Optimization of Micro-Pin-Fin Heat Sinks: Influence of Fin Geometry and Slot Size on Thermal Performance

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Abstract:

Micro-pin-fin heat sinks have become an essential solution for effective thermal management in modern electronic devices, particularly microprocessors, which generate substantial heat during high-speed operations. The low thermal conductivity of gases poses challenges for effective heat dissipation, requiring the development of heat sinks with improved heat transfer capabilities. This research examines how variations in slot sizes and fin geometries influence the thermal and fluid flow performance of micro-pin-fin heat sinks. Various fin shapes—triangular, rectangular, elliptical, hexagonal, and square—were examined with different slot sizes (ranging from 0.1 mm to 0.5 mm) at varying Reynolds numbers (500, 1000, 5000, and 10,000). Results show that elliptical fins outperform other configurations in terms of heat transfer efficiency, with Nusselt numbers (Nu) reaching as high as 112 at a Reynolds number of 5000 and a slot width of 0.5 mm, and a performance evaluation criteria (PEC) ranging from 2.7 to 3.15. In contrast, triangular fins exhibited the lowest Nu values, ranging from 15.8 to 16.8 at Re = 500. These results emphasize the crucial role of fin geometry and slot width in thermal management, providing key insights for optimizing heat sink designs to enhance cooling efficiency in electronic systems.

Introduction

Micro-pin-fin heat[1] sinks have emerged as a crucial innovation for managing the thermal challenges faced by modern electronic devices. Rapid advancements in electronic and information technologies have enabled computers and other devices to process vast amounts of data at exceptional speeds. However, this progress has also resulted in significant heat generation, particularly in the microprocessor[2] (CPU). Effective thermal management is crucial for maintaining the reliability and extending the lifespan of these systems.

The low thermal conductivity of gases, with heat transfer coefficients approximately 100 times lower than liquids, adds complexity to heat dissipation in electronic systems. Consequently, specialized designs incorporating highly efficient enhancement elements are indispensable. Micro-pin-fin heat sinks[3], known for their compact size[4], high heat transfer efficiency[5], and large surface area per unit volume, are among the most effective solutions[6] for managing heat in microelectronic chips.

These heat sinks play a pivotal role in cooling processors, maintaining optimal operating temperatures, and preventing issues such as component deterioration, increased resistance, insulation breakdown, and potential circuit failure. By mitigating these risks, micro-pin-fin heat sinks improve the reliability and performance of electronic systems, ultimately prolonging their service life.

Furthermore, the integration of micro-pin-fin arrays in heat sinks facilitates effective heat dissipation in constrained spaces, like those in computer processors [7] and motherboards. Incorporating fins into channel covers significantly increases the heat transfer rate, effectively managing localized hotspots[8] and improving overall thermal performance[9]. These advancements ensure the stability[10] of electronic circuits and contribute to the structural integrity of computing systems, which is critical for maintaining service continuity and minimizing disruptions.

This study examines the potential of micro-pin-fin heat sinks to optimize heat dissipation in electronic devices. By analyzing their hydrodynamic and thermal characteristics, this research aims to enhance cooling efficiency, supporting the development of more reliable and high-performance electronic systems.

2. Methodology

2.1. Geometric details

The geometry was modelled in 3D using Autodesk Fusion software. The model geometries[11] and their dimensions are shown in Figure 1. The microchannel's upper surfaces and outer walls, measuring (13 × 21), were assumed to be adiabatic. The heat sink base (13×11), positioned beneath the pin fins, experienced a constant heat flux of 2 kW/m². The air, acting as the refrigerant, enters through the inlet, flows over the pin fins, and exits through the outlet. Micro pins, designed with slots of varying sizes to enhance performance, are detailed in Figure 1. The flow was analyzed within a Reynolds number range of 500 to 10,000. Thermal performance was evaluated for a finless model and five distinct fin designs, as illustrated in Figure 1. The analysis also considered various Reynolds number values. These models were referred to as circular-finned, square-finned, triangular-finned, elliptical-finned, and hexagonal-finned. The fins are made of aluminum, exposing their base to the cooling channel. A steady-state 3D turbulence heat transfer problem was addressed to assess the flow and thermal performance of the fins with different meshing techniques.

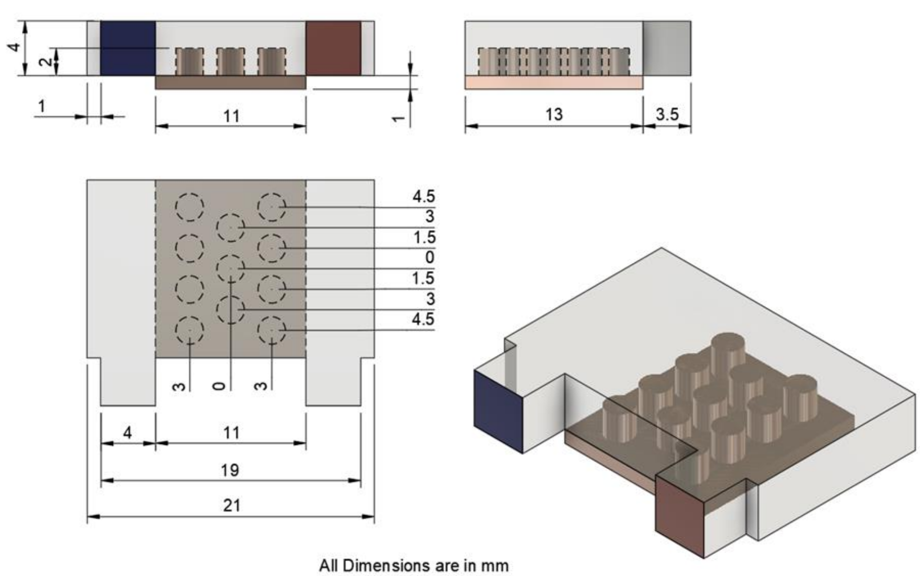

Figure 1: Details of the Analysis Domain

This study thoroughly investigates the impact of slot sizes on the thermal performance and fluid flow characteristics of fin arrays with various fin shapes[12]. The figure 2 shows the line diagram of different types fin geometries with slots. The results offer valuable insights into optimizing fin design for efficient heat dissipation while maintaining a compact structure. Through experiments and analysis, it offers a detailed understanding of how different slot sizes[13] influence thermal management and flow behavior in fin assemblies.

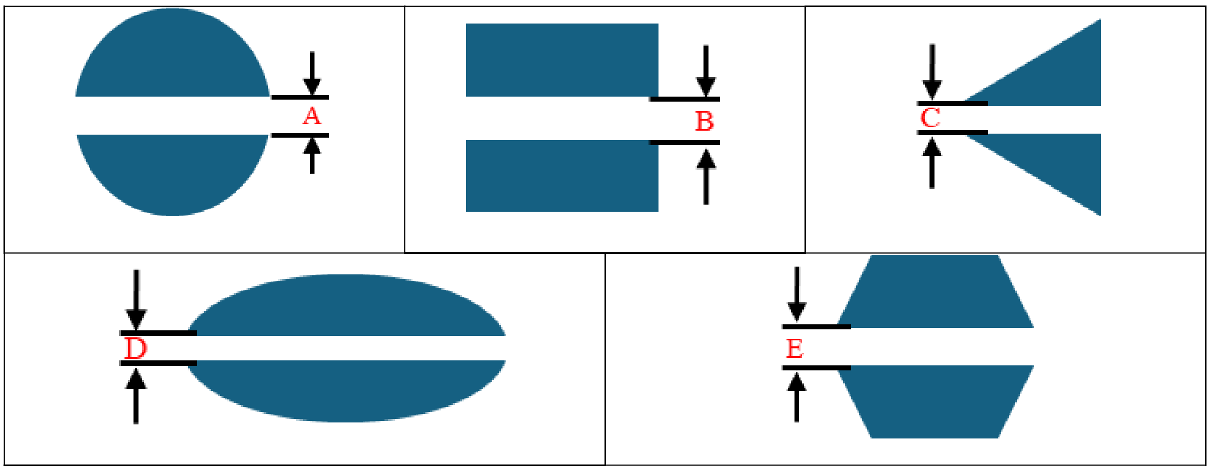

Figure 2: Different types of Fins shape (A) Circular (B) Rectangular (C) Triangle (D) Elliptical (E) Hexagonal with slots

The Table 1 presents the different types of fins and their corresponding slot sizes used in the study of heat transfer performance through CFD techniques. techniques. Example A1 is related to circular fin with no slot and as its moves A2, A3, A4, A5, and A6 are the circular fins with 0.1mm, 0.2mm, 0.3mm, 0.4mm, and 0.5mm slots. Similarly, B, C, D, and E are the series of Square, Triangle, Elliptical, and Hexagon fins.

Table 1: Designation of Fin type with Slot Size

Fin Type	Without Slot	0.1mm Slot	0.2mm Slot	0.3mm Slot	0.4mm Slot	0.5mm Slot
Circular	A1	A2	A3	A4	A5	A6
Square	B1	B2	B3	B4	B5	B6
Triangle	C1	C2	C3	C4	C5	C6
Elliptical	D1	D2	D3	D4	D5	D6
Hexagon	E1	E2	E3	E4	E5	E6

2.2. Boundary conditions

"Heat generated by a CPU[14] is distributed through a process that involves conduction, convection, and sometimes radiation. The amount of heat generated by a processor (CPU) depends on several factors, such as its power consumption, design, workload, and cooling solution. CPU heat dissipation occurs primarily through conduction, convection, and occasionally radiation. CPU cooling methods encompass air cooling (using heatsinks and fans), liquid cooling (employing water or coolant-based systems), passive cooling (through heat spreaders and natural airflow), and advanced techniques like thermoelectric or phase-change cooling for high-performance applications. "As a result, the fins are not directly attached to the CPU but are integrated into the heat sink base, which comes into direct contact with the CPU's Integrated Heat Spreader (IHS). Figure-1 clearly illustrates this situation and the corresponding boundary conditions.

All micro-pin–fin channel flow simulations were conducted using ANSYS Fluent, a computational fluid dynamics (CFD) software. The realizable k−ε turbulence model[15], as defined in the mathematical expressions, was used for the simulations. The regions close to the boundary walls and fin surfaces were meshed with high density to guarantee precise results. Additionally, a 20-layer inflation was applied to the fin surfaces, where heat transfer is most effective, as well as to the contact surfaces between the fins and the heat sink. In the solution settings, the "Second Order Upwind" scheme was utilized for momentum discretization, while the "Simple" scheme was employed for the numerical method. A no-slip boundary condition was applied by setting the fluid velocity to zero at all solid surfaces, with a turbulence intensity of 0.5%. All heat sink surfaces were considered adiabatic, except for the bottom surface in contact with the fins.**2.3. Meshing details**

The meshing process utilized polyhedral meshing techniques[16] in Ansys Fluent, known for enhancing the accuracy and efficiency of computational simulations. Five different mesh sizes were generated: 105,056, 125,586, 175,256, 205,286, 221,753, and 252,357. The Nusselt number and friction factor were calculated across the full range of meshes. Figure 3 depicts the variation in the Nusselt number and friction factor for all mesh sizes at a Reynolds number of 5000. A mesh size of 221,753 was ultimately chosen for the simulations, as the variation in results was less than 2%.

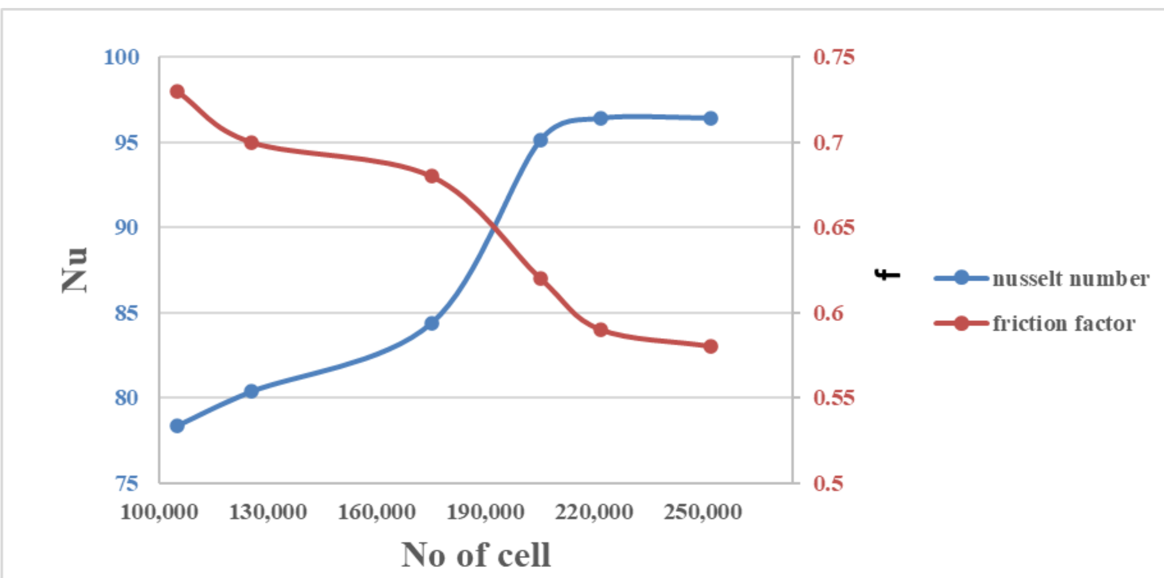

Figure 3: Grid Analysis test for Fins

2.4. Numerical scheme and boundary conditions

Simulations were conducted using ANSYS Fluent's pressure-based solver to address the steady-state 3D heat transfer flow problem. The Realizable K-Epsilon turbulence model was employed to accurately capture complex fluid interactions, while the SIMPLE algorithm[17] facilitated pressure-velocity coupling for convergence. Discretization was performed using the Second Order Upwind scheme to ensure precise representation of flow and thermal fields.

Boundary conditions included a velocity inlet defined by the Reynolds number and a pressure outlet to control exit pressure distribution. Adiabatic walls with no-slip conditions were applied to model heat transfer and fluid-solid interactions. A heat flux of 2000 W/m² was applied to the base of the fins, and the operating pressure was set to atmospheric. Air properties were defined with a density of 1.177 kg/m³, a viscosity of 1.85×10^{-5} Pa·s, and a thermal conductivity of 0.02624 W/m·K.

The simulation modelled the governing equations for continuity, momentum, energy, and turbulence, as detailed in equations (1) to (5).

$$Continuity\ equation\ \nabla.(\overrightarrow{\rho v}) = 0 \tag{1}$$

$$Momentum\ equation\ \overrightarrow{\rho v}.\ \nabla\vec{v} = \mu\nabla^2\vec{v} - \Delta P \tag{2}$$

$$Energy\ equation\ \rho C_p \nabla.(\vec{v}\, T) = \nabla.(k\, \nabla T) \tag{3}$$

$$Turbulence\ Kinetic\ Energy \frac{\partial}{\partial x_j}(\rho k u_j) = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right] + G_k + G_b - \rho_e + Y_M \tag{4}$$

$$Rate\ of\ Dissipation\ \frac{\partial}{\partial x_j}(\rho \varepsilon u_j) = \frac{\partial}{\partial x_j}\left[\left(\left(\mu + \frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_j}\right)\right] - \rho C_2 \frac{\varepsilon^2}{k+\sqrt{v\varepsilon}} + C_{1\varepsilon}\frac{\varepsilon}{k}C_{3\varepsilon}G_b \tag{5}$$

2.5. Calculation Procedure

The Reynolds number (Re) for a flow regime is calculated using Equation (6), which depends on the fluid density (ρ), inlet velocity (ui), hydraulic diameter (Dh), and fluid viscosity (μ).

$$D_h = \frac{4.A_{in}}{P_{in}} \tag{6}$$

The hydraulic diameter (D_h) is a defining length used to characterize the flow geometry in ducts or channels. It is determined by taking four times the cross-sectional area (A_{in}) and dividing it by the wetted perimeter (P_{in}) of the flow. For circular pipes, the hydraulic diameter is equivalent to the pipe's diameter. However, for non-circular geometries, it plays a key role in describing the flow and is crucial for calculating the Reynolds number and other fluid dynamic properties.

The heat transfer coefficient, “h” quantifies the rate of heat transfer between a solid surface and a fluid. The wall heat flux (q_W)is derived from the difference in energy flow rates between the inflowing and outflowing fluids, The bulk fluid temperature (T_b) is determined as the arithmetic average of the inlet (T_i) and outlet (T_o) fluid temperatures. The base wall temperature (Tw) refers to the temperature of the surface where heat is transferred between the solid and the fluid.

$$h = \frac{q_w}{T_w - T_b} \tag{7}$$

The Nusselt number (Nu) represents the heat transfer rate within the fluid. It is calculated using the heat flux applied at the base of the fins (q), the thermal conductivity of air (kair), the base wall temperature of the fin (Tw), and the inlet (Tin) and outlet (Tout) temperatures of the air.

The Nusselt number (Nu) describes the heat transfer rate along the fluid flow. It is defined in Equation (7), where h is the heat transfer coefficient, Dh is the hydraulic diameter, and kair is the thermal conductivity of air. Additionally, q denotes the heat flux applied at the base of the fins, Tw is the base wall temperature of the fin, Tin is the inlet air temperature, and Tout is the outlet air temperature.

$$Nu = \frac{qD_h}{k_{air}\left(T_w - \frac{T_{in}+T_{out}}{2}\right)} = \frac{hD_h}{k_{air}} \tag{8}$$

The pressure drop of the fluid along the fins is calculated as the difference between the inlet pressure (P_{in}) and the outlet pressure (P_{out}).

$$\Delta P = P_{in} - P_{out} \tag{9}$$

The friction factor along the flow regime is determined by the length of the domain "L" and other flow parameters.

$$f = \frac{2}{\frac{L}{D_h}} \frac{\Delta P}{\rho v^2} \tag{10}$$

The thermal performance factor, referred to as the PEC (Performance Evaluation Criteria) number, is influenced by both the Nusselt number and the friction factor. The PEC number gauges the effectiveness of heat transfer enhancement while considering the impact of hydrodynamic forces. As such, it serves as an important measure of a cooling surface's efficiency. Higher PEC values indicate improved heat transfer performance. Additionally, a PEC value greater than 1 signifies that the heat transfer rate exceeds the increase in pressure drop. The PEC number is calculated using the equation

$$PEC = \frac{Nu/Nu_0}{f/f_0^{\left(\frac{1}{3}\right)}} \tag{11}$$

Results:

The results present the temperature distribution and streamlines along the mid-plane of the heat sink fin region. Contour plots illustrate heat dissipation and flow patterns from the CPU surface to the surrounding environment for various fin configurations. Additionally, the dimensionless parameters—friction factor, Nusselt number, and performance evaluation criteria[18]—are analyzed and plotted for Reynolds numbers of 500, 1000, and 5000. The study investigates six slot configurations, ranging from no slot (Type 1) to incremental slot sizes of 0.1 mm, 0.2 mm, 0.3 mm, 0.4 mm, and 0.5 mm, corresponding to Types 2 through 6, respectively.

Temperature Contours:

The contour plots are generated using a common temperature range for all fin geometries to ensure a consistent comparison. Figure 4 presents the temperature contour plots for various fin configurations, as detailed in Table 1, alongside a plain surface without fins. All sections are analyzed at the mid-plane of the geometry and along the fin length. The contours indicate that elliptical fins[19] exhibit improved temperature distribution, lower core temperatures, and an extended cooling fluid travel length. In contrast, triangular fins experience higher core temperatures, shorter fluid travel lengths, and poorer temperature distribution.

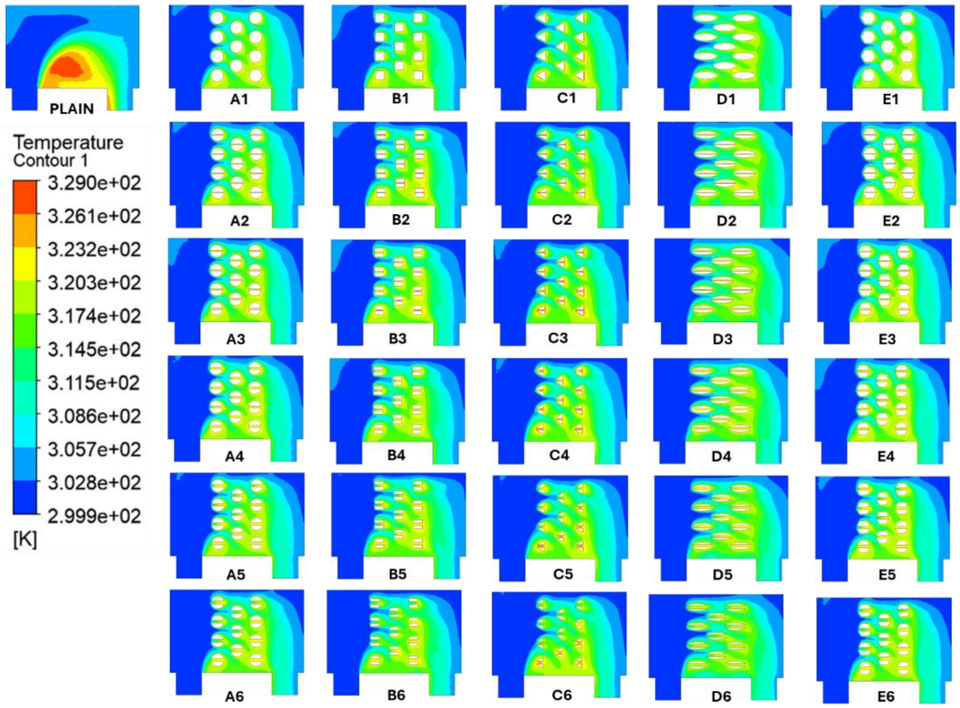

Figure 4: Temperature Distribution over CPU surface

Streamlines:

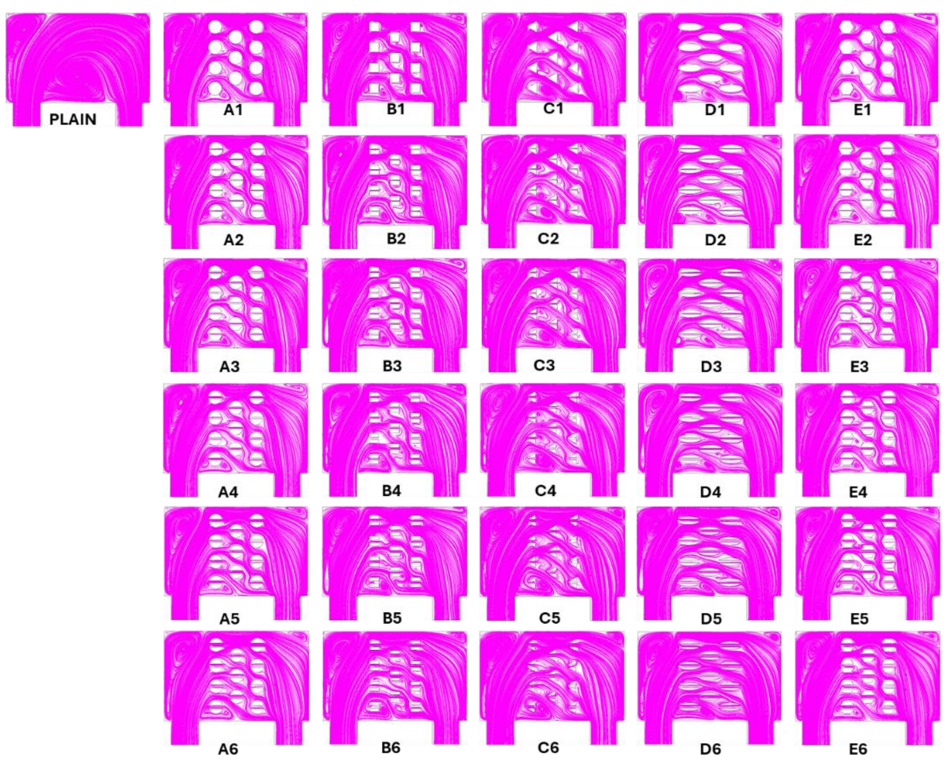

Figure 5: Flow pattern Distribution over CPU surface

Streamlines illustrate the flow characteristics within the fluid domain, providing insight into circulation and flow disturbances. It is observed that rectangular and triangular fin surfaces generate higher circulation and increased turbulence, leading to disrupted flow patterns. In contrast, elliptical fins facilitate a smoother and more continuous fluid motion, with an extended travel path for fluid particles. This streamlined flow enhances heat transfer efficiency by promoting better thermal exchange[20] and reducing localized overheating. The results suggest that minimizing disturbances and optimizing flow paths, as seen with elliptical fins, can significantly improve overall cooling performance.

Nusselt Number (Nu):

The Nusselt number (Nu) is a critical parameter in convective heat transfer, where higher values indicate enhanced heat dissipation. To achieve efficient cooling performance, maximizing the Nusselt number is desirable. As illustrated in Figure 6, Nu increases with an increase in slot width across all fin geometries on the CPU surface. This trend suggests that introducing and enlarging slots improves convective heat transfer by promoting better airflow[21] and thermal exchange.

Among the examined fin configurations, triangular fins exhibit the lowest Nu values, ranging from 15.8 to 16.8, while elliptical fins demonstrate the highest values, varying from 18.6 to 20.3 for a Reynolds number (Re) of 500. The superior performance of elliptical fins can be attributed to their streamlined shape, which facilitates smoother airflow, reduces flow separation, and enhances convective heat transfer. In contrast, the sharper edges and abrupt geometry of triangular fins contribute to higher flow resistance and localized thermal buildup, leading to lower heat dissipation efficiency. These findings highlight the significance of fin geometry and slot design in optimizing heat transfer performance for electronic cooling applications.

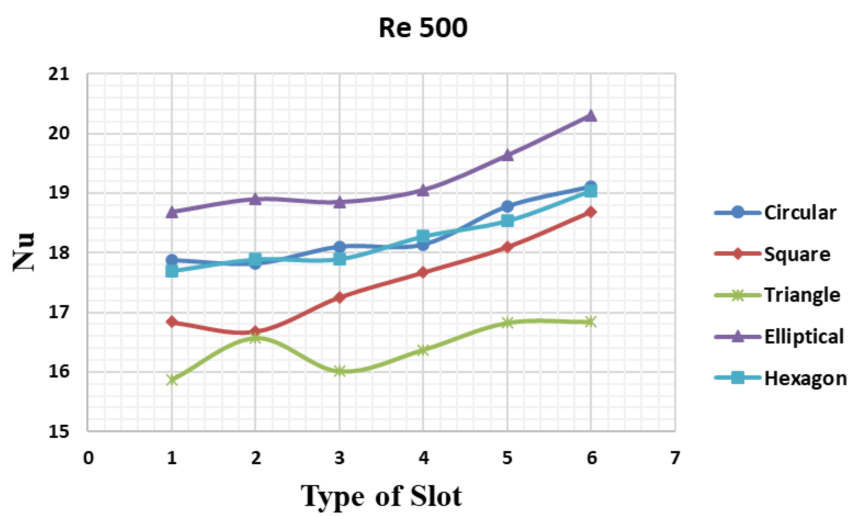

Figure 6: Nusselt Number at Re 500

Figure 7 illustrates the variation in Nusselt number (Nu) with increasing slot width for different fin geometries on the CPU surface at a Reynolds number (Re) of 5000. The results indicate that, except for elliptical fins, the Nu value decreases beyond a slot width of 0.3 mm. This suggests that while moderate slot widths enhance heat transfer by improving airflow and turbulence, excessive slotting may lead to flow separation or reduced surface area for heat exchange, thereby diminishing thermal performance.

Among the different fin configurations, triangular fins exhibit the lowest Nu values, ranging from 83 to 87, indicating comparatively poor heat transfer efficiency. In contrast, elliptical fins achieve the highest Nu values, varying from 94 to 112, demonstrating their superior ability to enhance convective heat transfer.

The remaining fin geometries show intermediate performance, with Nu values ranging between 88 and 105. These findings highlight the critical role of fin geometry and optimized slot width in maximizing heat dissipation, particularly under high Reynolds number conditions.

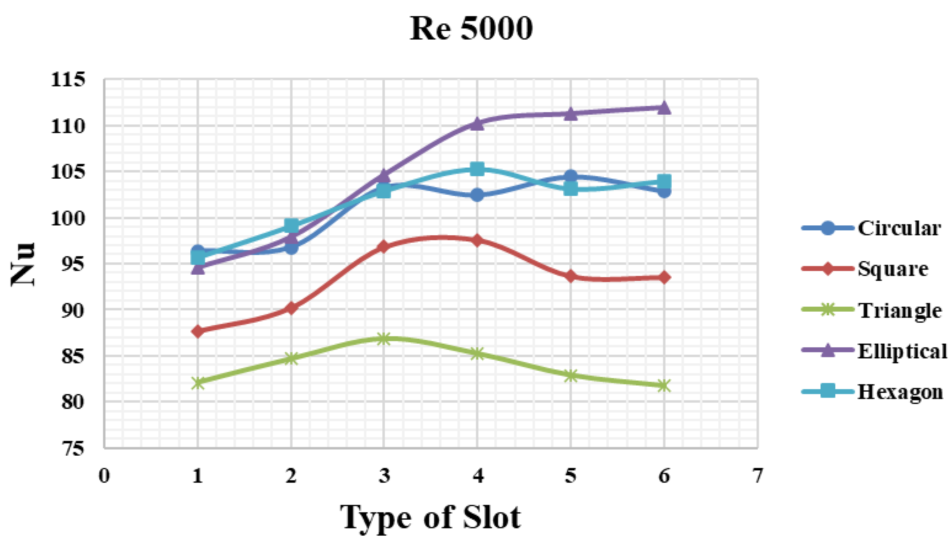

Figure 7: Nusselt Number at Re 5000

Figure 8 presents the variation of the Nusselt number (Nu) with slot width for different fin geometries on the CPU surface at a Reynolds number (Re) of 10,000. The results indicate that fins without slots exhibit superior heat transfer performance compared to their slotted counterparts. This suggests that, at higher Reynolds numbers, the presence of slots may disrupt the thermal boundary layer and reduce the effective heat transfer area, leading to diminished convective cooling efficiency.

Among the examined fin geometries, hexagonal fins achieve the highest Nu value, reaching up to 450, demonstrating their effectiveness in enhancing convective heat transfer. In contrast, triangular fins exhibit the lowest Nu values, ranging from 130, indicating relatively poorer thermal performance. These findings emphasize the importance of selecting appropriate fin geometries and design parameters, particularly at high flow conditions, to optimize heat dissipation and cooling efficiency in electronic applications.

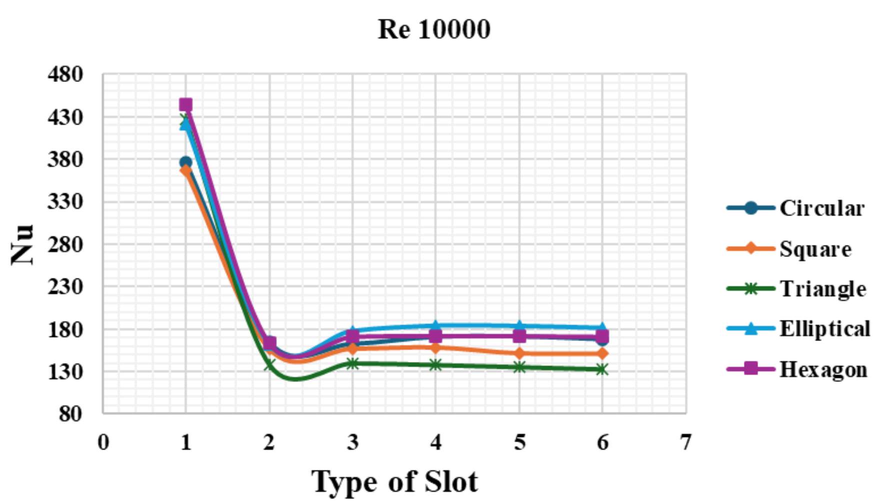

Figure 8: Nusselt Number at Re 10000

Friction Factor(f):

The friction factor (f) should be minimized to reduce the energy required to pump or blow fluid from the inlet to the outlet, thereby improving overall system efficiency. As shown in Figure 9, the friction factor decreases with increasing slot width from 0.1 mm to 0.5 mm at a Reynolds number (Re) of 500. This reduction suggests that larger slot widths facilitate smoother fluid flow by reducing flow resistance and turbulence.

Among the different fin geometries, elliptical fins exhibit the lowest friction factor, ranging from 0.985 to 0.97, indicating minimal flow resistance and efficient fluid motion. In contrast, hexagonal fins demonstrate the highest friction factor, varying from 1.065 to 1.05, suggesting greater obstruction to fluid movement. These findings highlight the influence of fin geometry and slot width on fluid dynamics, emphasizing the need to balance heat transfer performance with flow resistance for optimal thermal management.

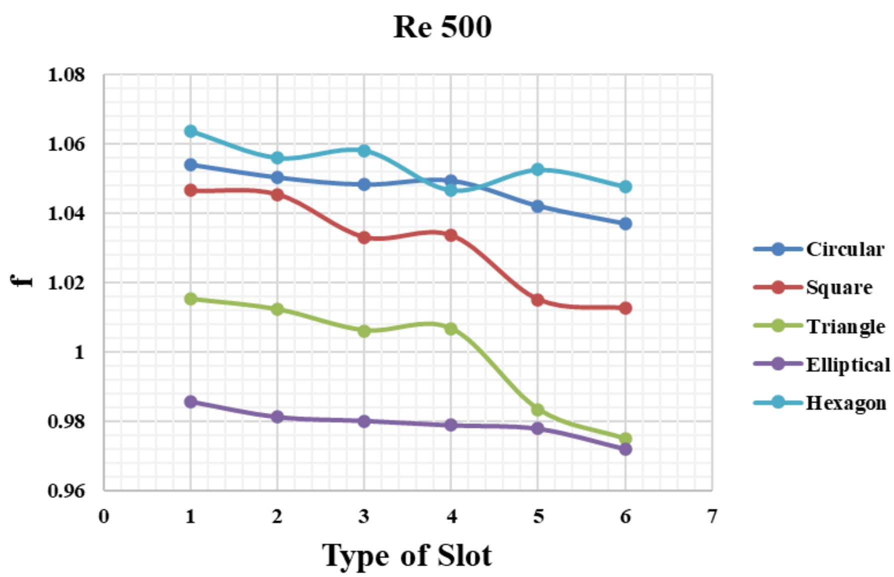

Figure 9: Friction factor at Re 500

Figure 10 illustrates the variation in the friction factor (f) with increasing slot width from 0.1 mm to 0.5 mm for different fin geometries at a Reynolds number (Re) of 5000. The results indicate a decreasing trend in the friction factor as the slot width increases, suggesting that larger slots contribute to smoother fluid flow by reducing flow resistance and pressure drop.

Among the examined fin configurations, elliptical fins exhibit the lowest friction factor, reaching a minimum value of 0.51 at a slot width of 0.4 mm. This highlights their superior aerodynamic efficiency, which minimizes energy losses associated with fluid movement. In contrast, square fins without slots exhibit the highest friction factor, with a value of 0.65, indicating greater resistance to fluid flow. These findings emphasize the impact of fin geometry and slot design on flow characteristics, underscoring the importance of optimizing these parameters to balance heat transfer efficiency and pressure drop in thermal management systems.

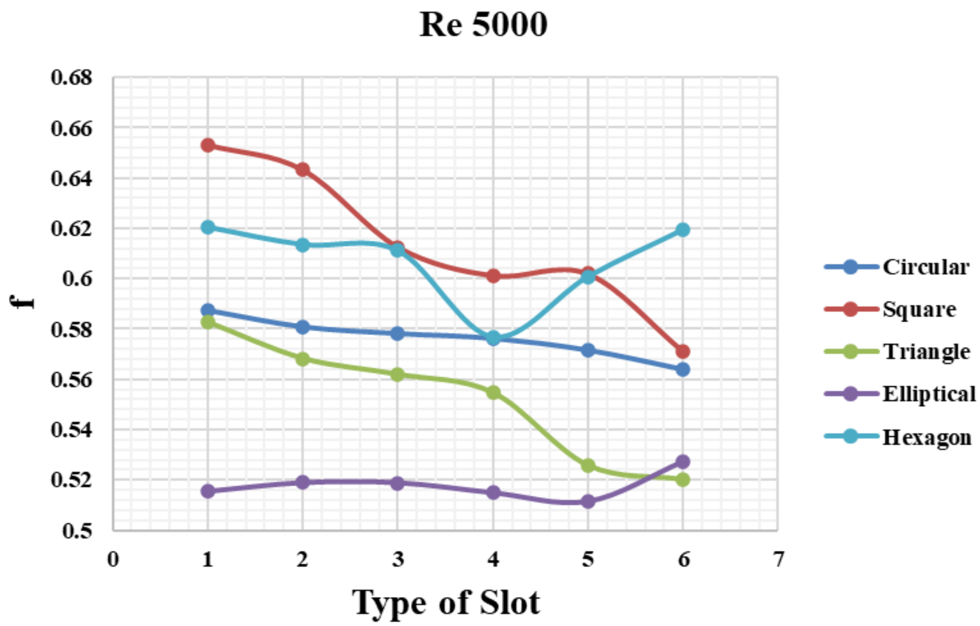

Figure 10: Friction factor at Re 5000

Figure 11 presents the friction factor (f) variation for various fin geometries at a Reynolds number (Re) of 10,000. The results show that for all slotted fin configurations—circular, square, triangular, elliptical, and hexagonal—the friction factor remains relatively unchanged, suggesting that the presence of slots does not significantly affect flow resistance under these conditions.

In contrast, for the unslotted fins, elliptical fins exhibit the lowest friction factor, with a value of 0.15, indicating minimal flow resistance. On the other hand, circular fins exhibit the highest friction factor, with a value of 2.6, signifying a higher energy cost for fluid movement due to increased flow obstruction. These findings highlight the significant influence of fin geometry on flow dynamics, particularly in unslotted configurations, and underscore the importance of selecting appropriate fin designs for optimizing thermal and flow performance in high Reynolds number applications.

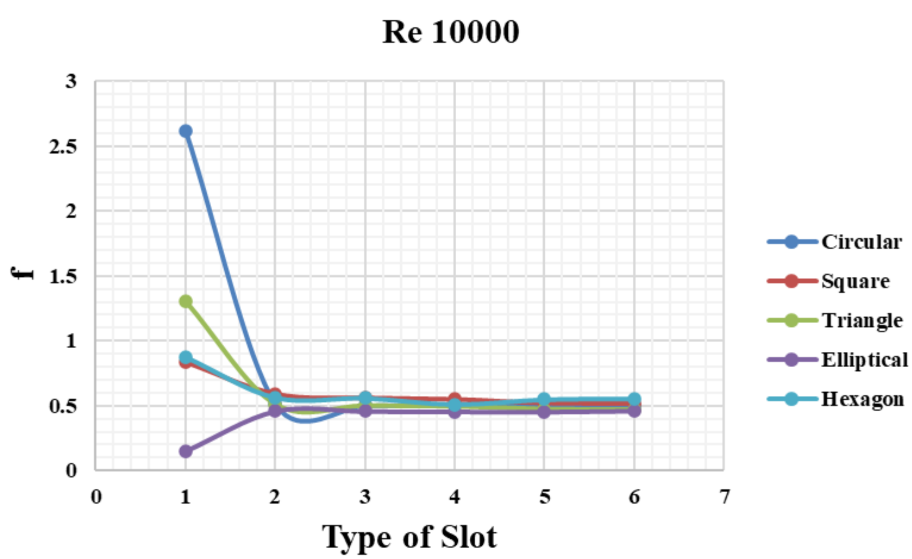

Figure 11: Friction factor at Re 10000

PEC:

The performance evaluation criteria (PEC) is a key indicator of fin efficiency, with higher PEC values reflecting better thermal performance. Figure 12 illustrates the variation of PEC at a Reynolds number (Re) of 500 for different fin configurations. Among the various geometries, elliptical fins achieve the highest PEC values, ranging from 2.26 to 2.48, indicating their superior heat transfer performance. In contrast, triangular fins exhibit the lowest PEC values, ranging from 1.9 to 2.06, suggesting that their thermal performance is less efficient compared to other fin designs. These findings underscore the importance of fin geometry in optimizing cooling efficiency and highlight the potential for elliptical fins to deliver the best performance in thermal management applications

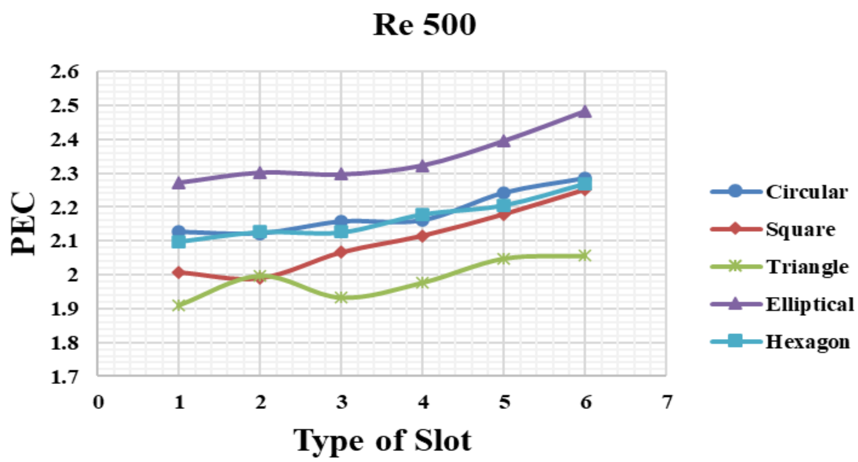

Figure 12: PEC at Re 500

Figure 13 presents the variation of the performance evaluation criteria (PEC) at a Reynolds number (Re) of 5000 for different fin configurations. Elliptical fins achieve the highest PEC values, ranging from 2.7 to 3.15, indicating their exceptional thermal performance and efficiency in heat transfer. In contrast, triangular fins exhibit the lowest PEC values, ranging from 2.25 to 2.4, suggesting relatively lower heat dissipation capabilities compared to other fin geometries. These results reinforce the superior performance of elliptical fins and highlight the importance of fin configuration in optimizing thermal management in high Reynolds number conditions.

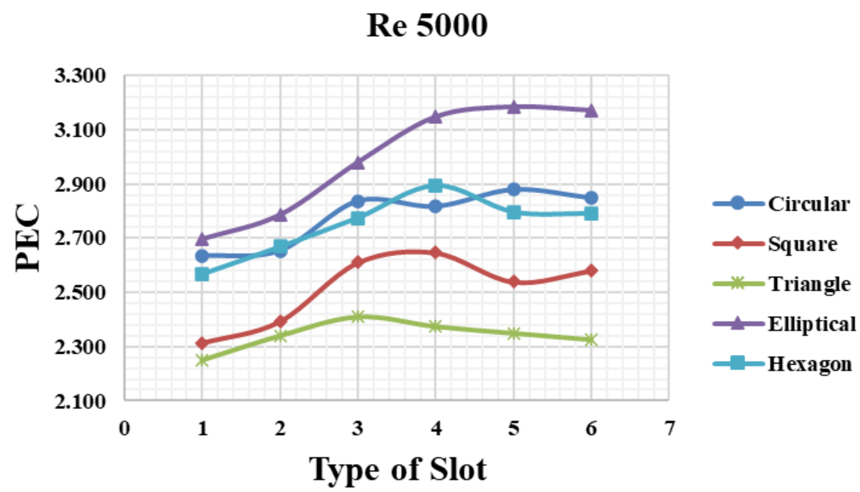

Figure 13: PEC at Re 5000

Figure 14 illustrates the variation of the performance evaluation criteria (PEC) at a Reynolds number (Re) of 10,000 for different fin configurations. Elliptical fins attain the highest PEC values, ranging from 2.7 to 3.15, indicating their superior efficiency in heat transfer and overall performance. In contrast, triangular fins exhibit the lowest PEC values, ranging from 2.25 to 2.4, reflecting relatively lower thermal efficiency. These results highlight the significant impact of fin geometry on heat dissipation performance at high Reynolds numbers, with elliptical fins demonstrating the best performance across the configurations studied.

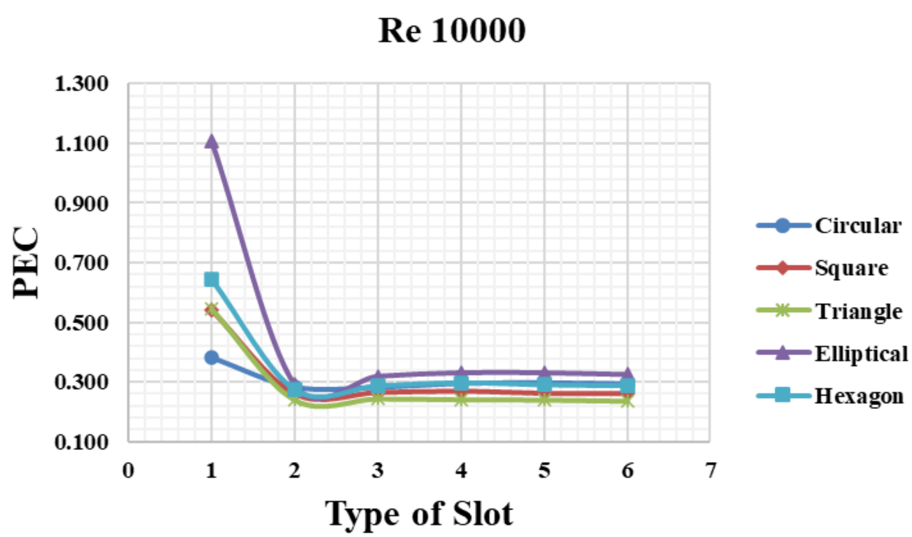

Figure 14: PEC at Re 10000

Conclusion:

This study provides a comprehensive analysis of the thermal and fluid flow characteristics of micro-pin-fin heat sinks with varying fin geometries and slot sizes. The results underscore the importance of fin geometry in optimizing heat dissipation and fluid dynamics. Elliptical fins were found to offer superior performance in enhancing heat transfer, with Nusselt numbers (Nu) ranging from 18.6 to 20.3 at a Reynolds number of 500, and a reduction in the friction factor (f) to as low as 0.51 at a slot width of 0.4 mm at Re = 5000. These fins also exhibited the highest performance evaluation criteria (PEC), reaching values of 2.7 to 3.15 at Re = 5000, indicating their excellent thermal performance. In contrast, triangular fins exhibited the least efficient thermal performance, with Nu values between 15.8 and 16.8 at Re = 500, and the highest friction factors, ranging from 1.065 to 1.05 at Re = 5000. Additionally, while introducing slots improved convective heat transfer to some extent, excessively large slots were found to negatively impact thermal performance, with a decrease in Nu beyond 0.3 mm slot width at high Reynolds numbers. These findings contribute valuable insights into the design and optimization of micro-pin-fin heat sinks, emphasizing the need for a balanced approach to maximize both heat dissipation and flow efficiency in electronic cooling systems.

Conflicts:

No conflicts between the authors.

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