Heat Sinks and Heat Exchanger

Fin Optimization

In electronics cooling, often separately managed Thermal/Mechanical (TM) and Software/Electrical (SE) engineering teams are finding themselves facing common challenges, as they are being driven towards similar business goals, such as product differentiation, company growth and profitability. More so than ever today, these teams are being directed to find ways to increase component performance, particularly on highly populated boards within complex systems, at an acceptable cost of manufacturing. They are also discovering that their goals are being held back by governing specifications, environmental conditions, mechanical limitations and budget restrictions. TM's design thermal solutions based on airflow, envelope size, power dissipation, etc. and migrate (as expected) to the lower cost "standard solutions" whenever possible. If adequate margin is not met, reliability implications are more apparent as engineers will have to optimize solutions. This is because, in most cases, the form factor, layout, boundary conditions, etc. are set. Thermal solutions become the gatekeeper, and in some cases, the determining factor in product deployment.

Many leading companies design their products by using technologies that will sustain long product life cycles for increased market share and brand awareness. As products are refined through the design cycle, thermal solutions may have to be optimized and this requires many investigations to be undertaken. As the electronics industry continues to use components dissipating more and more power, new heat sink solutions must be able to accommodate large heat fluxes while keeping the same spatial dimensions [1]. Finned heat sinks and heat exchangers are largely employed in many engineering fields, and this demand spurs researchers into devising and testing new geometries for the heat sinks. Engineers constantly try to develop new designs to enhance the performance of heat exchangers. One such effort is the design of the wavy fins to enhance the surface area.

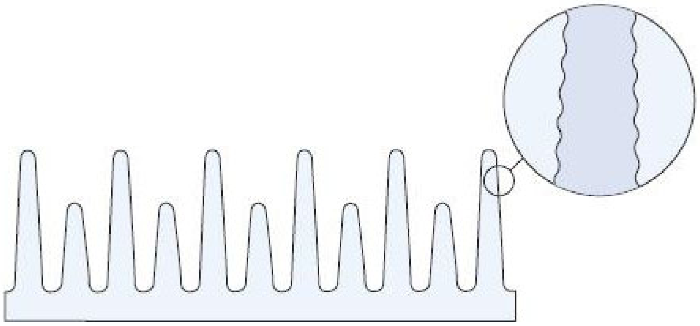

Figure 1. Close-Up View of Simply Wavy Fin Geometry [1]

Figure 1 shows a close up view of an extrusion type thermal solution where the profile has a feature of undulated fins. In general, a wavy fin heat sink should perform better under natural and forced convection due to the increased surface area created by the fins. This feature can easily be manufactured with a die. The "waviness" can be adjusted to increase surface area resulting in a positive impact on thermal performance.

Theoretical models have been devised to find the pressure drop and the heat transfer from wavy fin geometries. Figure 2 shows the schematic of a wavy fin.

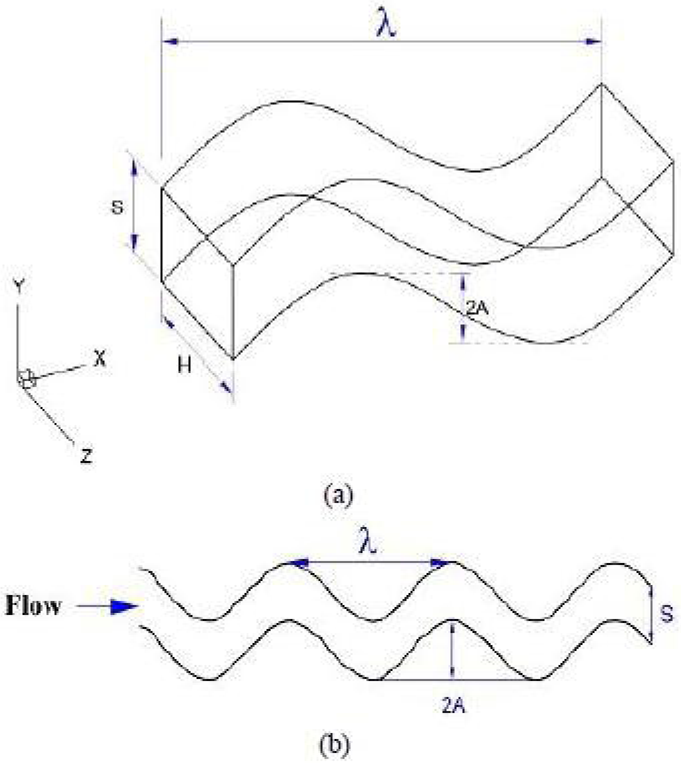

Figure 2. Schematic of a Wavy Fin Geometry [2]

In this figure, the fins are assumed to have a sinusoidal geometry where

λ = Wave length (m)
H = channel width (m)
S = channel height
2A = twice the amplitude of the wave
The shape of the curve is assumed to be

$$f(x)=A\sin(2\pi x/\lambda)$$

The length of the curve can be found from the following equation:

$$L_e = \int_0^{\lambda/2} \sqrt{1+[f'(x)]^2}dx$$

Shah and London [3] came up with the following equation for the friction and Nusselt number in channels:

$$fRe=24(1-1.3553\alpha + 1.9467\alpha^2 - 1.7012\alpha^3 - 0.9564\alpha^4 - 0.2537\alpha^5)$$

Where,
f = fanning friction factor
α=S/H aspect ratio

The same equation applies for a wavy fin based on the correct length.

$$fRe_{wavy}=fRe(\frac{Le}{\lambda})$$

The Nusselt number for the straight fins and wavy fins is the same as long as the correct surface area is used:

$$Nu_T=8.235(1-2.0421\alpha + 3.0853\alpha^2 - 2.4765\alpha^3 + 1.0578\alpha^4 - 0.1961\alpha^5)$$

The above equations are for the low Reynolds number.

For high Reynolds number Shapiro et. al [4] derived the following equations:

$$f_{app}Re_{D_h}=L/(D_hRe_{D_h})$$

$$Nu_L=0.664Re_L^{1/2}Pr^{1/3}$$

Where,
D_h = hydraulic diameter (m)
Re_{D_h} = Reynolds number based on hydraulic diameter
L = half length of the channel (Le/2)
Pr = prandtl number
D_h = 2SH/(S+H)

The combined asymptotic for the friction and Nusselt number is as follows:

$$f_{asy}=[(f_{wavy})^2 + (f_{app})^2]^{1/2}$$

$$\frac{Nu_{asy}}{Re_LPr^{1/3}} = \left[(\frac{Nu_T}{Re_LPr^{1/3}})^5 + (\frac{Nu_L}{Re_LPr^{1/3}})^{1/5}\right]$$

Figure 3 compares the results of the above analytical equations with the results from Kays and London [5]. In the graph, the Colburn j factor is shown and is defined as:

$$j = \frac{Nu}{Re_L Pr^{1/3}}$$

The results show that the experimental values of Shah and London are within 20% band of the values obtained from the above relations. The data is for the fin type 11.44-3/8W

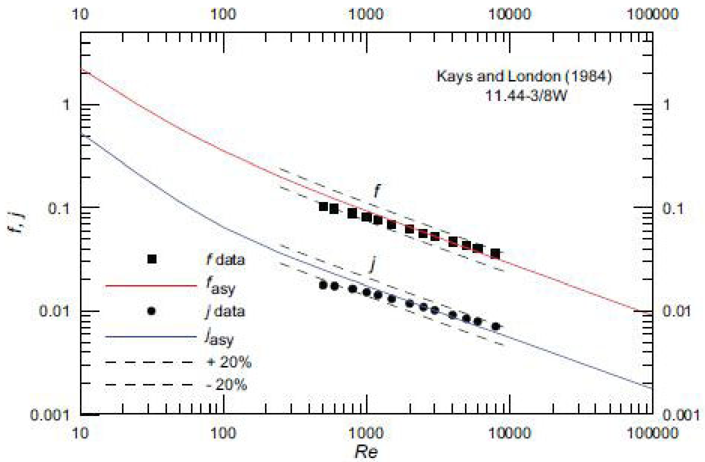

Figure 3. f and j Values as a Function of Reynolds Number [2]

Marthinuss et al. [6] reviewed published data for air-cooled heat sinks, primarily from Compact Heat Exchangers by Kays et al [5] and concluded that for identical fin arrays consisting of circular and rectangular passages, including circular tubes, tube banks, straight fins, louvered fins, strip or lanced offset fins, wavy fins and pin fins, the optimum heat sink is a compromise among heat transfer, pressure drop, volume, weight and cost. Figure 4 shows that if the goal is to get a higher value of heat transfer per unit of pressure drop, the straight fin is the best. Figure 5 shows that when heat transfer per unit height is of concern pin fin is the best.

Sikka et al. [7] performed experiments on heat sinks with different fin geometries. Figure 6 shows 3 different categories of heat sinks tested. The conventional fins, such as straight and pin fins, are shown in (a); (b) shows the fluted fins and (c)

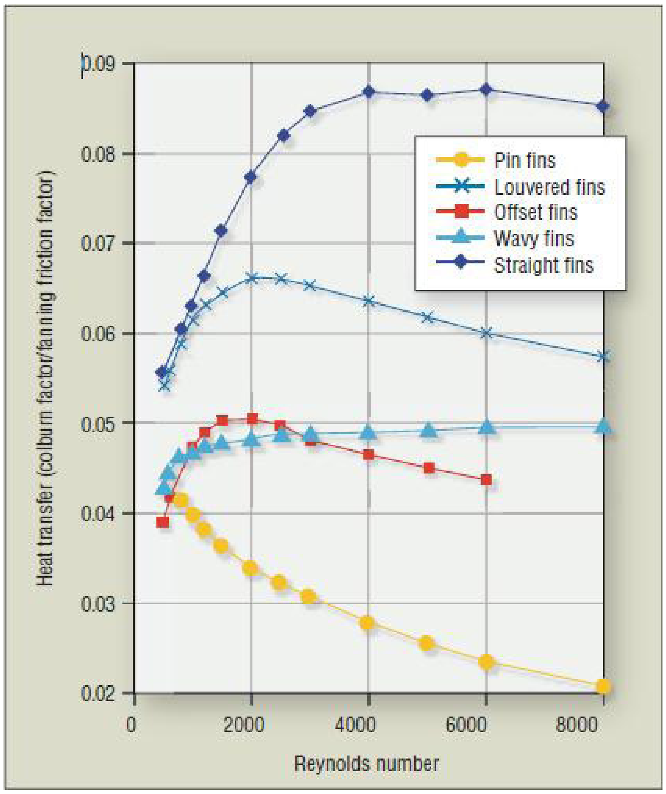

Figure 4. Profile Comparisons Based on Heat Transfer/ Pressure Drop [6]

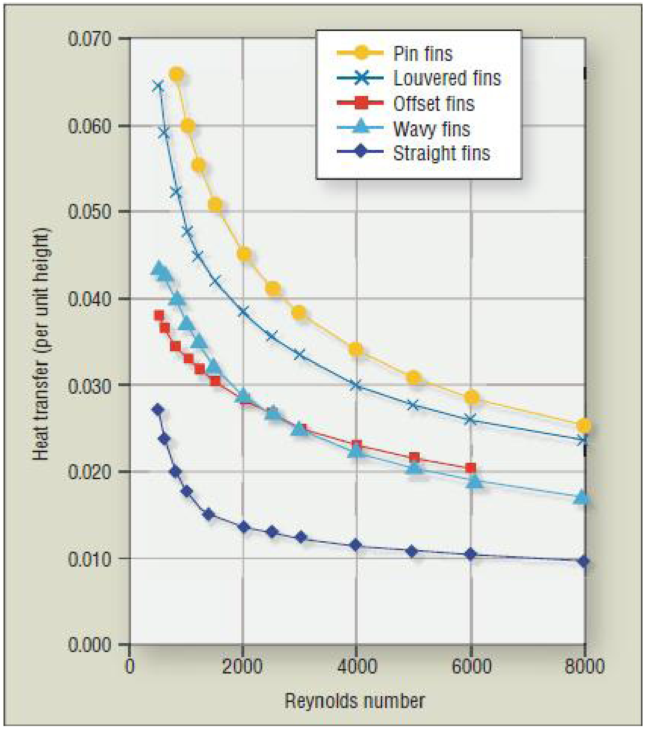

Figure 5. Profile Comparisons Based on Heat Transfer/ Volume [6]

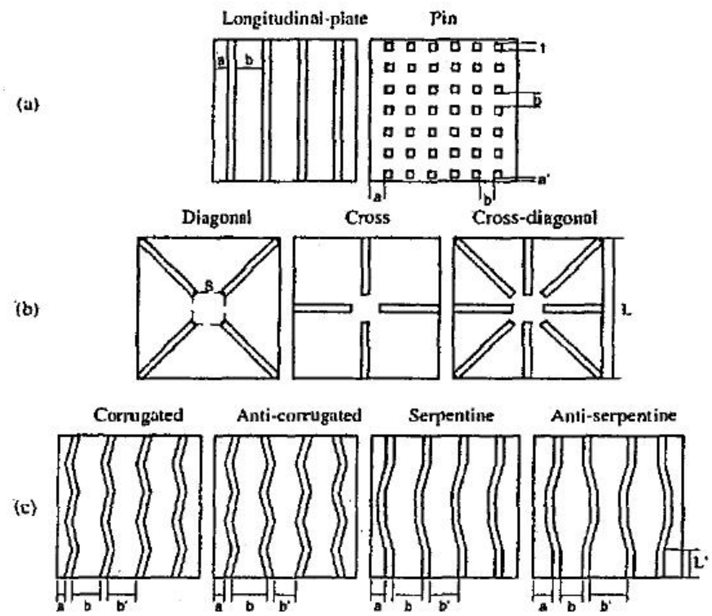

Figure 6. (a) Traditional Fins, (b) Fluted Fins, (c) Wavy Fins [7]

Heat Sink	b (mm)	b' (mm)	a (mm)	a' (mm)	L' (mm)	S (mm)	A_t/A_b
Longitudinal-Plate	12.7	-	6.35	-	-	-	4.36
Pin	6.35	-	6.35	1.59	-	-	4.26
Diagonal	-	-	-	-	-	12.7	2.89
Cross	-	-	-	-	-	12.7	2.44
Cross-diagonal	-	-	-	-	-	12.7	4.33
Corrugated	12.7	12.7	3.17	-	12.7	-	4.46
Anti-corrugated	15.88	9.53	4.76	-	12.7	-	4.46
Serpentine	12.7	12.7	6.35	-	12.7	-	4.4
Anti-serpentine	9.53	15.88	7.94	-	12.7	-	4.4

Table 1. Geometries and Dimensions of the Heat Sinks [7]

shows the wavy fin design. The tests were done for both horizontal and vertical direction of air flow at natural convection and low Reynolds number forced flow. Table 1 shows the dimensional values of each of these heat sinks. The last column shows the values of A_t/A_b (total surface area/base surface area)

The values of the Nusselt number were reported based on the following relation:

$$Nu = q/[k_f L A_t/A_b(T_h - T_a)]$$

Figure 7 shows that for natural convection in the horizontal direction, the pin fin has the best performance. The fluted fins have, in general, a better performance compared to longitudinal fins. The lower graph in figure 7 shows that the wavy fins are essentially the same as the longitudinal fins.

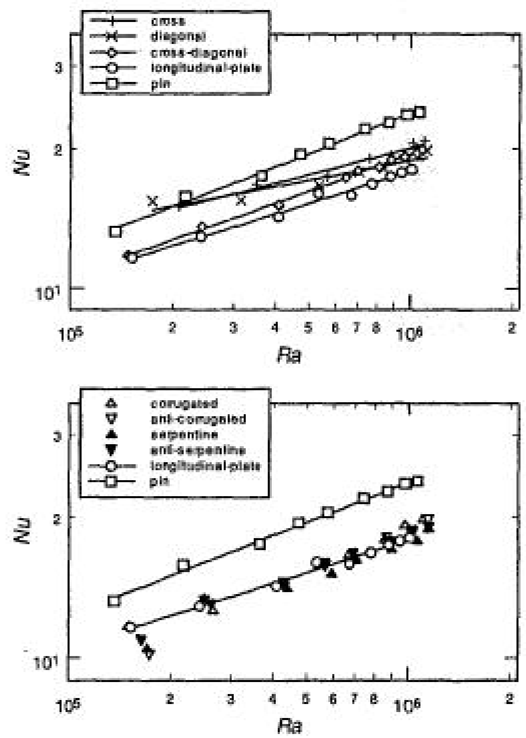

Figure 7. Nusselt Number As a Function of Rayleigh Number for Natural Convection-Horizontal Direction [7]

Figure 8 shows the natural convection cases for the vertical direction. The figure shows that heat transfer decreases for the pin fin, but increases for the plate fin. The pin fin still is better than the plate fin, but the difference is only 4-6%. Figure 8 also shows that the cross heat sink has the best performance. The bottom figure in 8 confirms that the wavy fins do not have much better heat transfer compared to plate fins.

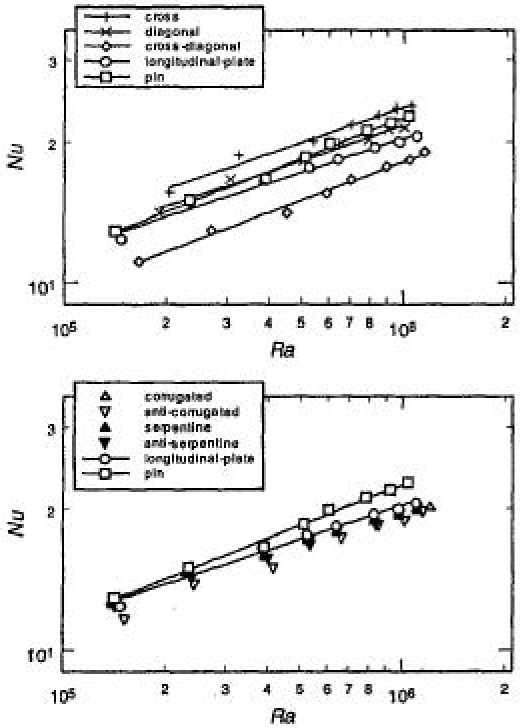

Figure 8. Nusselt Number as a Function of Rayleigh Number for Natural Convection-Vertical Direction [7]

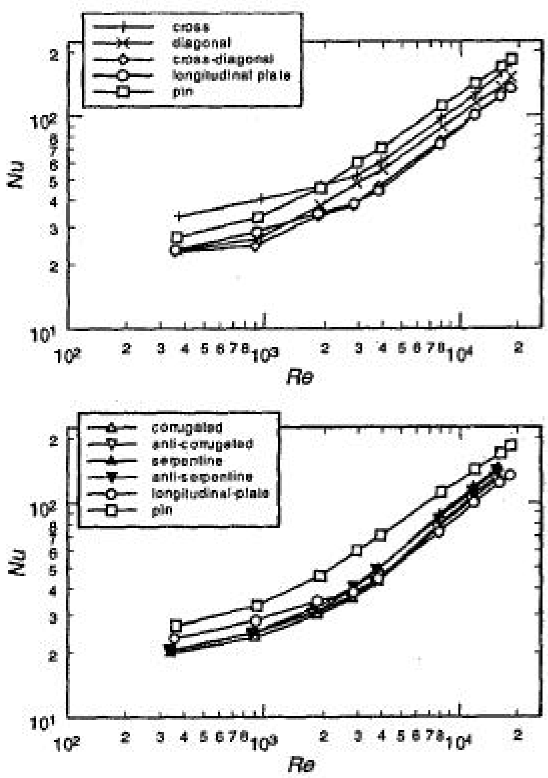

Figure 9. Nusselt Number as a Function of Reynolds Number for Forced Convection-Horizontal Direction [7]

Figure 9 shows the Nusselt number for forced convection over a horizontal plate as a function of Reynolds number. This figure indicates that, for very low Reynolds numbers, the cross fin is better than the pin fin; but, around Re = 2000, the situation reverses and the pin fin gets better than the cross heat sink. For low Reynolds numbers, the longitudinal pins are better than the wavy fins; but, at higher Reynolds numbers, the performance of the wavy fins gets better by almost 12-18%.

Figure 10 provides the Nusselt numbers for the vertical direction for forced flow. In comparing the results with the horizontal direction, the results are almost the same, with the difference being that the wavy fin heat sinks perform better than the plate fin heat sinks, by about 14-20%.

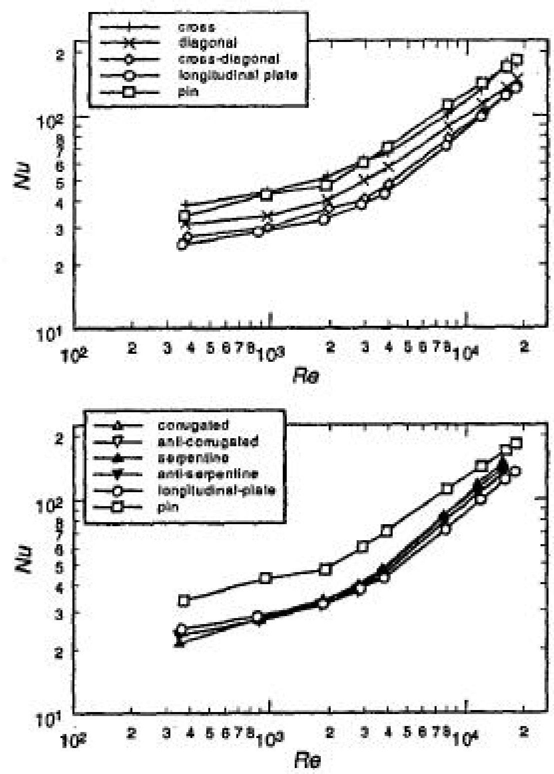

Figure 10. Nusselt Number as a Function of Reynolds Number for Forced Convection-Vertical Direction [7]

The results presented in this article strengthen our understanding about how heat exchangers and heat sinks can be made more compact and efficient. The results show that the design of the fin field is still an issue and much remains to be investigated for optimization, depending on the conditions and application. Further empirical testing is warranted for the evaluation of the effects of wavy fin heat sinks, as fine meshing and a high degree of confidence is not easily obtained through simulating these profiles using commercial CFD tools.

References:

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