Proceedings of the 5th International Conference on Materials Chemistry and Environmental Engineering
DOI: 10.54254/2755-2721/2025.23607

The Effect of Cross-Sectional Shape on the Thermo-Hydraulic Performance of Microchannel Heat Sinks

Yuanqi Mai

Southampton Ocean Engineering Joint Institute at HEU, Harbin Engineering University, Harbin, China
1587592803@qq.com

Abstract. Microchannel heat sinks are widely used in thermal management systems due to their compact design and efficient heat transfer. To further enhance performance, this study uses numerical simulations to explore the heat transfer characteristics of rectangular, triangular, trapezoidal, and circular microchannels while maintaining equal cross-sectional areas. At Re = 900, the triangular microchannel demonstrates optimal heat transfer performance (HTP) but the weakest hydraulic performance, with a 5.56% reduction in thermal resistance and a 54.45% increase in pumping power compared to the rectangular microchannel. The circular microchannel exhibits the worst HTP but the best hydraulic performance, with a 12.78% increase in thermal resistance and a 32.04% reduction in pumping power. The trapezoidal microchannel's HTP and hydraulic performance are similar to those of the rectangular one. This paper provides a theoretical basis for optimizing microchannel heat sink designs by evaluating the HTP and hydraulic performance of different cross-sectional shapes. The findings offer insights into the trade-offs between heat transfer efficiency and hydraulic performance, helping guide future design improvements for microchannel heat sinks in thermal management applications.

Keywords: Microchannel heat sink, cross-sectional shape, thermal resistance, pumping power

1. Introduction

With the continuous miniaturization of electronic devices and the increasing demand for high power density, the heat generation of such devices exceeds $10^6 \mathrm{W/m^2}$. Traditional thermal management methods have gradually become inadequate to achieve the heat dissipation performance (HDP) required by modern electronic devices [1]. The concept of the microchannel heat sink (MCHS) was first introduced by Tuckerman and Pease [2] in 1981. Owing to its compact structure, high heat transfer efficiency, minimal size, and ease of integration, the microchannel heat sink has become a critical technology in the field of thermal management, demonstrating significant application potential, particularly in high power density systems [3]. However, despite its notable advantages, two key performance indicators—thermal resistance and pumping power—remain critical bottlenecks that affect its application effectiveness. Excessive thermal resistance can hinder effective heat transfer, while excessive pumping power can reduce the energy efficiency of the system. Consequently, enhancing the comprehensive efficiency of MCHSs by reducing thermal resistance and improving pumping power has become a core focus in current research [4].

© 2025 The Authors. This is an open access article distributed under the terms of the Creative Commons Attribution License 4.0 (https://creativecommons.org/licenses/by/4.0/).

DOI: 10.54254/2755-2721/2025.23607

To enhance the HDP of heat sinks, Liang et al. [5] analyzed the thermal and flow properties of MCHS with different geometric structures. Their study revealed that structures incorporating ribs and secondary channels exhibited the lowest entropy generation and the best overall performance, significantly improving heat dissipation efficiency and temperature uniformity. Zhao et al. [6] discovered that periodically staggered micro-pin fin structures, while sacrificing some pressure drop, greatly enhanced the HTP of MCHS. Yuan et al. [7] examined the heat transfer performance (HTP) of novel single-crystal diamond straight microchannels with symmetric inclination, rotational symmetry inclination, X-shaped, and V-shaped designs. Compared to conventional rectangular straight microchannels, these four novel microchannel designs demonstrated significantly improved HTP. Pan et al. [8] used numerical simulations to analyze the impact of different reentrant cavity structures on fluid flow and HTP in microchannel heat exchangers. They found that reentrant cavities significantly enhanced heat transfer efficiency, reduced pressure drop, and enhanced overall performance. Deng et al. [9] developed pin-fin interconnected reentrant microchannels (PFIRMs), combining the advantages of reentrant cavities and micro-pin fins for enhanced heat transfer, resulting in a notable improvement in the HDP of MCHS. An et al. [10] examined TSV structural effects on MCHS, finding that staggered pin-fin layouts markedly increased convective heat transfer intensity compared to aligned configurations, albeit with higher flow resistance."

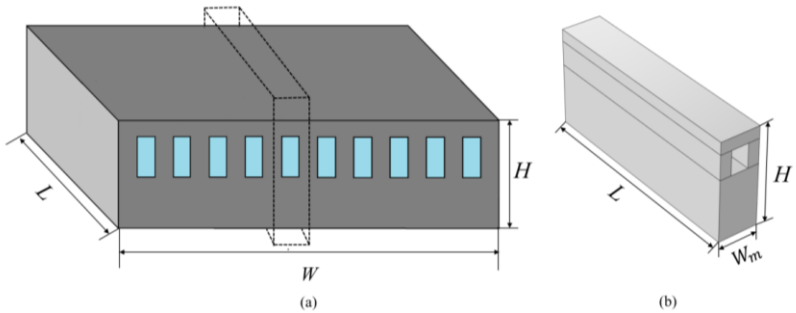
In summary, the geometric configuration of MCHS plays a crucial role in their HTP. To further optimize the HDP of MCHS, this study designs three types of channels—trapezoidal, triangular, and circular—based on rectangular microchannels, ensuring consistent cross-sectional areas. Using numerical simulations, the study investigates how varying cross-sectional geometries influence the HTP of microchannels and explores their potential in comprehensive thermal management performance. The findings of this research will offer theoretical insights to guide the design of MCHS..

2. Model description

2.1. Physical model

The microchannel heat sink (MCHS) measures 20 mm × 9 mm × 2 mm (length × width × height) and consists of a base plate, top plate, microchannels, and side walls. Each microchannel is 20 mm long, 0.9 mm wide, and 2 mm high. The heat sink comprises 10 microchannels in total. Relevant parameters are listed in Table 1 (where Wm, Hb, Hc, Ht, Wc, Wb, Wt, a, R represent the width of a single microchannel, height of the base plate, height of the channel, height of the top plate, width of a rectangular cross-section channel, lower width of a trapezoidal cross-section channel, upper width of a trapezoidal cross-section channel, width of a triangular cross-section channel, and radius of a circular cross-section channel, respectively). To save computational time, a single microchannel is selected for numerical simulation since the MCHS exhibits periodicity. The corresponding model schematic is shown in Figure 1. Based on rectangular cross-section microchannels, three additional designs were developed: trapezoidal, triangular, and circular cross-sections, all with identical cross-sectional areas but different shapes. These structures are illustrated in Figure 2. The HTP of these three microchannels is assessed in this study.

DOI: 10.54254/2755-2721/2025.23607



(a) Overall schematic of the MCHS (b) Schematic of a single MCHS

Figure 1. Schematic of MCHS

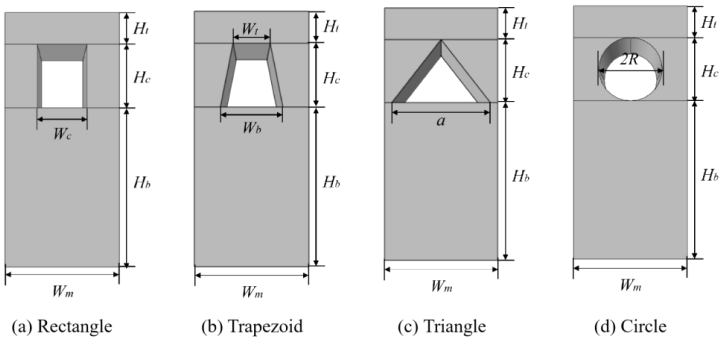


Figure 2. Microchannels with various cross-sections

Table 1. Relevant parameters of a single microchannel (unit: mm)

W_m	H_b	H_c	H_t	W_c	W_b	W_t	a	R
0.900	1.250	0.500	0.250	0.392	0.492	0.292	0.784	0.250

2.2. Mathematical model and calculation methods

Deionized water serves as the cooling medium in the oxygen-free copper MCHS.Their thermodynamic parameters are presented in Table 2. The following assumptions are adopted to simplify the calculations: 1) The cooling medium within the microchannel is a continuous, incompressible, steady-state laminar flow with isotropic properties. 2) Materials maintain consistent physical characteristics regardless of external temperature changes. 3) The effects of viscous dissipation, natural convection, and radiative heat transfer are negligible. 4) The solid-liquid contact surface follows a no-slip boundary condition. The governing equations are solved using the finite element method. Convergence is assumed when the residuals of the equations are below 10^{-4}.

Under the stated assumptions, the control equations for the fluid domain inside the microchannels are:

Continuity Equation:

$$\nabla \cdot \vec{V} = 0 \tag{1}$$

Momentum Conservation Equation:

$$\rho(\vec{V} \cdot \nabla\vec{V}) = \mu\nabla^2\vec{V} - \nabla P \tag{2}$$

Energy Conservation Equation for Fluid:

DOI: 10.54254/2755-2721/2025.23607

$$\rho C_p \vec{V} \cdot \nabla T_f = k_f \nabla^2 T_f \tag{3}$$

Energy Equation for Solid:

$$k_s \nabla^2 T_s = 0 \tag{4}$$

Where $\vec{V}$ is the fluid velocity, P is the pressure in the fluid domain, T_f is the fluid temperature, and ρ, μ, C_p, k_f represent the density, dynamic viscosity, specific heat capacity, and thermal conductivity of the fluid, respectively. k_s denotes the thermal conductivity of the solid material, and T_s represents the wall temperature.

2.3. Boundary conditions

The corresponding boundary conditions are defined as follows:

(1) Inlet velocity: $u_{in} = u,\ v = w = 0,\ T_{in} = 298.15K,\ x = 0\ m$

(2) Outlet pressure: $P = P_{out}, x = L$

(3) Solid-liquid contact surface: $T_f = T_s, -k_f T_f = -k_s T_s$

(4) Heat flux at the bottom of the microchannel: $q_0 = 10^6 W/m^2$

(5) All other external surfaces: Adiabatic (thermal insulation).

Table 2. Thermodynamic parameters of the MCHS and cooling medium

Material	Density ($kg \cdot m^{-3}$)	Thermal Conductivity ($W \cdot m^{-1} \cdot K^{-1}$)	Specific Heat Capacity ($J \cdot kg^{-1} \cdot K^{-1}$)
Copper	8960	400	385
Deionized Water	1000	0.6	4178

2.4. Evaluation metrics

For microchannels with a fixed cross-section, the Reynolds number at the inlet increases with velocity, defined as:

$$Re = \frac{\rho u_{in} D_h}{\mu} \tag{5}$$

where ρ is the fluid density, u_{in} and μ are the inlet fluid velocity and dynamic viscosity, respectively, and D_h is the hydraulic diameter of the MCHS, calculated as:

$$D_h = \frac{4A_c}{P_c} \tag{6}$$

where A_c and P_c represent the cross-sectional area and wetted perimeter of the microchannel, respectively.

Pumping Power:

$$W_{pp} = Q_v \cdot \Delta p = N \cdot A_c \cdot u_{in} \cdot \Delta p \tag{7}$$

where W_{pp} is the pumping power, Q_v is the volumetric flow rate of the fluid, N is the number of channels, and Δp is the total pressure drop of the MCHS.

Thermal Resistance:

$$R_{th} = \frac{T_{max} - T_{in}}{A_b \cdot q_0} \tag{8}$$

In the equation, R_{th} represents the thermal resistance of the microchannel, T_{max} is the maximum temperature of the MCHS, T_{in} is the inlet temperature, and A_b is the area of the heat exchange base.

DOI: 10.54254/2755-2721/2025.23607

3. Results and discussion

3.1. Heat sink performance analysis

Figure 3 illustrates the relationship between the thermal resistance (R_{th}) of heat sinks with various microchannel shapes and the inlet Reynolds number (Re) under equal channel areas. The R_{th} of all types of MCHS decreases as the Re increases. This is because higher Reynolds numbers lead to an increased flow velocity of the coolant within the channels, enhancing the HDP of the heat sink. At the same Reynolds number (Re = 900), the triangular microchannel exhibits the lowest thermal resistance, approximately 5.56% lower than that of the rectangular microchannel. Conversely, the circular microchannel has the highest thermal resistance, about 12.78% higher than that of the rectangular microchannel. The R_{th} of the trapezoidal microchannel is almost equivalent to that of the rectangular microchannel, indicating that the triangular microchannel has the best HDP, while the circular microchannel performs the worst.

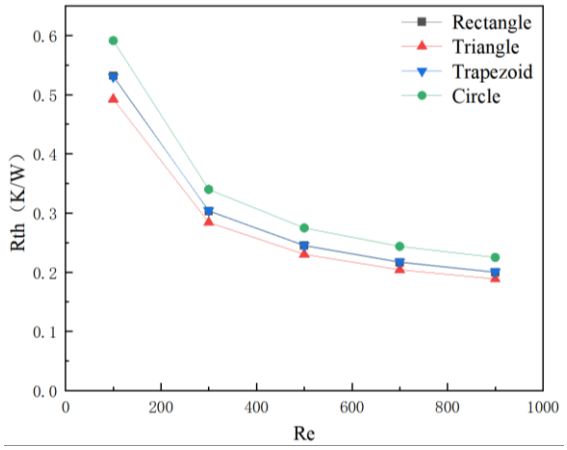


Figure 3. Correlation between thermal resistance and reynolds number in microchannels with various shapes

Pumping power is a critical indicator of energy consumption and hydraulic performance for MCHS. Figure 4 illustrates the relationship between pumping power (W_{pp}) and Reynolds number for heat sinks with different microchannel shapes under equal channel areas. As Re increases, the pumping power of all MCHS rises correspondingly. At Re = 900, the triangular microchannel has the highest pumping power, approximately 54.45% greater than that of the rectangular microchannel. In contrast, the circular microchannel has the lowest pumping power, about 32.04% lower than the rectangular design,while the trapezoidal microchannel's pumping power is nearly identical to that of the rectangular one. These results indicate that the circular microchannel offers the best hydraulic performance, while the triangular microchannel has the worst hydraulic performance.

An inverse correlation between pumping power and thermal resistance is revealed through a comprehensive analysis: higher pumping power corresponds to lower thermal resistance, while lower pumping power results in increased thermal resistance. This trend suggests the need to strike a balance between enhancing heat dissipation performance (reducing thermal resistance) and minimizing pumping power in the design of MCHS.

DOI: 10.54254/2755-2721/2025.23607

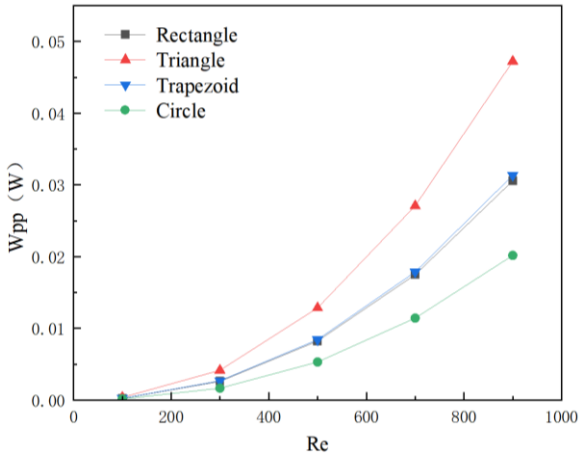


Figure 4. Correlation between pumping power and reynolds number in microchannels with various shapes

3.2. Mechanism behind Heat Transfer Performance

To analyze how channel shapes influence the HTP of heat sinks with equal cross-sectional areas, the temperature distribution at a cross-section (x = 0.45 mm) was examined at Re = 700, as shown in Figure 5. Under a constant heat flux density, heat is conducted from the base of the heat sink to the channel wall and then exchanged with the coolant inside the channel via convection. Microchannels of different cross-sectional shapes exhibit a common feature: the inlet temperature is the lowest, while the outlet temperature is the highest. This phenomenon is attributed to the lower inlet temperature of the coolant, which leads to better cooling at the front region of the microchannel. However, as the coolant flows through the channel, its temperature gradually rises, reducing the temperature difference between the coolant and the channel wall, thereby decreasing the heat transfer capability and resulting in a higher outlet temperature. The temperature field distribution of different MCHS indicates that the circular microchannel reaches the highest temperature of 347 K, while the triangular microchannel achieves the lowest temperature of 339 K. The temperatures of the rectangular and trapezoidal microchannels are comparable at 342 K and 341 K, respectively. These observations can be further explained by analyzing the flow field distribution.

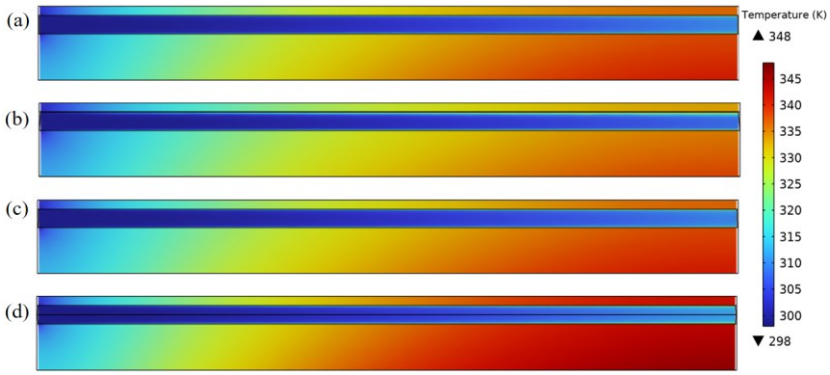


Figure 5. Cross-sectional temperature distribution of different microchannels (Re = 700, x = 0.45 mm):
(a) Rectangular microchannel; (b) Triangular microchannel; (c) Trapezoidal microchannel; (d) Circular microchannel

DOI: 10.54254/2755-2721/2025.23607

Further analysis of the velocity distribution shows that increased velocity directly enhances the convective heat transfer coefficient, thereby improving HDP. As illustrated in Figure 6, the velocity in the triangular microchannel reaches 2.4 m/s, surpassing other designs and contributing to its enhanced heat dissipation and lowest temperature. By contrast, the circular microchannel has the lowest velocity at 1.74 m/s, resulting in the weakest HDP and the highest temperature. The velocities in the rectangular and trapezoidal microchannels are similar, at 2 m/s and 1.97 m/s, respectively, leading to comparable HDP and temperatures. Additionally, among the three microchannel designs with equal cross-sectional areas in this study, the triangular microchannel exhibits the thinnest boundary layer, further enhancing its HTP. The circular microchannel has the thickest boundary layer, leading to the weakest HTP. The boundary layer thickness of the rectangular and trapezoidal microchannels is nearly identical, resulting in similar HTP.

The relationship between temperature and velocity distribution is comprehensively analyzed, showing that MCHS with higher velocities tend to achieve improved HDP, evidenced by reduced temperature distributions. This is because higher flow velocities enhance convective heat transfer between the coolant and the channel walls, reducing the wall temperature and improving heat transfer efficiency.

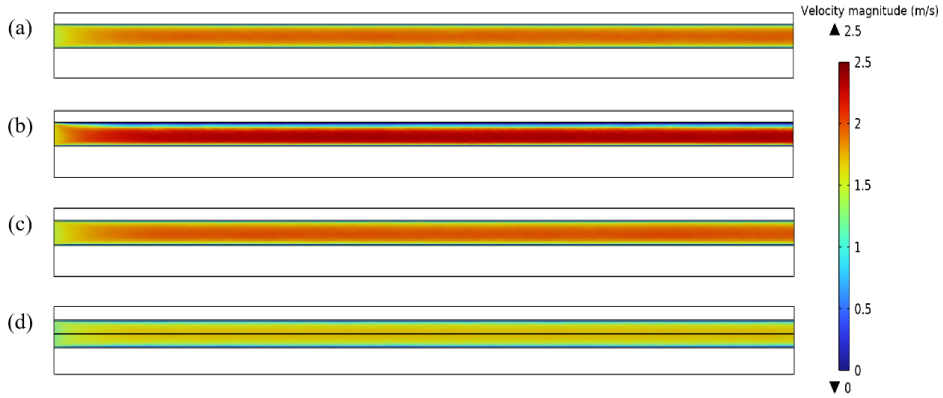


Figure 6. Cross-sectional velocity distribution of different microchannels (Re = 700, x = 0.45 mm): (a) Rectangular microchannel; (b) Triangular microchannel; (c) Trapezoidal microchannel; (d) Circular microchannel

3.3. Mechanism behind pumping power

Figure 7 depicts the pressure distribution along the flow direction in microchannels with various cross-sectional shapes. The results show that the triangular microchannel has the largest pressure drop, reaching 8700 Pa, while the circular microchannel has the smallest pressure drop at only 4760 Pa. The pressure drops of the rectangular and trapezoidal microchannels are nearly identical at 6340 Pa and 6490 Pa, respectively. This is because the streamlined design of the circular microchannel minimizes the flow resistance encountered by the coolant, resulting in the smallest pressure drop. In contrast, the triangular microchannel, lacking a streamlined design, has the highest flow resistance, leading to the largest pressure drop. The similar cross-sectional shapes of the rectangular and trapezoidal microchannels result in comparable flow resistance and pressure drops.

The correlation between pressure distribution and pumping power is thoroughly analyzed, revealing that MCHS with greater pressure drops typically require higher W_{pp} to maintain coolant flow. This increases the energy consumption and reduces the hydraulic efficiency of the heat sink.

DOI: 10.54254/2755-2721/2025.23607

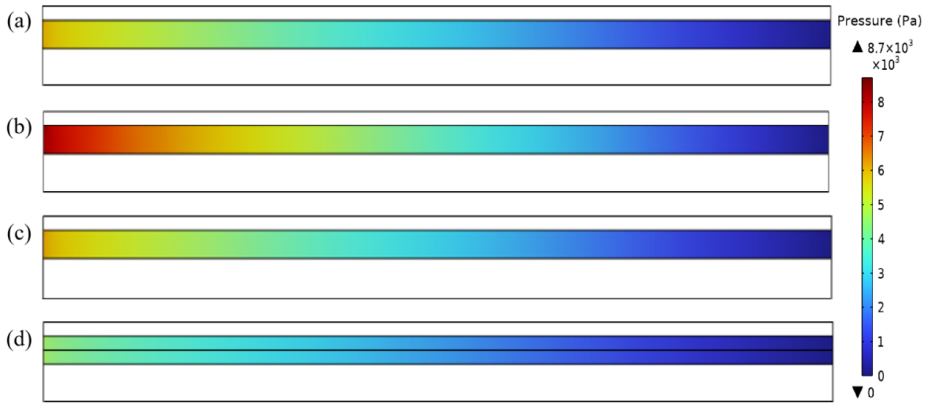


Figure 7. Cross-sectional pressure distribution of different microchannels (Re = 700, x = 0.45 mm): (a) Rectangular microchannel; (b) Triangular microchannel; (c) Trapezoidal microchannel; (d) Circular microchannel

4. Conclusion

This study simulates the HTP of MCHS with different cross-sectional shapes and investigates the flow characteristics and HTP of heat sinks with the same channel area but different cross-sectional geometries. The conclusions are as follows:

(1) Under the condition of equal cross-sectional area, the thermal resistance of heat sinks with different shapes decreases as Re increases. Among them, the triangular microchannel exhibits the lowest thermal resistance (R_{th}), while the circular microchannel has the highest. The R_{th} of rectangular and trapezoidal microchannels is nearly identical.

(2) For the same cross-sectional area, the triangular microchannel demonstrates the best HTP due to its higher flow velocity and the thinnest boundary layer, achieving the lowest temperature of 339 K. The circular microchannel, with the lowest flow velocity (1.74 m/s) and the thickest boundary layer, exhibits the weakest heat transfer capability, resulting in the highest temperature of 347 K. The flow velocities and boundary layer thicknesses of rectangular and trapezoidal microchannels are nearly the same, leading to similar HTP.

(3) Under the condition of equal cross-sectional area, the pumping power of heat sinks with different shapes increases as the Reynolds number rises. The triangular microchannel, with its higher flow resistance, generates the largest pressure drop and thus has the highest pumping power. Conversely, the circular microchannel, benefiting from its streamlined design, has the smallest pressure drop and the lowest pumping power. The rectangular and trapezoidal microchannels exhibit similar pressure drops and pumping power, showing comparable characteristics.

References

[1] Dewan, A., & Srivastava, P. (2015). A review of heat transfer enhancement through flow disruption in a microchannel. Journal of Thermal Science, 24(3), 203–214.
[2] Tuckerman, D. B., & Pease, R. F. W. (1981). High-performance heat sinking for VLSI. IEEE Electron Device Letters, 2(5).
[3] Wang, Q., Tao, J., Cui, Z., Zhang, T., & Chen, G. (2024). Passive enhanced heat transfer, hotspot management and temperature uniformity enhancement of electronic devices by micro heat sinks: A review. International Journal of Heat and Fluid Flow, 107.
[4] Dixit, T., & Ghosh, I. (2015). Review of micro- and mini-channel heat sinks and heat exchangers for single phase fluids. Renewable and Sustainable Energy Reviews, 41, 1298–1311.
[5] Liang, Z., Wen, H., Lv, Q., Su, W., & Wang, C. (2025). Comparative performance analysis of microchannel heat sink with different geometric structures. International Journal of Thermal Sciences, 208.

DOI: 10.54254/2755-2721/2025.23607

[6] Zhao, G., Xiang, L., Luo, Z., Wang, W., & Ni, L. (2023). Heat transfer performance of MCHS with different structural forms. Chemical Engineering, 11, 7-12.
[7] Yuan, J., Qu, Y., Deng, N., Du, L., Hu, W., Zhang, X., Wu, S., & Wang, H. (2023). Three-dimensional numerical investigation of flow and heat transfer performances of novel straight microchannel heat sinks. Diamond & Related Materials, 140.
[8] Pan, M., Wang, H., Zhong, Y., Fang, T. and Zhong, X. (2019), "Numerical simulation of the fluid flow and heat transfer characteristics of microchannel heat exchangers with different reentrant cavities", International Journal of Numerical Methods for Heat & Fluid Flow, Vol. 29 No. 11, pp. 4334-4348.
[9] Deng, D., Chen, L., Wan, W., Fu, T., & Huang, X. (2019). Flow boiling performance in pin fin- interconnected reentrant microchannels heat sink in different operational conditions. Applied Thermal Engineering, 150, 1260–1272.
[10] An, H., Li, J., & Gong, L. (2024). The Impact of TSV Structural Features on the Performance of Microchannel Heat Sinks. Journal of Engineering Thermophysics, 10, 3111-3121.