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CONFERENCE PAPER

NUMERICAL SIMULATION OF OPTIMIZED HEAT SINK DESIGNS FOR ENHANCED THERMAL MANAGEMENT OF ELECTRONICS

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Abstract

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Modern electronic devices pack increasing amounts of power into shrinking form factors, and the heat they generate remains a primary cause of premature failure and performance degradation. Among the various thermal management strategies available, passive heat sinks remain a go-to solution because they require no external power and require virtually no maintenance. In this work, two rectangular-fin heat sink layouts are examined side by side: a baseline design with 10 fins and a modified variant with 12 thinner, taller, and more closely spaced fins. Each geometry is evaluated in both aluminum and copper, producing four distinct configurations. Three-dimensional steady-state thermal simulations are carried out in ANSYS Workbench under a uniform base heat flux of 4 W/cm², a convective coefficient of 80 W/m²·K across all exposed fin surfaces, and an ambient temperature of 32°C. The heat flux magnitude was deliberately chosen as the highest value that still keeps peak base temperatures below the 100°C safety threshold across every configuration. Simulation results indicate that the modified geometry consistently outperforms the baseline: base temperatures range from 75.1°C (copper) to 79.5°C (aluminum), compared with 93.0°C and 96.0°C for the conventional layout. The modified designs also exhibit superior fin effectiveness, though their fin efficiencies are somewhat lower owing to the taller fin profile. On balance, the modified aluminum variant emerges as the most practical choice, delivering strong thermal performance while remaining lightweight and cost-effective.

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Introduction:-

As electronic components continue to shrink while handling higher power densities, managing the heat they generate has become an unavoidable engineering challenge. Microprocessors, power transistors, and high-brightness LEDs all concentrate significant thermal energy into small areas, and if that energy is not drawn away quickly enough,

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junction temperatures climb, performance suffers, and components can fail long before their rated lifetimes [1]. This reality has elevated thermal design from an afterthought to a central pillar of modern electronics development.Passive heat sinks remain one of the most dependable answers to this problem. The operating principle is straightforward: a thermally conductive base plate contacts the heat source, and an array of extended fins radiates that heat into the surrounding air. Because there are no fans, pumps, or thermoelectric elements involved, these devices are inherently reliable and add almost nothing to operating costs [2]. Their performance, however, hinges on a handful of interrelated factors: the thermal conductivity of the chosen material, the fins' geometry and surface area, and the airflow characteristics over them [3].

Material selection typically boils down to aluminum versus copper. Aluminum is the workhorse: it is inexpensive, easy to machine, and light enough for applications where weight matters. Copper, with a thermal conductivity roughly 1.9 times that of aluminum (about 385 W/m·K compared to 205 W/m·K), spreads heat more uniformly across the fin surfaces, but its higher density and cost often relegate it to applications where every degree counts [4]. Beyond material choice, the geometric parameters of the fins, height, thickness, spacing, and base plate dimensions, exert a strong influence on how much heat the sink can ultimately reject. Simulation tools such as ANSYS Workbench enable the computational exploration of these design variables, saving considerable time and money compared to building and testing physical prototypes [5].

The theoretical foundation for fin-based heat dissipation is well established. Bergman et al. [1] provide a comprehensive treatment of conduction and convection principles that underpin every aspect of heat sink analysis, from the one-dimensional fin equation to multidimensional numerical solutions. Building on these fundamentals, Bar-Cohen and Wang [2] demonstrated that structured microchannel fin arrays can yield dramatic improvements in local heat transfer coefficients, particularly near concentrated hot spots on a chip surface.Geometry-driven optimization has received extensive attention. Kakaç et al. [3] surveyed a wide range of surface configurations for heat exchangers. They concluded that carefully tailored fin layouts can substantially cut thermal resistance without a proportional increase in pumping power. In a related vein, Arularasan [7] used computational fluid dynamics to study rectangular fin arrays and observed that increasing fin height while tightening the spacing generally improved thermal performance, though at the expense of higher pressure drop across the array.Interface effects also matter. Yovanovich [4] demonstrated, over four decades of experimental and analytical work, that thermal contact resistance between a heat source and its sink can become the dominant bottleneck if the mating surfaces are not properly prepared, underscoring the importance of thermal interface materials in practice.

On the broader question of cooling strategy, Mjallal and Farhat [5] benchmarked several active and passive approaches and concluded that, for moderate heat loads, passive heat sinks remain competitive due to their simplicity and negligible operating costs. The numerical methods used in all of these computational studies owe a debt to Patankar [6], whose finite-volume framework for coupled fluid-flow and heat-transfer problems remains the backbone of most commercial CFD solvers.A recurring limitation in much of the existing literature is the choice of thermal boundary conditions. Many studies either prescribe a fixed temperature at the heat sink base or impose a lumped total heat input without tying it explicitly to a surface heat flux. In practice, however, a chip dissipates a known power over a known footprint, making a heat flux condition the more physically faithful representation. The present work adopts this approach and calibrates the applied flux to the highest value that produces physically realistic temperatures across all four design-material combinations under investigation.

A noteworthy aspect of this study is the choice of boundary conditions. Rather than fixing the base temperature at an assumed value, a common simplification in academic work, a uniform heat flux of 4 W/cm² is imposed on the base plate that better reflects real operating conditions, where the chip's power output is a known design parameter. Still, the resulting temperature is precisely what the engineer needs to determine. The 4 W/cm² figure was selected because it represents the upper limit at which all four configurations tested still maintain base temperatures below 100°C. The study pursues two primary objectives: first, to develop three-dimensional models of a conventional and a modified heat sink design, and second, to compare their steady-state thermal behavior in terms of temperature profiles, fin effectiveness, and fin efficiency.

Materials and Methods:-

This section outlines the geometric modeling, material properties, mesh strategy, and boundary conditions used in the simulations. All models were built in three-dimensional CAD software and imported into ANSYS Workbench for steady-state thermal analysis.

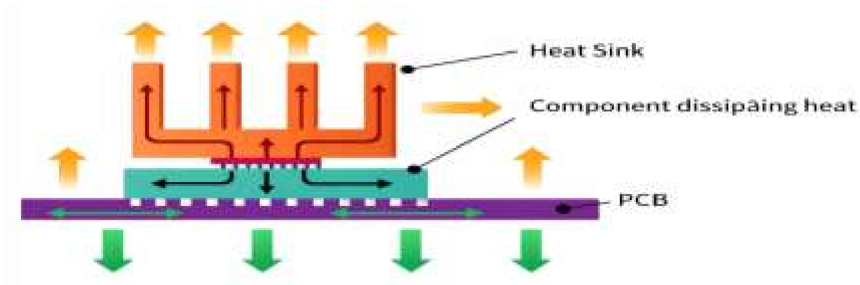

Fig. 1. Typical finned heat sink assembly used in electronic cooling applications.

Geometric Models (Fig 2):-

Conventional design: The baseline heat sink consists of ten rectangular fins, each 2 mm thick and 30 mm tall, mounted on a square base plate measuring 80 × 80 mm.

Modified design: The revised layout increases the fin count to 12, reduces fin thickness to 1.5 mm, and raises fin height to 40 mm. The base plate dimensions remain unchanged. These adjustments collectively increase the total fin surface area and reduce the gap between adjacent fins, both of which favor heat dissipation.Both geometries were analyzed in two materials, aluminum (k = 205 W/m·K, ρ = 2,700 kg/m³) and copper (k = 385 W/m·K, ρ = 8,960 kg/m³), yielding a total of four configurations for comparison.

Mesh Generation and Boundary Conditions:-

Each geometry was discretized using hexahedral elements, which provide a good balance between accuracy and computational cost for regular block-like domains. The resulting mesh comprised approximately 12,247 elements and 21,593 nodes (Fig. 2).

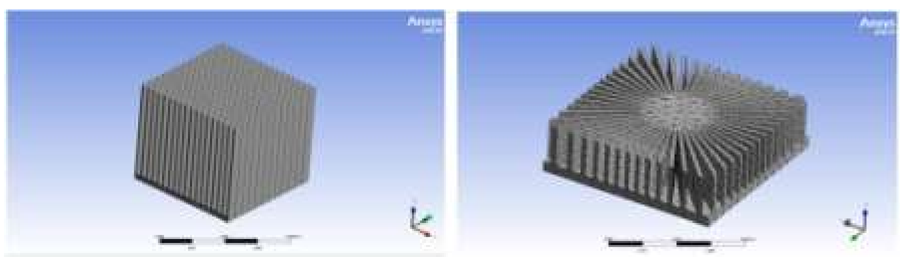

Fig. 2. Hexahedral mesh generated for the conventional and modified heat sink models.

Rather than fixing the base at a predetermined temperature, a uniform heat flux q'' was applied to the bottom face of the base plate. This reflects how electronic components actually operate: the chip's power dissipation is a known quantity, but the resulting temperature is not, and finding that temperature is the whole point of the simulation. To determine an appropriate flux level, the total thermal resistance of each configuration was first calculated analytically. The conventional aluminum design exhibits the highest resistance (R_{total} = 0.250 K/W), which means it is the thermal bottleneck. Setting the maximum allowable base temperature at 100°C and the ambient at 32°C gives a headroom of 68°C, yielding Q_{max} = 68/0.250 = 272 W, or about 4.25 W/cm² over the 64 cm² base. Rounding down to 4 W/cm² provides a safety margin that keeps every configuration comfortably below the ceiling. Table 1 summarises the maximum permissible flux for each variant.

Table 1. Maximum allowable heat flux for each configuration (T_{base} ≤ 100°C, h = 80 W/m²·K, T_a = 32°C)

Configuration	R_total (K/W)	Q_max (W)	q_max (W/cm²)
Conv. Aluminum	0.2500	272	4.25
Conv. Copper	0.2384	285	4.46
Mod. Aluminum	0.1856	366	5.73
Mod. Copper	0.1685	404	6.31

The boundary conditions applied uniformly to all four models were as follows:
Heat flux: q″ = 4 W/cm² (40,000 W/m²) over the 80 × 80 mm base, delivering a total heat input of Q = 256 W.
Convection: h = 80 W/m²·K on all exposed fin surfaces, representative of moderate forced-air cooling in an ambient environment.
Ambient temperature: T_a = 32°C.

Results and Discussion:-

The complete set of simulation outputs for all four configurations is collected in Table 2. The subsections that follow examine each performance metric individually.

Table 2. Simulation results for all configurations (q″ = 4 W/cm², h = 80 W/m²·K, T_a = 32°C)

Configuration	T_{base} (°C)	T_{tip} (°C)	η (%)	ε
Conv. Aluminum	96.01	85.44	88.9	7.81
Conv. Copper	93.03	87.29	93.7	8.19
Mod. Aluminum	79.51	64.26	78.3	10.52
Mod. Copper	75.14	66.70	86.8	11.59

Simulation Results: Contour Plots:-
Temperature and heat flux contour plots from the ANSYS steady-state solver are presented below for the aluminum variants. These provide a visual snapshot of how thermal energy distributes itself through each geometry.

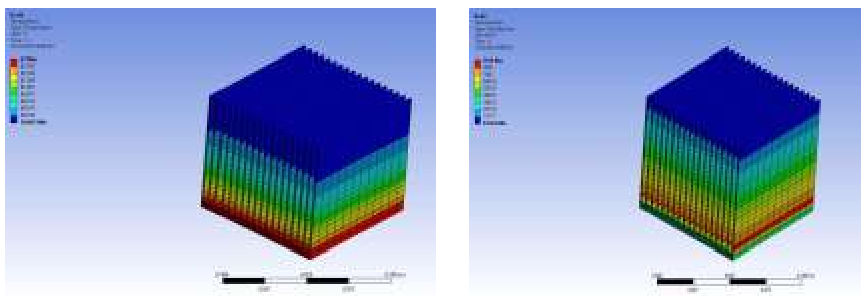

Fig. 3. Conventional Al heat sink: (a) temperature distribution, (b) heat flux contour.

In the conventional aluminum model, the base stabilized at 96.0°C while the fin tips settled at 85.4°C. The heat flux contour reveals a peak of approximately 160 kW/m² near the fin roots, which is expected: this is where the full cross-section of the fin must carry the conducted heat before it begins to spread out along the fin's length.

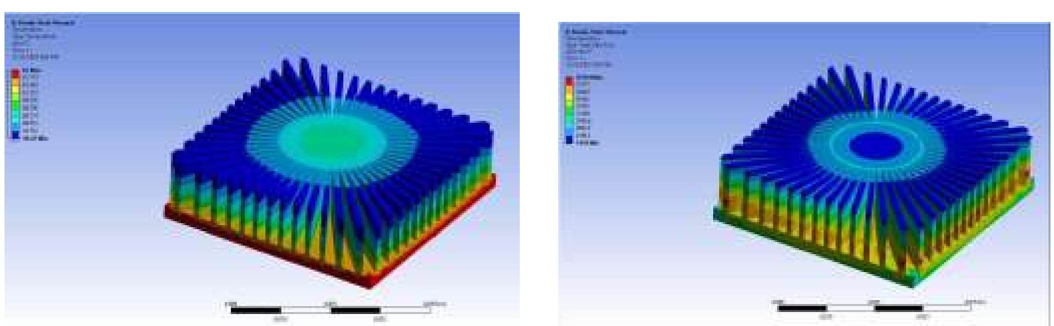

Fig. 4. Modified Al heat sink: (a) temperature distribution, (b) heat flux contour.

The modified aluminum model tells a markedly different story. The base temperature fell to 79.5°C, and the fin tips reached only 64.3°C, reductions of 16.5°C and 21.1°C, respectively, compared to the conventional layout. The peak heat flux at the fin roots was slightly higher, around 178 kW/m², a consequence of the denser fin packing that concentrates heat flow into each root cross-section.

Temperature Distribution: Conventional Model:-

Fig. 5 plots temperature as a function of position along the fin for both materials in the conventional layout. In both cases, there is a steady decline from the base to the tip, which is the expected behavior for a conductively cooled extended surface. The aluminum variant registered a base temperature of 96.0°C and a tip temperature of 85.4°C, producing a gradient of about 10.6°C across the fin length. Copper fared slightly better at the base (93.0°C), but its tip was warmer (87.3°C), yielding a much shallower gradient of only 5.7°C. That smaller drop is a direct fingerprint of copper's superior conductivity: heat spreads more evenly along the fin, so the tip stays closer to the base temperature. Crucially, both base temperatures sit below the 100°C threshold, confirming that the selected heat flux is appropriate for both materials.

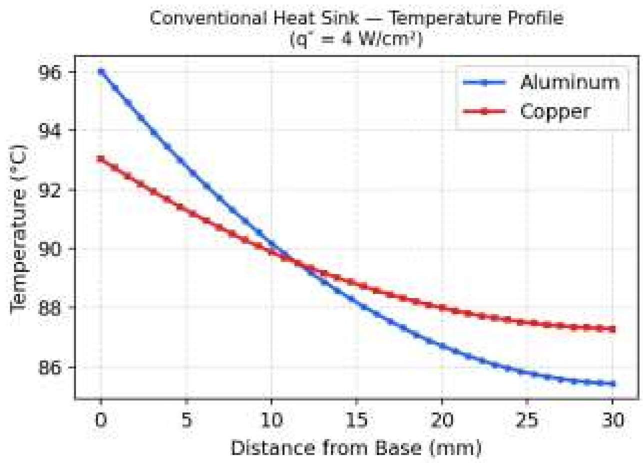

Fig. 5. Temperature along fin length for the conventional heat sink model (q″ = 4 W/cm²).

Temperature Distribution: Modified Model:-

Fig. 6 presents the analogous profiles for the modified design. The combination of additional fins, reduced thickness, and increased height lowers the array's overall thermal resistance, with a pronounced effect on temperatures. The aluminum variant dropped to a base of 79.5°C and a tip of 64.3°C, representing reductions of roughly 16.5°C at the base and 21.1°C at the tip relative to the conventional aluminum design. The copper variant achieved the lowest temperatures across all four cases, with a base temperature of 75.1°C and a tip temperature of 66.7°C. The base-to-tip gradients were 15.3°C for aluminum and 8.5°C for copper, once again illustrating that copper distributes heat more uniformly along the fin. Every temperature in the modified design falls well within safe operating limits.

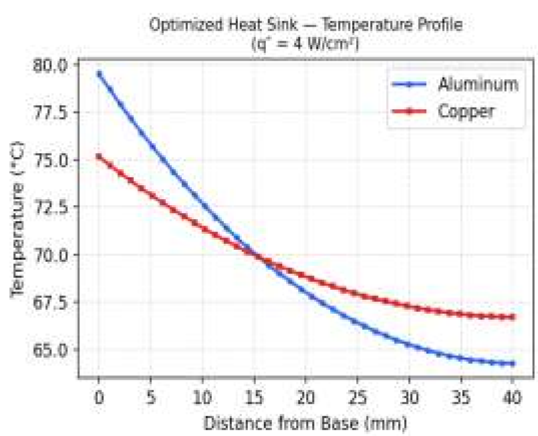

Fig. 6. Temperature along fin length for the modified heat sink model (q″ = 4 W/cm²).

Fin Effectiveness:-

Fin effectiveness (ε) quantifies the benefit of adding fins by comparing the heat rejected by the finned surface to the heat that the same base area would reject if left bare under identical conditions. Any value greater than unity means the fins are doing useful work. The results, plotted in Fig. 7, show that the modified copper design leads with an effectiveness of 11.59, followed by modified aluminum at 10.52. The conventional designs trail at 8.19 (copper) and 7.81 (aluminum). All four values sit comfortably above unity, confirming that fins are worthwhile in every case. The jump from the conventional to the modified layout is attributable primarily to the larger total fin surface area. It is also worth noting that effectiveness values computed under a heat flux boundary condition tend to run higher than those obtained with a fixed base temperature, because the base temperature is free to respond to changes in fin geometry, which penalizes the unfinned reference case.

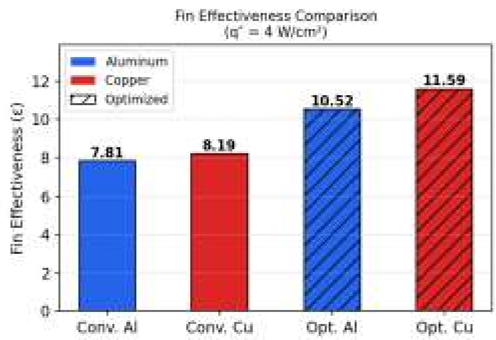

Fig. 7. Fin effectiveness for all four heat sink configurations (q″ = 4 W/cm²).

Fin Efficiency:-

Fin efficiency (η) measures how well the fin's actual surface area is being utilized relative to an idealized scenario in which the entire fin sits at the base temperature. A perfectly conducting fin would have an efficiency of 100%; real fins always fall short because temperature drops along their length. As shown in Fig. 8, the conventional copper model achieved the highest efficiency of 93.7%, followed by the conventional aluminum model at 88.9%. The modified designs registered lower values—86.8% for copper and 78.3% for aluminum—which is expected, since taller and thinner fins develop steeper temperature gradients, rendering the tip region less effective at shedding heat. Even the lowest recorded efficiency (78.3%) is perfectly respectable, indicating that the fins are utilized reasonably well across all configurations. The critical insight here is that the modified design sacrifices a modest efficiency gain in exchange for a much larger effectiveness gain, resulting in a net improvement in overall thermal performance.

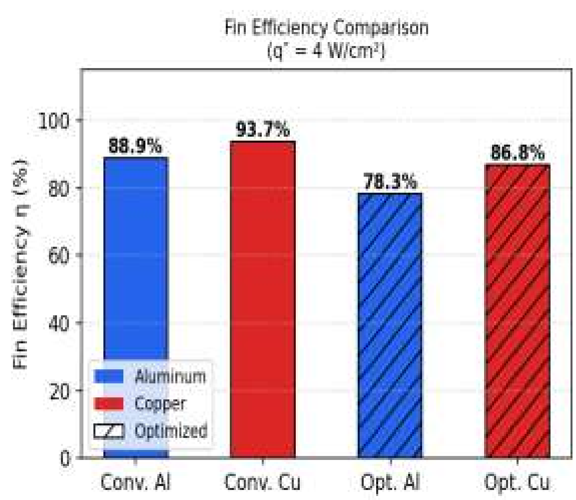

Fig. 8. Fin efficiency for all four heat sink configurations (q″ = 4 W/cm²).

Conclusion:-

This study set out to compare a conventional rectangular-fin heat sink and a geometrically modified rectangular-fin heat sink, both fabricated from aluminum or copper, under steady-state thermal conditions simulated in ANSYS Workbench. A base heat flux of 4 W/cm² was applied—the highest level at which all four configurations maintain base temperatures below 100°C with passive convective cooling at h = 80 W/m²·K and an ambient of 32°C.

The principal findings are summarised below:-

1. Peak base temperatures across the four configurations ranged from 75.1°C to 96.0°C, all of which fall safely within the operational envelope for typical power electronics.
2. The modified fin geometry brought base temperatures down by approximately 16–18°C compared to the conventional design, a substantial gain achieved through a relatively modest change in fin count, thickness, and height.
3. Copper consistently yielded lower base temperatures and higher fin efficiencies than aluminum in both designs, but these advantages must be weighed against its higher density and material cost.
4. Fin effectiveness exceeded 7.8 in every case and surpassed 10.5 for the modified layouts, reinforcing the value of finned surfaces for passive thermal management.
5. Among the four variants, the modified aluminum design offers the most attractive balance of thermal performance, weight, and cost, making it the recommended choice for most practical applications.

Several avenues remain open for future investigation. Transient simulations would reveal how quickly each design reaches thermal equilibrium after a sudden power-on event. Exploring enhanced forced-air conditions or alternative fin profiles, such as pin fins, tapered fins, or wavy geometries, could further improve performance. The influence of thermal interface materials on contact resistance also warrants closer examination. Finally, controlled experimental testing under known heat flux conditions would provide valuable validation data for the numerical predictions reported here.

Nomenclature:-

k Thermal conductivity (W/m·K)
h Convective heat transfer coefficient (W/m²·K)
q'' Applied surface heat flux (W/m² or W/cm²)
Q Total heat input to base plate (W)
m Fin parameter, $m = \sqrt{hP/kA_c}$ (m^{-1})
P Fin perimeter (m)
A_c Fin cross-sectional area (m²)
L Fin length (m)
L_c Corrected fin length, $L_c = L + t/2$ (m)
R_{total} Total thermal resistance of fin array (K/W)
T_0 Base temperature (°C)
T_a Ambient temperature (°C)
ε Fin effectiveness (–)
η Fin efficiency (%)

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