Journal of Brilliant Engineering 4 (2025) 41053

Brilliant Engineering

journal home: www.acapublishing.com/journals/4/ben

Research Article

Heat Transfer Performance Analysis of Microchannel Heat Sinks with Porous Substrate Integration

Abdüssamed Kabakuş*,[1], Mansur Mustafaoğlu[2,3]

[1] Artvin Çoruh University, Department of Machinery and Metal Technologies, Artvin, Turkey

[2] Mechanical Engineering, Engineering Faculty, Atatürk University, Erzurum, Turkey

[3] Institute of Science, Nano-Science and Nano-Engineering, Atatürk University, Erzurum, Turkey

Keywords

Microchannel heat sink,
Porous fins,
Numerical analysis,
Heat convection.

Abstract

This research focuses on improving the thermal efficiency of microchannel heat sinks (MCHSs) by integrating porous materials into the channel architecture. Numerical simulations were carried out in COMSOL Multiphysics, utilizing the Brinkman–Forchheimer extended Darcy model to comprehensively represent momentum and heat transfer within the porous structure. Various simulations assessed how different porous layer thicknesses influence critical performance metrics such as the heat transfer coefficient, pressure loss, Nusselt number, and Reynolds number. The incorporation of porous media was found to markedly boost thermal performance by expanding the fluid–solid interface area and promoting enhanced convective heat transfer. When compared with traditional MCHS designs, the porous-enhanced configurations showed a 15–30% improvement in both heat transfer coefficient and Nusselt number, though accompanied by an approximately 20% increase in pressure drop. The optimal performance was achieved with a porous layer thickness of 0.1 mm, as evidenced by the peak Figure of Merit (FOM), reflecting the most favorable trade-off between thermal enhancement and hydraulic resistance. These outcomes highlight the potential of porous-augmented microchannels as a compact and efficient thermal management strategy for electronics subjected to high heat fluxes. The study adds to the current body of knowledge by offering detailed insights into design factors affecting the effectiveness of porous microchannel systems.

1. Introduction

Microchannel heat sinks are a cooling technology used to remove a lot of heat from a small area. The heat sink is manufactured from highly thermally conductive solids like silicon or copper, and microchannels are embedded in its surface using precision machining or microfabrication technology. These microchannels increase the surface area of the liquid coolant, allowing for more efficient heat dissipation. This technology allows for more efficient cooling of systems that generate high heat, such as electronic devices. Furthermore, the compact design of microchannel heat sinks allows for smaller, lighter cooling systems. These micro channels, 10-1000 µm in size, serve as flow channels for the coolant. Microchannel heat sinks offer features such as a very high surface area-to-volume ratio, a high convective heat transfer coefficient, small mass and volume, and low coolant retention. Furthermore, this technology increases the operating reliability of electronic devices and minimizes overheating problems. Therefore, microchannel heat sinks are considered a potential solution for improving thermal management in contemporary technological applications. Therefore, these heat sinks are highly suitable for cooling devices such as high-performance microprocessors, radars, high-energy laser mirrors, and laser diode arrays. Microchannel heat sinks also contribute to a more compact design, helping to reduce device size. This, in turn, increases the portability and ease of use of electronic devices. Consequently, microchannel heat sinks are widely used in many industries today. A summary of some of the past and present studies on microchannels is provided in Table 1.

This study investigates the impact of porous materials—whose application in heat transfer systems has attracted growing interest recently—on the thermal performance of microchannel configurations. Through numerical analysis, the pressure drop and heat transfer behavior of a single-phase microchannel heat sink were examined. The findings highlight how variations in flow rates and fluid viscosities influence both pressure loss and heat transfer enhancement. The results offer valuable data to support the design optimization and efficiency improvement of microchannel heat sinks. A comprehensive evaluation of the average heat transfer characteristics is provided and analyzed in detail.

These outcomes contribute novel and fundamental understanding of the intricate three-dimensional thermal phenomena occurring within such heat sinks.

Table 1. MCHS literature summary

Researcher	Research Type	Main focus	Findings
Sasaki and Kishimoto (1986) [1]	Experimental	Optimization of micro-groove cooling fins	They found the optimum channel widths and determined that cooling pressures of 200 kg/m² were sufficient for high-power chips.

*Corresponding Author: akabakus@artvin.edu.tr
Received 04 Aug 2025; Revised 07 Aug 2025; Accepted 07 Aug 2025

https://doi.org/10.36937/ben.2025.41053

Nayak et al. (1987) [2]	Experimental	Advanced Thermal Control Unit for Computer System Packaging	The channel geometries were tailored for both deep and shallow configurations to achieve a high heat transfer coefficient, facilitate fabrication, and enhance the structural integrity of the module.
Phillips (1988) [3]	Experimental	Basic principles of micro-channel coolers	The researcher presented a thermal and fluid performance model of a heat sink and described microchannel fabrication techniques for InP/aluminum.
Samalam (1989) [4]	Numeric	Convective heat transfer in micro-channels	The researcher obtained the optimum dimensions for channel width and spacing analytically.
Knight et al. (1991) [5]	Numeric	Optimization of forced convection coolers	They achieved optimization of fin sizes and spacing
Knight et al. (1992) [6]	Numeric	Heat transfer optimization in micro-channels	The governing equations for fluid flow and coupled conduction–convection heat transfer within the heat sink were expressed in a dimensionless format, covering both laminar and turbulent flow regimes.
Weisberg et al. (1992) [7]	Numeric	Micro-channel analysis for integrated cooling	The authors determined a design algorithm for selecting the dimensions of the heat exchanger.
Bejan and Morega (1993) [8]	Numeric	Pin and plate fin arrangements in laminar flow	The authors identified the advantage of pin fins over plate fins.
Lee and Vafai (1999) [9]	Comparative Analysis	Comparison of jet impingement and micro-channel cooling	The authors determined that jet impingement was more effective than microchannel at high heat fluxes.
Fedorov and Viskanta (2000) [10]	Numeric	Conjugate heat transfer in micro-channel cooler	The study described the intricate heat transfer behavior within the channel, resulting from the interaction of convection and conduction mechanisms in a three-dimensional domain.
Qu and Mudawar (2002) [11]	Experimental & Numeric	Pressure drop and heat transfer in micro-channel cooler	The researchers emphasized that conventional Navier–Stokes and energy equations are sufficient to accurately model fluid dynamics and thermal behavior in microchannel heat sinks.
Kishimoto and Ohsaki (2003) [12]	Experimental	Use of liquid-cooled channels in VLSI packaging	The researchers achieved approximately 10 times more cooling with a microchannel cooler than with a conventional cooler.
Keyes (2005) [13]	Numeric	Heat transfer in forced convection fins	The author studied the effect of fin geometry on heat dissipation.
Sung and Mudawar (2006) [14]	Experimental & Numeric	Hybrid jet-impingement/micro-channel cooling	The researchers achieved lower surface temperatures by reducing the jet width and microchannel height.
Poggi et al. (2009) [15]	Experimental & Numeric	Flow distribution in mini channels	Researchers investigated the effect of microchannel heat transfer in single-phase flow.
Goli et al. (2019) [16]	Numeric	Flow physics in diamond-shaped convergent-divergent) microchannels	The authors determined that diamond channels provided better thermal performance than conventional channels.
Luo et al. (2020) [17]	Numeric	Effect of minichannel arrangements on two-phase flow	The authors determined that mini-channel design has a critical effect on heat transfer and pressure drop.
Loganathan and Gedupudi (2021) [18]	Numeric	Heat transfer in mini/micro-channels with micro-nozzles	The authors stated that mini channels increase heat transfer and balance pressure drop.
Kumar and Singh (2022) [19]	Numeric	Micro-channel heat transfer enhancement (with micro-inserts)	The authors concluded that while micro-inserts enhance heat transfer, they also lead to a higher pressure drop.
Yang et al. (2024) [20]	Experimental & Numeric	Heat transfer with water in a horizontal mini-annular channel	The authors stated that annular channels provide high heat transfer coefficients.

2. Materials and methods

The design and operating conditions were determined for a porous microchannel heat sink. This study investigated the effects of design changes to the size, shape, and configuration of the microchannels on heat transfer. Within porous spaces, the governing equation is the Brinkman equation (Equation 1) with the Forchheimer correction term. This equation accounts for the friction and inertia effects of flow in porous spaces. The Brinkman equation is as follows.

$$-\nabla p = \frac{\mu}{\kappa}\mathrm{u} + \frac{c_F}{\sqrt{\kappa}}\rho\mathrm{u}|\mathrm{u}| \tag{1}$$

The pressure drop depends on the velocity field u, where μ is the fluid viscosity, ρ is the density, and κ is the permeability of the porous substrate.

Determining performance parameters is important to increase the efficiency of the MCHS and ensure more effective operation. Pressure drop is a key factor determining the fluid velocity and pressure within the MCHS. Similarly, the average heat transfer coefficient indicates the heat transfer efficiency of the MCHS, which can help reduce energy losses in the system. Accurately monitoring and optimizing these parameters can improve the performance of the MCHS and increase energy efficiency.

The average heat transfer coefficient of the MCHS (Equation 2) is given by the average wall temperature at the lower centerline, T_w, and the given heat flux, q_{in}. T_b is the average of the fluid inlet and outlet temperatures. Optimizing the cooling system design and fluid properties plays a significant role in increasing the average heat transfer coefficient. This allows the MCHS to maximize energy efficiency and operational performance.

$$h = \frac{q_{in}}{T_w - T_b} \quad (2)$$

The Nusselt number was calculated using the following equation (Equation 3). In this equation, h represents the heat transfer coefficient, L represents the flow path length, and k represents the heat conduction coefficient of the fluid.

$$Nu = \frac{hL}{k} \quad (3)$$

Reynolds Number is expressed as Equation 4. In this equation, the average speed of the fluid U_m, L element of the flow path length, kinematic viscosity of the fluid refers to the ν. Depending on the number obtained from the Reynolds equation with the coefficient of friction is given for the fins.

$$Re = \frac{u_m L}{\nu} \quad (4)$$

In this equation where the Darcy friction factor is calculated (Equation 5), represents the density of the air fluid, ΔP represents the pressure difference between the inlet and outlet, the average speed of the fluid is U_m, L represents the flow path length, and D_h represents the hydraulic diameter of channel.

$$f = \frac{\Delta P}{\frac{L}{D_h} \rho_h \frac{U_m^2}{2}} \quad (5)$$

FoM (Performance Criterion) can be used to examine the heat transfer performance of microchannels. FoM is used to compare porous finned channels with respect to empty channels and is defined as the ratio of the Nusselt number to the friction factor [21]. The FoM equation is shown in Equation 6.

$$FoM = \frac{\frac{Nu_{enhanced}}{Nu}}{\sqrt[3]{\frac{f_{enhanced}}{f}}} \quad (6)$$

The design and operating conditions are shown in Figure 1. The problem was reduced to modeling only half of a single channel. This simplifies the structure by modeling only half of the channel. This simplification allows for easier analysis and analysis of the design and operating conditions.

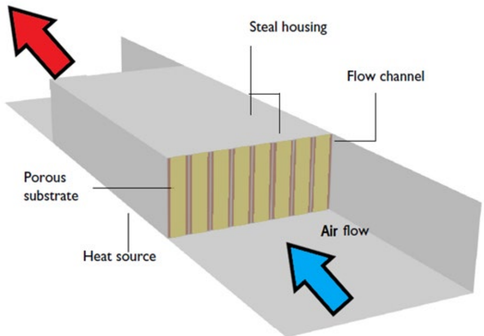

Figure 1. Geometry and operating conditions of porous MCHS.

The geometry of the modeled area is shown in Figure 2. The necessary calculations were made to optimize the design, taking into account the air flow rate and temperature within the modeled area. The geometry shown in Figure 2 was designed to maximize air flow direction and temperature distribution. This aims to improve duct performance and ensure energy efficiency.

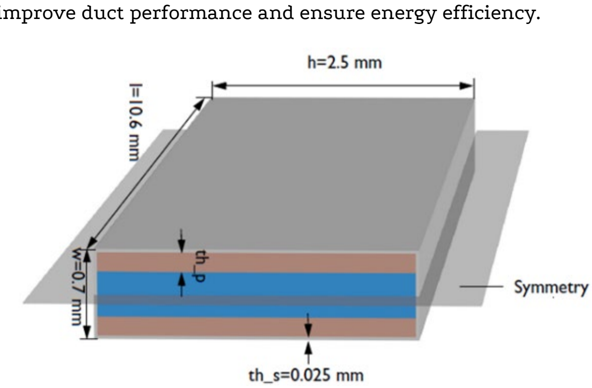

Figure 2. Geometry of the modeling area. (The free-flow domain used to provide the inlet profile for the MCHS is not shown.)

The flow channels are composed of sintered porous metal blocks with a porosity of $\varepsilon = 0.404$ on each side. A constant heat flux of $q_{in} = 100\ W/cm^2$ is applied at the base. The porous structure facilitates enhanced heat transfer within the metal blocks by increasing the surface area and promoting fluid-solid interaction, thereby enabling more effective dissipation of thermal energy from the heat source. Air is used as the cooling fluid, entering the channels with an inlet velocity of u = 0.6 m/s and a temperature of Tin = 300 K. The flow is assumed to be steady, laminar, incompressible, and isothermal, with fluid properties considered independent of the temperature field. Flow and heat transfer in the porous region are governed by the Brinkman equation, which incorporates the Forchheimer correction to account for inertial effects. This model allows for a more accurate representation of momentum transport within the porous structure. Heat conduction within the porous matrix is modeled considering effective thermal conductivity, enabling reliable analysis of temperature distribution and thermal performance. Overall, the use of porous metal blocks improves heat transfer efficiency in microchannel systems, contributing to better thermal management under high heat flux conditions.

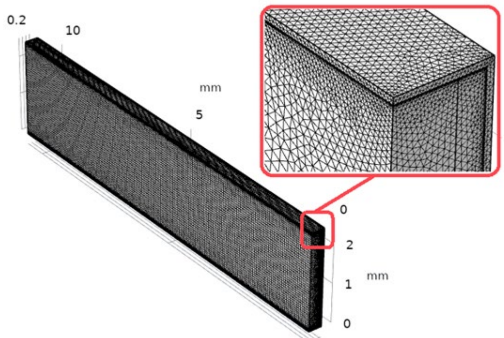

Figure 3. Model geometry and mesh construction

The model geometry presented in Figure 3 represents the solution space and the implemented mesh structure of the porous microchannel heat sink. Due to the symmetric nature of the system, the modeling was simplified to include only half of a channel. This reduces computational costs and allows for the efficient analysis of key performance parameters (pressure drop and heat transfer coefficient). The mesh construction was designed with high resolution and distinct physical regions. Using a denser mesh in the porous media regions allows for more accurate modeling of boundary layer effects and temperature/viscosity gradients. The model aims to precisely calculate pressure drop and heat transfer characteristics by considering physical parameters such as fluid viscosity (μ), density (ρ), and porous substrate permeability (k).

3. Results and discussion

A numerical study was conducted to determine the optimum porous material layer thickness in the Porous Microchannel Heat Sink Performance Model. Once the optimum porous material layer thickness is determined, the performance of the MCHS can be significantly improved. This study is considered a significant step in the design and optimization of microchannel heat sinks. Consequently, these parameter studies aimed at improving the efficiency of the MCHS contribute to the development of innovative heat transfer technologies.

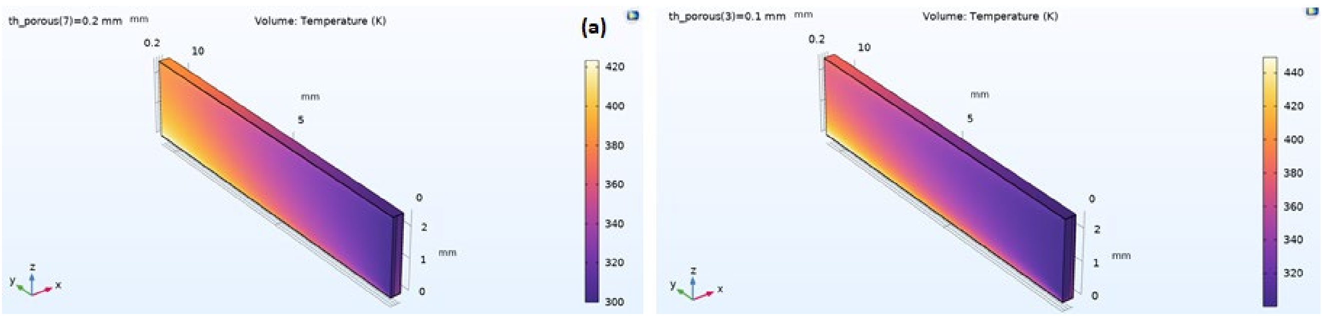

Figure 4. Temperature distribution for different porous structures in the channel cross-section.
a) th_porous=0.2mm b) th_porous=0.1mm

An examination of Figure 4 reveals that the temperature distribution exhibits a thermal load characteristic that increases from the channel inlet to the outlet. It appears that the 0.1 mm thick porous material provides more effective cooling than the 0.2 mm thick porous material. As the porous structure increases in thickness, its surface area increases, which is expected to increase heat transfer. However, increasing thickness also increases pressure drop and reduces fluid velocity.

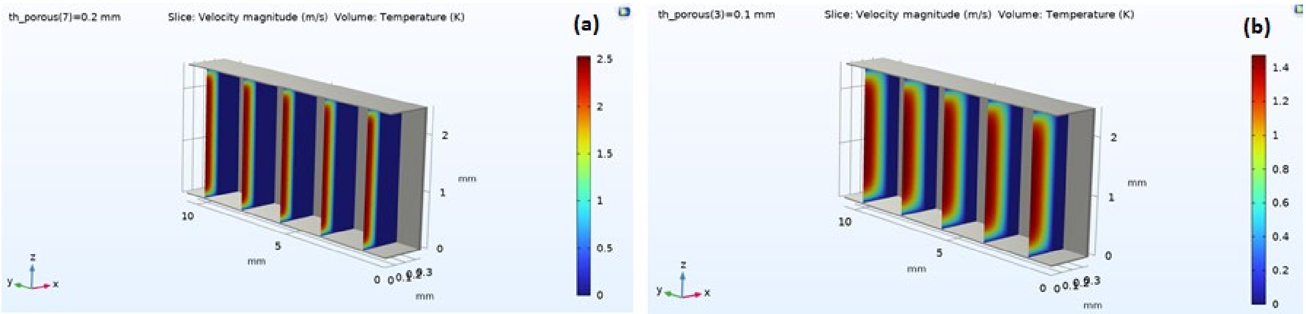

Figure 5. Velocity field for different porous structures in the channel cross-section. a) th_porous = 0.2 mm b) th_porous = 0.1 mm

Figure 5 shows the change in fluid velocity with increasing porous layer thickness (th_p). As the porous layer thickness increases, the pressure drop also tends to increase. Therefore, the fluid velocity affects a narrower region. These results demonstrate that thicker porous structures increase the heat transfer surface area, dissipating heat more effectively, but also increase pumping requirements due to the resistance encountered by the fluid. The images in Figure 5 highlight an important engineering trade-off: higher heat transfer efficiency comes at the expense of greater energy consumption. Therefore, an optimization approach that considers not only heat transfer but also energy efficiency is required in MCHS design. As seen in the images, a thickness of approximately th_p = 0.1 mm represents the ideal design range, offering the best balance between a high heat transfer coefficient and acceptable pressure loss.

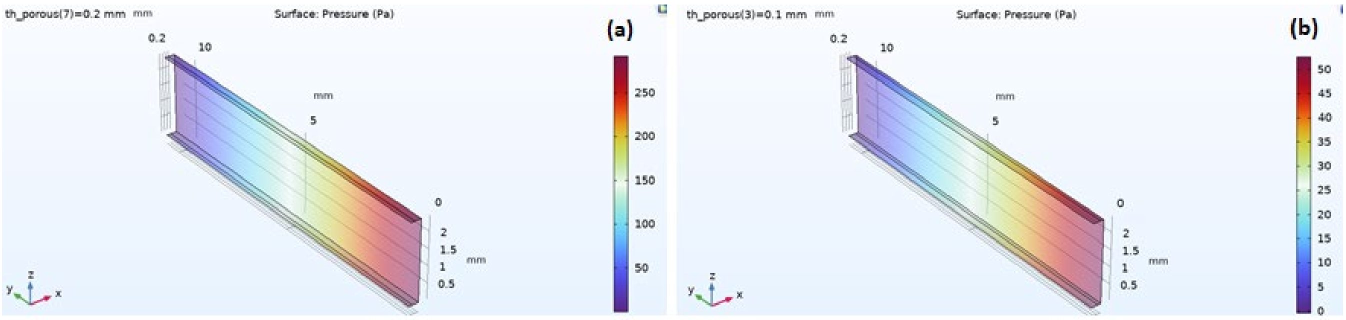

Figure 6. Pressure distribution for different porous structures in the channel cross-section. a) th_porous=0.2mm b) th_porous=0.1mm

Figure 6 presents a comparative comparison of the pressure distribution in the microchannel heat sink (MCHS) for different porous substrate thicknesses (th_p = 0.2 mm and th_p = 0.1 mm). The information obtained from the images clearly reveals that the pressure gradient (∇P) increases as the porous substrate thickness increases. This indicates that the fluid encounters greater resistance as it passes through the porous medium, resulting in a larger pressure drop between the inlet and outlet. In Figure 6(A), the pressure values at the 0.2 mm thickness vary over a wider range, increasing the system's pumping requirements and energy consumption. In contrast, the 0.1 mm thick structure in Figure 6(B) exhibits a more balanced and lower-gradient pressure distribution. This indicates that both pressure loss and hydraulic power requirements in the system are optimized. When numerically evaluated, the Darcy–Brinkman–Forchheimer model used for fluid flow through porous media physically supports these pressure distributions. The intensity of the friction and inertia terms in the model becomes more dominant in thick porous layers, which amplifies the pressure difference. Because the "Figure of Merit" defined in the literature is defined by considering both the average heat transfer coefficient and the pressure drop, these differences observed in Figure 6 directly impact performance. The data confirm that the system operates optimally in terms of both energy efficiency and thermal management for a porous thickness of 0.1 mm.

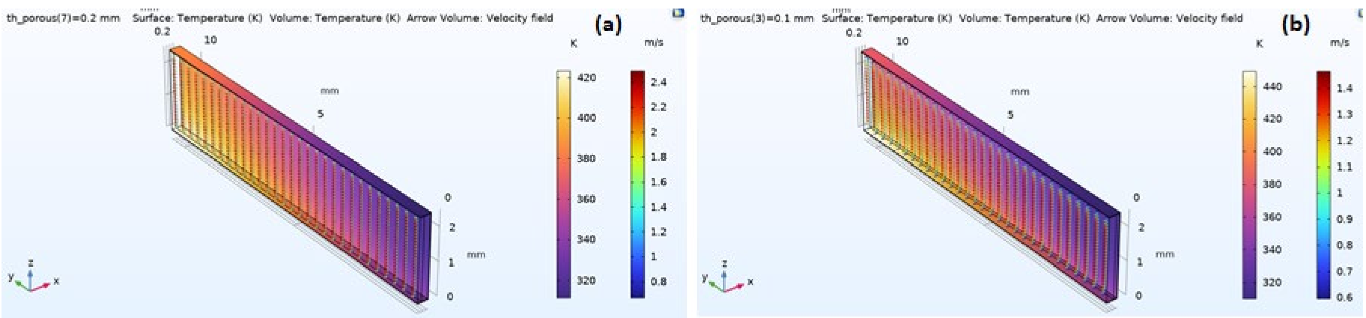

Figure 7. Velocity distribution according to temperature for different porous structures in the channel cross-section. a) th_porous=0.2mm b) th_porous=0.1mm

The image in Figure 7 analyzes the temperature-dependent velocity distribution of a porous microchannel heat sink (MCHS) and compares the system behavior for two different porous substrate thicknesses (th_p = 0.2 mm and 0.1 mm). In the porous structure with a thickness of th_p = 0.2 mm, it is observed that the temperature values spread more homogeneously as the fluid velocity decreases. This indicates that the thicker porous structure spreads the heat over a larger surface area, increasing thermal diffusivity and enabling convective heat transfer. However, the same thickness limits fluid movement within the channel, increasing pressure drop and potential pumping costs. On the other hand, when examining the structure with a thickness of th_p = 0.1 mm, it is observed that the temperature gradient is higher and the velocity vectors exhibit sharper directional changes. This structure provides lower hydrodynamic resistance, allowing the fluid to circulate more freely, thus creating a significant dynamism in the flow regime. However, these temperature gradients also reveal faster heat transfer, albeit with a slight decrease in uniformity. Numerically, comparing the two structures, it was determined that the 0.1 mm thickness offered the best compromise between heat transfer efficiency and pumping requirements.

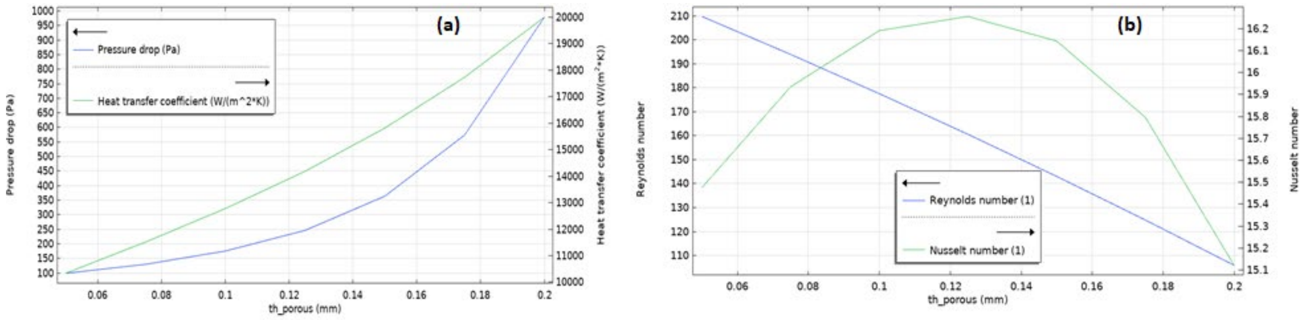

Figure 8. (a)Pressure drop and average heat transfer coefficient change for different porous structures. (b) Variation of Reynolds number and Nusselt number for different porous structures

As shown in Figure 8(a), the graph provides a quantitative analysis of how the thickness of the porous layer affects pressure drop and the mean heat transfer coefficient in porous microchannel heat sinks. An examination of the graph reveals a significant increase in both parameters as the th_p value increases from 0.05 mm to 0.2 mm. Figure 8(b) presents a comparative analysis of the Reynolds (Re) and Nusselt (Nu) numbers calculated for different porous substrate thicknesses. The numerical data reveal that changes in thickness directly affect the Re-Nu relationship. The Reynolds number reflects the effects of fluid inertia and viscosity, while the Nusselt number represents heat transfer efficiency. As seen in the figure, the Nu/Re ratio reaches its highest value at a thickness of th_p = 0.1 mm, demonstrating that this thickness offers the most effective heat transfer-performance balance. Increasing thickness causes a decrease in the Reynolds number. This limits heat transfer despite the increased surface area. This indicates that the flow regime weakens and heat transfer is limited due to the increased hydraulic resistance at higher thicknesses. Consequently, the 0.1 mm porous layer thickness is at the optimum point between low pressure loss and a high heat transfer coefficient, providing an ideal design from both thermal and hydrodynamic perspectives.

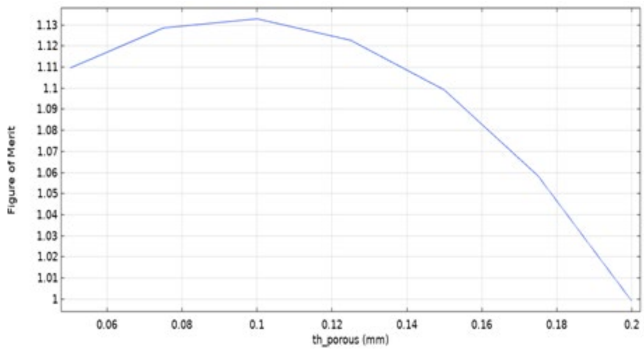

Figure 9. Performance graph comparing the performances of porous and non-porous MCHS.

Figure 9 presents a numerical success table directly comparing the thermal performance of porous and hollow channel heat sink systems (MCHS). This table, based on numerical data, compares key performance parameters such as the heat transfer coefficient (h), pressure drop (ΔP), Nusselt number (Nu), Reynolds number (Re), and, most importantly, the Figure of Merit (FOM). The analysis shows that the MCHS with integrated porous structure has a 15–30% higher heat transfer coefficient and correspondingly higher Nusselt number compared to the conventional design. This can be explained by the increased surface area and more effective flow direction oriented by the porous structure. However, as a result of this advantage, a significant increase in pressure drop in the porous system of approximately 20% occurred. Considering all these parameters, the FOM value, the measure of success, reached its highest level in the porous system, demonstrating that the performance/energy efficiency balance has been optimized.

The results obtained in this study align with and further expand upon the findings reported in previous works, as summarized in Table 1. For example, Qu and Mudawar (2002) demonstrated that classical Navier–Stokes and energy equations can reliably predict the flow and thermal behavior in MCHS configurations. Our results, obtained using the extended Brinkman–Forchheimer model, are in agreement with their experimental values, particularly in terms of pressure drop behavior and laminar flow regime stability. Similarly, the study by Kumar and Singh (2022) showed that adding micro-inserts increased the heat transfer rate but also significantly raised pressure losses—a trade-off also observed in our work, where increasing porous thickness enhanced heat transfer by 15–30% but caused a 20% increase in pressure drop. Yang et al. (2024) emphasized the importance of geometrical optimization in channels with high heat flux applications. Their work on mini-annular channels highlighted that the highest thermal performance is achieved when convective and pressure loss characteristics are balanced—a conclusion that aligns with our identification of 0.1 mm porous thickness as the optimal design based on maximum Figure of Merit (FOM). Additionally, Loganathan and Gedupudi (2021) highlighted the role of micro-nozzle structures in improving heat transfer in mini-channels, noting the importance of managing flow resistance. This matches the engineering trade-off discussed in our work, where thinner porous layers enabled better energy efficiency. These comparisons validate the accuracy of our model and reinforce the relevance of porous layer optimization in microchannel cooling design.

4. Conclusion

This study numerically investigated the effects of integrating porous substrates into microchannel heat sinks (MCHS) on thermal performance, with a particular focus on varying porous layer thicknesses (th_p). The results revealed that porous structures enhance the average heat transfer coefficient by increasing the effective surface area. Among the tested configurations, the structure with a porous thickness of 0.1 mm demonstrated the best balance between thermal performance and pressure drop, yielding the highest Figure of Merit (FOM). Therefore, this configuration can be considered optimal under the tested conditions. Additionally, analysis of flow patterns and temperature distributions showed that thinner porous layers lead to lower flow resistance and more dynamic velocity fields, contributing to improved cooling performance. However, the observed thermal benefits must be weighed against the accompanying hydraulic penalties such as increased pressure drop. It is important to note that the current study is limited to steady, laminar, single-phase air flow under fixed boundary conditions. The influence of varying flow rates, fluid types, or transient effects was not addressed. Therefore, future studies could explore the behavior of porous microchannel structures under turbulent or two-phase flow regimes, as well as investigate alternative geometries and materials to further refine the design space. This study provides valuable insights for designing compact and efficient thermal management systems, particularly for high heat flux electronic applications. However, further experimental validation is recommended to fully assess the applicability of the proposed configurations in practical systems.

Declaration of Conflict of Interests

The authors declare that there is no conflict of interest. They have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

References

[1.] Sasaki, S., Kishimoto, T., Optimal structure for microgrooved cooling fin for high-power LSI devices. Electronics Letters 22(25) (1986) 1332–1334.

[2.] Nayak, D., Hwang, L., Turlik, I., Reisman, A., A high-performance thermal module for computer packaging. Journal of Electronic Materials 16(5) (1987) 357–364.

[3.] Phillips, R.J., Microchannel heat sinks. The Lincoln Laboratory Journal 1(1) (1988) 31–48.

[4.] Samalam, V.K., Convective heat transfer in microchannels. Journal of Electronic Materials 18(5) (1989) 611–617.

[5.] Knight, R.W., Goodling, J.S., Hall, D.J., Optimal thermal design of forced convection heat sinks-analytical. Journal of Electronic Packaging 113(3) (1991) 313–321.

[6.] Knight, R.W., Hall, D.J., Goodling, J.S., Jaeger, R.C., Heat sink optimization with application to microchannels. IEEE Transactions on Components, Hybrids, and Manufacturing Technology 15(5) (1992) 832–842.

[7.] Weisberg, A., Bau, H.H., Zemel, J.N., Analysis of microchannels for integrated cooling. International Journal of Heat and Mass Transfer 35(10) (1992) 2465–2474.

[8.] Bejan, A., Morega, A.M., Optimal arrays of pin fins and plate fins in laminar forced convection. ASME Journal of Heat and Mass Transfer 115(1) (1993) 75–81.

[9.] Lee, D.Y., Vafai, K., Comparative analysis of jet impingement and microchannel cooling for high heat flux applications. International Journal of Heat and Mass Transfer 42(9) (1999) 1555–1568.

[10.] Fedorov, A.G., Viskanta, R., Three-dimensional conjugate heat transfer in the microchannel heat sink for electronic packaging. International Journal of Heat and Mass Transfer 43(3) (2000) 399–415.

[11.] Qu, W., Mudawar, I., Experimental and numerical study of pressure drop and heat transfer in a single-phase micro-channel heat sink. International Journal of Heat and Mass Transfer 45(12) (2002) 2549–2565.

[12.] Kishimoto, T., Ohsaki, T., VLSI packaging technique using liquid-cooled channels. IEEE Transactions on Components, Hybrids, and Manufacturing Technology 9(4) (2003) 328–335.

[13.] Keyes, R.W., Heat transfer in forced convection through fins. IEEE Transactions on Electron Devices 31(9) (2005) 1218–1221.

[14.] Sung, M.K., Mudawar, I., Experimental and numerical investigation of single-phase heat transfer using a hybrid jet-impingement/micro-channel cooling scheme. International Journal of Heat and Mass Transfer 49(3) (2006) 682–694.

[15.] Poggi, F., Bontemps, A., Macchi-Tejeda, H., Marechal, A., Leducq, D., Experimental and numerical study of the distribution of a single-phase flow in a small channel heat exchanger. IIR 1st Workshop on Refrigerant Charge Reduction (2009).

[16.] Goli, S., Saha, S.K., Agrawal, A., Three-dimensional numerical study of flow physics of single-phase laminar flow through diamond (diverging–converging) microchannel. SN Applied Sciences 1(11) (2019) 1353.

[17.] Luo, Y., Zhang, J., Li, W., A comparative numerical study on two-phase boiling fluid flow and heat transfer in the microchannel heat sink with different manifold arrangements. International Journal of Heat and Mass Transfer 156 (2020) 119864.

[18.] Loganathan, R., Gedupudi, S., Numerical study of single-phase heat transfer performance of a mini/micro-channel integrated with multiple bypass micro-nozzles. Journal of Thermal Science and Engineering Applications 13(4) (2021) 041029.

[19.] Kumar, S.R., Singh, S., Numerical analysis for augmentation of thermal performance of single-phase flow in microchannel heat sink of different sizes with or without micro-inserts. Fluids 7(5) (2022) 149.

[20.] Yang, Y., Yuan, X., Li, Z., Experimental and numerical study on heat transfer performance of single-phase water in a horizontal mini-annular channel. International Journal of Heat and Fluid Flow 108 (2024) 109483.

[21.] Bahiraei, M., Mazaheri, N., Hassanzamani, S.M., Efficacy of a new graphene–platinum nanofluid in tubes fitted with single and twin twisted tapes regarding counter and co-swirling flows for efficient use of energy. International Journal of Mechanical Sciences 150 (2024).

How to Cite This Article

Kabakuş, A., Mustafaoğlu, M., Heat Transfer Performance Analysis of Microchannel Heat Sinks with Porous Substrate Integration, Brilliant Engineering, 4(2025), 41053.
https://doi.org/10.36937/ben.2025.41053