Thermal performance of perforated pin finned heat sinks: A simulation based study

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Abstract

One of the major concerns of electronic systems is their ever-increasing demand for a competent thermal management solution at a miniaturized scale to evade failures due to formation of hot spots. Thus, employing heat sinks with better thermal performance can boost the efficiency of the system and helps in improving its reliability. Higher efficiency in heat transfer can enhance the hitherto poor coefficient of performance (COP) of thermoelectric cooling and heating. Many researches have suggested that by modifying the heat sink design, the heat transfer can be improved radically. This study aims to add necessary understanding of unidirectional heat transfer flow in heat sink for ensuring better regulation of temperature in the heating devices without thermal-runaway. In this investigation, the heat transfer improvement is studied in naturally convected heat sink with perforated fin and their thermal characteristics are compared with the non-perforated/solid pin fin heat sink made of copper and aluminium. The heat sinks are designed to study the difference in thermal behaviour with increment in number of perforations in presence of air as the working fluid. These heat sink cases are simulated for Computational Fluid Dynamics (CFD) analysis in COMSOL 4.4 environment. From the results, it is evident that the improvement of heat transfer rates with increased number of perforation is consistent in case of both copper and aluminium. A relationship is established between optimized number of perforations and material with its effect on their corresponding heat transfer rates. The outcome of the research will be useful in designing heat sink for cooling CPUs, microcontrollers, various electrical implements and improving COP of thermoelectric cooling and heating systems. However, further investigation seems necessary to achieve better performance.

Keywords: Perforated pin fin heat sinks, heat transfer rate, COMSOL, finite element method

Introduction

The removal of excessive heat from system components is essential to avoid the damaging effects of burning or overheating. Therefore, improving the system to deal with high heat transfer is of great importance. Thus, by increasing the heat transfer coefficient between a surface and its surroundings, by increasing the heat transfer area of the surface, or by both, can be used to enhance the heat transfer characteristics. In most cases, the area of heat transfer is increased by using extended surfaces in the form of fins attached to walls and surfaces [1].

Convection is one of the primary modes of heat transfer between a surface and a fluid moving over it. The energy transfer in convection is predominantly due to the bulk motion of the fluid particles though the molecular conduction within the fluid itself. If this motion is primarily caused due to the density variations associated with temperature gradient within the fluid, the mode of heat transfer is said to be due to free or natural convection. On the other hand fluid motion produced by some superimposed velocity field (like a fan or a pump), the energy transport is said to be due to forced convection [2, 3]. Heat sinks are sustainable, robust, and cost effective natural convection cooling solutions for electronic and photonic systems. Enhancing the heat transfer rate from air-cooled heat sinks is necessary due to the low thermal performances of air as the coolant. Thus, increasing heat transfer coefficient requires installation of a pump or fan, or to increase the surface area by attaching to the surface extended surfaces called fins made of highly conductive materials (such as aluminium or copper) to cumulatively boost the heat transfer [4, 5]. However, the performance of natural convection cooled heat sinks is considerably dependent on different structures of fins attached to it. Also, a number of recent studies have explored the performance benefits of perforated plate fins heat sinks (PFHSs) and perforated pinned heat sinks (PHSs). From such investigations, it is evident that perforations can increase simultaneously heat transfer and reduce the mechanical fan power needed to overcome the pressure losses. Also, elliptical perforations also demonstrate encouraging results where turbulence plays its role, but still the circular perforations were found to be better [6, 7]. The plate fins in PFHSs can be perforated longitudinally, along the fins, or laterally across them. On the other hand, perforations in pin fin heat sink can be of various geometries and number along the pins. Heat sinks with perforation can dissipate the heat rapidly to surrounding fluid by disturbing the fluid flow along with reduction of size, weight and cost as other additional paybacks.

Literature Review

The experimental and computational studies done so far have shown that perforating plates and pins on heat sinks can offer significant benefits as compared to that of the solid fins. Shaeri & Yaghoubi [8, 9] and Shaeri & Jen [10, 11] linked the effect of the number of perforations with their size, while Farhad Ismail et al. [12, 13] considered the influence of perforation shape on heat transfer and frictional drag on the air for both laminar and turbulent flow cases. It is found that a single perforation in a plate fin could increase the heat transfer rate by up to 80%. These studies found that increasing the number of perforations in plate fins leads to reductions in the size of the fin and the length of the recirculation zone around the fin. Dhanawade and Dhanawade [14] experimentally determined the effect of lateral circular perforations for plate fins on heat transfer and found that perforations generally increase the Nusselt number and that the optimum perforation diameter is a function of the applied heat flux density with larger perforations being beneficial for low heat fluxes and smaller perforations better for high heat fluxes. These studies establish that increasing the number of perforations in plate fins leads to reductions in the size of the fin and the length of the recirculation zone around the fin.

Conventionally, the performance parameters of a heat sink are primarily reliant on its geometrical considerations viz. fin length, dimension, design and material. Peterson and Ortega [15] have reviewed the use of natural convection, among other approaches, to cooling electronic equipment. Fisher and Torrance [16] presented analytical solutions free convection limits for pin fin cooling. They suggested that proper choosing of both fin diameter and porosity could optimize the pin fin heat sink design. Sparrow and Vemuri [17] studied the natural air-cooled heat transfer characteristics of pin fin arrays in three different orientations. Huang et al. [18] provided experimental study on natural convective performance of square pin fin heat sinks with various orientations. Shen et al. [19] studied the effects of the orientation angle on the fluid flow and heat transfer of rectangular fin heat sinks under natural convection. Sertkaya et al. [20] experimentally investigated heat transfer by natural convection in air from pin-finned surface under various orientation angles. As per experimental analysis conducted by Elshafei [21] and Kou et al. [22], heat sinks with perforated pin fins exhibits better cooling performance than their solid pin fin heat sink. Al-Damook et al. [23] studied the significance of multiple perforations of the pin fin heat sinks and found that perforations can increase the heat transfer as well as reduce the overall power consumption needed for mechanical fan. Ahmed et al. [24] studied the thermal design optimization on a ribbed flat-plate fin heat sink. Kobus and Oshio [25] considered effect of radiation effect and carried out a theoretical and experimental study that is capable of forecasting the influence of several physical and thermal parameters affecting the thermal resistance of a pin-fin array heat sink.

The above literature review suggest that several studies supported the physical modification of fin arrays in terms of their geometry and arrangement, but still there is a lack of knowledge for the natural convection heat transfer of the heat sink from multiple perforation perspective. The current work is associated with simulating perforated copper and aluminium heat sink and their respective thermal characteristics on natural convection.

Model Geometry

The models are simulated using COMSOL software to determine the thermal characteristics for different pin perforations. The model consists of a heat sink with 49-pin fin array. The geometric assembly of the model is represented in below diagram:

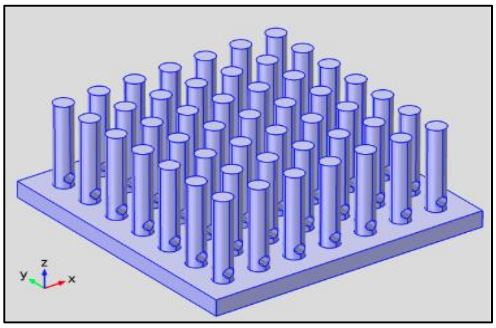


Fig 1: Geometry of perforated pin fin Heat sink

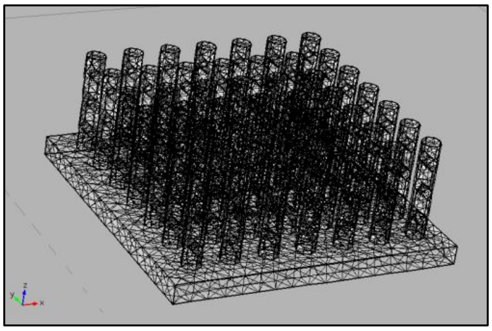


Fig 2: Mesh structure of perforated pin fin Heat sink

Design parameters:

The model geometry consists of a heat sink base of dimension 7.5 cm * 7.5 cm * 0.5 cm. The cylindrical pin fin heat sink have radius = 0.25cm, height = 2.5cm and are located at an equidistant from each other with seven in number in each row or column. For perforated heat sink fins, the perforations are equidistant depending on number of perforations, starting from the base point and its radius is 0.15cm.

Calculation of total surface area of heat sinks with different type of fins:

Base plate surface area of heat sink:

Surface area of base plate (S1) = Cuboid surface area – (49 * Area of the base of cylinder/cone)

$$= 7.5 * 7.5 – 49 * \pi * 0.25^{2} \text{ cm}^2$$

$$= 46.63 * 10^{-4} \text{ m}^2$$

Cylindrical fin surface area (S2) = Surface area of cylinder – area of base of cylinder

$$= 2\pi rh + 2\pi r^2 - \pi r^2$$

$$= 4.12 * 10^{-4} \text{ m}^2$$

Since 49 fins are present,

$$S2 = 49 * 4.12 * 10^{-4} \text{ m}^2$$

$$= 201.88 \text{ m}^2$$

∴ Total surface area of heat sink with solid fins = S1+ S2 = **$248.33 * 10^{-2}$ m^2**

And, cylindrical fin with 1 perforation surface area = Surface area of cylinder – area of base of cylinder + surface area of cylindrical perforation – 2 * area of circular perforation

$$S = 4.12 + (2\pi rh + 2\pi r^2) – 2\pi * r^2$$

$$= 4.12 + 0.57 – 0.26$$

$$= 4.43 * 10^{-4} \text{ m}^2$$

Since 49 fins are present,

$$S2 = 49 * 4.43 * 10^{-4} \text{ m}^2$$

$$= 217.07 * 10^{-4} \text{ m}^2$$

∴ Total surface area of heat sink with one perforation = S1+ S2 = **$263.7 * 10^{-4}$ m^2**

Similarly, the total surface area of heat sink can be tabularized in Table1:

Table1: Perforation and effective surface area of the heat sink

Perforation	No Perforation (S0)	1-P (S1)	3-P (S2)	4-P (S3)
Total surface area of heat sink (m^2)	248.33 * 10^{-4}	263.7 * 10^{-4}	308.8 94 * 10^{-4}	324.87 * 10^{-4}

The formula used for the heat transfer is:

$$Q = h * S * (T_p - T_a)$$

where, Q =heat transferred, J/s = W
h = heat transfer coefficient, W / (m^2 K)
S = transfer surface, m^2
T_p= Temperature at a point on the heat sink, K
T_a=Ambient Temperature, K (say T_a=300K)

The convection heat transfer coefficient (h_c), W/ (m^2 K) is calculated using the Nusselt number Nu, which is the ratio between the convective and the conductive heat transfer:

$$Nu = \frac{Convective\ heat\ transfer}{Conductive\ heat\ transfer} = (hc * L)\ /\ k \quad \textbf{Eq. 1}$$

where, Nu = Nusselt number
h_c = convective heat transfer coefficient
k = thermal conductivity, W/m K
L = characteristic length, m

The convective heat transfer coefficient can be written as following:

$$Nu = \frac{Nu * k}{L} \quad \textbf{Eq. 2}$$

The Nusselt number depends on the geometrical shape of the heat sink and on the airflow.

Now, if the heat flux given at the base of heat sink is maintained constant

q = 12 Volt × 6 Amp = **72 Watt**

Thus, for laminar flow and vertical fins,

$$Nu = 0.59 \times R_a^{0.25} \quad \textbf{Eq. 3}$$

where, $Ra = Gr * Pr$ **Eq. 4**

Rayleigh number is defined in terms of Prandtl number (Pr) and Grashof number (Gr).

If Ra < 109, the heat flow is laminar, while if Ra > 109, the flow is turbulent.

The Grashof number, Gr is defined as following:

Gr = (g * L3 * β * (Tp-Ta)) /η2 **Eq. 5**

where, g = acceleration of gravity = 9.81, m/s2
L= longer side of the fin, m
β = air thermal expansion coefficient. For gases, the reciprocal of the temperature, K
β = 1/Ta, 1/K
T_p= Plate temperature, °C.
T_a= Air temperature, °C
η = air kinematic viscosity,
= 1.56×10-5 (at 27 °C).

And, Prandtl number, Pr is defined as:

Pr = (μ × Cp) / k **Eq. 6**

where, μ = air dynamic viscosity, is 1.846×10-5 at 27 °C.
C_p = air specific heat = 1005 J/ (Kg*K) for dry air
k = air thermal conductivity = 0.026 W/ (m*K) at 27 °C

From governing equations of the convective heat transfer in fluid, the obtained h_c and T_p values for different surface areas and materials can be tabularized in Table 2:

Table 2: Convective heat transfer coefficient (htc) and Temperature at a point on heat sink with different perforations in pin fin Heat Sink

Surface Area	Convective htc (Aluminium) (hc)	Temp. at Aluminium heat sink (K)	Convective htc (Copper) (hc)	Temp. at Copper heat sink (K)
S0	12.13	539.06	11.84	544.83
S1	11.96	527.792	11.6855	533.121
S2	11.552	501.776	11.306	506.17
S3	11.425	493.987	11.1864	498.123

Simulation & Results

The simulations are done considering following material properties

Property	Name	Aluminium	Copper
Heat capacity at constant pressure 900 J/(Kg*K)	Cp	900J/Kg*K	385J/Kg*K
Density	rho	3900Kg/m³	8700Kg/m³
Thermal Conductivity	k	207W/m*K	400W/m*K

Table 3: Properties of Aluminium and Copper

Using heat transfer module with laminar flow, the time-dependent simulations are studied in COMSOL. The heat flux and heat transfer coefficient values were specified for each iteration with the fluid considered as Air.

The various perforations that are considered for the simulation are mentioned below:

0p - No perforations on Heat sink fins

3p - Three perforations on Heat sink fins

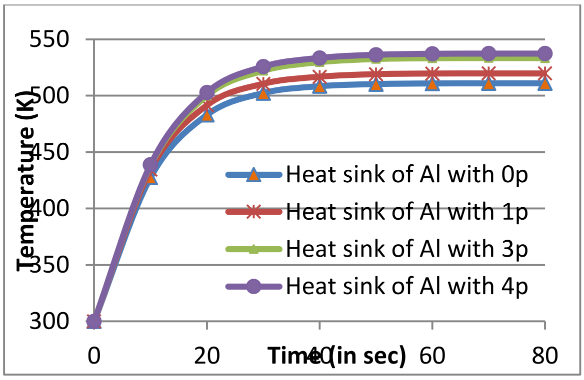
4p - Four Perforations on Heat sink fins

Case 1: At natural convection for Aluminium heat sink (hs)

Table 4: Temperatures at the tip of fin of an Aluminium Heat sink for different perforation conditions at a constant heat flux with respect to time

Time in min	0-p	1-p	3-p	4-p
0	300	300	300	300
10	427.50	434.46	438.0	438.75
20	482.96	491.50	500.0	502.79
30	502.33	510.59	522.08	525.76
40	508.39	516.85	529.68	533.46
50	510.31	519.12	532.49	536.28
60	510.856	519.70	533.2	537.14
70	510.96	519.76	533.317	537.3
80	510.94	519.74	533.29	537.29

Graph 1: Comparative heat transfer rate with respect to perforation



The results and comparative analysis of heat transfer for Aluminium heat sink can be seen under:

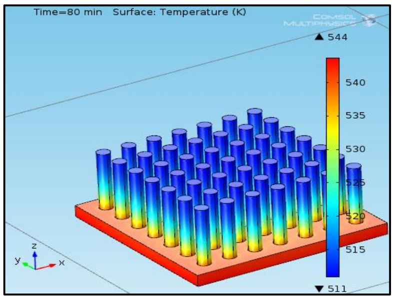


Fig 3: Al 0p at t = 80 min

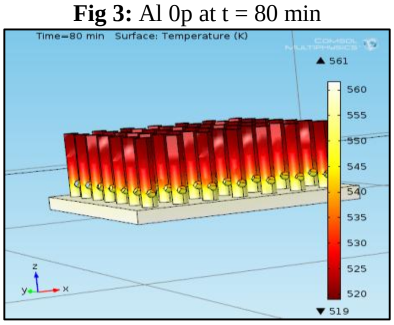


Fig 4: Al 1p at t= 80 min

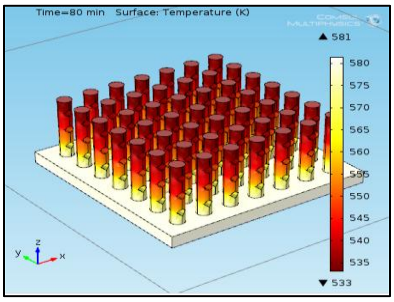


Fig 5: Al 4p at t= 80 min

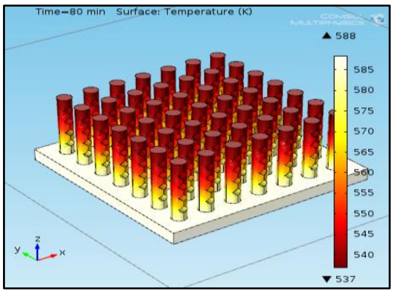


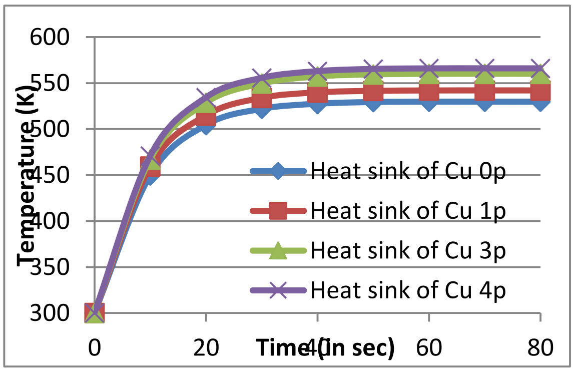
Fig 6: Al 3p at t = 80 min

Case 2: At natural convection for Copper heat sink

Table 5: Temperature at the tip of fin on copper Heat sink for different perforation conditions at a constant heat flux with respect to time

Time in min	0-p	1-p	3-p	4-p
0	300	300	300	300
10	449.73	459.28	467.08	470.76
20	504.41	514.91	528.59	534.39
30	521.93	533.99	549.53	555.64
40	527.44	539.96	557.17	562.94
50	529.33	541.62	559.4	565.34
60	529.71	542.05	559.99	565.94
70	529.72	542.12	560.1	566
80	529.68	542.1	560.09	565.95

Graph 2: Comparative heat transfer rate with respect to perforation



Graph 2: Comparative heat transfer rate with respect to perforation

The results and comparative analysis of heat transfer for Copper heat sink can be seen under:

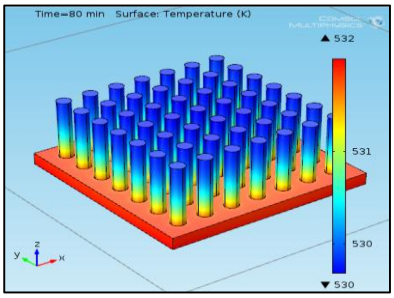


Fig 7: Cu 0p at t = 80 min

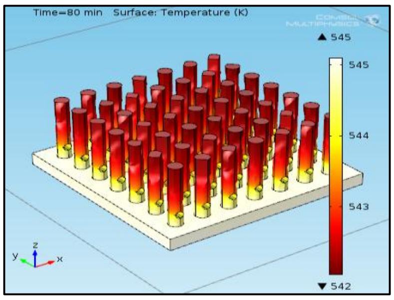


Fig 8: Cu 1p at t= 80 min

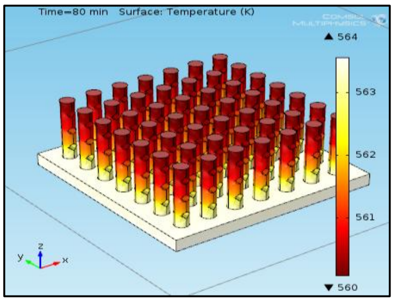


Fig 9: Cu 3p at t = 80 min

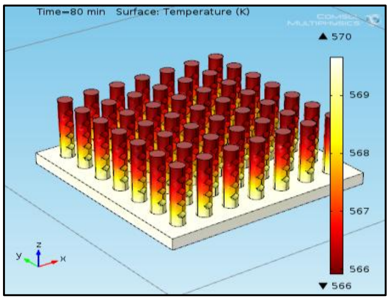


Fig 10: Cu 4p at t= 80 min

Conclusion

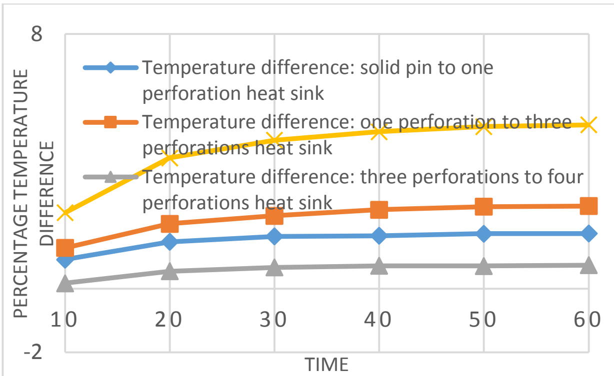
The simulation results demonstrate that pin fin heat sink with four circular perforations have better heat transfer characteristics than solid pin fin (zero perforation) or pin fins with less number of perforations for both Copper and Aluminium material. This higher performance may be attributed to larger surface area for convection and higher Nusselt number arguably due to the change in airflow directions. The construction process of the perforated pin-fin heat sink is not complex and can be accomplished easily and economically. Applying this knowledge in cooling electronics components and circuit, and CPUs is likely to help achieving improved cooling performance and thus higher reliability. Besides improving cooling performance of electronic circuits, the perforated pin fin heat sink can improve COP of thermoelectric systems. This inference will aid in improving the COP of the thermoelectric cooler by reduction of temperature between the hot and cold side of thermoelectric module. Further experimental research needs to be performed to compare the simulation results with performance in real applications.

At steady state, the improvement in temperature with respect to the geometry of the heat sink fins is presented as under:

Table 6: Improvement %

No. of perforations	Improvement % w.r.t. solid fins (Al)	Improvement % w.r.t. solid fins (Cu)
One	1.72	2.34
Three	4.37	5.74
Four	5.16	6.85

Graph 3: Comparative Representation of Performance of Heat Sink with Different Perforations



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