Optimized Heatsinks for Forced Convection

by
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Synopsis

Cooling of processors in servers and workstations has changed to accommodate increasing power, approaching 100 Watts. Dedicated fan-duct-heatsink combinations are becoming state-of-the-art for packaging of the processor(s). Extruded heatsinks, standard for many years, require larger space, pressure drop, and/or fan/blower power than necessary. In this study, optimum dimensions of fin thickness and pitch are calculated for a variety of realistic operating conditions. These dimensions are somewhat smaller than those achievable by forged or bonded fin heatsinks.

1. Introduction

Heatsinks for processors have evolved in recent years as power has increased. The first personal computers required large enclosures for the low density packaging of the early 1980's. Cooling of the processor was achieved by the airflow induced by the cooling fan of the power supply. As packaging density and power increased, heatsinks for processors were developed. Initially, these were used without any directed airflow, but recently in many systems another fan has been added for cooling the processor and motherboard. Currently, servers and workstations are changing to configurations with fan-duct-heatsink assemblies dedicated to cooling the processor(s). In such a fully ducted configuration, a heatsink of fixed size can be optimized to minimize thermal resistance at a given pressure drop, fan/blower power or fan curve.

The minimum fin thickness and pitch currently available in bonded fin heatsink technology is 0.8mm and 3.0mm, independent of height. Recently, state-of-the-art extruded and forged heatsinks have become able to offer such small dimensions, but only up to about 25mm fin height. Narrower fin pitches and, especially, thinner fins require other technologies, such as corrugated fins. In this case, all fins are formed from a single sheet of aluminum, then brazed to the base, resulting in a heatsink such as that shown in **Fig. 1**. Assemblies of stamped plate fins are also under development for taller heatsinks.

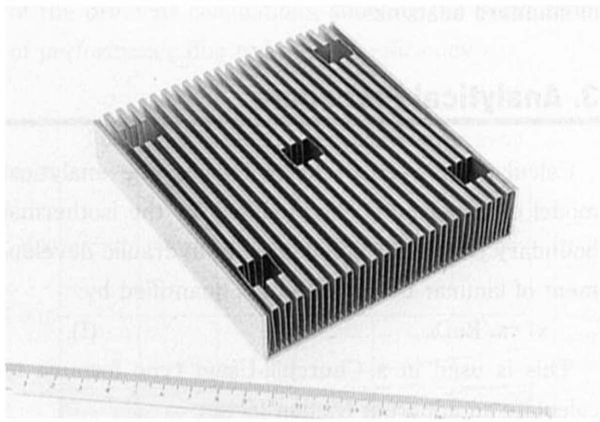

Fig. 1 Corrugated fin type heatsink, 25mm fin height

2. Previous Work

Knight et al. (1992) extended previous analyses of microchannel heatsinks for turbulent as well as laminar flow. They demonstrated improvement of previous studies by relaxing constraints on fin thickness/pitch ratio and allowing turbulent flow.

Copeland (1995) modified previous analyses for developing flow and calculated optimum fin thickness and pitch for silicon heatsinks cooled by fluorocarbon liquid. As channel length increased, optimum fin thickness and pitch increased.

Lee (1995) analyzed flow through parallel fin heatsinks in fully ducted and partially ducted flows. Unlike a fully ducted configuration, in a partially ducted configuration at a fixed approach velocity, an optimum fin and pitch exists. When the bypass path is eliminated, thermal performance improves monotonically as fin pitch is decreased.

Aranyosi et al. (1997) showed isocurves of pressure drop and fan power at fixed thermal resistance in addition to isocurves of thermal resistance at fixed pressure drop and fan power. As pressure drop or fan/blower power increased, optimum fin thickness and pitch decreased, resulting in reduced thermal resistance. In addition to analysis, experimental and numerical studies were performed.

Tasaka et al. (1997) performed experimental studies of compact heatsinks with fin thickness and pitch as small as 0.34mm and 0.70mm. Results correlated well with results from compact heat exchanger data. The compactness factor, defined as thermal conductance per unit volume, was three to seven times that of standard heatsinks.

3. Analytical Procedure

Calculations were performed using the analytical model of Copeland (1995) modified for the isothermal boundary condition. The degree of hydraulic development of laminar flow in a duct is quantified by:

$$x^{+}=x/ReD_h \tag{1}$$

This is used in a Churchill-Usagi type formula to calculate an apparent friction factor:

$$f_{app}Re = [(3.2x^{+\,-0.57})^2+(fRe)^2]^{1/2} \tag{2}$$

in which fRe is that of fully developed flow. In a similar manner, the degree of thermal development of laminar flow in a duct is quantified by:

$$x^{*}=x/ReD_hPr \tag{3}$$

and the resulting average Nusselt number is given by:

$$Nu_m = [(2.22x^{*\,-0.33})^3+Nu^3]^{1/3} \tag{4}$$

in which Nu is that of fully developed flow.

Using the remaining formulas, factors such as pressure drop, fan power, mean air temperature rise, average heat transfer coefficient, fin efficiency and thermal resistance are calculated. These calculations assume a uniform heat flux into the base of the heatsink. To produce more realistic calculations, an additional spreading resistance, which is a function of the size and location of the heat source(s), must be added to the convective resistance.

4. Results and Discussion

A case study of 125mm square heatsinks with 25mm and 50mm heights was performed. This is the approximate size of near-future processor modules (containing the processor and SRAMs). The minimum fin thickness and pitch of 0.2mm and 1.8mm represent practical minima for corrugated fin technology, while the maxima of 0.8mm and 3.0mm are near the limits of other manufacturing technologies. Realistic air flow rates and pressure drops, typical of 92mm and 120mm fans, were chosen as operating conditions

Pressure drop fixed at a constant value is the first operating condition to be considered. **Figure 2** shows the thermal resistance of all combinations of fin thickness and pitch at a constant pressure drop of 25 Pascals. At small values of fin pitch, increasing fin thickness greatly increases thermal resistance, as air flow becomes severely constrained. At higher values of fin pitch, the optimum thickness increases to about 0.6mm. The global optimum is found at a fin thickness near 0.3mm and pitch near 2.2mm. Other fin thickness and pitch combinations offer nearly equal performance. Note that each value of fin thickness has its corresponding optimum pitch, and each value of fin pitch has its corresponding optimum thickness.

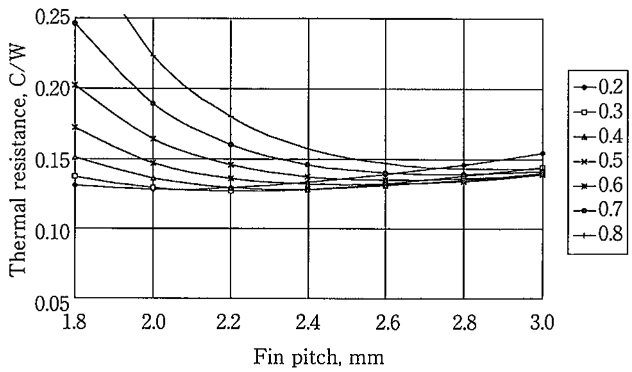

Fig. 2 Pressure drop=25Pa

Increasing the fin height to 50mm and keeping the same pressure drop constraint increases the optimum values of fin thickness and pitch. In **Fig. 3**, similar trends in thermal resistance occur, due to excessive fin thickness at small pitch and insufficient fin thickness at large pitch. The global optimum has shifted to a fin thickness near 0.5mm and pitch near 2.4mm.

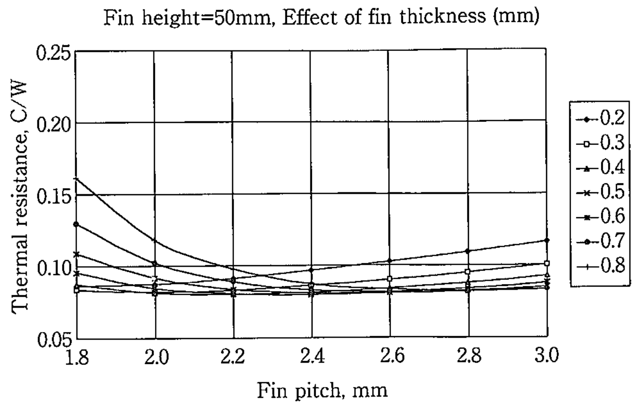

Fig. 3 Pressure drop=25Pa

Fan/Blower power can also be fixed at a constant value. In many cases this is representative of the middle portion of the operating curve of the fan/blower. Returning to the 25mm height, **Fig. 4** shows effects of heatsink dimensions at a moderate fan/blower power of 0.10 Watts. A minimum thermal resistance of 1.45 C/W is achieved near a fin thickness of 0.3mm and pitch of 2.0mm. The rate of increase in thermal resistance as fin pitch is changed (as long as the local optimum fin thickness for the pitch is maintained) is somewhat weaker than that for fixed pressure drop, resulting in shallower curves.

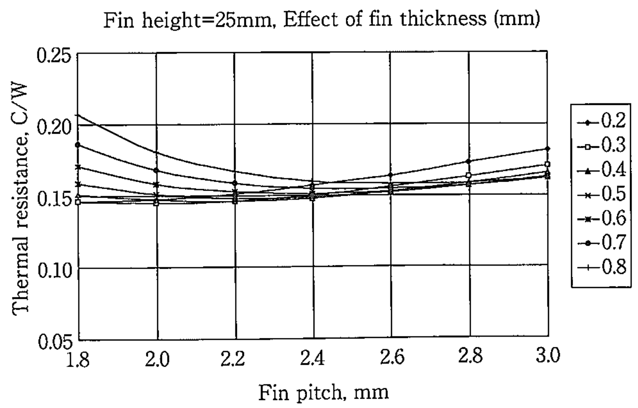

Fig. 4 Fan/Blower power=0.10W

Figure 5 shows a similar plot for a higher fan/blower power. In this case, the optimum fin thickness stays about the same, while optimum pitch is decreased to near 1.8mm. An 50% increase in fan/blower power results in only a 12% improvement in thermal resistance.

The taller 50mm height was also considered, with results shown in **Fig. 6**. At a fan/blower power of 0.25W, the average of the two previous values in terms of specific power per unit heatsink volume, the optimum values of fin thickness and pitch are near 0.5mm and 2.0mm. The thermal resistance of 0.82 C/W is considerably higher than half the average value of the previous calculations, showing the degradation of performance due to lower fin efficiency.

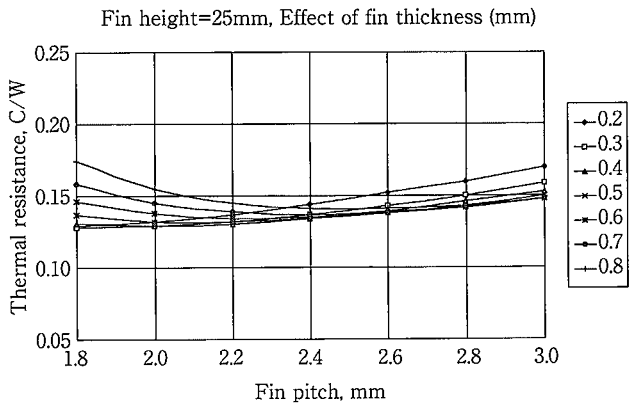

Fig. 5 Fan/Blower power=0.15W

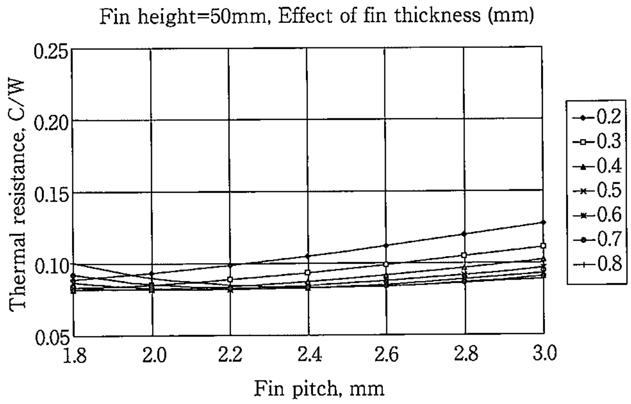

Fig. 6 Fan/Blower power=0.25W

Thermal resistance is the final parameter to be set at a fixed value in this study. **Figure 7** shows pressure drop for various combinations of fin thickness and pitch for a 50mm tall heatsink providing a thermal resistance of 0.08 C/W. The global minimum is reached at the same fin thickness and pitch as those of **Fig. 3**. Other combinations of fin thickness and pitch provide nearly equal performance, provided the fin thickness is closely matched to its corresponding local optimum value of fin pitch.

The effect on fan/blower power of non-optimum fin dimensions can be seen in **Fig. 8**. The global optimum fin thickness and pitch are near as those of **Fig. 6**, in which the minimum thermal resistance was slightly higher. As in the previous figure, several

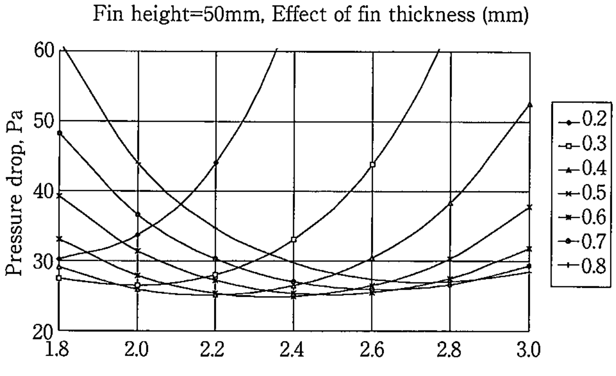

Fig. 7 Thermal resistance=0.08C/W

combinations of fin thickness and pitch offer near-optimum performance, but the cost (in terms of increased fan/blower power) of deviating from matching the fin pitch to its local optimum thickness is high.

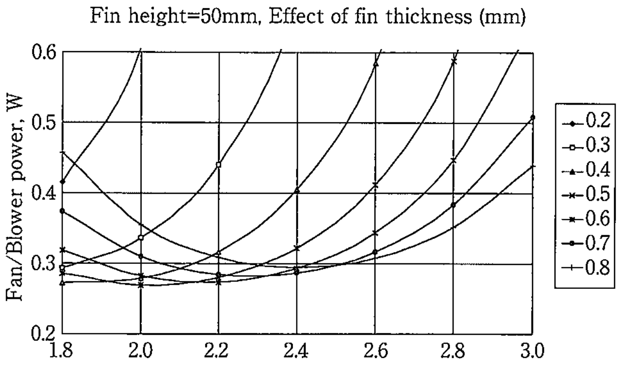

Fig. 8 Thermal resistance=0.08C/W

5. Conclusions and Recommendations

The results of the previous calculations provide the fin thickness and pitch which minimize thermal resistance for a given application, and allow comparison to extruded, forged, bonded and other heatsink technologies. Additionally, comparisons of the required size, pressure drop and/or fan power for different technologies offering equal performance can be made.

For a specific operating condition (pressure drop, fan/blower power or fan/blower curve) and heatsink outer dimensions, optimum values can be calculated. While fin thickness or pitch need not be exactly optimum to achieve high performance, it is important that the value of fin thickness or pitch be near its corresponding optimum value of pitch or thickness.

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