

Research Paper

Investigation of the Effect of Fin Geometry on the Thermal Performance of Microelectronic Devices

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This study systematically evaluates the optimal heat sink geometry for enhanced thermal performance in electronic cooling applications. ANSYS-Icepak and COMSOL Multiphysics software assessed four distinct fin geometries (square, rectangular, circular, and conical), all uniformly sized for accurate comparison. Vital performance parameters were analyzed, including maximum temperature reduction, pressure drop, and airflow velocity. These indicators provide a holistic assessment of cooling efficacy and aerodynamic characteristics. The findings highlight the effectiveness of rectangular fins, which significantly reduce maximum temperatures due to their efficient balance of conductive and convective heat transfer. At the same time, conical fins display lower pressure drops and circular fins achieve higher airflow velocities, these attributes do not consistently enhance cooling capabilities. The importance of this study resides in its provision of definitive guidance for selecting fin geometry in designing efficient heat sinks, crucial for electronic devices. This knowledge is particularly vital for developing advanced cooling systems in compact, high-power electronics, underscoring the significant impact of fin geometry on overall thermal management efficiency. Quantitative data supporting these findings are available in the full study.

Keywords: fin geometry, electronic cooling, heat sink, thermal management, ANSYS-Icepak.



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1. INTRODUCTION

The progress in electronic chip technology has reached a stage where highly potent chips can now be accommodated within more compact packages [1]. Consequently, the resulting increase in power densities in these devices makes it imperative to implement more efficient thermal management solutions to prevent

these devices from potentially surpassing their maximum allowable operating temperatures. To mitigate heat dissipation from the components, it is conceivable to incorporate fins made of materials exhibiting high thermal conductivity [2, 3]. However, to achieve even greater heat dissipation rates, forced air or liquid cooling techniques involving these fins are employed, although these methods entail the utilization of additional components such as pumps or fans. Such approaches can lead to increased system costs and physical dimensions, making them potentially unsuitable for all applications [4]. Passive approaches, such as employing phase-change materials (PCMs), provide a method to delay the accumulation of sensible heat during a chip's operational cycle. These PCMs are composed of materials characterized by a substantial latent heat of fusion and are selected in such a way that their melting point is lower than the maximum operational temperature of the electronic device components. Instead of experiencing a gradual increase in sensible temperature, the PCM undergoes a phase change, transitioning from a solid to a liquid state by absorbing the heat generated by the chips [5]. As long as the PCM has not completely transformed into a liquid, the assembly will maintain a temperature close to the material's melting point. It is only when the latent heat storage within the PCM is exhausted will the temperature begin to rise once more. This approach is well-suited for devices with established duty cycles wherein the PCM undergoes a phase change, melts during operation, and resolidifies during shutdown.

Due to the inherent limitations in thermal conductivity of PCM materials, they may not efficiently dissipate heat quickly enough to prevent chip overheating [6]. To enhance the effective thermal conductivity of the system, fins made of highly conductive materials, such as aluminum, are introduced. These fins aid in the efficient transfer of heat from the chip to the PCM. Consequently, the addition of fins results in an increase of thermal conductivity, albeit at the expense of some latent heat storage capacity [7–10]. In the cooling enhancement of electronic devices, heat sinks are used to achieve optimum cooling performance in the electronics industry [8–17]. The challenge for designers lies in effectively dissipating the heat generated by the components while simultaneously maintaining the component's temperature at the necessary level to ensure reliable system operation [18]. Over the past two decades, researchers have devised a solution to improve heat dissipation by using different shapes of micro pin-fin heat sinks [19–22]. The objective is to achieve the most efficient cooling performance for the designed system, taking into account both cost and size considerations. To pursue this goal, various geometries must be investigated to evaluate the influence of heat sink designs on cooling performance. JALAL *et al.* [23] numerically investigated the impact of thermal performance of micro pin fins. Several fins with different configuration such as in-line and staggered layouts were analyzed. The results indicated an enhancement in cooling performance ranging from 7.9 %

to 9.3 % with an increase in the number of fins from 902 to 1312, and an improvement of 10.2 % to 11.1 % when the number of fins increased from 902 to 1640.

OGUZHAN *et al.* [24] conducted experimental investigations to assess the cooling performance of plate-fin and pin-fin heat sinks utilizing both water (as the base fluid) and Al_2O_3 nanofluid as cooling mediums. For the plate fin-heat sink, the results showed that the maximum enhancement in heat transfer, compared to an unmodified surface, was approximately 64.25 % for the base fluid and 82.8 % for the nanofluid. EKPU [25] studied the influence of fin arrangement on the thermal performance of microelectronic devices by using ANSYS finite volume design software. The findings clearly demonstrated that the configuration of heat sink fins has a substantial influence on both the thermal resistance and the overall efficiency of the microelectronic device. Furthermore, the rectangular fin configuration showed advantages over the other fin configurations studied. TANDA [26] conducted experimental research to explore heat transfer and pressure drop in a rectangular channel equipped with arrays of diamond-shaped fins. The study investigated both in-line and staggered fin arrangements. The results demonstrated that incorporating diamond-shaped fins resulted in enhanced heat transfer compared to a rectangular channel lacking fins. This improvement reached a maximum of 4.4 times when maintaining an equal mass flow rate and 1.65 times when keeping the pumping power constant.

DEBICH *et al.* [27] conducted a parametric study to assess the influence of various factors on the overall thermal efficiency of a PCM heat sink assembly. The study utilized plate-style fins and varied parameters such as the PCM type, the number of cavities in the assembly, and the materials of both the matrix and the fins. According to their findings, the geometry of the fins had the most significant effect on the chip's ability to transfer heat into the PCM. CHOUDARI *et al.* [28] performed a numerical analysis to examine different cross-sectional shapes for fins encircling a high-temperature battery. The study considered five distinct shapes, each with different PCM thicknesses and fin counts. Their findings revealed that by implementing the optimal design, the temperature of the battery was reduced by 9.28 °C. SHATIKIAN *et al.* [29] conducted a numerical investigation focusing on a design featuring internal fins with exposed tips, allowing contact with the surrounding air. In their study, they varied both the thickness of the fins and the proportion of the fins exposed to the surrounding air. The computational findings indicated that the transient phase-change process, as quantified by the volume melt fraction of the PCM, is influenced by the thermal and geometrical characteristics of the system, including the thickness and exposed portion of the fins.

JAWORSKI [30] observed that to achieve a more uniform heat distribution, employing a heat sink featuring thin-walled pipes filled with paraffin wax was advantageous. This design offered a significant surface area directly in contact

with the PCM. When compared to a larger assembly filled with wax, the results showed that this approach lowered the heat resistance between the fins and the wax. The enlarged contact area expedited the rapid distribution of higher heat density into the wax, resulting in a noticeable improvement, particularly in situations with higher heat flux. KAMKARI and SHOKOUHMAND [31] conducted an experimental study that focused on the operation of a PCM heat sink assembly with fins located on the side wall. Their study involved varying the number of fins, and it was found that the configuration with three fins exhibited the highest overall effectiveness. PAKROUH *et al.* [32] optimized a square housing with pin fins by varying the number of fins from 0 to 100 to find the ideal fin-to-wax ratio. They considered parameters such as the number of fins, fin height, fin thickness, and base thickness for optimization. The simulations factored in natural convection and PCM volume changes during melting at different critical temperatures (50 °C, 60 °C, 70 °C, and 80 °C). For a critical temperature of 50 °C, the optimal configuration featured 100 pin fins with a thickness of 4 mm, resulting in a 60.61 % PCM volume fraction.

In a study conducted by BONDAREVA and SHEREMET [33], the heat source was placed in close proximity to the PCM. This approach could be beneficial for electronic packaging designs that need to minimize the risk of PCM degradation. ALI *et al.* [34] conducted a study to investigate the influence of rectangular, round, and triangular pin fins on the heat storage ratio of a heat sink, while considering a range of energy levels from 5 W to 8 W. Their findings indicated that triangular pin fins were particularly effective because they allowed for more fins to be packed within the same area while simultaneously reducing the surface area ratio. Consequently, this configuration resulted in a higher heat storage ratio. SENTHILKUMAR *et al.* [35] examined three extruded fin shapes combined with PCMs possessing a 28 °C to 44 °C melting range. They improved melting duration and thermal conductivity through PCM volume fraction adjustments. In a related study, they analyzed a rectangular brass fin for free convection cooling in diverse applications. Thermocouples measured temperature and heat transfer rates. They enhanced heat transfer by coating the brass with carbon nanotubes. Using the Taguchi method, they investigated temperature and heat transfer characteristics, finding a 12 % increase in the heat transfer rate for the carbon-nanocoated rectangular brass fins. JI *et al.* [36] proposed a heat sink with varying fin lengths to ascertain the optimal length ratio between adjacent fins for PCM melting. Additionally, non-standard fin shapes, such as branch-shaped fins, were explored by XIE *et al.* [37]. However, it is worth noting that these non-standard fin shapes may entail higher manufacturing costs compared to more conventional solutions.

The work in this article will analyze the effect of fin geometry on the dissipated heat generated from heat sinks using ANSYS-Icepak, a common compu-

tational software for the thermal analysis and design of electronic devices and systems.

SANA *et al.* [38] investigated heat sinks with in-line and staggered fin arrangements using computational fluid dynamics (CFD) software to explore the optimal heat sink design for microelectronic devices. Their study provides an analysis of the thermal performance and pressure drop of two different types of fin cross sections. Two fin configurations, in-line (CIA), and staggered (CSA), are analyzed with 36 fins each. When comparing the performance of circular pin and cone fins at the same power inputs and mass flow rates, the results indicated the maximum temperature for circular pin fins was 0.46 % of the maximum temperature for cone fins, and they had a higher pressure drop, especially in staggered arrangements. Moreover, the maximum temperature for staggered arrangements was lower than that for in-line configurations by a factor of up to 1.17 % and 2.035 % for circular fins and conical pin fins, respectively.

Based on the described findings, several previous studies dealt with the same topics, in particular shape, the number of fins in the heat sink, and their subsequent effects on heat transfer. In the current research, five profiles of fins are studied and compared. Specifically, fin geometries, including longitudinal, square, rectangular, circular, and conical are assessed. These five configurations are considered among the most common fin profiles and are used in electronic devices. The developed design methodology is versatile enough to be applied to different circuit boards, and provides manufacturers with highly applicable, comparative performance data of the different fin assemblies. The results obtained include the cooling performance of each heat sink, maximum temperature reduction, pressure drop, and airflow velocity selected as performance indicators. In the current study, numerical simulation considering cross-sectional variations at a constant fin thickness and base height is conducted to determine, optimal dimensions of the heat sink to achieve the minimum peak operating temperature of the heat sink. Furthermore, the significance of this research lies in its ability to offer clear recommendations for choosing appropriate fin shapes when designing effective heat sinks, which is essential for electronic devices. This understanding becomes particularly crucial for the advancement of cooling systems in compact, high-power electronics, highlighting the substantial influence that fin shapes have on the overall effectiveness of thermal management.

2. MATERIALS AND METHODS

2.1. PHYSICAL MODEL AND PROBLEM DESCRIPTION

In this research, a heat sink with fins of different cross-sections is studied. A schematic diagram of the heat sink with different fin cross-sectional con-

figurations is presented in Fig. 1. All the walls of the heat sink are insulated except for two regions: the inlet, where air enters as the cooling fluid at a specified velocity and temperature and the outlet, which is exposed to atmospheric pressure. The current study is conducted to investigate the influence of fin cross-sectional geometry on overall heat sink performance.

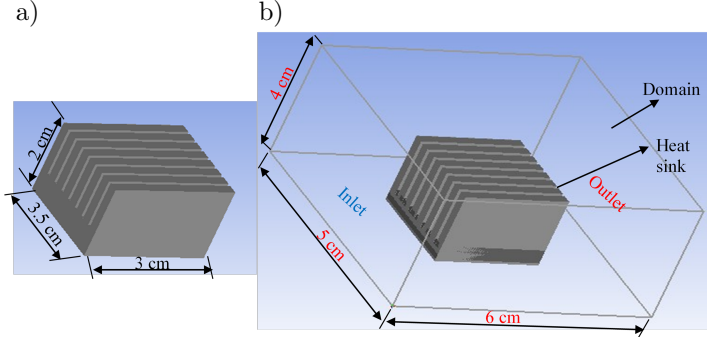


FIG. 1. Schematic diagram of the physical model:
a) heat sink, b) computational domain enclosing the heat sink.

The physical model is an enclosure containing a heat sink mounted on an electronic heat source (chip). The heat sink is made of aluminum and air serves as the cooling fluid. The total dissipated power is set at 8 W, and the heat sink dimensions are given in Table 1. The dimensions of the computational domain are based on the investigation presented in [39].

TABLE 1. Geometric dimensions of the computational domain and the heat sink assembly.

Parameter	Value			
Enclosure	6 cm $\times$ 5 cm $\times$ 4 cm			
Source	0.3 cm $\times$ 0.8 cm $\times$ 0.1 cm			
Heat sink	3 cm $\times$ 3.5 cm $\times$ 2 cm			
Fins	Number of fins = 8			
Base height	0.2 cm			
Cross section	Circular	Square	Rectangular	Conical
	$D = 0.0845$ cm	$L = 0.3$ cm	0.25 cm $\times$ 0.36 cm	$R_{\text{large}} = 0.2$ cm $R_{\text{small}} = 0.05$ cm

Five shapes of fins are studied: longitudinal, rectangular, square, circular, and finally conical as shown in Fig. 2. In these five cases, the inlet conditions are constant and the amount of heat flux was applied by the heat source located at the base of the heat sink. The five considered cases vary by their fin cross sections and all the other boundary conditions (BCs) are fixed and do not change.

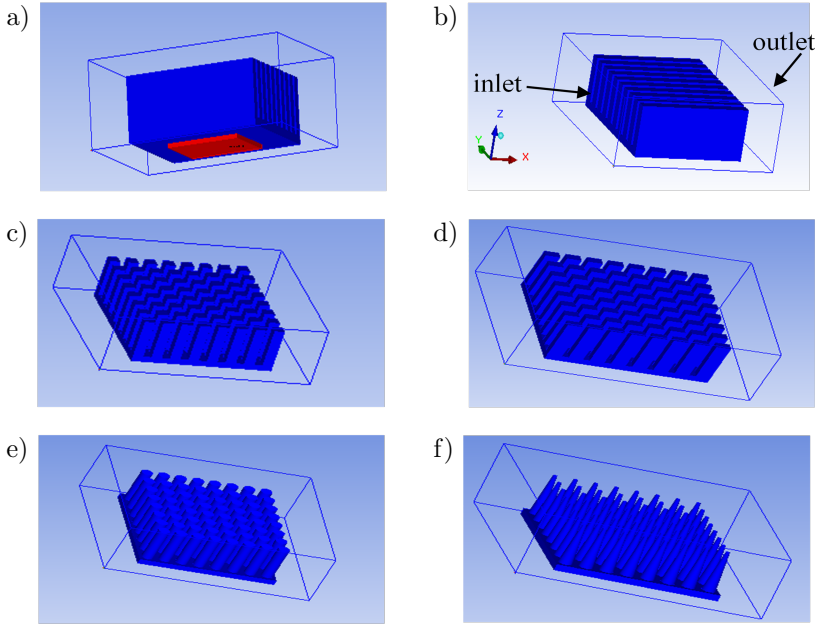


FIG. 2. Schematic of the studied fin geometries: a) whole domain, b) longitudinal fins, c) square fins, d) rectangular fins, e) circular fins, f) conical fins.

ANSYS-Icepak uses the FLUENT solver. In this study, a hexa-dominant mesh was generated in the preprocessing module. The aim of the obtained results is to identify the optimal fin cross section that can be chosen for the fins in order to reduce the temperature of the heat sink and prevent exposing it to high temperatures that exceed what the metal of the heat sink can withstand. In addition, the pressure gradient along the section was evaluated, as it varies depending on the profile change of each fin profile.

2.2. NUMERICAL ANALYSIS ASSUMPTIONS AND GOVERNING EQUATIONS

A numerical analysis is conducted using ANSYS-Icepak software to assess the cooling of the heat sink with air as the cooling fluid. The Icepak software relies on the ANSYS FLUENT solver to compute the thermal and fluid flow fields and to create a diagram, add BCs, and extract graphics and values for all the studied cases. The problem under consideration concerns the flow through the cabinet and heat sink. Heat transfer in the heat sink unit is a conjugate process, which combines conduction heat transfer in the solid parts and convection heat transfer through the fluid. The simulation utilizes the 3D Navier–Stokes and energy equations, in addition to the conservation of mass, to model temperature and flow fields. This involves the application of simplifying assumptions to the momentum and energy equations:

1. 3D, incompressible, laminar, and steady-state fluid flow condition is assumed.
2. A no-slip BC is assumed at all solid walls.
3. Viscous dissipation in the energy equation is negligible.
4. Thermal adiation is negligible.
5. The effect of gravity is negligible.
6. Properties of fluid and solid phases are constant.

After applying this set of assumptions, the 3D equation system, which governs the single-phase model, includes the continuity, momentum, and energy equations, described as [40]:

$$(2.1) \quad \frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} = 0.$$

The momentum conservation equations in x -, y -, and z -directions, respectively, are given by [40, 41]:

$$(2.2) \quad u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} + w \frac{\partial u}{\partial z} = -\frac{1}{\rho} \frac{\partial P}{\partial x} + \frac{\mu}{\rho} \left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2} \right),$$

$$(2.3) \quad u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} + w \frac{\partial v}{\partial z} = -\frac{1}{\rho} \frac{\partial P}{\partial y} + \frac{\mu}{\rho} \left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} + \frac{\partial^2 v}{\partial z^2} \right),$$

$$(2.4) \quad u \frac{\partial w}{\partial x} + v \frac{\partial w}{\partial y} + w \frac{\partial w}{\partial z} = -\frac{1}{\rho} \frac{\partial P}{\partial z} + \frac{\mu}{\rho} \left(\frac{\partial^2 w}{\partial x^2} + \frac{\partial^2 w}{\partial y^2} + \frac{\partial^2 w}{\partial z^2} \right).$$

The energy equation is

$$(2.5) \quad u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} + w \frac{\partial T}{\partial z} = \frac{k}{\rho c_p} \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} \right),$$

where the variables T and P represent the fluid temperature and pressure, respectively, whereas u , v , w , represent the fluid velocity components, and ρ , μ , c_p , and k denote the fluid density, dynamic viscosity, specific heat capacity, and thermal conductivity, respectively.

The heat conduction equation for the solid heat sink walls, is given by [9]:

$$(2.6) \quad \frac{\partial^2 T_s}{\partial x^2} + \frac{\partial^2 T_s}{\partial y^2} + \frac{\partial^2 T_s}{\partial z^2} = 0,$$

where subscript s refers to the solid domain.

2.2.1. AVERAGE NUSSELT NUMBER

The average Nusselt number (Nu_{avg}) is a dimensionless parameter that characterizes the efficiency of convective heat transfer relative to conductive heat

transfer across a boundary layer. Higher Nusselt numbers indicate more effective convective heat transfer.

The local Nusselt number indicates how the local rate of heat transfer varies along the hot surface and can be defined as [42]:

$$(2.7) \quad \text{Nu}_{\text{avg}} = \frac{hl}{k_f},$$

where h is the average heat transfer coefficient, l is the characteristic length of the sink, and k_f is the thermal conductivity of the fluid.

The average forced convection heat transfer coefficient (h) of the air inside the heat sink is determined using Newton's law of cooling, where h is determined by the convective heat transfer rate (Q), the heat sink surface area (A), and the temperature difference (ΔT) between the surface and the air.

The following three steps are used to determine the heat transfer coefficient (h).

Step 1: Determine the temperature difference (ΔT).

Since the air heats up as it passes through the heat sink:

$$(2.8) \quad \Delta T = T_s - T_{\text{in}},$$

where T_s is the surface temperature of the heat sink, and T_{in} is the fluid inlet temperature.

Step 2: Measure the heat transfer rate (Q).

This can be calculated using the air mass flow rate (m) and the specific heat capacity of the air (c_p) as follows:

$$(2.9) \quad Q = m c_p (T_{\text{in}} - T_{\text{out}}),$$

where T_{out} is the fluid outlet temperature.

Step 3: Calculate the heat transfer coefficient (h), from Newton's law of cooling as follows:

$$(2.10) \quad h = \frac{Q}{A \Delta T},$$

where A is the surface area of the heat sink exposed to the airflow.

Note: The forced convection heat transfer coefficient in air-cooled heat sinks typically ranges from 20 W/m²K to 250 W/m²K.

The BCs used for this problem are stated as follows. Air is uniformly drawn, with constant temperature and velocity, into the cabinet inlet. At the downstream boundary of the computational domain, which is located at the cabinet outlet along the x -direction, a pressure BC is applied. No-slip conditions with a thermal insulation are enforced on all the other confined walls. At the base of the heat sink, a uniform constant heat flux is applied from the source heat. Adiabatic thermal BC are applied to the outer perimeter of the heat sink base, except for the heated area:

1. BCs for the flow field: a uniform velocity is prescribed at the channel inlet. A no-slip condition is used for all the walls, i.e., $u = 0$, $v = 0$, $w = 0$.
2. Adiabatic BCs are applied to all the boundaries of the solid region except for the heat sink bottom wall, where a heat source with a constant power of 8 W is applied. At the inlet, the temperature is set to $T = T_{\text{in}} = 298$ K, $v_{\text{in}} = 0.1$ m/s, $-k_s \frac{\partial T_s}{\partial y} = -k_f \frac{\partial T_f}{\partial y}$ at the fluid–solid interface, $\frac{\partial T}{\partial y} = 0$ at the outlet.

2.3. VERIFICATION OF GRID INDEPENDENCE

The effect of grid refinement on the numerical solutions is investigated for the cases of 8 solid circular fins in an in-line arrangement. Grid independence is investigated by testing five different mesh sizes, as shown in Table 2. The grid independence test is first conducted by using several different numbers of elements and nodes. The results obtained from these meshes are summarized in Table 2.

TABLE 2. Mesh configurations and corresponding maximum source temperatures.

Test no.	Number of elements	Number of nodes	Temperature of the source [°C]
1.	40 560	45 738	138.688
2.	82 784	91 800	136.888
3.	90 902	100 320	136.112
4.	111 585	122 496	135.998
5.	136 912	249 600	135.948

Table 2 reveals that increasing the number of nodes beyond 122 496 results in a negligible temperature variation of less than 0.04% for the heat source. In the case of solid fins, a significantly lower number of nodes, specifically 122 496 nodes, demonstrates grid independence. Beyond this point, further grid refinement does not appreciably impact the solution. As a result, the fourth mesh size is selected for calculations across all fin geometries to maintain maximum accuracy. Figure 3 shows the 3D computational quadratic mesh of the domain.

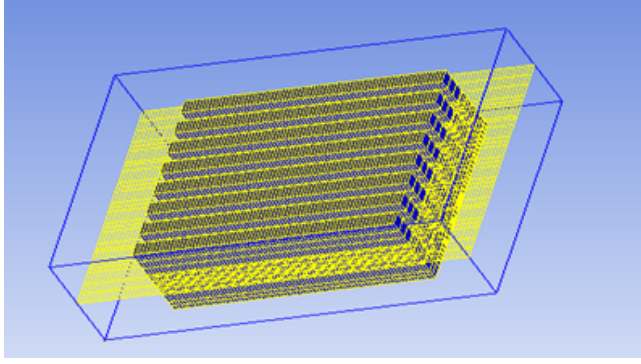


FIG. 3. 3D computational quadratic mesh of the domain.

3. SIMULATION RESULTS

3.1. VALIDATION OF THE NUMERICAL MODEL

In computational thermal analysis, ensuring the accuracy of numerical models is essential, and this accuracy can be established through rigorous validation against well-established benchmarks. This subsection aims to validate the precision of the current computational framework by directly comparing it with the numerical results obtained by ROSLI *et al.* [39]. To achieve this, a model was employed, featuring a heat sink with vertical fins. This model meticulously adhered to the dimensions and BCs outlined in [39], including a pin fin height of 0.2 cm, a length of 4 cm, and a width of 57 μm . Additionally, the model considered an inlet temperature of 25 $^{\circ}\text{C}$, an inlet velocity of 0.1 m/s, and a constant thermal load of 5 W applied to the heat source located at the base of the heat sink.

Figure 4 illustrates a top-view depiction of the temperature distribution within the pin-fin heat sink featuring in-line circular fins, akin to the configuration examined by ROSLI *et al.* [39], denoted as Model CIA 3. In the present

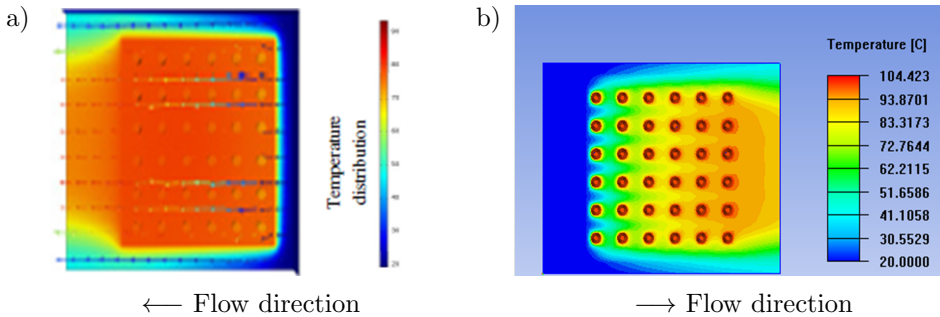


FIG. 4. Top-view distribution plots of temperature for the circular pin-fin heat sink model with an in-line arrangement. A comparison between the result presented by (a) ROSLI *et al.* [39], and (b) current simulation result.

model, the maximum temperature reached 104.423°C , closely mirroring the reference value of 107.43°C . This slight temperature variance of 2.8% falls well within the widely acknowledged error tolerance range for thermal simulations within this field [36], affirming the high accuracy of the simulation results. This slight variance can typically be attributed to minor differences in mesh discretization, numerical methods, or the precise implementation of boundary conditions between the two models. A summary of the temperature distribution between the current model and the corresponding data from ROSLI *et al.* [39], which is performed for pin fins with CIA arrangements, is presented in Table 3.

TABLE 3. Comparing the temperature distribution between our current results and the corresponding ones by ROSLI *et al.* [39] which is performed for pin fins with CIA arrangements.

Temperature in $^{\circ}\text{C}$ [starting from maximum to minimum]	Current simulation results	Numerical results by ROSLI <i>et al.</i> [39]
Maximum temperature	104.423	107.43
Intermediate temperatures	83.32	84
	72.76	70
	62.21	60
	30.55	30
Minimum temperature	20	20

The close proximity between the outcomes of the present simulation and those of the reference study validates the present numerical approach, affirming its capability to accurately predict the thermal behavior of heat sink configurations. While this validation establishes the accuracy of the model for the specific scenario examined, it is acknowledged that further validation tests are necessary to enhance the model's broader applicability. Furthermore, further validation efforts should encompass different fin geometries, flow rates, and thermal loads to expand the model's suitability for a wider spectrum of thermal management scenarios. In conclusion, this validation study underscores the reliability of the employed computational approach and reinforces the subsequent findings of the study.

Figure 5 displays the maximum temperature of the heat sink at different inflow velocities, serving to further verify the precision of our numerical model. We juxtaposed this data with the findings of ADHIKARI *et al.* [43]. Their study utilized a combination of experimental and numerical techniques to examine heat transfer from rectangular fins under forced convection conditions, specifically at low Reynolds numbers. This comparison is an essential step in showcasing the reliability of our computational technique. Fortunately, the data demonstrate outstanding concurrence. The highest discrepancy between our computational

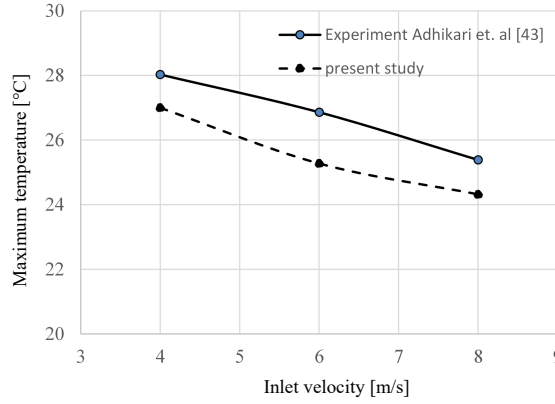


FIG. 5. Maximum temperature of the heat sink at different inlet velocities.

forecasts and the data obtained from ADHIKARI *et al.* [43] is below 3.68 %. This strong correlation confirms that our selected numerical approach effectively simulates forced convection heat transfer in rectangular fin geometries.

3.2. TEMPERATURE DISTRIBUTION RESULTS

3.2.1. TEMPERATURE DISTRIBUTION WITHIN THE CABINET

In thermal management systems, especially in electronic enclosures, it is crucial to consider the significant consequences of both high and low temperatures. Higher temperatures can result in reduced efficiency and the risk of electronic component failure, whereas lower temperatures typically indicate effective heat dissipation [1]. Therefore, analyzing the temperature distribution within the cabinet provides essential information about the effectiveness of the cooling system in use.

In Fig. 6, the temperature distribution within the computational domain is illustrated, showing a clear gradient that stems from the cooler inlet region toward the warmer outlet region. This gradient indicates the progressive heating of the cooling air as it absorbs heat from the cabinet's components. The temperature increase aligns with the direction of airflow, supporting the understanding of convective heat transfer mechanics, where air, entering at a lower temperature, absorbs heat from the solid electronic components and the heat sink.

The cooler temperatures at the cabinet inlet suggest that the incoming air effectively absorbs heat, a positive indicator of the initial thermal management strategy. As the air moves through the cabinet, the increasing temperature gradient toward the exit signifies that the air becomes less efficient at removing heat due to its own temperature rise. Figure 6 visually represents this temperature change, showing a transition from cooler blue regions to warmer red regions.

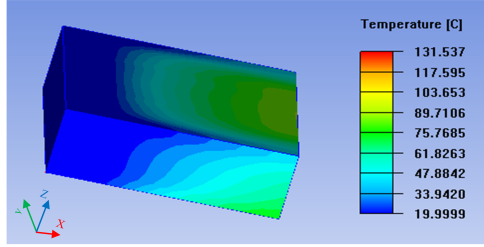


FIG. 6. Temperature distribution within the electronic enclosure domain.

This distribution pattern is critical as it reflects the thermal load within the cabinet and assesses the cabinet's design effectiveness in facilitating airflow and heat transfer. The elevated temperatures near the outlet raise concerns about potential hotspots, which are localized regions of concentrated heat that could surpass the operational temperature limits of the electronic components housed within the cabinet.

Furthermore, these high-temperature regions imply the possibility of optimizing the cabinet's cooling design. Potential enhancements might include improving airflow through strategic placement of vents or fans, optimizing the layout of internal components to achieve more uniform heat distribution, or incorporating heat sinks with more efficient fin designs for enhanced heat dissipation.

The observed temperature distribution offers valuable insights into the thermal behavior within the cabinet, underscoring the significance of considering both the design of the thermal management system and the layout of electronic components to ensure the sustained effectiveness of cooling air throughout the entire cabinet.

3.2.2. TEMPERATURE DISTRIBUTION OF DIFFERENT FIN GEOMETRIES

The thermal performance of heat sinks is significantly influenced by the temperature distribution within different fin designs. Elevated temperatures observed on the fins may signal suboptimal heat dissipation, raising the risk of component overheating. Conversely, lower temperatures typically signify efficient heat removal. Consequently, analyzing the temperature profiles of diverse fin configurations offers valuable insights into their cooling effectiveness. Figure 7 to Fig. 11 illustrate a comparative examination of temperature distributions across various fin geometries (longitudinal, square, rectangular, circular, and conical) implemented on heat sinks, indicating the nuanced impact of fin design on thermal performance.

Starting with the examination of longitudinal fins, as depicted in Fig. 7, the temperature distribution pattern shows that the temperature distribution

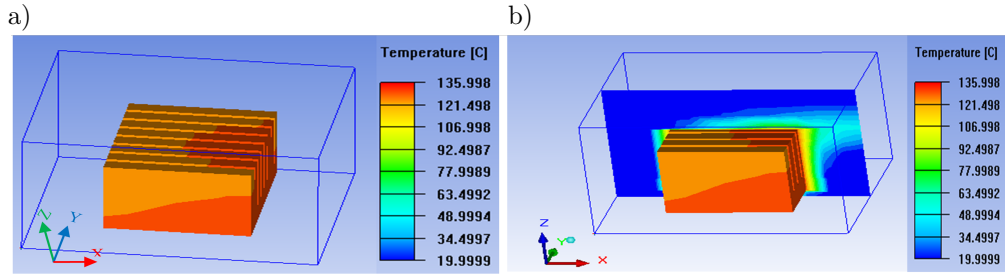


FIG. 7. Temperature distribution of longitudinal fins on: a) heat sink, b) Y-cut plane.

along the fins is continuous and has a large value along the heat sink. The entire heat sink has warmer extremes, peaking at around 135.99°C . This consistent thermal gradient along the length of the fins implies an effective heat conduction pathway from the heat source to the fin tips. It is observed that the temperature distribution along the fins ranges between approximately 121°C and 136°C . The figure also shows that the temperature is highest at the base of the fins because it is close to the heat source located below the heat sink, and the temperature decreases from the fin base to the fin tip. Furthermore, Fig. 7b illustrates that along a single fin, the temperature decreases as the end of the sink is approached. This is because the air absorbs heat through convection from the fins, causing the air temperature to rise and the fin temperature to decrease from the inlet to the outlet.

In contrast, Fig. 8 presents the temperature profile for square fins, where an evident central heating zone is discernible, characterized by a sharper gradient reaching its maximum at approximately 131.54°C . This indicates a concentration of heat directly above the heat source. The square fin design, while effective, may benefit from optimization strategies, such as alterations to fin spacing or increased airflow, to address the relatively less efficient heat transfer in this particular region. Figure 9 illustrates the temperature distribution for rectangular fins, with the highest temperature recorded at around 124.55°C . The temperature gradient observed across the fins is notably uniform, suggesting efficient heat distribution. The design of rectangular fins seems to maximize

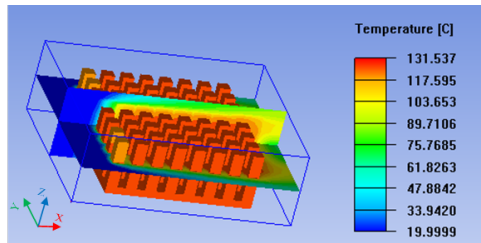


FIG. 8. Temperature distribution of square fins on the heat sink and through y- and z-planes.

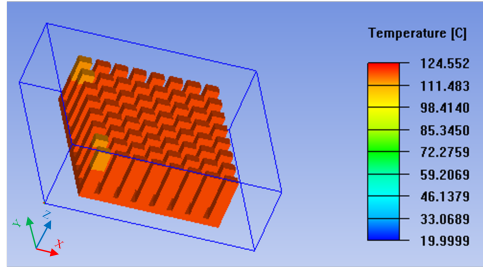
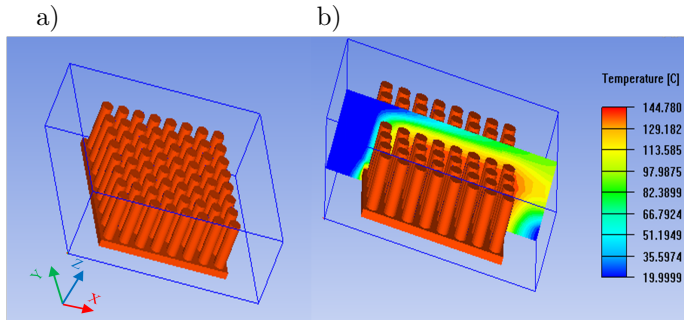
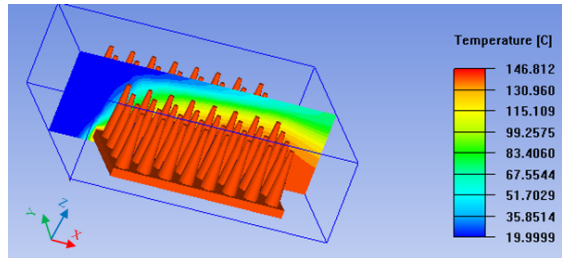


FIG. 9. Temperature distribution of rectangular fins on the heat sink.

the surface area in contact with the cooling air, thereby promoting effective convective heat transfer. This is reflected in the relatively cooler temperatures observed across the entire heat sink. Upon examining circular fins in Fig. 10, a higher maximum temperature is evident, particularly noticeable in the z -plane, where it reaches up to 144.78°C . The circular design might be impeding airflow, leading to a less effective convective cooling process in comparison to the more streamlined rectangular fins. The thermal pattern suggests that while circular fins provide an efficient conductive pathway, the convective cooling may not be as effective, likely due to airflow disruption caused by the fins' shape. Lastly, Fig. 11 presents conical fins, exhibiting the highest maximum tempera-

FIG. 10. Temperature distribution through: a) the circular fins on the heat sink, b) the z -plane.FIG. 11. Temperature distribution of conical fins on the heat sink and through the z -plane.

ture among the various geometries, ascending to 146.81°C . The conical shape, though visually distinctive, appears to yield diminishing returns in terms of cooling performance. As the fins taper towards their tips, the available surface area for heat dissipation decreases, which likely contributes to the reduced effectiveness of convective heat transfer.

The temperature distribution of square fins in Fig. 8 reveals a distinctive thermal pattern when analyzed from two perpendicular planes along the y - and z -axes. This dual-plane analysis offers a more intricate understanding of the heat sink's thermal behavior, highlighting the anisotropic nature of heat transfer in different directions. In the y -direction, which likely corresponds to the direction of airflow, there is a gradual increase in temperature from the cooler inlet, depicted by blue-green colors at approximately 33.94°C , to the warmer outlet regions represented by yellow and red hues, peaking at around 131.54°C . This temperature gradient is significant as it illustrates how air absorbs heat while passing over the fins. The square fins' geometry, despite providing a substantial surface area for heat transfer, may induce airflow patterns resulting in uneven cooling. Specifically, the central region directly above the heat source exhibits a noticeable hotspot, indicating that the air passing through this region becomes saturated with heat, becoming less effective at cooling the downstream fins. This phenomenon suggests that there may be a limit to the cooling capacity of square fins due to airflow dynamics, which could potentially be optimized by adjusting fin spacing or height to enhance cross-flow and improve convective heat dissipation. In the z -direction, the temperature distribution along the height of the fins is observed, with lower sections of the fins closer to the heat source having higher temperatures, as expected due to conductive heat transfer. However, the temperature profile indicates a rapid decrease in temperature when moving away from the base, transitioning from red to green on the color scale. This decline suggests that while the fin bases effectively conduct heat away from the source, the tips of the fins may not be as efficient in dissipating this heat into the surrounding air. The heat distribution in the z -direction is particularly significant as it impacts the overall thermal resistance of the heat sink. The observed temperature variation along the height of the fins may imply the need for a design that encourages better airflow around the fin tips or the use of materials with higher thermal conductivity to ensure more uniform heat dissipation along the entire fin length.

The temperature stratification observed in both the y - and z -directions highlights the intricate interaction between heat conduction along the fins and convective heat transfer to the surrounding air. An effective heat sink design must take into consideration these directional variations to prevent performance limitations. The implications derived from Fig. 8 can provide valuable insights for design modifications, including adjustments to fin aspect ratios, alterations in

material selection, or improvements to the thermal interface between the fin base and the heat source, all aimed at enhancing the thermal management capabilities of square fin heat sinks.

In conclusion, this comparative analysis underscores the critical role of fin geometry in the field of thermal management. Among the examined designs, rectangular fins emerge as the most efficient, striking a harmonious balance between conductive and convective heat transfer. In contrast, circular and conical fins, which exhibit higher temperatures, may necessitate design modifications or supplementary cooling strategies to achieve comparable levels of thermal performance. The rectangular fins provided a reduction of the maximum temperature by approximately 8.42% compared to longitudinal fins, 5.29% compared to square fins, 13.97% compared to circular fins, and 15.16% compared to conical fins, respectively. These insights bear significant relevance in the context of designing cooling systems for microelectronic devices, where the effective management of peak temperatures is a fundamental requirement for ensuring reliability and optimal performance.

3.3. PRESSURE DISTRIBUTION RESULTS

The pressure distribution patterns across various fin geometries, as illustrated in Fig. 12, provide valuable insights into the aerodynamic efficiency and its consequential impact on thermal performance. Elevated pressure levels detected on the inlet side of the heat sink fins suggest resistance to incoming air. This resistance may be attributed to either insufficient spacing between the fins or suboptimal fin shapes, resulting in increased airflow hindrance. While this situation may imply enhanced forced airflow through the heat sink, potentially leading to improved heat transfer rates, it could also necessitate higher energy consumption by fans or pumps handling air or coolant circulation within the system. Furthermore, excessive pressure could induce airflow bypass, wherein air circumvents the fin array rather than passing through it, thereby diminishing the heat sink's effectiveness [31]. Conversely, lower pressure values signify smoother airflow through the fin array with reduced resistance. This is generally desirable as it indicates that less energy is required to move air through the system, leading to increased system cooling efficiency. However, excessively low pressure could imply insufficient contact time between the air and the fins for effective heat transfer, particularly if the airflow velocity is too high [32]. Striking the right balance is crucial; airflow should be adequate to facilitate efficient heat removal but not so swift that it fails to absorb adequate heat from the fins.

The pressure distribution data for square fins in Fig. 12 a show a peak pressure of 0.176982 N/m^2 near the inlet, gradually decreasing toward the outlet. This pattern suggests that square fins induce a higher pressure drop, possibly

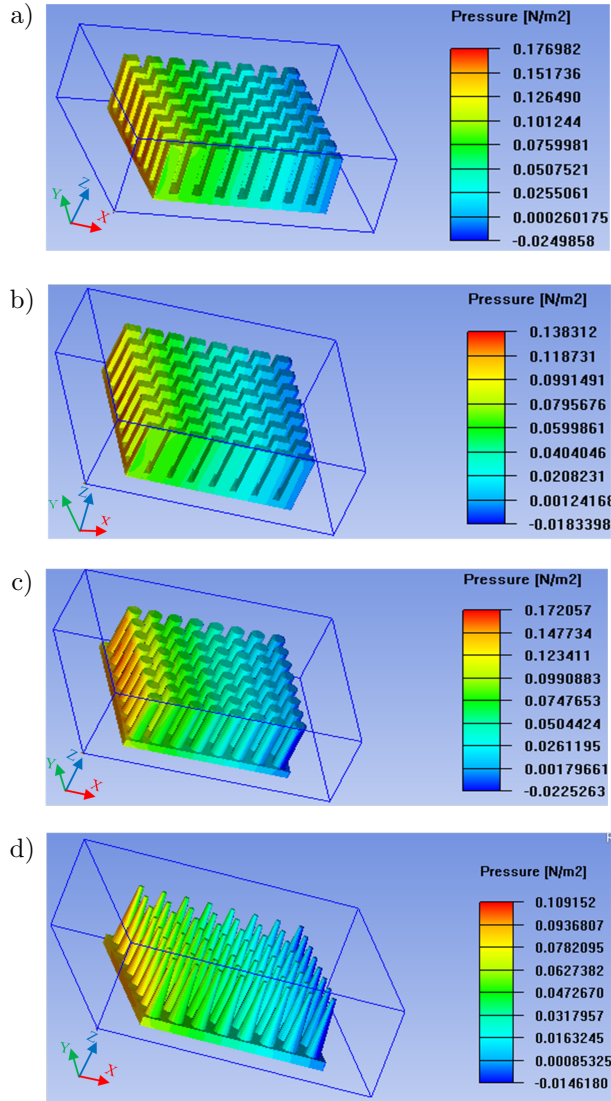


FIG. 12. Pressure distribution over: a) square fins, b) rectangular fins, c) circular fins, d) conical fins.

due to airflow disruption caused by the sharp edges and corners of these fins, resulting in separated flows and vortices that elevate overall aerodynamic drag. In Fig. 12b, the pressure drop over the rectangular fins is slightly lower, with a maximum of 0.138312 N/m², indicating relatively smoother airflow compared to square fins. The elongated shape of rectangular fins likely guides airflow more smoothly between the fins, minimizing flow separation and the associated pressure drop.

This suggests a better-balanced design in terms of thermal and aerodynamic performance. Circular fins in Fig. 12c exhibit a higher-pressure region with a maximum of 0.172057 N/m^2 at the inlet, similar to square fins, but the pressure gradually decreases toward the outlet. The rounded shape of circular fins likely contributes to maintaining a more laminar flow, reducing the occurrence of flow separation and consequently the pressure drop. Interestingly, conical fins in Fig. 12d show the lowest maximum pressure at 0.109152 N/m^2 . This can be attributed to the tapered design, which facilitates a streamlined airflow with reduced resistance. This shape aids in efficient redirection of the flow, minimizing aerodynamic drag and resulting in a lower pressure drop across the fins.

When comparing these fin profiles, square fins exhibit the highest pressure drop, indicating lower aerodynamic efficiency. Rectangular and circular fins offer improvements in airflow management, with circular fins particularly enhancing flow smoothness. Conical fins stand out with the lowest pressure drop, indicating superior airflow resistance, but this may not necessarily translate to improved heat dissipation, as observed in the temperature distribution analysis. Relating these pressure findings to earlier temperature distribution observations reveals a complex trade-off between achieving low thermal resistance and low aerodynamic drag. Conical fins exhibit a 38.07 % reduction in pressure drop compared to square fins, a 21.01 % reduction compared to rectangular fins, and a 36.63 % reduction compared to circular fin geometries, indicating a more streamlined airflow through the fin array and reduced aerodynamic resistance. However, despite the reduced pressure drop, conical fins also exhibit the highest temperatures, suggesting that while airflow through the fins encounters less resistance, it may not be the most effective for heat transfer. In contrast, square and circular fins, despite causing higher pressure drops, do not necessarily result in higher temperatures compared to conical fins, indicating a complex interplay between airflow patterns and thermal efficiency. Rectangular fins, with their balanced pressure drop and lower temperatures, appear to offer an optimal geometry for effective heat dissipation in this study.

This analysis underscores the importance of considering both aerodynamic and thermal aspects when designing heat sinks. The goal is to identify a configuration that minimizes both thermal resistance for efficient heat transfer and aerodynamic drag for lower pressure drops, ultimately enhancing the overall performance of the cooling system.

3.4. VELOCITY DISTRIBUTION RESULTS

A comprehensive examination of airflow patterns across various fin configurations plays a pivotal role in gaining insights into heat sink behavior, a critical determinant of convective heat transfer efficiency. Within the array of fins in a heat

sink, elevated velocity values suggest the rapid movement of air through the inter-fin channels. This accelerated airflow can be advantageous for heat transfer, as it promotes convective cooling by directing more air over the fin surfaces and reducing the thickness of the thermal boundary layer [44]. Consequently, this promotes the efficient extraction of heat from the heat sink. However, excessively high velocities may compromise heat transfer efficiency by diminishing the time air spends in contact with the fins, limiting heat absorption. Furthermore, heightened velocities can elevate noise levels in the cooling system and potentially lead to increased wear on mechanical components such as fans, owing to greater forces involved [45]. Conversely, lower velocity values indicate a moderate airflow, implying less efficient heat transfer due to a thicker thermal boundary layer, which allows more heat to accumulate within the heat sink. While this reduced airflow speed may result in quieter operation and less mechanical stress, it may not suffice for effective cooling, particularly under high thermal loads [46]. It is imperative to strike a balance and ensure that the velocity does not dip too low to maintain adequate cooling performance, especially in compact or high-powered electronic devices where thermal management holds paramount importance. In essence, the optimization of the velocity distribution is a meticulous task, necessitating careful design considerations to align with the specific cooling requirements of the electronic device or system under investigation.

Figure 13 reveals that the velocity distribution over different shapes of fins. Figure 13a shows that the square fins peak velocity at approximately 0.327 m/s, indicating a substantial airflow through the fin channels, which can augment convective heat transfer. However, this high-velocity airflow may also lead to an increased pressure drop and turbulence, particularly in the vicinity of the square fin edges, as elucidated in the earlier discussion on pressure distribution. Figure 13b depicts slightly lower peak velocities of around 0.263 m/s for rectangular fins, suggesting a more moderate airflow. This velocity profile could reduce the power requirements of fans and potentially yield quieter operation while still delivering sufficient airflow for cooling and minimizing energy consumption for air movement. In Fig. 13c, circular fins exhibit the highest peak velocity at approximately 0.340 m/s, indicating that their rounded design may streamline airflow, reducing resistance. Nevertheless, as observed in the temperature analysis, this does not invariably translate into the most effective cooling, possibly due to the circular fins' suboptimal surface area for heat transfer. Lastly, Fig. 13d presents conical fins with a peak velocity of roughly 0.272 m/s. The tapered configuration of these fins may aid in guiding airflow smoothly through the heat sink, contributing to the lowest pressure drop among the various fin geometries. Nonetheless, and as previously noted, this does not align with the best thermal performance, as conical fins exhibit the highest temperatures compared to other fin designs.

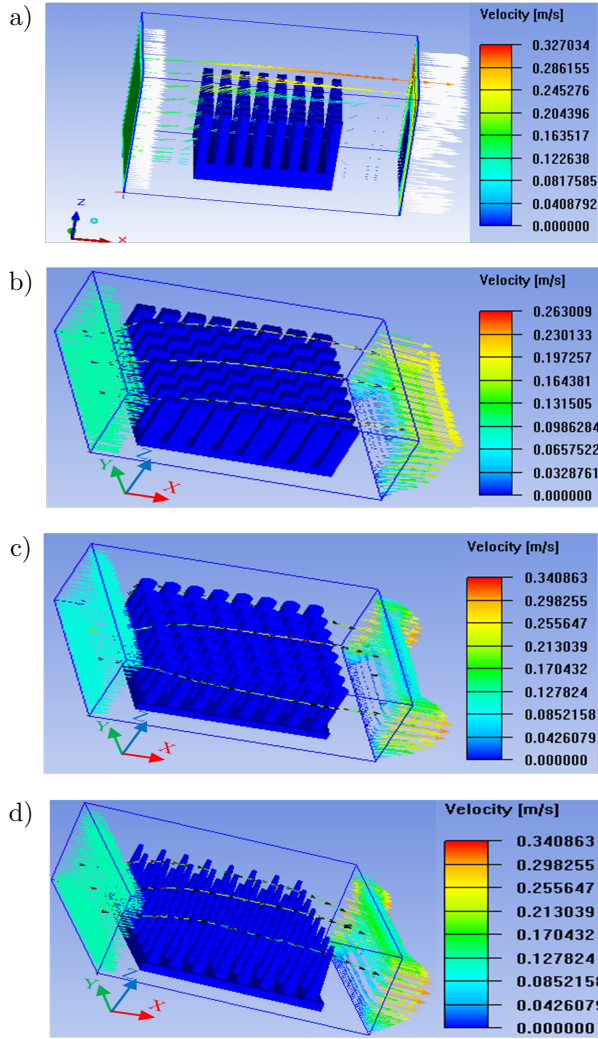


FIG. 13. Velocity distribution over the different types of fins circular fins: a) square fins, b) rectangular fins, c) circular fins, d) conical fins.

Circular fins demonstrate approximately a 29.28 % increase in peak airflow velocity over rectangular fins, a 3.98 % increase over square fins, and a 25.0 % increase over conical fins. This indicates that their rounded shape may facilitate a more streamlined airflow path, contributing to the acceleration of air through the fin array. Rectangular fins exhibit slightly reduced velocity, signifying potentially smaller airflow turbulence, while conical fins display a moderate velocity profile. Nevertheless, the high velocities observed with circular and square fins do not necessarily translate into lower temperatures, underscoring that merely augmenting airflow velocity does not invariably enhance heat sink performance.

Instead, heat transfer efficiency also hinges on the available surface area and the behavior of the thermal boundary layer.

The overarching objective is to optimize heat dissipation while concurrently minimizing both aerodynamic drag and the power needed to sustain airflow, which is a pivotal consideration for the overall performance and energy efficiency of cooling systems in electronic devices. Combining data on velocity, pressure, and temperature furnishes a holistic perspective on cooling performance. Although conical fins offer the lowest pressure drop and moderate velocities, they lead to the highest temperatures, suggesting that their shape may not be the most suitable for thermal management, despite favorable aerodynamic attributes. In contrast, rectangular fins strike an optimal equilibrium, with moderate velocities corresponding to both lower temperatures and pressure drops, signifying their superior suitability for cooling applications. These findings underscore the necessity of an integrated approach that takes into account all three parameters in designing efficient heat sinks. The ultimate aim is to maximize heat dissipation while simultaneously minimizing both aerodynamic resistance and power consumption, which are critical for the overall performance and energy efficiency of cooling systems in electronic devices.

3.5. AVERAGE NUSSLETT NUMBER

The Nusselt number (Nu_{avg}) is calculated for different types of fins, and the results are listed in [Table 4](#).

TABLE 4. Average Nusselt number for each type of fins.

Type of fin	Average Nusselt number
Rectangular	10.0380
Longitudinal	9.1690
Square	8.7390
Circular	7.8870
Conical	6.1340

Given the average Nusselt numbers for the different types of fins presented in [Table 4](#), their thermal performance can be interpreted as follows:

1. Rectangular fins ($Nu_{avg} = 10.0380$) exhibit the highest Nusselt number, indicating superior convective heat transfer efficiency. This suggests that the rectangular fins have the most effective surface area and geometry for promoting both conductive and convective heat transfer. The large surface area of rectangular fins facilitates better heat dissipation from the fin surface to the surrounding air, maximizing thermal performance.
2. Longitudinal fins ($Nu_{avg} = 9.1690$) have a slightly lower Nusselt number than rectangular fins. They still provide efficient heat transfer, but

their geometry may not promote as much airflow interaction or provide as much surface area as rectangular fins. Nevertheless, they are still quite effective in enhancing heat transfer compared to simpler or less optimized geometries.

3. Square fins ($Nu_{avg} = 8.7390$) demonstrate a good level of heat transfer, but smaller than that of rectangular and longitudinal fins. The edges and geometry of square fins might cause some flow separation or less optimal airflow patterns compared to the more streamlined shapes, reducing their overall efficiency.
4. Circular fins ($Nu_{avg} = 7.8870$), while promoting high airflow velocity, have a lower Nusselt number, indicating less effective convective heat transfer. The round shape may lead to a reduced surface contact area for heat transfer compared to flat or elongated shapes, which can diminish the overall heat dissipation capability.
5. Conical fins ($Nu_{avg} = 6.1340$) have the lowest Nusselt number, signifying the least effective convective heat transfer. The tapered shape reduces the surface area available for heat transfer as the air moves along the fin, limiting their thermal performance. Additionally, the pointed tips may not contribute significantly to heat dissipation, further decreasing their efficiency.

3.5.1. OVERALL INTERPRETATION

Rectangular fins are the most efficient for heat transfer, making them ideal for applications requiring maximum cooling efficiency. Longitudinal and square fins also perform well, making them suitable for situations where slightly lower performance can be tolerated or where geometric constraints favor their use. Circular fins have lower efficiency but might be preferred in scenarios where airflow velocity needs to be maximized or where design constraints require a streamlined shape. Conical fins are the least efficient in terms of heat transfer, and their use would therefore be more specialized or limited to applications where other factors (such as minimal pressure drop) are more critical than thermal performance.

In conclusion, the physical interpretation of the Nusselt number for each fin type provides insight into their relative effectiveness for heat dissipation in electronic cooling applications, guiding the design and selection of heat sinks based on specific thermal management needs.

4. COMPARISONS AND DISCUSSION

This section presents a comparison of the various fin geometries assessed in this study based on the results of the analyses, including longitudinal, square, rectangular, circular, and conical fin shapes, with a specific emphasis on their performance across three key performance metrics: maximum temperature re-

duction, pressure drop, and airflow velocity. Furthermore, [Table 5](#) offers a concise summary of this comparative analysis based on the results of the analyses.

TABLE 5. Comparative ranking of fin geometries.

Performance indicator	1st (best)	2nd	3rd	4th	5th (worst)
Maximum temperature reduction (higher reduction is better)	Rectangular	Square	Longitudinal	Circular	Conical
Pressure drop (lower is better)	Conical	Rectangular	Circular	Square	Longitudinal
Peak airflow velocity (descending magnitude; not an overall-performance ranking)	Circular	Square	Conical	Rectangular	Longitudinal
Overall cooling performance (thermal performance with hydraulic consideration)	Rectangular	Square	Circular/ conical/ longitudinal	–	–

4.1. MAXIMUM TEMPERATURE REDUCTION

It can be noticed that the rectangular fins demonstrated the highest efficacy in reducing heat sink temperatures, achieving reductions of 8.4 % compared to longitudinal fins, 5.3 % compared to square fins, 14.0 % compared to circular fins, and 15.1 % compared to conical fins. This high performance can be attributed to their larger surface area, which facilitates both conductive and convective heat transfer. At the same time, longitudinal fins exhibited a moderate level of performance in terms of temperature reduction. Square fins offered temperature control that was close to that of longitudinal fins, but they were slightly less efficient than rectangular fins. Although circular fins exhibited high airflow velocity, they were less effective in reducing temperature, possibly due to suboptimal surface area contact with the surrounding air. Conical fins displayed the lowest effectiveness in temperature control, likely because of their reduced surface area at the tips, which hindered efficient heat dissipation.

4.2. PRESSURE DROP

It can be noticed that the conical fins stood out with a significant reduction in pressure drop, indicating more streamlined airflow. They achieved reductions

of 38.1 % compared to square fins, 21.0 % compared to rectangular fins, and 36.6 % compared to circular fins. Rectangular fins offered a well-balanced pressure drop, neither too high to cause excessive energy consumption for airflow nor too low to compromise effective heat transfer. Both square and circular fins experienced higher pressure drops, suggesting potential airflow resistance due to their respective shapes.

4.3. AIRFLOW VELOCITY

It can be noticed that the circular fins demonstrated the highest increase in airflow velocity, implying that their shape promotes streamlined airflow. They showcased a 29.3 %, 4.0 %, and 25.0 % increase in peak airflow velocity compared to the rectangular, square, and conical fin geometries, respectively. Conical fins exhibited a moderate airflow velocity, aligning with their reduced pressure drop and aerodynamic design. Rectangular fins displayed a balanced airflow velocity, conducive to effective cooling without excessive energy consumption. Square fins showed slightly lower airflow velocities than circular fins, possibly due to airflow disturbances caused by their sharp edges.

4.4. DISCUSSION

This comprehensive analysis yields crucial insights into how the geometry of fins influences the thermal management of heat sinks. It emphasizes the necessity of considering multiple factors, including surface area, airflow dynamics, and pressure characteristics, when selecting a fin geometry for heat sinks, particularly in applications demanding efficient cooling in compact, high-powered electronic devices.

The findings from this study can serve as a valuable guide for designing and choosing heat sinks in various electronic cooling applications, striking a balance between effective temperature control and considerations of aerodynamic efficiency and energy efficiency.

Given these performance metrics, rectangular fins stand out as the optimal choice for such applications. Their ability to reduce maximum temperatures surpasses that of the other fin shapes, demonstrating their effectiveness in thermal management. Moreover, their balanced airflow velocity and moderate pressure drop indicate their efficiency in maintaining aerodynamic stability while minimizing energy consumption. This makes rectangular fins particularly well-suited for high-performance electronic cooling applications, where optimizing heat dissipation and energy efficiency are crucial. Their versatility and effectiveness under various operating conditions reaffirm their status as a preferred choice for advanced electronic cooling solutions.

5. CONCLUSION AND FUTURE WORK

In this study, the effect of fin geometry on heat sink performance was investigated. The evaluation of the cooling effectiveness of each heat sink relied on comprehensive performance indicators, including maximum temperature reduction, pressure drop, and airflow velocity distribution across the fins. These metrics collectively contributed to the assessment of each design's thermal performance. The main outcomes of this investigation are as follows:

1. Fin geometry is a key determinant of the heat sink temperature, pressure drop, and airflow velocity, crucially affecting its thermal and aerodynamic efficiency.
2. Rectangular fins outperform longitudinal, square, circular, and conical geometries in reducing the maximum temperature by 8.4 %, 5.3 %, 14.0 %, and 15.1 % respectively. This suggests a more efficient heat dissipation capability, striking the best balance between conductive and convective heat transfer.
3. Conical fins exhibit a 38.1 %, 21.0 %, and 36.6 % reduction in pressure drop compared to square, rectangular, and circular fin geometries, respectively, indicating a more streamlined airflow through the fin array and reduced aerodynamic resistance.
4. Circular fins demonstrate a 29.3 %, 4.0 %, and 25.0 % increase in peak airflow velocity over rectangular, square, and conical fin geometries, respectively, implying that their shape may promote a more streamlined airflow path, accelerating air passage through the fin array.
5. The comprehensive analysis of temperature, pressure, and airflow velocity data reveals that rectangular fins offer the best balance, achieving optimal heat dissipation with efficient aerodynamics and lower energy requirements, thus emerging as the most effective design for heat sink applications.
6. Balancing thermal resistance to enhance heat transfer efficiency while reducing aerodynamic drag to minimize pressure drops and increase airflow velocity is crucial for the optimization of cooling system performance.

In future work, it is crucial to investigate the effects of varying environmental conditions, including changes in ambient temperature and airflow patterns, to gain deeper insight into heat sink performance. The utilization of advanced computational models capable of simulating dynamic thermal loads and real-world operational scenarios is pivotal for further heat sink design refinement. Additionally, experimenting with innovative fin geometries, potentially drawing inspiration from biomimicry or novel engineering concepts, could unlock new possibilities for managing thermal concerns in compact, high-powered electronic devices. It is also recommended to study hybrid or irregular geometric shapes

of fins, or combinations of two or more types of fins to investigate/evaluate the effect of these combinations on the performance of heat sinks.

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CONFLICT OF INTERESTS

The authors declare that there are no known competing financial interests or personal relationships that could have influenced the work described in this paper.

AUTHORS' CONTRIBUTIONS

All authors contributed equally to the paper, reviewed and approved the final manuscript.

REFERENCES

1. LUO W. *et al.*, Recent progress in quantum photonic chips for quantum communication and internet, *Light: Science & Applications*, **12**(1): 175, 2023, <https://doi.org/10.1038/s41377-023-01173-8>.
2. AL-OMARI S.A.B., QURESHI Z.A., ELNAJJAR E., MAHMOUD F., A heat sink integrating fins within high thermal conductivity phase change material to cool high heat-flux heat sources, *International Journal of Thermal Sciences*, **172**(Part A): 107190, 2022, <https://doi.org/10.1016/j.ijthermalsci.2021.107190>.
3. GAIKWAD A., SATHE A., SANAP S., A design approach for thermal enhancement in heat sinks using different types of fins: A review, *Frontiers in Thermal Engineering*, **2**: 980985, 2023, <https://doi.org/10.3389/ftther.2022.980985>.
4. BLINOV A., VINNIKOV D., LEHTLA T., Cooling methods for high-power electronic systems, *Scientific Journal of Riga Technical University. Power and Electrical Engineering*, **29**(1): 79–86, 2011, <https://doi.org/10.2478/v10144-011-0014-x>.
5. FALLAHI A., GULDENTOPS G., TAO M., GRANADOS-FOCIL S., VAN DESSEL S., Review on solid-solid phase change materials for thermal energy storage: molecular structure and thermal properties, *Applied Thermal Engineering*, **127**: 1427–1441, 2017, <https://doi.org/10.1016/j.applthermaleng.2017.08.161>.
6. EIVARI H.A., SOHBATZADEH Z., MELE P., ASSADI M.H.N., Low thermal conductivity: fundamentals and theoretical aspects in thermoelectric applications, *Materials Today Energy*, **21**: 100744, 2021, <https://doi.org/10.1016/j.mtener.2021.100744>.
7. YANG T., KING W.P., MILJKOVIC N., Phase change material-based thermal energy storage, *Cell Reports Physical Science*, **2**(8): 100540, 2021, <https://doi.org/10.1016/j.xcrp.2021.100540>.

8. BEN SALAH S., BEN HAMIDA M.B., Alternate PCM with air cavities in LED heat sink for transient thermal management, *International Journal of Numerical Methods for Heat & Fluid Flow*, **29**(11): 4377–4393, 2019, <https://doi.org/10.1108/HFF-02-2019-0099>.
9. IZADI M., FAGEHI H., IMANZADEH A., ALTNJI S., BECHIR BEN HAMIDA M., SHEREMET M.A., Influence of finned charges on melting process performance in a thermal energy storage, *Thermal Science and Engineering Progress*, **37**: 101547, 2023, <https://doi.org/10.1016/j.tsep.2022.101547>.
10. BEN HAMIDA M.B., HAJLAOUI K., ALMESHAAL M.A., A 3D numerical analysis using phase change material for cooling circular light emitting diode, *Case Studies in Thermal Engineering*, **43**: 102792, 2023, <https://doi.org/10.1016/j.csite.2023.102792>.
11. SARAVANAN V., HITHAISH D., UMESH C.K., SEETHARAMU K.N., Numerical investigation of thermo-hydrodynamic performance of triangular pin fin heat sink using nano-fluids, *Thermal Science and Engineering Progress*, **21**: 100768, 2021, <https://doi.org/10.1016/j.tsep.2020.100768>.
12. MASSOUDI M.D., BEN HAMIDA M.B., ALMESHAAL M.A., HAJLAOUI K., Numerical evaluation of MHD SWCNT-water nanoliquid performance in cooling an electronic heat sink featuring twisted hexagonal fins considering thermal emission impact: comparison between various fins shapes, *Sustainable Energy Technologies and Assessments*, **53**(Part A): 102350, 2022, <https://doi.org/10.1016/j.seta.2022.102350>.
13. MASSOUDI M.D., BEN HAMIDA M.B., Enhancement of MHD radiative CNT-50% water + 50% ethylene glycol nanoliquid performance in cooling an electronic heat sink featuring wavy fins, *Waves in Random and Complex Media*, **35**(6): 11357–11382, 2025, <https://doi.org/10.1080/17455030.2022.2122626>.
14. MASSOUDI M.D., BEN HAMIDA M.B., ALMESHAAL M.A., HAJLAOUI K., Effects of L-shaped fins on cooling an electronic heat sink fitted under magnetic field of CNT-water/ethylene glycol nanoliquid, *The European Physical Journal Plus*, **137**(7): 843, 2022, <https://doi.org/10.1140/epjp/s13360-022-03044-4>.
15. BEN HAMIDA M.B., HATAMI M., Investigation of heated fins geometries on the heat transfer of a channel filled by hybrid nanofluids under the electric field, *Case Studies in Thermal Engineering*, **28**: 101450, 2021, <https://doi.org/10.1016/j.csite.2021.101450>.
16. MASSOUDI M.D., BEN HAMIDA M.B., Combined impacts of square fins fitted wavy wings and micropolar magnetized-radiative nanofluid on the heat sink performance, *Journal of Magnetism and Magnetic Materials*, **574**: 170655, 2023, <https://doi.org/10.1016/j.jmmm.2023.170655>.
17. BEN HAMIDA M.B., Thermal management of square light emitting diode arrays: modeling and parametric analysis, *Multidiscipline Modeling in Materials and Structures*, **20**(2): 363–383, 2024, <https://doi.org/10.1108/MMMS-09-2023-0311>.
18. SHAFEIE H., ABOUALI O., JAFARPUR K., AHMADI G., Numerical study of heat transfer performance of single-phase heat sinks with micro pin-fin structures, *Applied Thermal Engineering*, **58**(1–2): 68–76, 2013, <https://doi.org/10.1016/j.applthermaleng.2013.04.008>.
19. YEOM T., SIMON T., ZHANG T., ZHANG M., NORTH M., CUI T., Enhanced heat transfer of heat sink channels with micro pin fin roughened walls, *International Journal of Heat and Mass Transfer*, **92**: 617–627, 2016, <https://doi.org/10.1016/j.ijheatmasstransfer.2015.09.014>.

20. AMBREEN T., SALEEM A., PARK C.W., Numerical analysis of the heat transfer and fluid flow characteristics of a nanofluid-cooled micropin-fin heat sink using the Eulerian-Lagrangian approach, *Powder Technology*, **345**: 509–520, 2019, <https://doi.org/10.1016/j.powtec.2019.01.042>.
21. ARSHAD W., ALI H.M., Experimental investigation of heat transfer and pressure drop in a straight minichannel heat sink using TiO₂ nanofluid, *International Journal of Heat and Mass Transfer*, **110**: 248–256, 2017, <https://doi.org/10.1016/j.ijheatmasstransfer.2017.03.032>.
22. ARSHAD W., ALI H.M., Graphene nanoplatelets nanofluids thermal and hydrodynamic performance on integral fin heat sink, *International Journal of Heat and Mass Transfer*, **107**: 995–1001, 2017, <https://doi.org/10.1016/j.ijheatmasstransfer.2016.10.127>.
23. JALIL J.M., REJA A.H., HADI A.M., Numerical investigation of thermal performance of micro-pin fin with different arrangements, [in:] *IOP Conference Series: Materials Science and Engineering*, **765**: 012037, 2020, <https://doi.org/10.1016/10.1088/1757-899X/765/1/012037>.
24. OZBALCI O., DOGAN A., ASILTURK M., Heat Transfer performance of plate fin and pin fin heat sinks using Al₂O₃/H₂O nanofluid in electronic cooling, *Processes*, **10**(8): 1644, 2022, <https://doi.org/10.3390/pr10081644>.
25. EKPUR M., Effect of fins arrangement on thermal performance in microelectronics devices, *Journal of Applied Sciences and Environmental Management*, **22**(11): 1797–1800, 2018, <https://doi.org/10.4314/jasem.v22i11.14>.
26. TANDA G., Heat transfer and pressure drop in a rectangular channel with diamond-shaped elements, *International Journal of Heat and Mass Transfer*, **44**(18): 3529–3541, 2001, [https://doi.org/10.1016/S0017-9310\(01\)00018-7](https://doi.org/10.1016/S0017-9310(01)00018-7).
27. DEBICH B., EL HAMI A., YAICH A., GAFSI W., WALHA L., HADDAR M., Design optimization of PCM-based finned heat sinks for mechatronic components: a numerical investigation and parametric study, *Journal of Energy Storage*, **32**: 101960, 2020, <https://doi.org/10.1016/j.est.2020.101960>.
28. ALI H.M. *et al.*, Advances in thermal energy storage: fundamentals and applications, *Progress in Energy and Combustion Science*, **100**: 101109, 2024, <https://doi.org/10.1016/j.pecs.2023.101109>.
29. SHATIKIAN V., ZISKIND G., LETAN R., Numerical investigation of a PCM-based heat sink with internal fins, *International Journal of Heat and Mass Transfer*, **48**(17): 3689–3706, 2005, <https://doi.org/10.1016/j.ijheatmasstransfer.2004.10.042>.
30. JAWORSKI M., Thermal performance of heat spreader for electronics cooling with incorporated phase change material, *Applied Thermal Engineering*, **35**: 212–219, 2012, <https://doi.org/10.1016/j.applthermaleng.2011.10.036>.
31. KAMKARI B., SHOKOUHMAND H., Experimental investigation of phase change material melting in rectangular enclosures with horizontal partial fins, *International Journal of Heat and Mass Transfer*, **78**: 839–851, 2014, <https://doi.org/10.1016/j.ijheatmasstransfer.2014.07.056>.
32. PAKROUH R., HOSSEINI M.J., RANJBAR A.A., BAHRAMPOURY R., A numerical method for PCM-based pin fin heat sinks optimization, *Energy Conversion and Management*, **103**: 542–552, 2015, <https://doi.org/10.1016/j.enconman.2015.07.003>.

33. BONDAREVA N.S., SHEREMET M.A., 3D natural convection melting in a cubical cavity with a heat source, *International Journal of Thermal Sciences*, **115**: 43–53, 2017, <https://doi.org/10.1016/j.ijthermalsci.2017.01.021>.
34. ALI H.M., ASHRAF M.J., GIOVANNELLI A., IFRAN M., IRSHAD T.B., HAMID H.M., HASSAN F., ARSHAD A., Thermal management of electronics: an experimental analysis of triangular, rectangular and circular pin-fin heat sinks for various PCMs, *International Journal of Heat and Mass Transfer*, **123**: 272–284, 2018, <https://doi.org/10.1016/j.ijheatmasstransfer.2018.02.044>.
35. SENTHILKUMAR R., PRABHU S., CHERALATHAN M., Experimental investigation on carbon nano tubes coated brass rectangular extended surfaces, *Applied Thermal Engineering*, **50**(1): 1361–1368, 2013, <https://doi.org/10.1016/j.applthermaleng.2012.05.040>.
36. JI C., QIN Z., DUBEY S., CHOO F.H., DUAN F., Simulation on PCM melting enhancement with double-fin length arrangements in a rectangular enclosure induced by natural convection, *International Journal of Heat and Mass Transfer*, **127**(Part C): 255–265, 2018, <https://doi.org/10.1016/j.ijheatmasstransfer.2018.07.118>.
37. XIE J., CHOO K.F., XIANG J., LEE H.M., Characterization of natural convection in a PCM-based heat sink with novel conductive structures, *International Communications in Heat and Mass Transfer*, **108**: 104306, 2019, <https://doi.org/10.1016/j.icheatmasstransfer.2019.104306>.
38. YASEEN S.J., RADHI Z.K., NATOOSH R.L., AL-SABUR R., HOMOD R.Z., MOHAMMED H.I., A computational search for the optimal microelectronic heat sink using ANSYS Icepak, *International Journal of Thermofluids*, **23**: 100759, 2024, <https://doi.org/10.1016/j.ijft.2024.100759>.
39. ROSLI R., MOHD ANNUAR K.A., ISMAIL F.S., Optimal pin fin heat sink arrangement for solving thermal distribution problem, *Journal of Advanced Research in Fluid Mechanics and Thermal Sciences*, **11**(1): 1–18, 2015.
40. YASEEN S.J., Numerical study of the fluid flow and heat transfer in a finned heat sink using Ansys Icepak, *Open Engineering*, **13**(1): 20220440, 2023, <https://doi.org/10.1515/eng-2022-0440>.
41. AL-SAAD M., YASSEN S.J., SUAREZ-AFANADOR C., ALSHARA A.K., CHAMKHA A.J., Simulation of blood flow in human arteries as porous media, *Waves in Random and Complex Media*, **36**(1): 1192–1202, 2026, <https://doi.org/10.1080/17455030.2022.2162151>.
42. RADHI Z.K., YASEEN S.J., ALSAHLANI A.A., AL-SABUR R., Exploring magneto-hydrodynamic influence on mixed convection within a vented enclosure containing a heat-conductive square column, *International Journal of Mechanical Engineering and Robotics Research*, **13**(1): 75–84, 2024, <https://doi.org/10.18178/ijmerr.13.1.75-84>.
43. ADHIKARI R.C., WOOD D.H., PAHLEVANI M., An experimental and numerical study of forced convection heat transfer from rectangular fins at low Reynolds numbers, *International Journal of Heat and Mass Transfer*, **163**: 120418, 2020, <https://doi.org/10.1016/j.ijheatmasstransfer.2020.120418>.
44. HU W.L., MA A.J., GUAN Y., CUI Z.J., ZHANG Y.B., WANG J., Experimental study of the air side performance of fin-and-tube heat exchanger with different fin material in dehumidifying conditions, *Energies*, **14**(21): 7030, 2021, <https://doi.org/10.3390/en14217030>.

45. WEI W., XU G., WANG Y., DING E., Study on air-side performance of air-cooled heat exchangers under large air velocity and wet conditions, *Thermal Science and Engineering Progress*, **48**: 102389, 2024, <https://doi.org/10.1016/j.tsep.2024.102389>.
46. WATBAN KHALID FAHMI A.T., REZA KASHYZADEH K., GHORBANI S., A comprehensive review on mechanical failures cause vibration in the gas turbine of combined cycle power plants, *Engineering Failure Analysis*, **134**: 106094, 2022, <https://doi.org/10.1016/j.engfailanal.2022.106094>.

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