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Effects of Ambient Temperatures on Natural Convection Heat Transfer from Finned Heat Sinks

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Abstract: The electronic equipment is often required to operate at various ambient temperatures in the real world conditions, especially for an outdoor system. A cfd analysis is performed to study the effects of the ambient temperatures for the finned heat sink on the heat transfer coefficient under the natural convection conditions. Four different ambient temperatures, 100, 120, 140 and 160 ºF are considered for the continuous fin, staggered fin and in-line fin heat sinks. The overall heat transfer coefficient of individual finned heat sinks is determined to facilitate the thermal analysis and design of the equipment

1. Introduction

The electronic equipment is often required operating at various ambient temperatures in the real world applications such as the outdoor electronics. Therefore, there is an urgent need to address these issues for the practical engineering applications.

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 outdoor electronic equipment such as those shown in Figure 1. The advantages of air cooling by natural convection are simple, safe and cost effective.

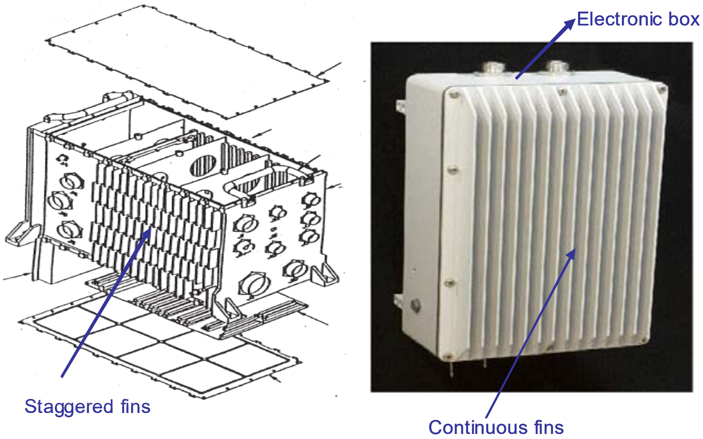

Figure 1 Finned Heat Sink for Outdoor Electronics

Figure 2 shows a typical configuration of a vertically 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 flow through corner regions which are formed by the heat sink base and fins results in a significant reduction in the heat transfer.

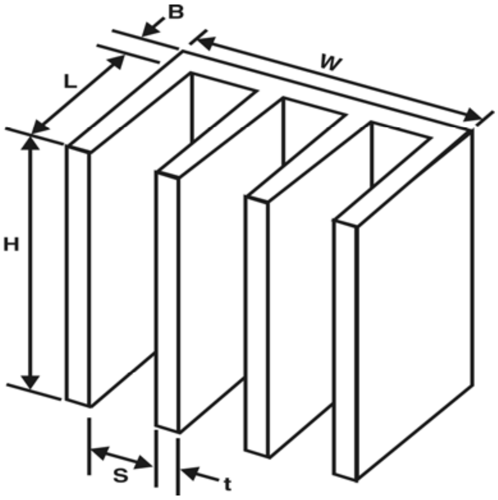

Figure 2 A Vertical Straight Fin Heat Sink

For vertically straight- fin heat sinks as shown in Figure 2, 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 to channel width. To overcome this problem, Van De Pol and Tierney (4) developed an empirical equation which is applicable to any channel depth to width ratios to compute the U-channel heat transfer coefficient. The correlation is limited to the constant wall temperature condition and is only applicable to the continuous straight fins.

Yeh et al. (5) performed a CFD analysis on the continuous fin, staggered fin and in-line fin heat sinks as illustrated in Figure 3 at the constant wall temperature conditions. The results indicate that the continuous fin configuration is the most efficient thermally, and is followed by the staggered fins and then by the in-line fins. Though the in-line fin array has the greatest surface area, it has the least heat loss because of the smallest fin spacing choking the flow.

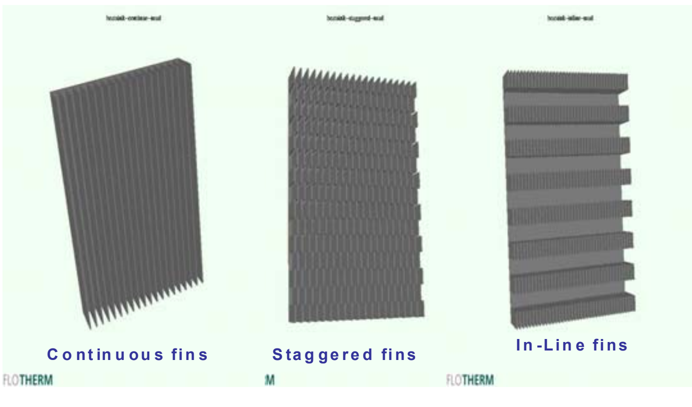

Figure 3 Three Types of Finned Heat Sinks

Yeh (6) extended the analysis to examine the effects of the cover/shroud on the heat transfer. 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 results indicate that there will be no effect of the cover on the heat loss and entrant flow rate as long as the distance between the cover and the heat sink tip is greater than 4.36” with the fin height of 2.0”. Based on the limited data in this work, it is concluded 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.

Thermal Analysis

The previous analyses (5, 6) are limited to the case with a vertical heat sink at the ambient temperature of 120 ºF (40 ºC). Those analyses were made to meet specific product development needs then. However, the electronic equipment is often required to operate at various ambient temperatures for the indoor and outdoor applications. Therefore, the thermal analysis is further extended to all three types of finned heat sinks at various ambient conditions (from 100ºF to 160ºF).

The overall dimensions of the finned heat sink are listed as follows:
Fin width (in) : 10.341
Fin length (in) : 15
Fin thickness (in) : 0.1
Fin height (in) : 2.0
Fin base plate thickness (in) : 0.2
Fin numbers : 20

In order to maintain the component junction temperature below 100ºC, the maximum temperature of the heat sink base must not be over 80ºC and thus it is assumed to be uniform at 176 °F (80 °C). In addition, the air density is considered to be constant and at the sea level. Because of light weight and high thermal conductivity, aluminum is often a preferred choice to be used in cooling of electronics. The thermal conductivity of the heat sink is assumed to be 80 Btu/hr-°F-ft (137 W/m-°C).

Results and Discussions

The heat losses from the heat sink versus the ambient air temperatures for various types of finned heat sinks are given in Figure 4. With the fixed temperature (176 °F or 80 °C) at the heat sink base, the heat loss decreases when the ambient temperature increases. As can been seen in Figure 4, the continuous fin heat sink with the least heat loss is the most effective thermally, then is followed by the staggered fin heat sink. On other hand, the in-line fin configuration has the worst thermal performance even though it has largest convection surface area. This is because the fin spacing of the heat sink is too small to be effective by chocking the air flow through the fins. Table 1 lists the geometrical data of each type of fin configurations.

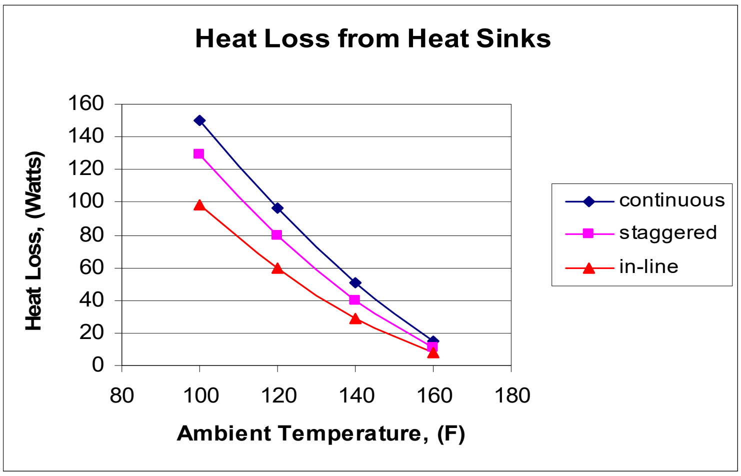

Figure 4 Natural Convection Heat Losses versus Ambient Air Temperatures

Table 1 Comparison among Various Types of Fin Configurations

	Continuous	Staggered	In-Line
Number of fins	20	20/19	39/0
Fin spacing (in)	0.439	0.439	0.1695
Convection area (in^2)	1373.2	1454.4	1537.9

The heat transfer coefficients for various types of fin configurations as functions of the ambient temperatures are calculated and presented in Figure 5. The heat transfer coefficient, h, which is defined as $h = Q/(A\ \Delta T)$ is computed based on the total convection (exposed) surface area as listed in Table 1 and the total heat loss from Figure 4. Figure 5 clearly shows the nonlinear functions of the temperature for the natural convection heat transfer coefficient. Based on the limited data available from the present analysis, the following correlation is developed.

$$h1/h2 = (\Delta T1/\Delta T2)^{0.5} \qquad (1)$$

Where hi is the heat transfer coefficient at ΔTi = Tsurface - Tambient .
The following equation exists for the most cases of the natural convection in laminar flow, including the flow over a vertical plate or horizontal plates,

$$Nu \ \alpha \ Ra_H^{\ 0.25} = (Gr\ Pr)^{0.25} \qquad (2a)$$

where

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

Equation (2a) implies

$$h \ \alpha \ (\Delta T)^{0.25} \qquad (2b)$$

As can be seen from Equations (1) and (2b), one can conclude that the flow through individual U-channels of the finned heat sinks is not a laminar flow. This is due to the fact that the U-channel flow field is complicated. For vertical heat sinks, the flow entering the heat sink from the bottom side is relatively uniform. However, combining the entrant flow between the fin tips at the front side of the heat sink as illustrated in Figure 6 makes the flow field relatively complicated. The flow distributions for the heat sinks operating at the ambient of 49ºC (120ºF) are listed on Table 2.

Table 2 Average flow rate to heat sink (CFM)

	Continuous	Staggered	In-Line
- Entering from bottom	1.8004	1.6026	1.2237
- Entering from fin tip (net entrant)	5.0825	3.4805	1.3625
- Exiting from top	6.9209	5.3127	3.0548

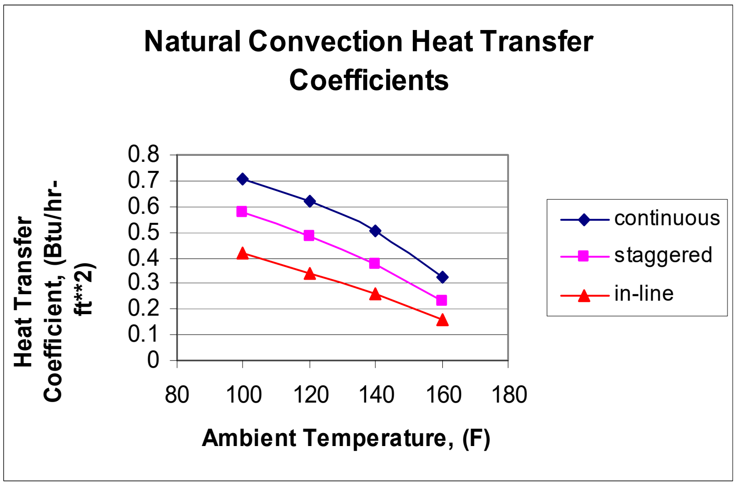

Figure 5 Heat Transfer Coefficients of Finned Heat Sinks

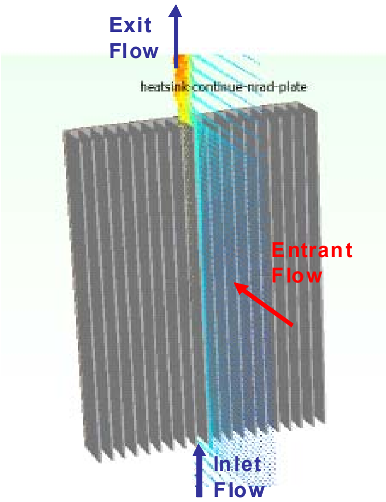

Figure 6 Flow Field for a Vertical Finned Heat Sink

The heat sink thermal resistance which is defined as ΔT/Q (=1/ (h A)) (ºC/W) is often used to measure the thermal efficiency of the heat sink. The thermal resistance of the heat sink as a function of the ambient temperatures is presented in Figure 7. The thermal resistance basically represents an inverse of the heat transfer coefficient.

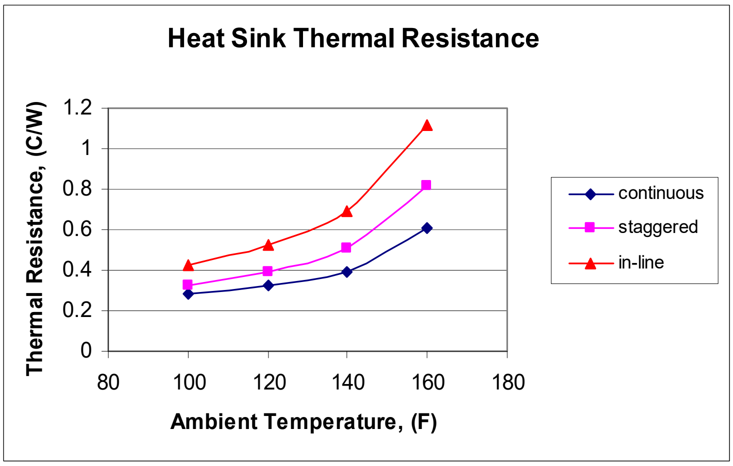

Figure 7 Thermal Resistance of Various Heat Sinks

The system thermal design generally utilizes the ambient temperature based on the worst case condition and it is industry dependent. In other words, individual industry has its standards to follow. For the outdoor equipment, the ambient temperature of 49 ºC (120 ºF) is applied to the system thermal design. The ambient temperature of 71 ºC (160 ºF) is employed if the equipment is located inside a non-air conditioning building which is also subjected to solar heating. Therefore, the temperature range of the ambient from 100ºF to 160ºFare chosen for this study. Table 3 listed the operating and non-operating environments for a wide range of components installations.

Table 3 Electronic Equipment Environments and Heat Sinks

Application	Nonoperating ambient temperature range, °C	Operating ambient temperature range, °C	Comments
Ground			
Inside without temperature control	< −40 to 71	10 to 40	General noncritical commercial installations; air-cooled
Inside with temperature control	< −40 to 71	15 to 25	Air-conditioned rooms; computer rooms air-cooled
World extremes	−62 to 95	−54 to 71	Military gear exposed to the elements
Shipboard	−54 to 71	0 to 50	Internal mounted equipment; for external equipment refer to specification
Aircraft			
Radome	−62 to 95	−54 to >100*	Antenna can be free-convection or liquid-cooled
Equipment bay	−62 to 95	−54 to 71	Coolant supplied in accordance with Fig. 2.13
Cockpit	−54 to 71	0 to 32	Cockpit units self-air-cooled
Missiles			Forced-convection skin cooling during flight heat storage
Captive flight	−54 to 95	54 to >100*	
Free flight	—	−54 to >100	
Spacecraft			Conduction to mounting structure is primary heat sink; external equipment subject to wide range of cold to hot, depending on orbit
Launch pad	−30 to 71	—	
In orbit	0 to 40	0 to 40	

*Short duration periods.

Summary and Conclusion

The electronics are often required operate at various ambient temperature ranges in the real world applications. The air cooling by the natural convection is the most simple, safe and cost-effective method. Therefore, the present study is focusing at the analysis of natural convection heat transfer over the wide range of ambient temperatures from 100ºF to 160ºF.

The continuous fin is the most efficient thermally, and is followed by the staggered fins and then by the in-line fins. The in-line fin array has the greatest surface area but it has the least heat transfer coefficient because of the smallest fin spacing choking the flow. As given in Figure 5, it clearly shows the nonlinear functions of the temperature for the natural convection heat transfer coefficient. Based on the limited data available from the present analysis, the heat transfer coefficient is a function of $(\Delta T)^{0.5}$ where ΔT is the temperature difference between the heat sink base and the ambient.

The radiation heat transfer must always be included in the analysis and design of any system under the natural convection (passive cooling method). This is especially true for the system operating at high altitudes where the radiation becomes even more important because the air density is smaller. In addition, the solar heating must also be considered for cooling of the outdoor equipment.

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