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Thermal Optimization and Comparative Analysis of Vertical Fin Arrays for Passive Electronic Cooling

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Abstract

Passive cooling using extended surfaces is a widely used thermal management method for electronic devices because it requires no external power and has high reliability. This article develops an extended thermal analysis of a vertical fin array subjected to natural convection in quiescent atmospheric air. The original fin-array problem is modified by including fin efficiency analysis, comparison of alternative fin geometries, parametric investigation of fin spacing and thickness, and thermal resistance network modeling. Four geometries are considered: rectangular plate fins, triangular tapered fins, trapezoidal tapered fins, and circular pin-fin equivalents. A Python-based computational model is used to calculate the Rayleigh number, Nusselt number, convection coefficient, fin efficiency, total heat transfer rate, and thermal resistance. For the baseline aluminum fin array, the optimum spacing is approximately $11.36\,\mathrm{mm}$, and the rectangular plate-fin configuration provides the largest heat dissipation, approximately $69.84\,\mathrm{W}$, with a thermal resistance of $0.716\,\mathrm{K/W}$. The results show that geometry and spacing strongly affect passive heat-sink performance, and that thermal resistance modeling provides a direct design metric for electronic cooling applications.

Keywords

Natural convection, Fin array, Thermal analysis, Heat transfer enhancement, Electronic cooling, Optimum fin spacing

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Introduction

Thermal management is essential for the reliability and performance of electronic devices. When the operating temperature of electronic components increases, material degradation, reduced efficiency, and premature failure may occur. Passive heat sinks with extended surfaces are commonly used because they enhance heat dissipation without fans, pumps, or additional power consumption.

A vertical rectangular fin array operating under natural convection is a classical configuration in electronic cooling. Its thermal performance depends not only on the available surface area but also on the airflow induced by buoyancy between adjacent fins. When fins are placed too close to each other, the thermal boundary layers merge and restrict the upward airflow. Conversely, excessive spacing reduces the number of fins and decreases the total heat-transfer area. Therefore, an optimum spacing exists.

The present work extends the standard fin-array problem into a more complete article-level thermal study. The analysis includes fin efficiency, comparison of multiple fin geometries, parametric variation of spacing and thickness, and thermal resistance modeling.

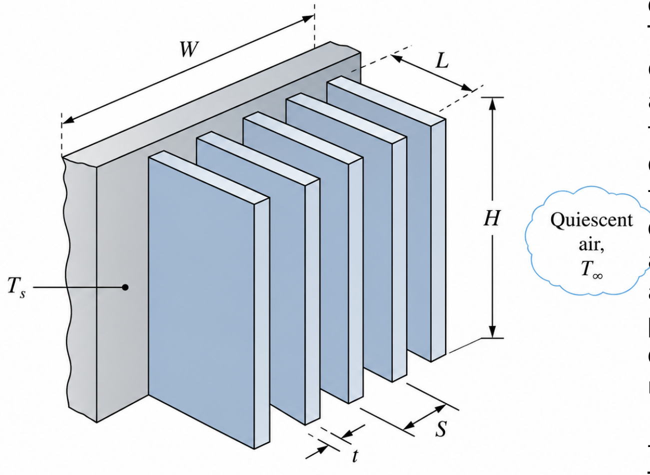

Figure 1: Schematic of the vertical rectangular fin array used in the present study (adapted from Incropera et al. [6]).

1. Physical Model

The considered heat sink consists of vertical fins attached to a heated base and exposed to quiescent air. The prescribed operating and geometric conditions are listed in Table 1.

Table 1: Input parameters for the baseline fin array.

Parameter	Symbol	Value
Fin length	L	20 mm
Fin height	H	150 mm
Array width	W	355 mm
Fin thickness	t	1.5 mm
Surface temperature	T_s	77 °C
Ambient temperature	T_∞	27 °C
Fin thermal conductivity	k_f	205 W/(m K)
Air thermal conductivity	k	0.028 W/(m K)
Kinematic viscosity	ν	$1.8 \times 10^{-5}\ \mathrm{m^2/s}$
Prandtl number	Pr	0.70

The analysis assumes steady-state heat transfer, constant properties, one-dimensional fin conduction, uniform base temperature, laminar natural convection, and negligible contact resistance.

2. Methods

The present study investigates the thermal performance of vertical fin arrays operating under natural convection conditions for passive electronic cooling applications. The methodology combines analytical heat transfer modeling, fin efficiency evaluation, comparative geometry analysis, thermal resistance modeling, and computational parametric analysis using Python. A vertical array of uniformly spaced fins attached to a heated base surface is considered, where the fin base is maintained at a constant temperature while the surrounding air remains at ambient conditions. Thermophysical properties of air are evaluated at the film temperature using standard property tables from Incropera and DeWitt. The natural convection characteristics of the fin array are determined using classical dimensionless correlations involving the Rayleigh number, Nusselt number, and convection heat transfer coefficient for laminar flow over vertical surfaces. The optimum fin spacing is estimated using established empirical correlations for vertical parallel plates under natural convection, accounting for the interaction between buoyancy-driven airflow and thermal boundary layer development. To incorporate conduction effects within the fins, fin efficiency analysis is performed using standard extended-surface formulations. A comparative thermal analysis is conducted for rectangular, triangular, trapezoidal, and circular pin-fin geometries under identical operating conditions to evaluate differences in heat transfer performance and thermal resistance characteristics. The overall thermal resistance of the heat sink is subsequently evaluated to assess cooling effectiveness for electronic applications. Furthermore, a parametric investigation is carried out to examine the influence of fin spacing, fin thickness, and fin geometry on heat transfer rate, fin efficiency, and thermal resistance. All analytical calculations and parametric studies are implemented using Python programming language with NumPy, Pandas, and Matplotlib libraries for numerical computation, data processing, and graphical visualization. The adopted methodology provides a comprehensive framework for analyzing and optimizing passive fin-array heat sinks under natural convection cooling conditions.

3. Natural Convection Formulation

The film temperature is evaluated as

$$T_f = \frac{T_s + T_\infty}{2}. \tag{1}$$

The Rayleigh number based on fin height is

$$Ra_H = \frac{g\beta(T_s - T_\infty)H^3}{\nu^2} Pr. \tag{2}$$

For laminar natural convection over a vertical plate, the average Nusselt number is approximated by

$$Nu_H = 0.59 Ra_H^{1/4}. \tag{3}$$

The corresponding convection coefficient is

$$h = \frac{Nu_H k}{H}. \tag{4}$$

The optimum spacing between vertical parallel plates is estimated using

$$S_{\text{opt}} = 2.714 \left(\frac{\nu^2}{g\beta(T_s - T_\infty)Pr} \right)^{1/4}. \tag{5}$$

The calculated baseline thermal parameters are reported in Table 2.

Table 2: Baseline natural convection parameters.

Quantity	Value	Unit
Rayleigh number, Ra_H	1.10×10^7	–
Nusselt number, Nu_H	33.98	–
Convection coefficient, h	6.34	$\mathrm{W/(m^2\,K)}$
Optimum spacing, S_{opt}	11.36	mm
Baseline fin count	28	–

4. Fin Efficiency Analysis

The fin efficiency accounts for the temperature drop along the fin due to conduction resistance. For a straight fin of uniform cross-section, the efficiency is

$$\eta_f = \frac{\tanh(mL_c)}{mL_c}, \tag{6}$$

where

$$m = \sqrt{\frac{hP}{k_f A_c}}. \tag{7}$$

Here, P is the fin perimeter, A_c is the conduction cross-sectional area, and L_c is the corrected fin length. The heat transfer from one fin is then

$$q_f = \eta_f h A_f (T_s - T_\infty). \tag{8}$$

For the baseline aluminum rectangular fin, the computed fin efficiency is approximately 0.994. This high value indicates that the fin remains nearly isothermal under the specified conditions.

5. Thermal Resistance Network

The total heat transfer from the fin array is modeled as the sum of heat transfer from all fins and from the exposed base surface:

$$Q = N q_f + h A_b (T_s - T_\infty), \tag{9}$$

where N is the number of fins and A_b is the exposed base area.

The overall thermal resistance is

$$R_{\text{th}} = \frac{T_s - T_\infty}{Q}. \tag{10}$$

Lower thermal resistance corresponds to stronger cooling performance. This metric is particularly useful in electronic heat-sink design because it directly relates the allowable temperature rise to the heat dissipation requirement.

6. Comparison of Fin Geometries

Four fin configurations were considered: rectangular plate fins, triangular tapered fins, trapezoidal tapered fins, and circular pin-fin equivalents. The geometry comparison is presented in Table 3. The rectangular plate-fin array provides the highest heat transfer rate because it has the largest effective convective area for the prescribed envelope.

Table 3: Comparison of fin geometries at the optimum spacing.

Geometry	η_f	Area per fin/pitch unit (m^2)	Number of fins	Q (W)	R_{th} (K/W)
Rectangular plate fins	0.994	0.006225	28	69.84	0.716
Triangular tapered fins	1.000	0.004482	28	54.69	0.914
Trapezoidal tapered fins	1.000	0.005353	28	62.43	0.801
Circular pin-fin equivalent	0.989	0.004801	28	57.04	0.877

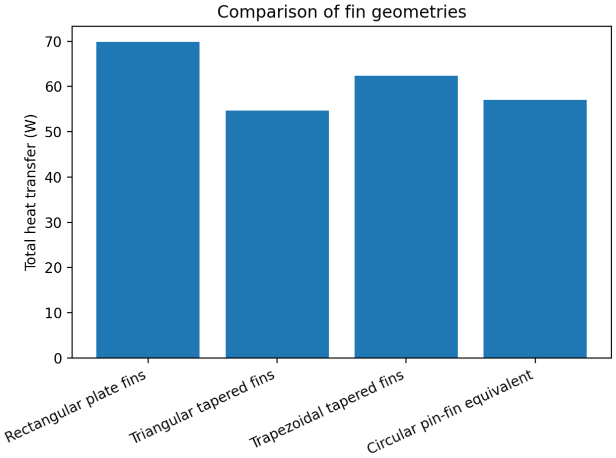

Figure 2: Total heat transfer comparison for different fin geometries.

7. Parametric Analysis

7.1 Effect of Fin Spacing

The effect of fin spacing on heat transfer and thermal resistance is shown in Figures 3 and 4. Reducing the spacing increases the number of fins, but very narrow passages can suppress buoyancy-driven flow. The optimum region occurs around the calculated spacing of approximately $11.36\,\mathrm{mm}$. In practical design, the spacing should be selected by balancing fin count, airflow resistance, manufacturing constraints, and mechanical robustness.

7.2 Effect of Fin Thickness

The influence of fin thickness is shown in Figure 5. Increasing thickness improves conduction through each fin, but it reduces the number of fins that can fit within the fixed array width. Under the specified conditions, thinner fins provide higher overall heat transfer because the aluminum fins already have high efficiency. However, very thin fins may be limited by manufacturability and structural strength.

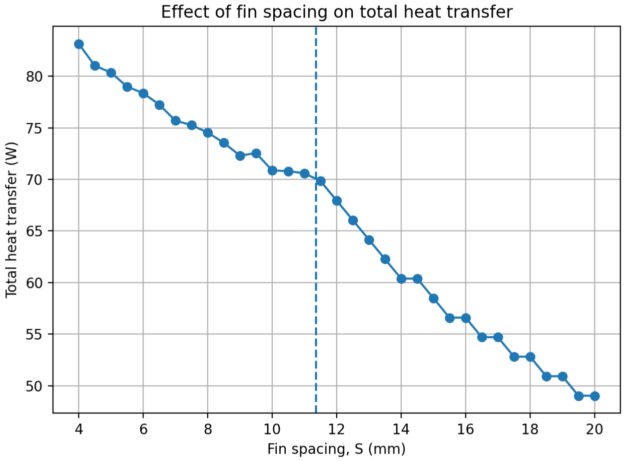

Figure 3: Effect of fin spacing on total heat transfer. The dashed line indicates the calculated optimum spacing.

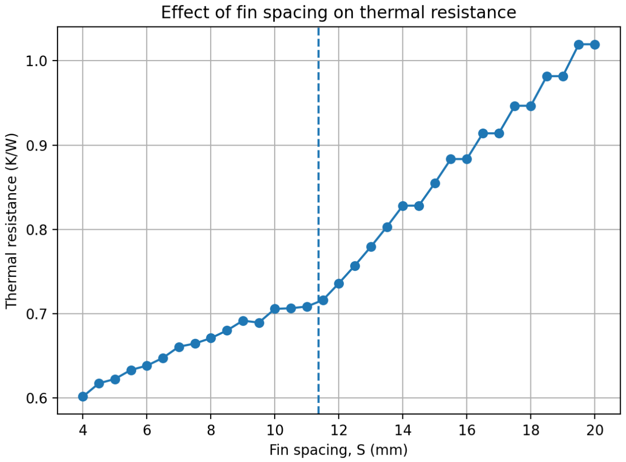

Figure 4: Effect of fin spacing on thermal resistance.

8. Computational Method

The calculations were implemented in Python. The computational procedure consisted of the following steps:

1. Define geometric dimensions, surface temperature, ambient temperature, and material properties.

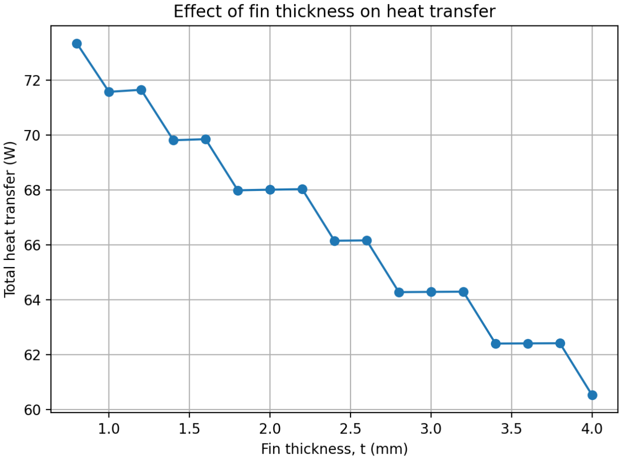

Figure 5: Effect of fin thickness on total heat transfer.

2. Calculate film temperature and air properties.
3. Compute Ra_H, Nu_H, h, and S_{opt}.
4. Determine the number of fins for each spacing and thickness.
5. Evaluate fin efficiency and total heat transfer.
6. Calculate the overall thermal resistance.
7. Repeat the calculations for alternative geometries and parametric sweeps.

This approach allows rapid early-stage heat-sink optimization without requiring computational fluid dynamics.

9. Results and Discussion

The results show that the rectangular plate-fin geometry dissipates the largest heat load among the investigated configurations. Its total heat transfer rate is approximately $69.84\,\mathrm{W}$, while the corresponding thermal resistance is $0.716\,\mathrm{K/W}$. The trapezoidal fin provides the second-best performance, followed by the circular pin-fin equivalent and the triangular tapered fin.

The fin efficiency analysis shows that the aluminum fins are highly efficient. Therefore, the dominant design factor is not conduction resistance inside the fin but rather external natural convection and available heat-transfer area. This explains why reducing fin thickness improves heat transfer in the studied range: thinner fins allow more fins to be installed within the same array width.

The spacing analysis confirms the importance of geometric optimization. Although smaller spacing increases the total surface area, overly small flow passages may restrict natural convection. Hence, heat-sink design should not simply maximize the number of fins; it should instead optimize the coupled conduction–convection system.

10. Conclusions

A classical vertical fin-array problem was modified into a broader thermal optimization study including fin efficiency, geometry comparison, parametric analysis, and thermal resistance modeling. The main conclusions are as follows:

1. The calculated optimum spacing for the specified conditions is approximately $11.36\,\mathrm{mm}$.
2. The rectangular plate-fin geometry gives the largest heat dissipation, approximately $69.84\,\mathrm{W}$.
3. The corresponding thermal resistance of the rectangular fin array is approximately $0.716\,\mathrm{K/W}$.
4. Fin efficiency for the aluminum rectangular fin is high, approximately 0.994.
5. Fin spacing and thickness significantly influence the thermal performance of passive heat sinks.
6. Python-based analysis is effective for preliminary heat-sink design and optimization.

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