Convective Cooling of Electronic Components

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Abstract:
In this study, the time-dependent fluid flow and thermal interactions have been solved for transient and steady state conditions of the conjugate heat transfer problem between the natural convective flow and heat sinks composed of arrays of pins.The increased ratio of convective to conductive heat transfer across the boundary of the heat exchanger leads to improved dissipation of heat generated by the attached components. The dimensions and geometric configuration of the pins in the array can be selected to provide beneficial heat transfer characteristics

Keywords: Conjugate heat transfer, Fluid flow, Conduction, Convection, Heat sink.

1. Introduction

The development and miniaturization of electronic circuits with increased functionality oftentimes results in significant increases in the required input power density to components. As a result higher thermal fluxes are developed that may affect performance and must be dissipated to maintain device functionality and provide long-term operation. While forced air cooling may solve some commonly encountered problems, many applications avoid forced convection because of concern about the reliability of fans and instead rely on natural convection cooling. New ideas to utilize convective cooling to meet today's demand for thermal dissipation of high power electronic circuits are being investigated.

Dissipation of thermal energy by natural convection can be enhanced by attaching heat sinks to critical components, and two primary designs of heat sink are used based on plate-fin or pin-fin arrangements. In certain applications, heat sinks with pin-fin arrangements have been shown to perform better than those with plate-fins. In this study, we examine the use of various modeling approaches and optimization studies to assess the design of pin-fin heat sinks cooled by natural convection. Vertical channels containing horizontal pin fins are subject to natural convection causing chimney-style flow upwards through the array of fins. The steady state and transient nature of the flow and the associated heat transfer have been investigated.

2. Problem description

The base of the heat exchanger connects directly to the heat source by high conductivity media, ambient air is drawn over the pin-fin array and heat is dissipated into the flowing column of air. Heat transfer from the powered component occurs by conduction through the pin-fin array into the surrounding gas fluid. The heated fluid at the surface has a lower density and thus rises creating a natural convection current. The surrounding cooler fluid then moves to replace it thus creating a buoyancy effect. This cooler fluid is then heated and the process continues transferring heat energy from the bottom of the convection cell to the top.

The overall geometry of the pin-fin heat exchanger analyzed during the course of this work is provided in Figure 1.

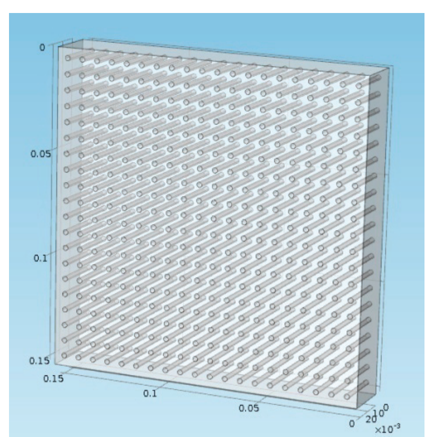

Figure 1: Pin-Fin heat exchanger

Using pin-fins in the path of the flowing column of air not only increases the surface area over which heat transfer occurs but can also increase the drawing power of the heat exchanger. The geometry of the pin-fin array considered in this work is given in Figure 2.

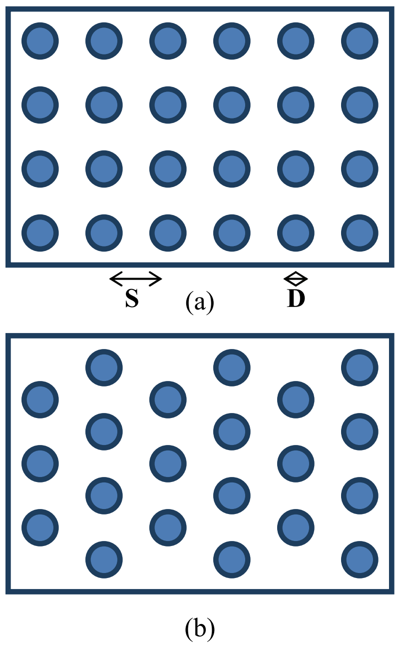

Figure 2: Geometry of pin-fin array: a) in-line, b) staggered.

Studies on the effectiveness of the pin-fin array geometry were conducted at heat flux values from 200 and 1000 $Wm^{-2}$and with spacing (S) to diameter (D) ratios ranging from 2.25 to 4.5.

In addition, this work also considered the influence of computational model size by comparing a model of the full heat exchanger with results using a unit cell approach, and compared the results of a steady state analysis with a transient analysis.

3. Computational model

Heat transfer within the solid domain is described by the well-known heat equation:

$$\rho\, c_p \frac{\partial T}{\partial t} = \nabla \cdot (\lambda \nabla T) \tag{1}$$

where, ρ is the density of the solid or fluid material, c_p is the specific heat capacity, T is the temperature, and λ is the thermal conductivity.

In the fluid domain, the physics are described by the conservation of mass, momentum, and energy according to the following equations:

$$\nabla \cdot (\rho\, \mathbf{u}) = 0 \tag{2}$$

$$\rho\, \mathbf{u} \cdot \nabla \mathbf{u} = -\nabla p + \nabla \cdot \left(\eta \left(\nabla \mathbf{u} + (\nabla \mathbf{u})^T \right) - \frac{2}{3} \eta (\nabla \cdot \mathbf{u}) \mathbf{I} \right) + \rho\, \mathbf{g} \tag{3}$$

$$\nabla \cdot (-k \nabla T) = Q - \rho\, c_p \mathbf{u} \tag{4}$$

The viscous heating and pressure work terms are neglected in the energy equation. In the above equations, ρ is the density, $\mathbf{u}$ is the velocity vector, p is the pressure, η is the dynamic viscosity, $\mathbf{g}$ is the gravitational acceleration vector, k is the thermal conductivity, T is the temperature, Q is a heat source term, and c_p is the specific heat capacity. The viscosity, thermal conductivity, and specific heat capacity are functions of temperature, while the density is a function of both temperature and pressure.

An open boundary condition is applied to top and bottom surface with normal stress (f_o) equal to zero. The energy equation is solved using a fixed temperature value for the fluid inflow and a temperature gradient equal to zero for the fluid outflow. A no slip wall boundary condition is applied to sides of the geometry; the heat flux boundary condition is defined for one side wall and other three side walls are thermally insulated.

4. Results

Three models are compared here

- Full geometry with a transient solution
- Full geometry with a steady state solution
- Unit cell model with a steady state solution

This section compares the results obtained from each model for a ratio of pinseparation(S) to pin diameter (D) of 3 and a power density of 1000 Wm^{-2} with air at an ambient temperature of 20°C.

Use of a repeating unit cell to represent the pin-fin array and analysis of the steady state condition reduces the computational resources required to solve the problem comsiderably. A summary of the computational resources required for the three conditions examined is provided in Table 1.

Condition	RAM, GB	# Cores	Analysis time, h
Unit cell/Steady State	8	4	6
Full/Steady State	96	8	20
Full/Transient	384	96	34

Table 1: Summary of computational requirements for steady state and transient analyses.

The use of a repeating unit cell to analyze the pin-fin array reduces the computational resoruces considerably but also leads to some differences compared to the results from the full geometry. Figures 3 and 4 show that the air and solid temperature predicted by the unit cell model deviate slightly from the results obtained from analyses of the full geometry. This is believed to be due to the inherently different boundary conditions associated with the finite boundary of the full geometry compared to the infinite boundary of the repeating unit cell analyses.

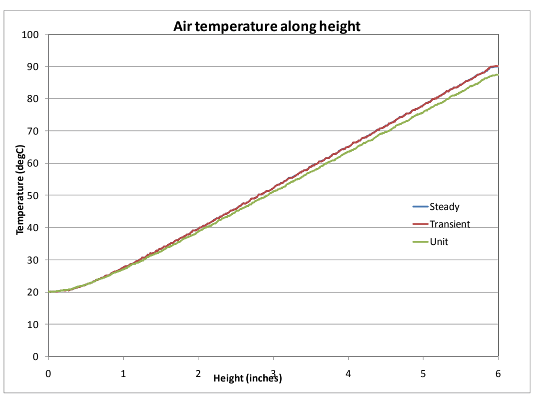

Figure 3: Air temperature as a function of height in the pin-fin array for three analysis conditions.

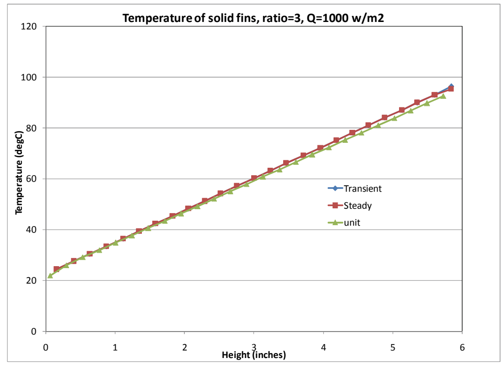

Figure 4:Solid temperature as a function of height in the pin-fin array for three analysis conditions.

The effectiveness of the pin-fin array geometry in providing cooling is a balance between increasing surface area to maximize heat transfer vs flow restriction through the pin-fin heat exchanger as the spacing of the pins decreases. To identify the most efficient balance of pin dimensions and spatial arrangement, i.e. in-line vs staggered, a comparison of the mass flow rate through the pin-fin array and the Fin Temperature Drop Ratio (FTDR) defined by:

$$FTDR = (T_{base} - T_{tip}) / (T_{base} - T_{amb}) \quad (5)$$

where, T_{base} is the average temperature at the base of the pins, $T_{tip}$the average temperature at the tip of the pins and T_{amb} is the ambient temperature. Hence lower values of FTDR suggest more efficient heat transfer.

For the range of S/D and unit cell geometries considered there was little variation in the mass flow rate through the array. The effect of S/D in the pin-fin array on FTDR along the vertical center line of the pin-fin array for applied heat fluxes of 200 and 1000 $Wm^{-2}$is shown in Figures 5 and 6.

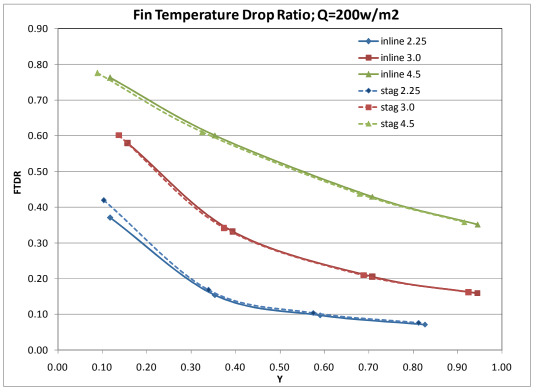

Figure 5: FTDR as function of height along the center line of the array for S/D of 2.25, 3.0, 4.5 for applied heat flux of 200 Wm^{-2}

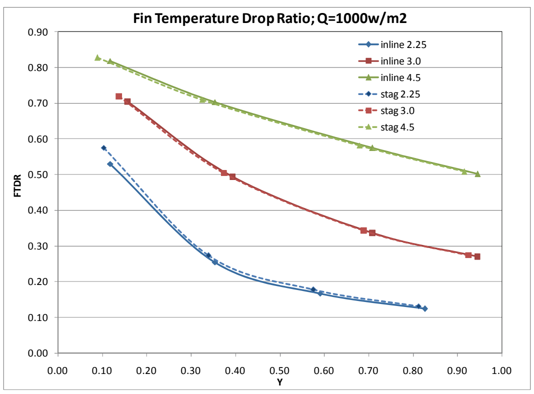

Figure 6: FTDR as function of height along the center line of the array for S/D of 2.25, 3.0, 4.5 for applied heat flux of 1000 Wm^{-2}

These results indicate the maximum heat flow is obtained for an array containing a pin spacing to diameter ratio of 2.25. The staggered pin-fin array provides better thermal performance than the in-line array but the effect is restricted to the lower pins in the array. With increasing height in the array the data trend to a common value independent of the unit cell geometry. Maximum heat transfer occurs in the lower pins where the temperature difference between the pins an air is greatest. With increasing height in the array the temperature of the air increases and the heat transfer drops. At the lower S/D value the data also approach an asymptotic response suggesting an optimum size of array exists after which further increasing the array size has limited effect on the heat extraction and dissipation into the surrounding environment. As expected, analyses of the full geometry in the steady state and transient analyses show agreement over a range of parameters once equilibrium is reached in the transient behavior, Table 2.

Condition	**Mass flow, kg/s**	**Ave.T 10^{th} pin, °C**	**Mean wall T, °C**	**T_{max}, °C**
Full/Steady	3.058 e-4	51.26	111.8	95.93
Full/Transient	3.06 e-4	51.24	111.7	96.89

Table 2: Comparison of results from transient and steady state analyses.

Despite the increased computational resources required to perform the transient analysis of the full scale pin-fin array,the results provide the most completeevaluation of the temperature history, temperature distribution and flow behavior. For example, the temperature evolution of the pin located 10 from the bottom and in the center of the horizontal axis, Figure 7, demonstrates that it takes approximately 700s to reach a steady state temperature.

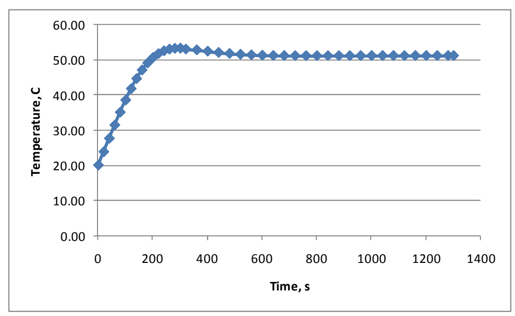

Figure 7: Temperature vs time for pin 10 from bottom and in the center of the horizontal axis.

5. Summary

This work has analyzed the thermal behavior of a pin-fin heat sink when subject to natural convection. The response of the array can be studied by using a repeating unit cell representationof the geometry. The results indicate that a limiting height of the array exists after which no further heat is extracted.