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Thermal Optimizations and CFD Analysis of Finned Heat Sinks for Natural Convection

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Abstract

It would save a lot of time and efforts if individual heat sinks are thermally optimized prior to using the CFD tools for system analysis and design. The practical example of such process which employs an existing correlation for optimization of finned heat sinks is presented. Several CFD simulations are first performed to compare with the results from the correlations. The good agreement of the U-channel heat transfer coefficient between the correlation and CFD results further validates the accuracy of the correlation.

The main focus of the present work is to perform a detailed CFD analysis on the heat sink with the fin optimal spacing of 0.439”. The flow field ultimately determines the heat transfer from the heat sink. Therefore, an effort is made to provide the insight view of the detailed flow fields which has never been done before. The velocity is relatively uniform when the air first enters the finned heat sink from the bottom side (low end). However, due to the entrant flow entering spacing between the fin tips at the face of the heat sink, the velocity of the air flow increases along the length of the heat sink when the air flows upwards. The CFD results indicate that the heat loss per zone decreases along the heat sink length (height). The total natural convection heat loss of this heat sink is 96.61 watts. The heat transfer coefficient of the entire heat sink is 0.7 Btu/hr-ft^2 while the heat transfer coefficient from the U-channels alone is 0.61 Btu/hr-ft^2.

The results indicate that system with the cover in contact with fins perform better thermally than that of the case without the cover. It is also found that there is no effect of the cover/shroud on the heat loss or entrant flow rate as long as the distance between the cover and the heat sink fin tips is greater than 4.36” with the fin height of 2.0”. Based on the limited data in this work, one may conclude that there is no effect of the cover on the heat transfer of a finned heat sink if the distance between the heat sink and the cover is greater than 2.5 times of the fin height.

Introduction

Heat transfer by natural (or free) convection has long been considered as one of the most cost effective and reliable cooling methods. Natural convection with air has many practical engineering applications and is of special interest to the cooling of electronic equipment such as those shown in Figure 1. The advantages of air cooling by natural convection are simple, safe and cost effective.

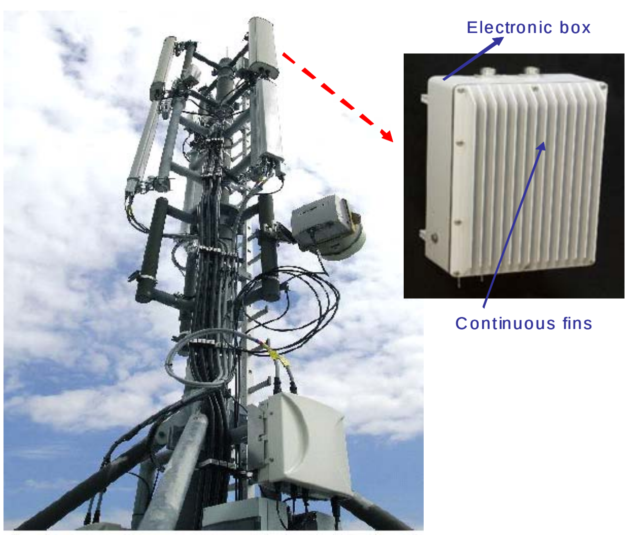


Figure 1 Tower Mounted Electronics with Finned Heat Sink

Figure 2 shows a typical configuration of a continuous fin array. The flow field over a finned heat sink is much complicated than the typical flow over a single plate or in parallel plates because of the involvement of the third surface (fin base). The finned heat sink as presented in Figure 2 consists of a number of U-shaped channels. Little fluid flows through the corners which are formed by the base plate and the fins results in a significant reduction in the heat transfer.

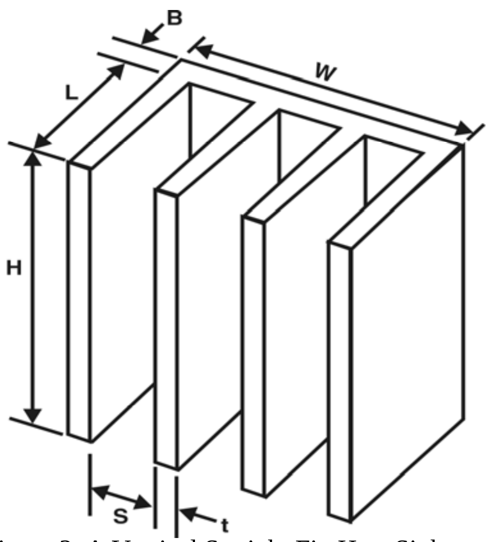


Figure 2 A Vertical Straight Fin Heat Sink

There are many studies of the natural convection from the multiple surfaces such as parallel plates or finned heat sinks by numerical analysis or by experiment. For vertically straight- fin heat sinks, several experimental data [1-3] are available. Among them, Izume and Nakamure [3] developed a mathematical relationship describing heat transfer from the finned heat sink, however, their equation does not hold in the limiting cases of very large or very small ratios of the channel depth (fin height, L) to channel width (fin spacing, S). To overcome this problem, Van De Pol and Tierney [4] developed the following empirical equation applicable to any channel depth to width ratios.

$$Nu_r = (Ra^* / \psi)\ \{1 - \exp[-\psi\ (0.5/Ra^*)^{3/4}]\} \qquad (1)$$

Where

- $Nu = h_u\ r / k$
- $r = 2LS/(2L+S)$,
- $Ra^* = (r/H)\ Gr_r\ Pr$,
- $Gr_r = g\beta(\rho/\mu)^2\ (T_w - T_a)\ r^3$,
- $\psi = 24\ (1 - 0.483\ e^{-0.17/a})/\{(1+0.5a)[1+(1-e^{-0.83a})(9.14a^{0.5}e^{V*S} - 0.61)]^3$,
- $a = S/L$,
- $V = -11.8$ (1/in).

When the ratio L/s approaches zero, Equation (1), U-channel correlation, is reduced to Equation (2a) for a single vertical plate. On the other hand, when r approaches s as $L/s \to \infty$, the Nusselt number in Equation (1) is reduced to the parallel plate correlation, Equation (2b) developed by Elenbaas in 1942.

$$Nu = 0.59\ Ra_H^{\ 0.25} \qquad (2a)$$

Where

- Nu = Nusselt number, $h\ H/k$,
- Ra_H = Rayleigh number, $Gr\ Pr$,
- Pr = Prandtl number, $c_p\ \mu / k$,
- Gr = Grashof number, $\beta\rho^2 g\ H^3\ \Delta T/\ \mu^2 = B\ H^3\ \Delta T$,
- H = Plate height,
- ΔT = temperature difference between the wall and ambient,

And,

$$Nu = (1/24)\ (b/H)\ Ra\ [1 - e^{-35\ H/(b\ Ra)}]^{3/4} \qquad (2b)$$

where the Nusselt and Grashof numbers are based on the plate spacing, b. All properties are evaluated at the wall temperature, T_w except β is computed at the ambient (or free stream) temperature, T_a.

Equation (1) is limited to the uniform wall temperature condition and is only applicable to the continuous straight fins as shown in Figure 2. In addition, h_u is the heat transfer coefficient for a U-channel. The total heat loss by natural convection from a vertical straight-fin array as shown in Figure 2 which includes the loss from every exposed surfaces of the heat sink can be expressed by the following equation

$$q_c = [(n-1)(2LH+HS)\, h_u + 2(L+B)H\, h_v + nHt\, h_v + BW\,(h_t + h_b) + ntL\,(h_t + h_b)]\,(T_w - T_a) \quad (3)$$

Where

hu – heat transfer coefficient from a U-channel

hv – heat transfer coefficient from a vertical plate

ht – heat transfer coefficient from a horizontal plate with heated surface facing up

hb – heat transfer coefficient from a horizontal plate with heated surface facing down

For horizontal plates, the following equations can be applied

(a) Heated plate facing upward

$Nu = 0.54\, Ra_L^{\;0.25}$ $\quad 10^5 < Ra_L < 4 \times 10^7$ (Laminar) (4.a)

$Nu = 0.162\, Ra_L^{\;1/3}$ $\quad 4 \times 10^7 < Ra_L$ (Turbulent) (4.b)

(b) Heated plate facing downward

$Nu = 0.27\, Ra_L^{\;0.25}$ $\quad 3 \times 10^5 < Ra_L < 3 \times 10^{10}$ (4c)

For a horizontal plate, the characteristic length, L for a square plate is taken as the length of the side of the square, for a finite rectangular plate as the arithmetic mean of the lengths of two sides, and for a circular disk as 0.9 x disk diameter. For other shaped plates, L is defined as A/P, where A is the surface area and P is the perimeter.

Thermal Optimization

With advancement of CFD analysis tools, engineers are often just simply applying CFD analysis tools for the system analysis and design without thermal optimization of the individual heat sinks on the components. It would save a lot of time and efforts if individual heat sinks on the components are thermally optimized prior to using the CFD analysis. The following practical example of a telecommunication rack as illustrated in Figure 3 is to demonstrate how to utilize Equation (1) for optimizing component heat sinks. As shown in Figure 4, the heat sink widths under consideration are 40mm, 65mm and 90mm and each width with four different heat sink lengths (30mm, 40mm, 50mm, and 60mm) in the flow direction. For given heat sink width and length, analyses are performed to determine the optimal fin spacing for the maximum natural convection heat loss. The fin thickness for all configurations is 1.2mm and the heat sink is made of anodized aluminum. The ambient is assumed at 50 °C and the heat sink base at 85 °C. The former represents inlet air temperature over a printed circuit board and the latter corresponds to the component case temperature.

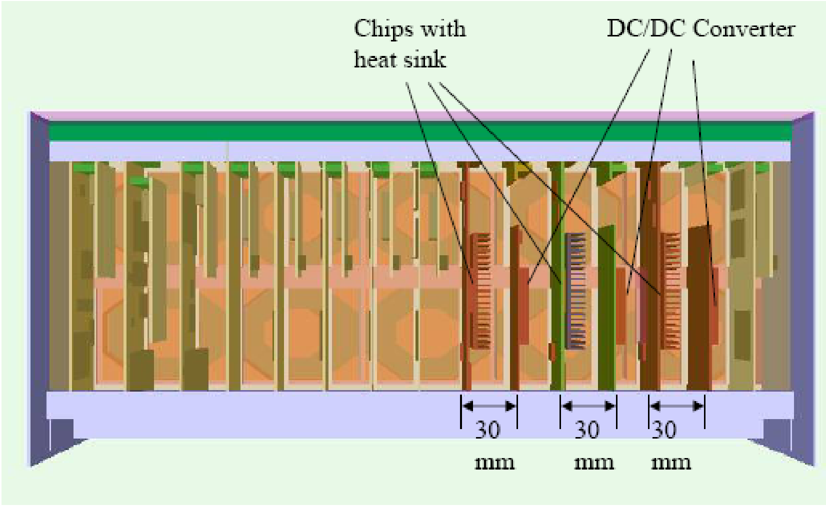


Figure 3 A Telecommunication Rack with Printed Circuit Boards

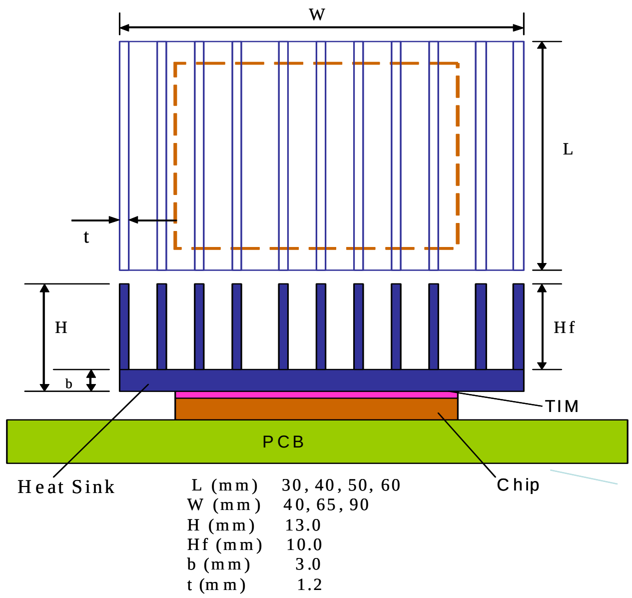


Figure 4 Heat Sink Assembly under Consideration

It should be noted that all other dimensions of the heat sink remain unchanged while varying the fin spacing to achieve the maximum heat loss which is the product of the heat transfer coefficient, heat transfer surface area and the overall surface efficiency. The heat loss versus fin spacing is shown in Figures 5, 6, and 7 for the heat sink widths of 40mm, 65mm, and 90mm, respectively. The summary results for all cases are given in Table 1.The optimal fin spacing is a function of fin thickness, heat sink width and length. From Table 1, the optimal fin spacings are determined to be 5.27mm, 5.18mm, and 5.14 mm for the heat sink width of 40mm, 60mm and 90mm, respectively. For practical engineering practice, a single optimal fin spacing of 5.1 2mm which is the average value of three optimal fin spacings, (5.27+5.18+5.14)/3, is selected for all heat sink configurations considered. By doing so, it greatly enhances the manufacturing and assembling processes while there is no noticeable impact on the thermal performance of all heat sink configurations.

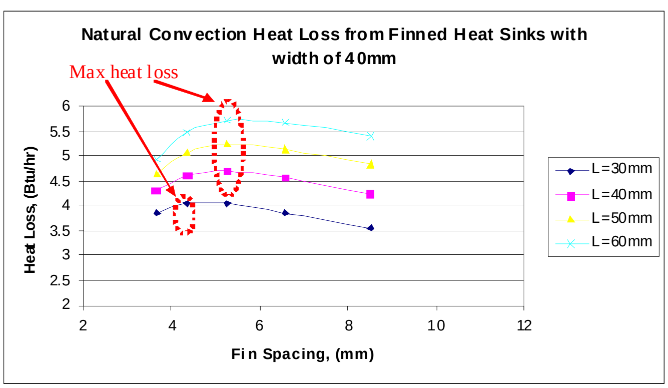


Figure 5. Natural Convection Loss for Heat Sink Width of 40mm

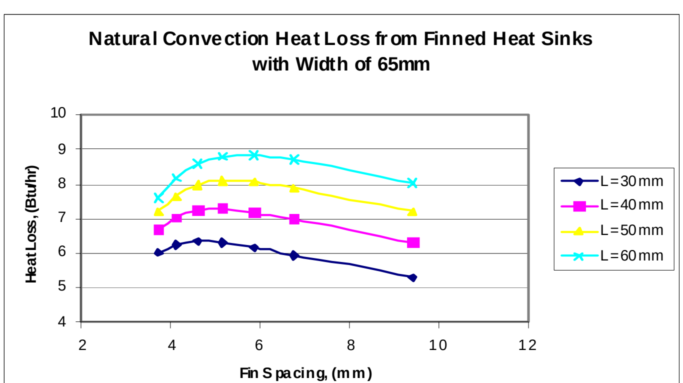


Figure 6 Natural Convection Loss for Heat Sink Width of 65mm

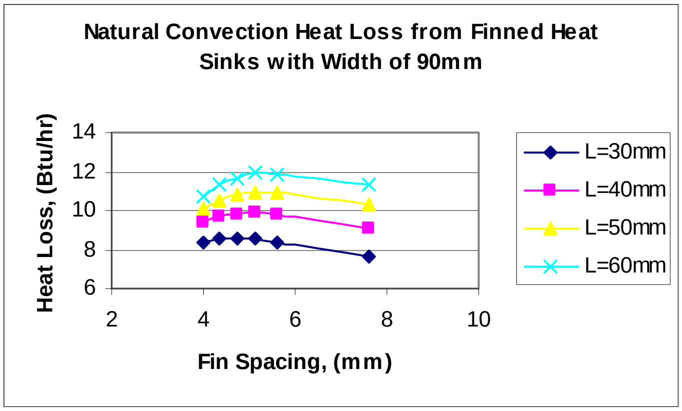


Figure 7 Natural Convection Loss for Heat Sink Width of 90mm

Table 1 Summary Results for All Heat Sink Configurations

Total Heat Loss (Btu/hr) for Heat sink Width of 40 mm

Optimal spacing: 5.27mm

fins	spacing(mm)	L=30mm	L=40mm	L=50mm	L=60mm
9	3.64998	3.853	4.291	4.637	4.923
8	4.3434	4.047	4.611	5.071	5.459
7	5.26796	4.039	4.69	5.243	5.722
6	6.56082	3.861	4.547	5.146	5.68
5	8.49884	3.564	4.24	4.844	5.393

Total Heat Loss (Btu/hr) for Heat sink Width of 65 mm

Optimal spacing: 5.18mm

fins	spacing(m	L=30mm	L=40mm	L=50mm	L=60mm
14	3.7084	6.017	6.674	7.182	7.593
13	4.11734	6.231	7.017	7.641	8.154
12	4.59994	6.32	7.216	7.949	8.563
11	5.17906	6.288	7.267	8.087	8.791
10	5.88772	6.151	7.179	8.059	8.828
9	6.77418	5.927	6.973	7.885	8.695
7	9.43356	5.295	6.305	7.21	8.035

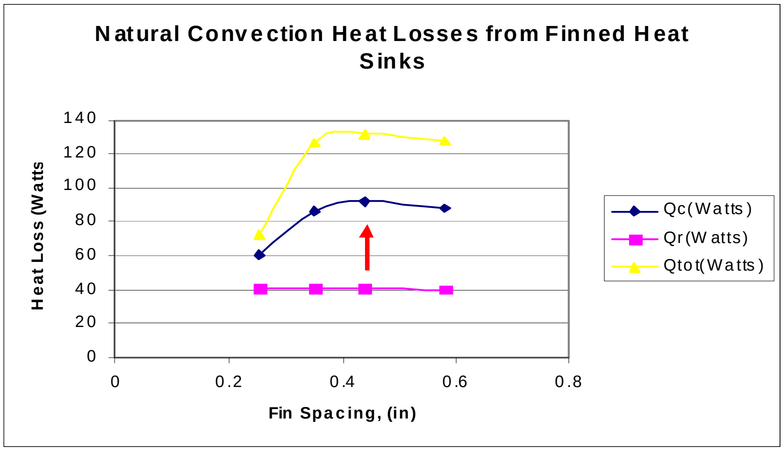
Total Heat Loss (Btu/hr) for Heat sink Width of 90 mm

Optimal spacing: 5.14mm

fins	spacing(mm)	L=30mm	L=40mm	L=50mm	L=60mm
18	4.0005	8.353	9.345	10.12	10.746
17	4.350004	8.534	9.659	10.559	11.299
16	4.71932	8.578	9.804	10.806	11.645
15	5.1435	8.539	9.919	10.933	11.969
14	5.6007	8.396	9.75	10.896	11.887
11	7.59968	7.671	9.059	10.281	11.375

The same process is also applied to a large heat sink with the overall dimensions of 15" (height) x 10.341" (width) x 2.2"(depth). The summary results are given in Figure 8

It should be noted that Equation (1) can be applied to various sizes of the heat sink. Radiation heat transfer must always be included under the natural convection conditions. This is especially true for the air at high altitudes where the effectiveness of natural convection is significantly reduced due to reduction of the air density.



	spacing(in)	Qc(Watts)	Qr(Watts)	Qtot(Watts)
Optimal	0.253	60.60193	40.54716	71.857938
Fin	0.35	86.28852	40.7853	127.07381
Spacing →	0.439	91.92677	40.29994	132.08319
	0.583	87.69977	40.17252	127.87229

Figure 8 Natural Convection Loss versus Fin Spacing of Finned Heat Sink

CFD Analysis

Several CFD simulations are first performed to compare with the results from the correlations. The good agreement of the U-channel heat transfer coefficient between the correlation, Equation (1) and CFD results as given in Figure 9 further validates the accuracy of Equation (1). Except for narrow fin spacing of 0.253 in, the percentage error for all other fin spacing is less than 4% as compared with the CFD results. Equation (1) is very useful for practical design and analysis because of its simplicity and ease for obtaining quick results.

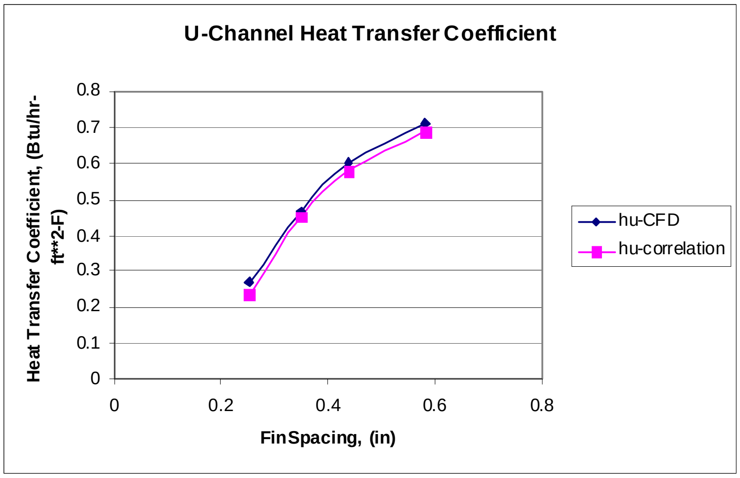
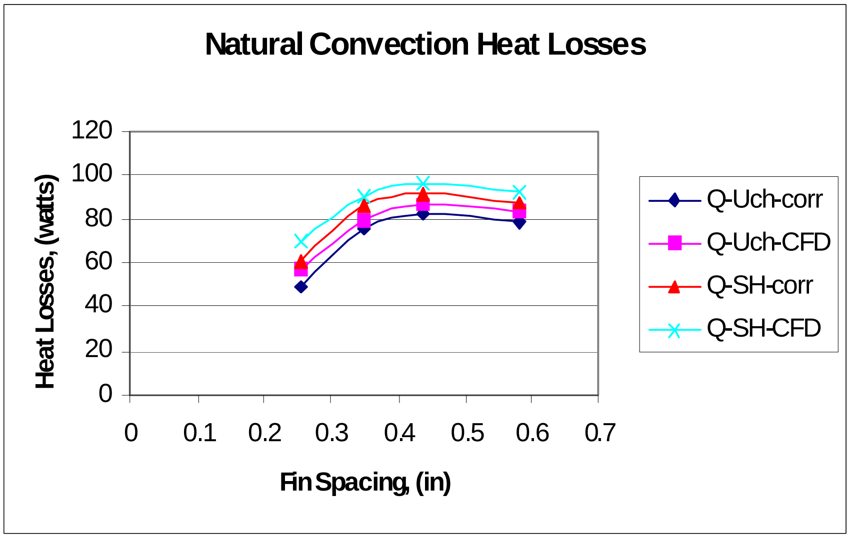


Figure 9 Comparison of U-Channel Heat Transfer Coefficient between Correlation and CFD

Comparisons of natural convection heat loss between the results from correlation and CFD are shown in Figure 10. The good agreement between the results from the correlations and CFD for various fin spacing is found. The ratio of the natural convection heat loss from the U-channels to the entire finned heat sink is given in Figure 11. As can be seen from the figure, this ratio increases as the fin spacing increasing. For the heat sink with fin spacing of 0.439”, this ratio is 0.898. Since the U-channels is the predominant factor in determining the heat loss from the heat sink, the optimal fin spacing can be determined by optimizing the U-channel heat transfer coefficient alone.



Q-Uch-corr : heat loss from U-channel by correlation
Q-Uch-CFD : heat loss from U-channel by CFD
Q-SH-corr : heat loss from entire heat sink by correlations
Q-SH-CFD : heat loss from entire heat sink by CFD

Figure 10 Comparison of Heat Loss between CFD and Equation (3)

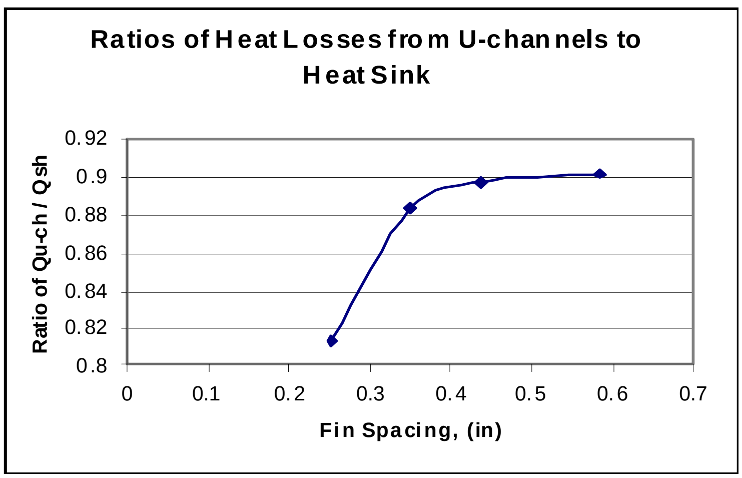


Figure 11 Ratio of Natural Convection Heat Loss to Finned Heat Sink

The U-channel correlation which has been validated is very useful for practical design and analysis because of its simplicity and ease for obtaining quick results.

The main focus of the present work is to perform a detailed CFD analysis on the heat sink with the fin optimal spacing of 0.439”.The overall fin dimensions of heat sink are 15” x 10.341” x 2.2”. The dimensions of the CFD model are as follows:

Fin width (in) : 10.341
Fin length (in) : 15
Fin spacing (in) : 0.439
Fin height (in) : 2.0
Fin thickness (in) : 0.1
Fin base plate thickness (in) : 0.2
Fin numbers : 20

The temperature of the heat sink base and the ambient air temperature are assumed to be at 176 °F (80 °C) and 120 °F (49 °C), respectively. In addition, the air density is considered to be at the sea level. Because of light weight and high thermal conductivity, the aluminum heat sink is often a preferred choice to be used in cooling of electronics. The thermal conductivity of the aluminum alloy is 80 Btu/hr-°F-ft (137 W/m-°C). The properties of the fluid (air) are considered to be constant and to be evaluated at 176 °F.

This heat sink model is vertically divided into is 15 1-inch sections along the length of the heat sink. The flow field for the vertical heat sink is given in Figure 12. As can been seen, the velocity is relatively uniform when the air first enters the finned heat sink from the bottom side (low end). However, due to the entrant flow entering spacing between the fin tips at the face of the heat sink, the velocity of the air flow increases along the length of the heat sink when the air flows upwards as illustrated in Figure 13. The flow rates entering and exiting from the heat sink are 1.8 and 6.92 cfm, respectively. The explored view of the entrain flow per zone is given in Figure 14. The entrant flow rate per ranges from 0.23 to 0.49 cfm and the total entrant flow rate over the entire heat sink is 5.08 cfm.

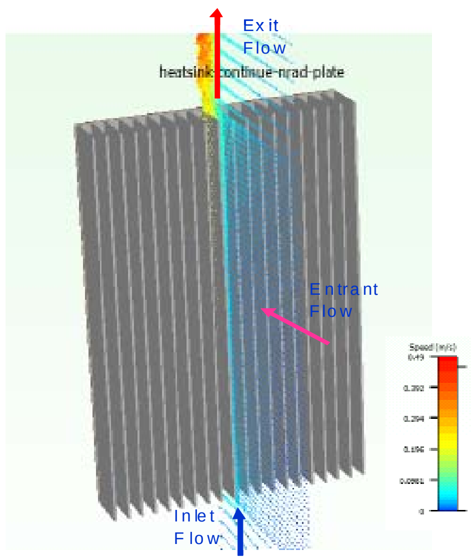


Figure 12 Velocity Profile for Vertical Finned Heat Sink

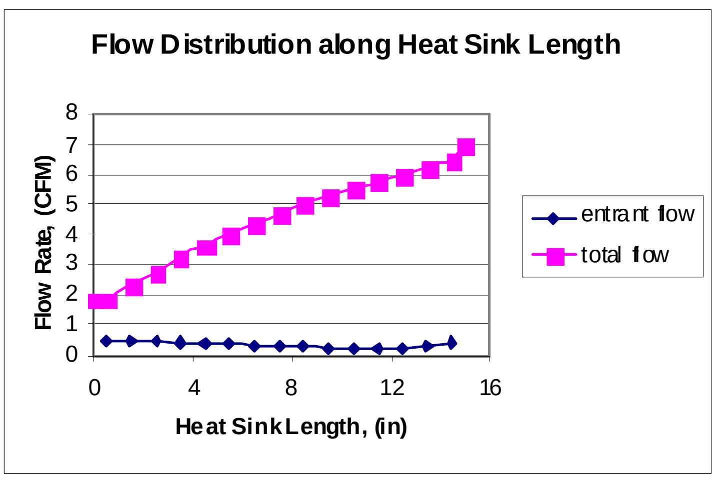


Figure 13 Flow Distribution along Heat Sink Length

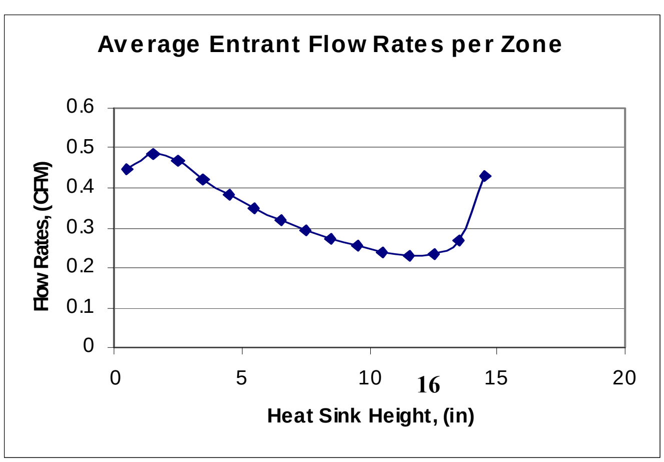


Figure 14 Average Entrant Flow per Zone

The average natural convection heat loss per zone is shown in Figure 15. The results indicate that the heat loss per zone decreases along the heat sink length (height). The reason is that the temperature of air increases as the air flow upwards along the heat sink. The total natural convection heat loss of this heat sink is 96.61 watts. The percentage difference of the heat loss between the results from CFD and the correlations is 4.8% (96.61 watts for the CFD versus 91.78 watts from the correlations). The total heat loss from the heat sink is 96.61 watts and the total exposed convection area is 1485.25 in^2. The heat transfer coefficient of the entire heat sink is 0.7 Btu/hr-ft^2.

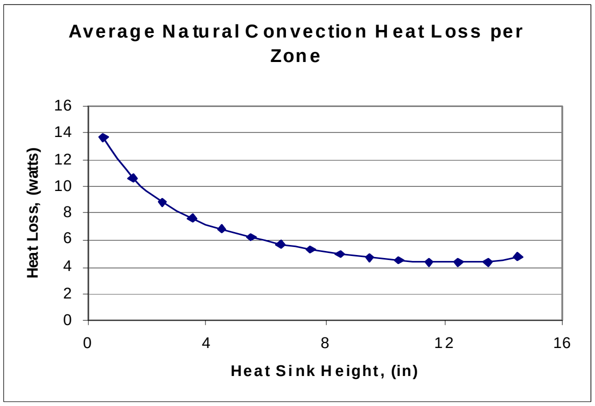


Figure 15 Average Heat Loss per Zone

The wireless operators employ a cover or shroud to conceal the equipment in order to meet local zoning regulations. The present analysis is extended to examine the effects of the cover/shroud on the heat transfer of a finned heat sink as illustrated in Figure 16. To further understand the effect of the distance between the cover and the fin tip of the heat sink, this distance is varied from zero to 99”. The zero distance represents the case with the cover directly in contact with the heat sink fins. The representative velocity profile for X=4.36” is shown in Figure 17.

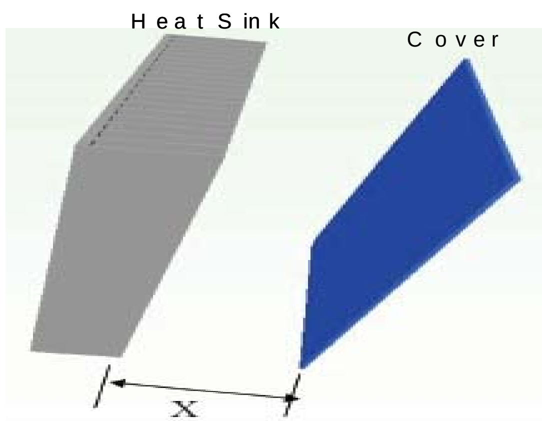


Figure 16 Heat Sink with Cover at X Distance

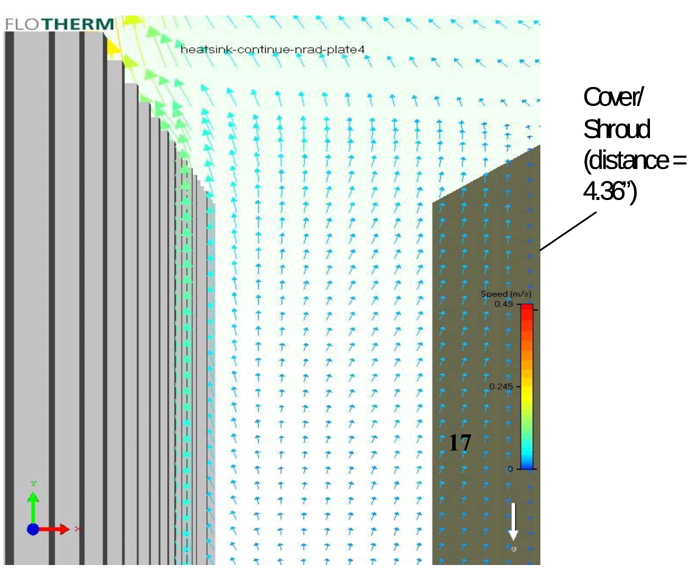


Figure 17 Velocity for Shroud at 4.36” from Fin Tip

The entrant flow rate per zone is given in Figure 18. For the case with the cover in contact with the fin tip, i.e., x=0, there is no entrant flow. In other words, the induced flow enters a set of parallel rectangular channels so that the inlet flow rate equals to the exit flow rate. Examining the results in details reveals that the curves for the cases with no cover and cover at x>4.36” collapse together except the lowest zone immediate next to the inlet. The detailed flow distribution is summarized in Table 2. In other words, there is no effect of cover/shroud on the entrant flow rate when the distance x is greater than 4.36”

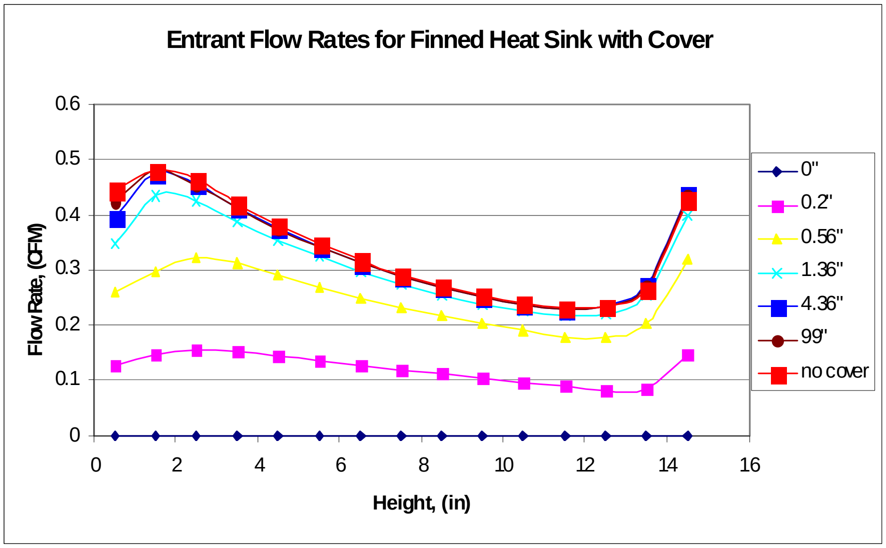


Figure 18 Average Zone Flow Rate for Heat Sink with Cover/Shroud

Table 2 Flow Distribution over Vertical Heat Sink with Cover/Shroud at X-distance

	************		entrant	flow	(cfm)		***********
height (in)	0"	0.2"	0.56"	1.36"	4.36"	99"	no cover
14.5	0	0.1478	0.31854	0.39971	0.43706	0.433692	0.428353
13.5	0	0.08344	0.20176	0.25485	0.27501	0.273653	0.267381
12.5	0	0.08259	0.17949	0.22029	0.23437	0.23456	0.233776
11.5	0	0.08894	0.18092	0.21701	0.2292	0.229856	0.231784
10.5	0	0.09644	0.18988	0.22516	0.23708	0.237738	0.240917
9.5	0	0.10416	0.20197	0.23849	0.25094	0.251469	0.25524
8.5	0	0.11199	0.21601	0.25506	0.2685	0.268822	0.272975
7.5	0	0.12	0.23203	0.27454	0.28927	0.289397	0.293867
6.5	0	0.12831	0.25035	0.29743	0.31351	0.313467	0.318319
5.5	0	0.13699	0.27101	0.3241	0.34192	0.341712	0.347115
4.5	0	0.14573	0.29316	0.35438	0.37544	0.374915	0.381229
3.5	0	0.15346	0.31362	0.38922	0.4145	0.41316	0.42117
2.5	0	0.15668	0.32256	0.42475	0.4549	0.453186	0.464225
1.5	0	0.14862	0.29798	0.43602	0.47645	0.479905	0.481134
0.5	0	0.12772	0.26153	0.34868	0.39714	0.423246	0.444944
Entrant	0	1.83287	3.73081	4.65969	4.99529	5.018778	5.082429
Inlet(low end)	5.8551	4.5312	3.115	2.25258	1.8655	1.7795	1.8004
Exit(high end)	5.8549	6.3961	6.8279	6.93022	6.89844	6.8353	6.9209

The heat losses per zone are summarized in Table 3. The results indicate that system with the cover in contact with fins perform better thermally than that of the case without the cover. The reason is that the cover provides an additional surface area for heat transfer from the heat sink to the ambient. In addition, the results also reveal that the total heat loss of the heat sink decreases as the distance between the fin tip and cover increasing. It is also found that there is no effect of the cover on the heat loss or entrant flow rate as long as the distance between the cover and the heat sink fin tips is greater than 4.36” with the fin height of 2.0”. Based on the limited data in this work, one may conclude that there is no effect of the cover on the heat transfer of a finned heat sink if the distance between the heat sink and the cover is greater than 2.5 times of the fin height.

Table 3 Heat Losses from Vertical Heat Sink with shroud at X Distance

height (in)	Heat Loss (watts) 0"	0.2"	0.56"	1.36"	4.36"	99"	no cover
14.5	2.8428	3.0012	4.0315	4.5292	4.7213	4.7312	4.7472
13.5	2.6136	2.7761	3.7156	4.159	4.3183	4.3214	4.3547
12.5	2.7573	2.911	3.7531	4.1274	4.2515	4.2474	4.2932
11.5	2.9987	3.1269	3.8824	4.2036	4.2993	4.2891	4.3462
10.5	3.3315	3.4095	4.0881	4.3668	4.4396	4.4236	4.49
9.5	3.7587	3.7589	4.3574	4.5979	4.6497	4.6283	4.7029
8.5	4.2929	4.1857	4.6895	4.8888	4.9183	4.8915	4.9737
7.5	4.9551	4.7104	5.0936	5.242	5.2214	5.2119	5.303
6.5	5.777	5.3645	5.5904	5.6698	5.6367	5.5994	5.7007
5.5	6.8033	6.1945	6.2139	6.1969	6.1174	6.0753	6.1886
4.5	8.1001	7.27	7.0215	6.8672	6.7259	6.679	6.8065
3.5	9.7723	8.701	8.1118	7.7614	7.5354	7.4831	7.627
2.5	12.014	10.681	9.6625	9.036	8.6907	8.6299	8.834
1.5	15.245	13.602	12.036	11.03	10.514	10.434	10.562
0.5	20.405	18.438	16.252	14.764	14.01	13.886	13.678
Total Loss	105.6673	98.1307	98.4993	97.44	96.0495	95.5311	96.6077

Summary and Conclusion

A CFD analysis is performed on a vertical heat sink with the continuous fins and the results compare well with those from the existing correlations. However, the correlations limit to the case with the constant wall temperature. The U-channel correlation which has been validated by present CFD results is very useful for practical design and analysis, especially in conceptual design phase because of its simplicity and ease for obtaining quick results. It would save a lot of time and efforts if individual heat sinks are thermally optimized by using the U-channel correlation, Equaiton (1), prior to utilizing the CFD analysis

The flow field ultimately determines the heat transfer from the heat sink. Therefore, an effort is made to provide the insight view of the detailed flow fields which has never been done before. The velocity is relatively uniform when the air first enters the finned heat sink from the bottom side (low end). However, due to the entrant flow entering spacing between the fin tips at the face of the heat sink, the velocity of the air flow increases along the length of the heat sink when the air flows upwards. The entrant flow rate per ranges from 0.23 to 0.49 cfm and the total entrant flow rate over the entire heat sink is 5.08 cfm. The flow rates entering and exiting from the heat sink are 1.8 and 6.92 cfm, respectively.

The CFD results indicate that the heat loss per zone decreases along the heat sink length (height). The total natural convection heat loss of this heat sink is 96.61 watts. The ratio of the natural convection heat loss from the U-channels alone to the entire finned heat sink is 0.898. The total heat loss from the heat sink is 96.61 watts and the total exposed convection area is 1485.25 in^2. The heat transfer coefficient of the entire heat sink is 0.7 Btu/hr-ft^2 while the heat transfer coefficient from the U-channels alone is 0.61 Btu/hr-ft^2.

The CFD results indicate that system with the cover in contact with fins perform better thermally than that of the case without the cover. It is also found that there is no effect of the cover/shroud on the heat loss or entrant flow rate as long as the distance between the cover and the heat sink fin tips is greater than 4.36" with the fin height of 2.0". Based on the limited data in this work, one may conclude that there is no effect of the cover on the heat transfer of a finned heat sink if the distance between the heat sink and the cover is greater than 2.5 times of the fin height.

Radiation heat transfer must always be included under the natural convection conditions. This is especially true for the air at high altitudes where the effectiveness of natural convection is significantly reduced due to reduction of the air density. The heat sink under consideration can be applied to both indoor and outdoor equipment. However, the direct solar heating must always be included in the system design and analysis for any outdoor systems.

REFERENCES

1. Starner, K.E., and McManus, H. N, "An Experimental Investigation of Free Convection Heat Transfer from Rectangular Fin Arrays", J Heat Transfer 85, 1963

2. Welling, J.R. and Wooldridge, C. R., "Free Convection Heat Transfer Coefficients from Rectangular Vertical Fins", J Heat Transfer 87, 1965
3. Izume, K, and Nakamura, H, "Heat Transfer by Convection on Heated surface with Parallel Fins", Jap. Soc. Mech. Eng., 34, 1969
4. Van De Pol, D. W., and Tierney, J.K., "Free Convection Nusselt Number for Vertical U-Shaped Channels", Journal of Heat Transfer, 95, 1973
5. Yeh, L.T., "Natural Convection from Finned Heat Sinks with/without Cover/Shroud", 19th International Symposium on Transport Phenomena, Reykjavik, Iceland, August 17th – 21st, 2008
6. Yeh, L.T., and Chu, R. C., Thermal Management of Microelectronic Equipment", 2nd Edition, ASME Press, 2016