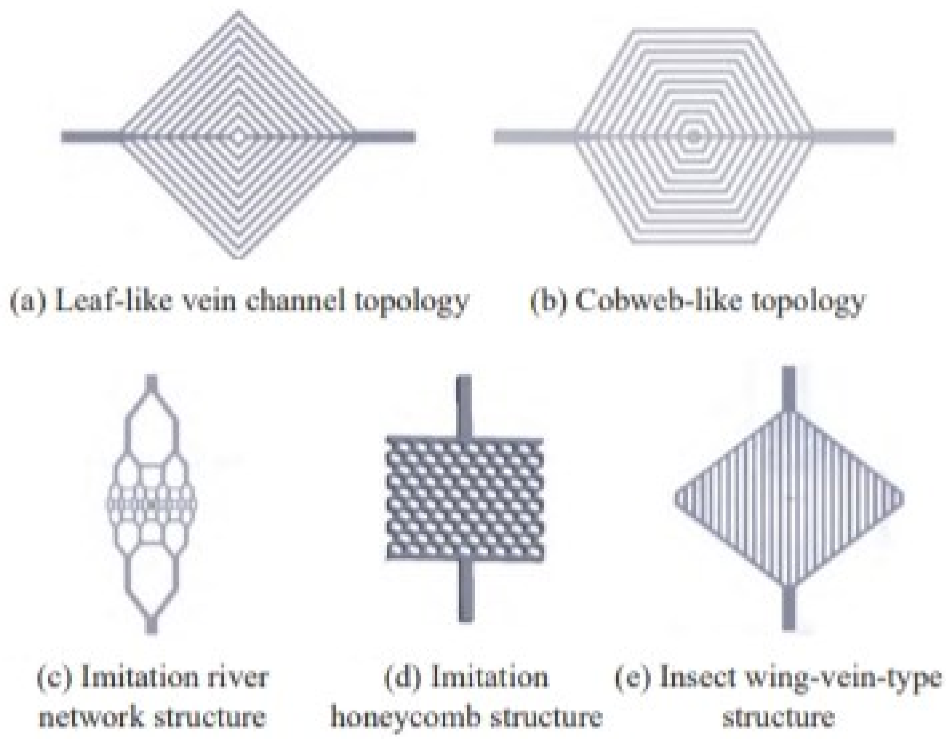

Figure 9: Schematic Diagram of Different Biomimetic Structure Heat Sink Designs.

For example, Figure 11 describes a venation-type MCHS design (based on the typical lateral vein structure of plants, a venation-type microchannel structure is designed) [13]. In the venation-type MCHS, the heat sources on the lateral veins are arranged in a “parallel” configuration. The microchannel length is reduced by approximately 50% compared to parallel MCHS, which is advantageous for microchannel fabrication and prevents clogging.

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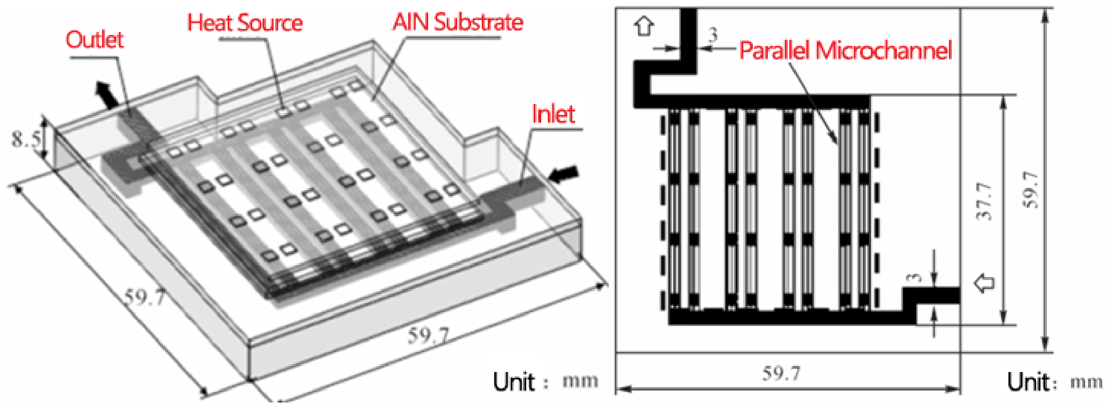

Figure 10: Schematic Diagram of the Parallel Microchannel Heat Sink Structure (Note: In the left diagram, the text from left to right reads as Outlet, Heat Source, AIN Substrate, and Inlet; in the right diagram, the text reads Parallel Microchannel).

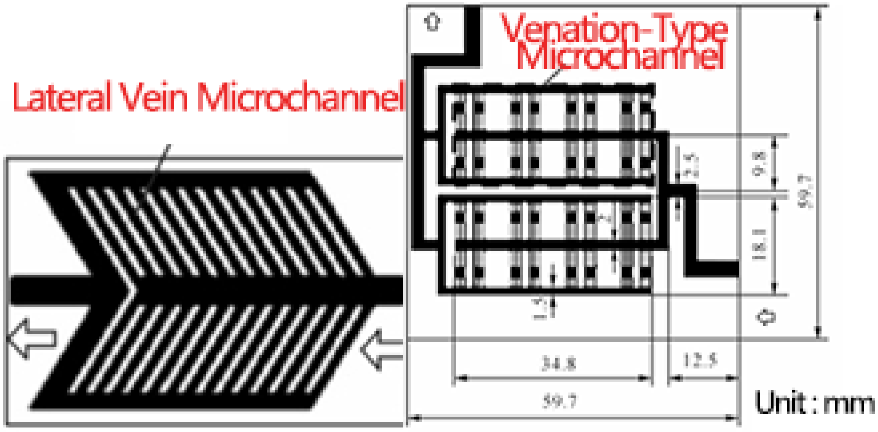

Figure 11: Diagram of Plant Leaf Veins and Venation-Type Microchannels (Note: The text in the left diagram reads Lateral Vein Microchannel, and the text in the right diagram reads Venation-Type Microchannel).

Below is a comparative analysis of the performance of MCHS. The heat sink material and size are the same, the wall surface is smooth, gravity is neglected, viscous dissipation and radiation heat transfer are ignored, and laminar flow is assumed within the microchannels. The fluid is incompressible with constant properties. The cooling medium is HFE7200, with inlet boundary conditions of velocity inlet at a flow rate of 0.4 L/min and a temperature of 50°C. The outlet boundary conditions are pressure outlet with static pressure at the outlet. The heat flux density of all heat sources is 100 W/cm². The grid division method uses a width of 0.05 mm for numerical simulations of the heat sink performance under these conditions, as shown in the table: (Left) Parallel Microchannels, (Right) Venation-Type Microchannels.

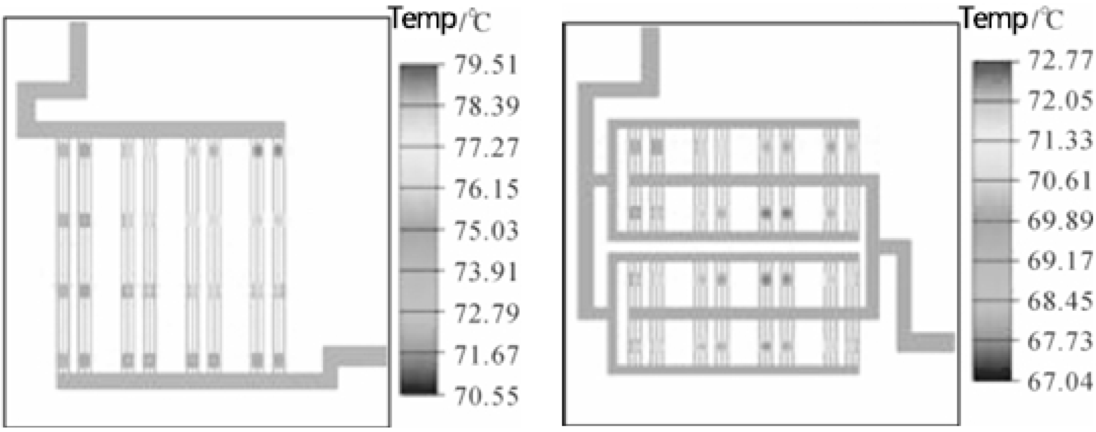

Figure 12: Temperature Distribution Contour Maps for Two Types of MCHS.

Table 1: The data comparison between Parallel Microchannel Heat Sink and Vein-Type Microchannel Heat Sink

Heat Sink Type	Temperature Standard Deviation (°C)	Maximum Temperature (°C)	Pressure Drop (kPa)
Parallel Microchannel Heat Sink	3.43	79.51	6.934
Vein-Type Microchannel Heat Sink	2.76	72.77	6.771

From the data in the table 1, it can be observed that the performance of the vein-type microchannel heat sink is superior to that of the parallel microchannel heat sink. The temperature standard deviation and maximum temperature of the heat source decrease by 19.5% and 8.5%, respectively, while the pressure drop changes by only 2.3%. This indicates that the vein-type microchannel heat sink enhances heat transfer performance, reduces the heat source temperature, and achieves a more uniform temperature distribution without increasing pressure loss.

Additionally, Figure 13 describes experimental research conducted on rectangular and fish-shaped microchannels fabricated through a microchannel performance test platform [14].

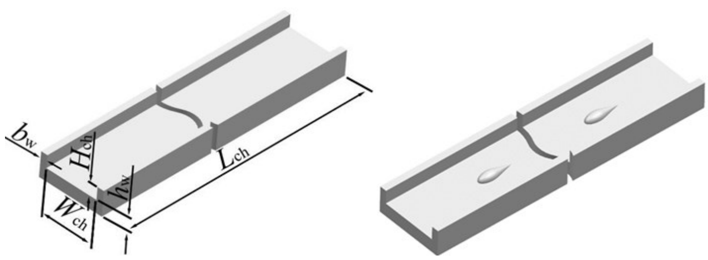

Figure 13: Rectangular Microchannel (Left), Fish-Shaped Microchannel (Right).

Under experimental conditions with a Reynolds number ranging from 400 to 1300, the pressure drop and average friction coefficient of the fish-shaped microchannel were both higher than those of the rectangular microchannel. The increase in resistance loss ranged from approximately 15.98% to 25.99%. The average Nusselt number increased by about 37.33% to 55.59%, the average measured wall temperature decreased by about 2.2 K to 2.8 K, and the maximum wall temperature difference decreased by about 0.8 K to 1.5 K. The introduction of the fish-shaped structure enhanced the

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convective heat transfer of the fluid within the microchannel and improved the temperature distribution characteristics of the microchannel substrate. Within the experimental Reynolds number range, the enhancement effect on convective heat transfer brought by the fish-shaped structure was greater than the increase in resistance loss. The flow entropy production rate of the fish-shaped microchannel was higher than that of the rectangular microchannel, while the irreversible heat loss was smaller. Furthermore, the overall performance factor of the fish-shaped microchannel was greater than 1, while the increase in entropy production was less than 1. The use of the fish-shaped structure helps reduce the total irreversible loss during the flow and heat transfer process of deionized water in the rectangular microchannel.

The optimization design of microchannel structures is usually achieved through numerical simulations. However, numerical simulations of complex structures involve large computational workloads. Therefore, to improve computational speed, multi-objective calculations [15], sorting genetic algorithms [16], and conjugate gradient methods [17] have been developed. For example, [18] used the number of flow channels and the height-to-width ratio of the channels as variables, with the objective of minimizing the overall thermal resistance to reach a stable optimal value using a simplified conjugate gradient method.

4. Conclusions

As integrated circuits and electronic devices advance towards higher performance and miniaturization, the heat generated per unit area has increased sharply. Traditional cooling methods can no longer meet the increasingly stringent cooling demands. Microchannel heat sinks, as an emerging high-efficiency heat dissipation technology, have received widespread attention in recent years. The microchannel heat sink technology, proposed by Tuckerman and Pease in 1981, integrates micron-scale channels into heat dissipation substrates, utilizing the cooling fluid flowing through these small channels to rapidly transfer and dissipate heat. From optimizing the basic dimensions, aspect ratio, and depth-to-width ratio of rectangular microchannels, to changing the cross-sectional shape, and further exploring the integration of biomimicry and topology, microchannel heat sink technology has made significant progress in the past few decades, showing great potential for application. However, further research is still needed in basic theory, design optimization, and manufacturing processes to meet the growing heat dissipation needs. With the strengthening of interdisciplinary collaboration and the integration of new technologies, MCHS are expected to play an important role in a wider range of fields.

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