Convective Cooling of Electronic Components

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Abstract:
Predictions of the steady state thermal performance of vertically oriented plate-fin heat sinks cooled by natural convection have been investigated. Thermal dissipation to the environment is considered to occur by heat transfer involving conduction, convection and radiation. The importance of including heat transfer from the fin tips and radiation from the heat sink is demonstrated. The effect of mesh refinement and selection of the element type for developing an accurate and converged solution has been examined. To reduce model size and computational expense a repeating unit cell of the heat sink geometry is used. These approaches allow definition of the optimum fin spacing for thermal dissipation over a range of heat fluxes and plate-fin thickness.

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

1. Introduction

In response to continued miniaturization and increased multi-functionality of electronic circuits, the number of integrated circuit (IC) packages on the circuit board continues to increase. As a consequence the operating power density increases and significant increases in the operating temperature of devices result. To maintain operation and long term performance device temperature must be maintained below specific limits thus necessitating improved thermal dissipation. For many devices forced convection is not an option due to small form factors and concern about long term reliability. For these applications passive approaches for the dissipation of thermal energy by natural convection can be enhanced by attaching heat sinks to critical components. Two primary designs of heat sink are commonly used based on plate-fin or pin-fin arrangements. Heat sinks are commonly attached to high power density components and act to reduce the thermal resistance between the device junctions and the ambient environment by increasing the surface area for convective cooling.

Plate-fin heat sinks contain vertical channels that are subject to natural convection causing chimney-style flow upwards through the array of plates. Analysis of a simple heat sink problem involves heat transfer by conduction, convection and radiation. Depending on the dimensions of the heat sink, operating conditions and material properties one of these mechanisms may be dominant. Increasing the number of fins of a heat sink increases the surface area for heat dissipation but also increases obstruction of flow and an additional pressure drop across the heat sink. Thus it is expected that an optimum distribution of fins exists for maximum cooling under different operating conditions.

In this study, we examine the behavior of plate-fin heat sinks and obtain the optimum fin spacing for a range of heat fluxes for cooling under steady state natural convection. The effect of heat transfer from the fin-tips, surface radiation and fin thickness is also investigated.

2. Problem description

The base of a heat sink connects directly to a uniform heat source, ambient fluid is drawn over the plate fins and heat is dissipated into the flowing column of ambient fluid. Heat transfer from the powered component occurs by conduction through the plate-fins into the surrounding 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. Fluid flowing in the vertical channel between the plate fins does not mix until it exits the heat sink. In addition to convection from the heat sink walls, convection from the fin-tips and radiation from the heat sink may also provide important heat dissipation mechanisms.

A typical thermal-fluid velocity and temperature distribution for a plate-fin heat exchanger is shown in Figure 1.
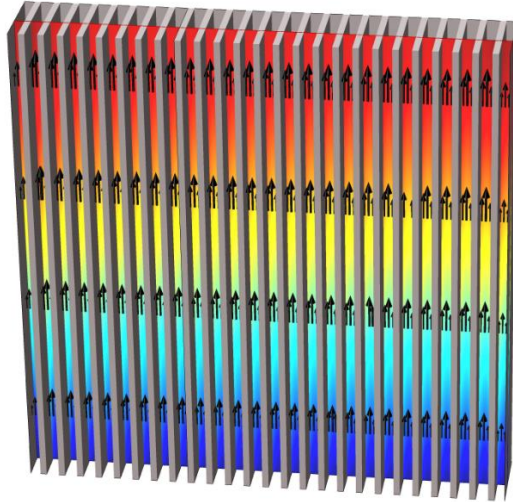

Figure 1: Plate-Fin heat exchanger

In this study, the size of the base of the plate-fin heat sink was kept constant and values of fin spacing (S), fin pitch and fin thickness (t) varied to determine their effect on thermal performance for different values of heat flux values (Q) ranging from 200 W/m^2 to 2000 W/m^2. The ambient temperature was maintained at 12 °C. Figure 2 shows the geometry of the plate-fin heat sink.
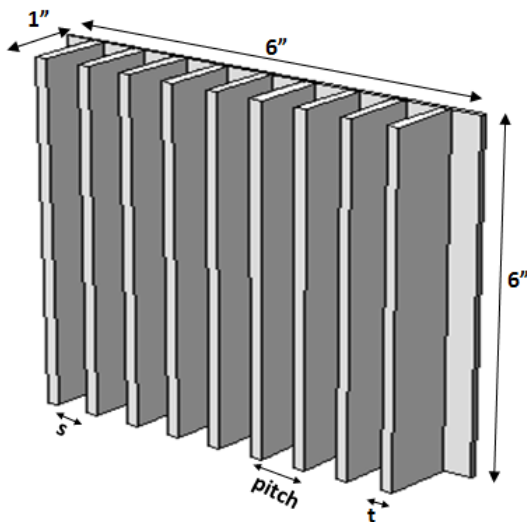

Figure 2: Geometry of plate-fin heatsink

In addition to analyses of the full heat sink geometry, analyses were also performed on a unit cell model to reduce the computational expense without loss of accuracy.

3. Computational model

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

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

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:

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

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

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

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. The fluid domain is modeled as air and solid domain as aluminum.

An open boundary condition was 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 the sides of the geometry; the heat flux boundary condition is defined for one side wall and other three side walls are thermally insulated.

The heat flux at the surface of the part due to radiation is modeled as

$$q_r = \varepsilon_{emis} \left(G_m + F_{amb} \sigma T_{amb}^4 - \sigma T^4 \right) \qquad (5)$$

Where ε_{emis} is the emissivity of the surface, G_m is the mutual irradiation from other surfaces, F_{amb} is the ambient view factor, σ is the Stefan-Boltzmann constant, T_{amb} is the far-away ambient temperature, and T is the temperature at the surface. G_m is a function of the radiosity. In the presence of mutually irradiation surfaces, COMSOL Multiphysics automatically computes the ambient view factor and mutual irradiation.

4. Results

Initially analyses were performed by considering convection from the walls of the plate fins. A steady state converged solution was obtained for both the full heat sink and unit cell models with a fin pitch to thickness ratio of 3, a fin thickness of 2mm and heat flux value of 1000 W/m². The element type and number of boundary layers were optimized to produce consistent solution methodologies. The results from these approaches were found to provide good agreement across the range of parameters of interest, example results are shown in Table 1. Significant differences in computational time and memory were observed for the two models and therefore the unit cell approach was used for a series of parametric studies based on heat flux and design of the plate-fins.

Model	Mass flow rate (kg/s)	Air exit temp. (degC)	Maximum temp. (degC)	Domain elements
Full heatsink	1.043 e-5	92.07	95.59	~5.e+6
Unit cell	1.042 e-5	92.08	95.62	~3e+5

Table1: Comparison between full heat sink and unit cell model

The unit cell model was used to determine the optimum plate-fin spacing with fixed fin thickness of 2mm for different heat flux values. The results are shown in figure 3; the x-axis shows the fin pitch to thickness ratio and the y-axis show the mean base temperature of the heat sink. The positive slope on the right side of the line-plots in figure 3 shows the effect of decrease in the surface area of the heat sink dissipating heat to the ambient environment. At lower fin pitch-thickness ratios obstruction of fluid flow

results in a negative slope on the left side of the line-plots. The optimum value of the fin pitch-thickness ratio is the balance between increasing the surface area and decreasing the pressure drop caused by the flow obstruction. The optimum fin pitch-thickness ratio is also seen to decrease with increases in the heat flux values.
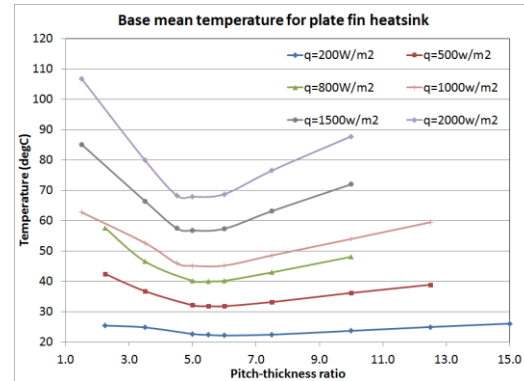

Figure 3: Effect of fin pitch to thickness ratio for different heat flux values for a plate fin heat sink

The effect of changing the plate fin thickness and spacing is shown in figure 4 for a single heat flux value of 2000 W/m². The x-axis shows the spacing between plate-fins and the y-axis shows the mean base temperature of the heat sink. The mean base temperature increases with increasing fin thickness and the optimum fin spacing decreases with increasing fin thickness.
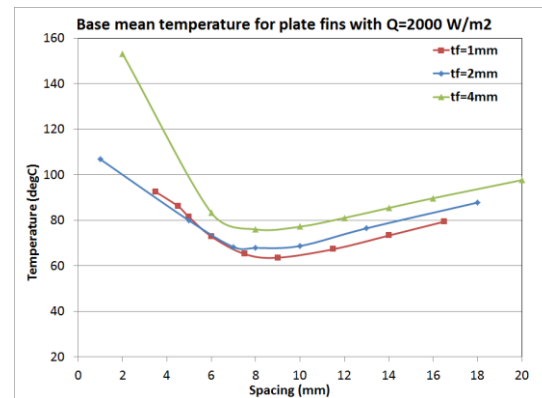

Figure 4: Effect of fin thickness and spacing on heat dissipation for a plate-fin heat sink

Figure 5 shows the combined effect of introducing convective heat transfer from the fin-tip and radiative heat transfer from heat sink on the base mean temperature of the heat sink. The

solid lines show the temperature without convection and radiation from heat sink; the solid points show the values including the convective and radiative heat transfer for different heat flux values.
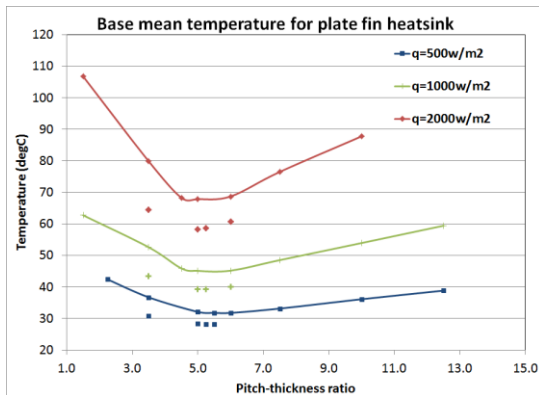

Figure 4: Combined effect of convective heat transfer from the fin-tip and radiative heat transfer from heat sink (solid points) vs convection from the walls (line)

The temperature differences with the addition of radiation and fin-tip convection increase with increasing heat flux values. This is because the higher absolute temperature of the heat sink surface corresponds to higher heat flux values. However the optimum values of the fin pitch to thickness ratio is not significantly affected by the introduction of radiation and fin-tip convection in the model.

5. **Summary**

This work has analyzed the steady-state thermal behavior of a plate-fin heat sink when subject to natural convection. An equivalent unit cell model was used to reduce the computational complexity of a full heat sink model. The study shows that the optimum plate-fin spacing decreases with increasing heat flux values. Increasing the fin thickness results in an increase in the mean base temperature, however the optimum fin spacing between the plate-fins does not change significantly. The effect of convective heat transfer from the fin-tip and radiation heat transfer from heat sink has the most significant effect on heat sink temperature for higher values of heat flux.