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Effects of System Orientations on Natural Convection Heat Transfer from Finned Heat Sinks

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Abstract: The electronic equipment is often required to operate at various orientations in the real world applications. This is especially critical to the tower or pole or roof top mounted electronics under natural convection. However, it appears that no study has ever been made to examine the effects of orientations of the heat sink over full 90 degrees range from the vertical to the horizontal positions. Therefore, the present study is focusing at the analysis of natural convection heat transfer over the wide range of heat sink orientations to fulfill the needs for the industry.

A CFD analysis is performed to obtain the heat transfer coefficient of individual cases in order to facilitate thermal analysis and design of the electronic equipment. 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 over full 90 degrees range of angles of inclinations which has never been done before. The results indicate that the velocity profile of the group with small angels of inclination for θ = 0º, 30º, and 45º is completely different from the group with large angels of inclination for θ = 75º and 90º. It is also found that the vertical heat sink has the best thermal performance. Examining the results in details also reveals that the heat transfer coefficient from the heat sinks decreases as the angle of inclination from the vertical plane increasing from 0º to 60º, and then increases with increasing angles of the inclination from 60º to 90º .

1. 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.

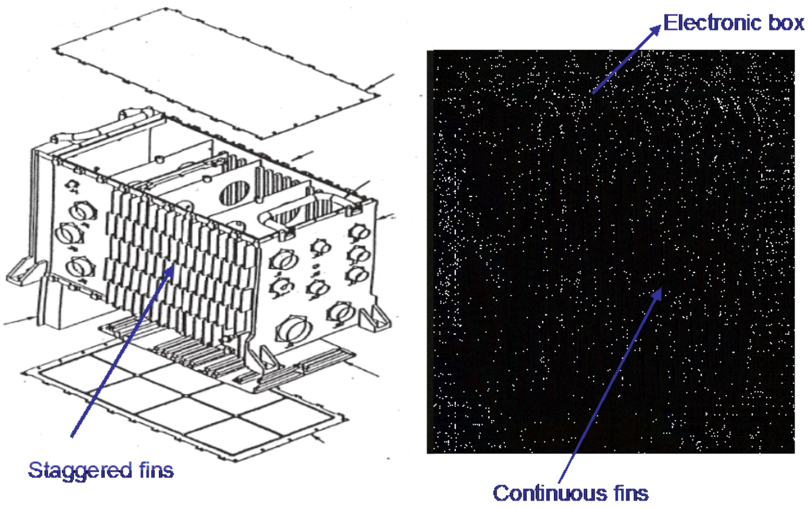


Figure 1 Finned Heat Sink for Outdoor Electronics

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.

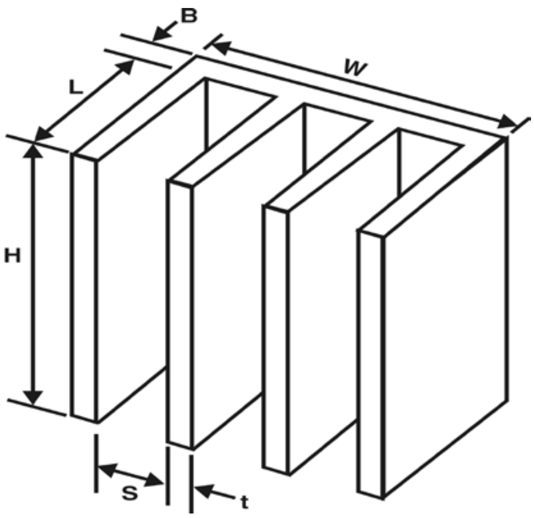


Figure 2 A Vertical Straight Fin Heat Sink

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 (fin height, L) to channel width (fin spacing, S). 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. (5,6) 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.

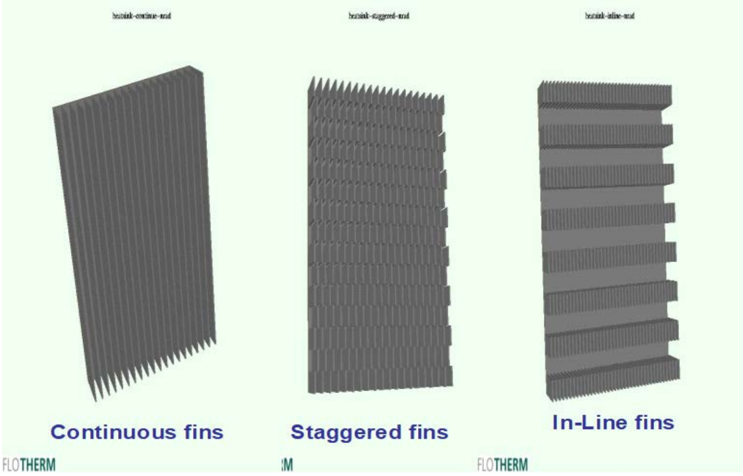


Figure 3 Three Types of Finned Heat Sinks

The electronic equipment is often required to operate at various orientations in the real world applications. However, it appears that no study has ever been made to examine the effects of orientations of the heat sink over full 90 degrees range from the vertical to the horizontal positions. The purpose of this study is to fulfill the needs from electronics industry, especially for those tower, or pole or roof-top mounted wireless communications systems. The results from the present study can be directly applied to the indoor as well as outdoor equipment under the natural convection environments.

2. Thermal Analysis

The previous analyses (5) are limited to the case with a vertical heat sink at the ambient temperature of 120 ºF (49 ºC). Those analyses were made to meet specific product development needs then. However, the electronic equipment is often required to operate at various orientations in the real world applications. This is especially true with recent wireless communication devices that the heat sink orientation is no longer limited to the vertical position.
The overall dimensions of this continuous fin 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

The temperature of the heat sink base is assumed to be uniform at 176 °F (80 °C). In addition, the air density is considered to be at the sea level. Furthermore, the heat sink is also assumed to be suspended in the air. 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). The properties of the fluid (air) are considered to be constants and are evaluated at 176 °F.
A CFD analysis is performed by using a commercial code of Flotherm. The thermal modeling including optimizing fin spacing and grid refinement processes are described in details in Reference 5.

3. Results and Discussions

Since the natural convection is strongly dependent of the shapes and orientations of the system under consideration, the analysis is performed to examine the effects of the orientations of the heat sink on heat transfer. The effects of the orientations on the heat transfer are limited to the case with the continuous fin arrays because it has the best thermal performance among all three fin configurations. The orientation of the heat sink is based on the inclined angles from the vertical plane as illustrated in Figure 7. The cases under consideration include the inclined angles from the vertical plane of 0º, 30º, 45º, 60º, 75º, and 90º. The vertical heat sink represents the angle of the inclination angle of zero while the horizontal heat sink corresponds to the case with the inclination angle of 90º.

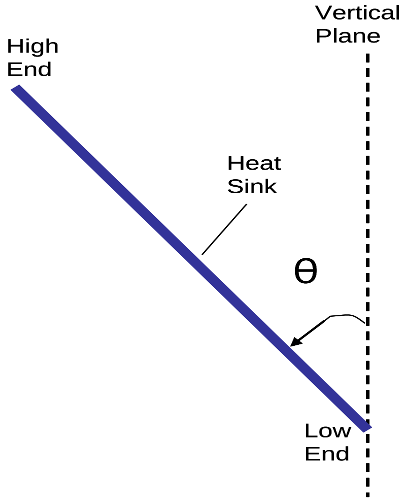


Figure 4 Orientations of Finned Heat Sink

A) Flow field

The flow fields for the vertical heat sink with inclination angles of 0º is given in Figure 4. 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 between the fin tips at the front of the heat sink, the velocity of the air flow increases along the length of the heat sink when the air flows upwards.
The flow field for the horizontal finned heat sink is shown in Figure 5. The air flow patterns for the vertical and horizontal heat sinks are completely different. For a vertical heat sink, the flow enters not only from the bottom but also from the front face between fins of the heat sink. The latter is referred to as the entrant flow. On the other hand, the air flow enters the horizontal heat sink from both ends and flows inside the U-channels formed by fins along the length of the heat sink toward to the center. The hot air finally exits near the middle of the heat sink and flows upwards. For a horizontal heat sink, a small amount of the cold entrant flow enters the heat sink near both end sections. However, the hot exhaust air flows out of the heat sink in the middle section which is completely different from the pattern found on the vertical heat sink.

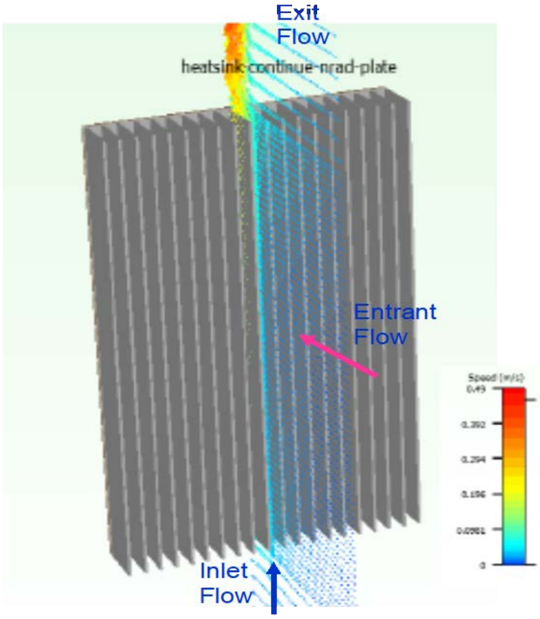


Figure 5 Flow Fields for Vertical Finned Heat Sink

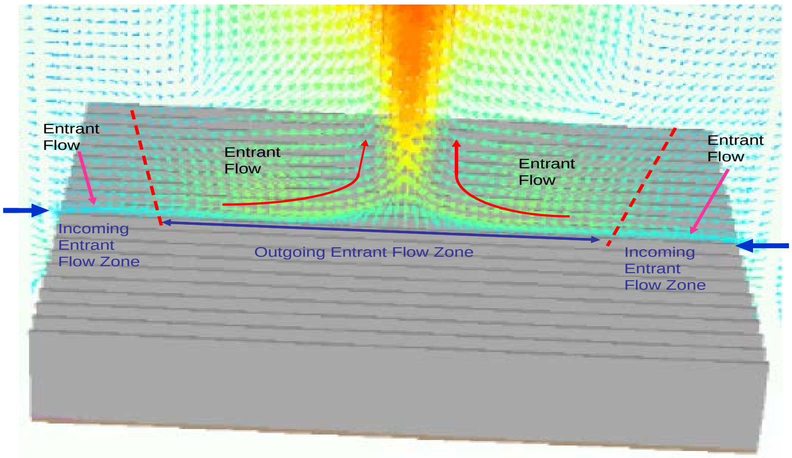


Figure 6 Flow Fields for Horizontal Finned Heat Sinks

The detailed flow distributions over the heat sinks for all cases are listed in Table 1. The model is broken into 15 1-inch sections along the length of the heat sink, starting with zone 0 at the high end down to the zone 14 at the low end of the heat sink. The minus sign ("-") represents the entrant flow exiting the heat sink. The middle section of the table describes the average zone entrant flow rates which are also plotted in Figures 6 and 7, for the entrant flow either entering or exiting from the spacing between the fin tips along the entire length of the heat sink. As can be seen from Figure 11, the symmetrical velocity profile is found for the case of horizontal heat sink with $\theta = 90^{o}$. This is because of the symmetrical thermal boundary condition with respect to the mid-plane between both ends of the heat sink.

Table 1 Detailed Flow Distribution over Finned Heat Sinks

Cases	θ=0º m**3/sec	θ=30º m**3/sec	θ=45º m**3/sec	θ=60º m**3/sec	θ=75º m**3/sec	θ=90º m**3/sec
Flow from High End of Heat Sink						
High End	-0.003223	-0.00302	-0.002549	-0.001236	0.000938	0.001054
Entrant Flow Enter or Exit from Spacings between Fin Tips of Heat Sink						
heatsink zone 0	0.000204	0.0001426	8.39E-06	0.0004761	0.000188	5.09E-05
heatsink zone 1	0.000129	9.36E-05	3.00E-05	2.24E-04	-4.56E-04	-6.00E-06
heatsink zone 2	0.000111	8.99E-05	5.54E-05	-6.44E-05	-7.11E-04	-3.31E-05
heatsink zone 3	0.000109	9.34E-05	6.98E-05	-8.01E-06	-5.15E-04	-7.94E-05
heatsink zone 4	0.000112	9.88E-05	7.89E-05	-3.79E-05	-2.68E-04	-1.41E-04
heatsink zone 5	0.000119	1.06E-04	8.64E-05	-5.15E-05	-1.07E-04	-2.38E-04
heatsink zone 6	0.000127	1.13E-04	9.36E-05	-6.00E-05	-3.12E-05	-3.75E-04
heatsink zone 7	0.000137	1.22E-04	1.01E-04	-6.77E-05	-2.63E-06	-4.80E-04
heatsink zone 8	0.000148	1.32E-04	1.10E-04	-7.56E-05	-1.07E-05	-3.66E-04
heatsink zone 9	0.000161	1.43E-04	1.19E-04	-8.34E-05	-2.09E-05	-2.29E-04
heatsink zone 10	0.000177	1.56E-04	1.30E-04	-9.21E-05	-3.14E-05	-1.37E-04
heatsink zone 11	0.000195	1.70E-04	1.42E-04	-1.02E-04	-4.29E-05	-7.80E-05
heatsink zone 12	0.000214	1.84E-04	1.55E-04	-1.14E-04	-5.67E-05	-3.41E-05
heatsink zone 13	0.000227	1.94E-04	1.66E-04	1.27E-04	7.43E-05	-3.53E-06
heatsink zone 14	0.000201	1.85E-04	1.67E-04	1.42E-04	1.03E-04	4.75E-05
Net Flow via Fins	0.002371	0.0020228	1.51E-03	0.0002124	-0.001887	-0.002103
Flow from Low End of Heat Sink						
Low End	0.000835	0.0009021	0.0009883	0.001003	0.001022	0.001054

Note : "-" represents flow exiting heat sinks

For the small inclined angel group (θ =0º to 45º), the flow rates exiting from the high end side represent the total induced air flow through the heat sink. On the other hand, for the large inclined angel group (θ =75º to 90º), the net flow rates through the spacing between fins correspond to the total induced air flow through the heat sink. The results from Table 2 indicate that the total induced flow through the heat sink decreases as the angle of inclination increasing from θ =0º to 60º. This is due to the fact that the induced flow rate for the natural convection depends on the available pressure head of the system (heat sink) which is a function of vertical height of the system. The vertical height of inclined heat sinks is equal to L * cos θ where L is the length of the heat sink which is also the height for the vertical heat sink (θ = 0º).

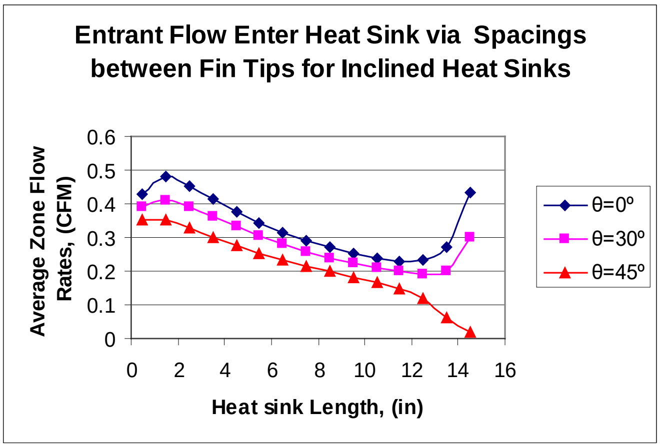


Figure 7 Entrant Flow Entering Heat Sinks

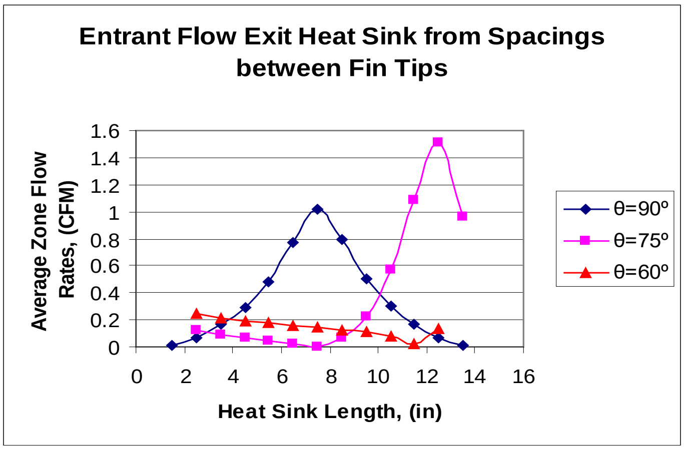


Figure 8 Entrant Flow Exiting Heat Sinks

The mass balance for all orientations of heat sinks is achieved as data listed in Table 1. In other words, the total flow rate enters the heat sink is always equal to the flow exit from the system. For the cases with inclination angle of 0º, 30º, and 45º, the inlet flow rates into the low end of the heat sink plus the total net entrant flow into the system along the length of the heat sink are equal to the exit flow rates at the high end of the heat sink. Similarly for the cases with inclination angle of 90º, and 75º, the total inlet flow rate entering from both ends of the heat sink is equal to the net entrant flow rate exiting the system over the middle section of the heat sink. The following examples are used to demonstrate the continuity equation of the system.

Examples from Table 1:

For vertical heat sink (0º case) :

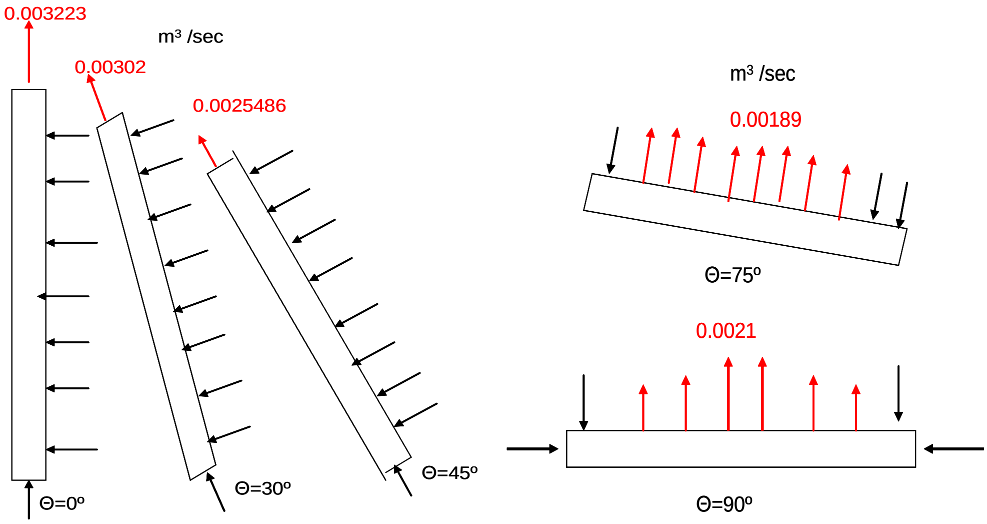
Inlet flow to low end + Net inlet entrant flow = Exit flow leaving high end
0.000835 + 0.002371 = 0.003206 (as compared with CFD result of 0.003223)

For horizontal heat sink (90º case)
Inlet flow to both ends = Net exit entrant flow
0.001054 + 0.001054 = 0.002108 (as compared with CFD results of 0.0021)

The rest of cases have also been checked out.

Data in Table 1 also reveals the different flow patterns between those small angles of inclination (0º to 45º) and large inclination angels (75º to 90º). The completely different flow fields for these two groups are amplified by two extreme cases as shown in Figures 8 and 9 for the vertical and horizontal heat sinks, respectively. Generally, the entrant flow from spacing between fins for small inclination angles is always entering into the heat sink while the majority of entrant flow from spacing between fins is exiting the heat sink.

The sketch of the general flow fields for these two groups is illustrated in Figure 12. The system flow rate for each case is also included in the figure. The flow field for the case with the angle of inclination of 60º which includes the characteristics of both groups appears to be in the transit from the group of small angles of inclination to that of the large inclination angles. The system flow rates as function of the angles of inclination is shown in Figure 13. As can been seen from the figure, the system flow rate decreases as the angles of inclination increasing from θ = 0º (the vertical heat sink) to 60º, and then increases as the angles of inclination increasing from θ = 60º to 90º.



b) For Small Angles of Inclination (Θ >70º)

a) For Small Angles of Inclination (Θ < 50º)

Figure 9 Flow Fields for Inclined Heat Sinks

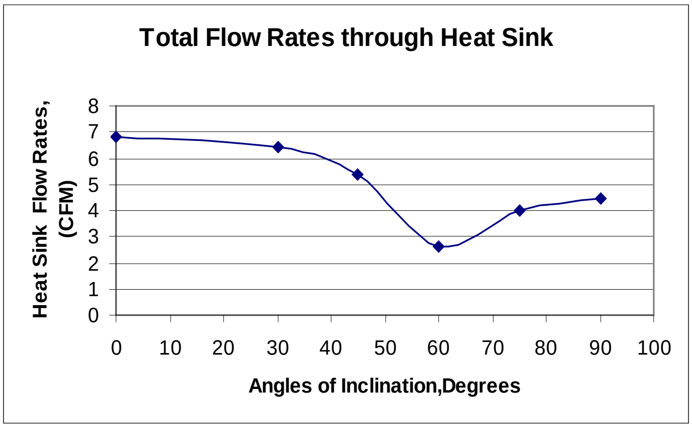


Figure 10 System Flow Rates versus Angles of Inclination

B) Heat transfer

The flow field over the inclined heat sinks has been discussed extensively in the previous sections. The attention here is to focus at the heat transfer aspects of the heat sinks with various angles of inclination from θ = 0º to 90º. The natural convection heat loss from the finned heat sink at various orientations is presented in Figure 10. As can been seen from the figure, the heat loss of the finned heat sink decreases as the angle of inclination increasing from θ = 0º to 60º, and then it starts to increase with the angles of inclination when the angles of inclination is greater than 60º. This trend is similarly to that of the total induced air flow through the heat sink as shown in Figure 9. This is because the heat transfer rate for convection is a function of the flow rates.

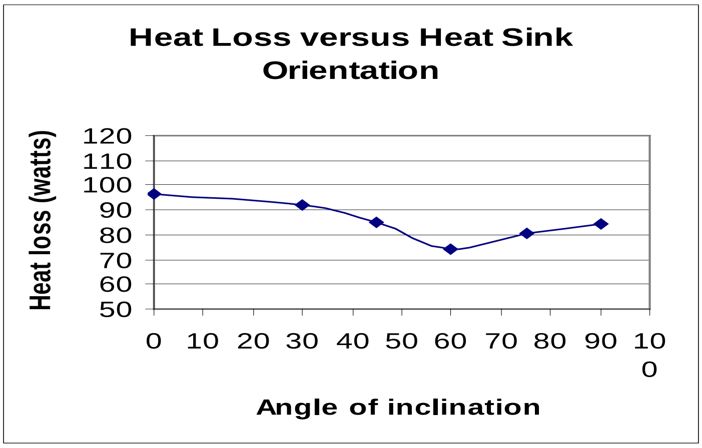


Figure 11 Heat Loss versus Heat Sink Orientations

The heat transfer coefficients (h) as a function of the inclination angles are plotted in Figure 11. The trend for the heat transfer coefficient is similar to the heat loss (Q) as shown in Figure 10. This is because the heat transfer coefficient is computed from the heat loss [$h = Q/(A\Delta T)$]. Examining results from Figures 14 and 15 in details reveals that the heat loss or heat transfer coefficient is function of $(\text{Cos } \theta)^{0.25}$ for the range of inclination angles between 0º and 60º (small angles of inclination group).

$$h1/h2 = (\text{Cos}\theta 1/ \text{Cos } \theta 2)^{0.25} \qquad \text{for } 0^{\circ} \le \theta \le 60^{\circ} \qquad (1)$$

where h2 and θ2 are often referred to the values for the vertical heat sink (θ = 0º)

This relationship is found to be very useful in the engineering applications. Since the data for the vertical heat sink is most readily available, one can easily apply this information to compute the values for the cases at other angles of inclination. In other words, Q or h at any inclination angle can be determined approximately using the values, (i.e. Q or h) of the vertical heat sink multiplying by $(\text{Cos } \theta)^{0.25}$. The percent errors between the actual CFD results and predicated values by using $(\text{Cos } \theta)^{0.25}$ relationship increase from 0.9% at 30º to 9% at 60º. It is believed that the percent errors increased with increasing in the inclination angles is due to the changes of flow patterns from the vertical to horizontal positions. It is also found that the accuracy of Equation (2) will increase if the ratio of (θ1/θ2) approaching to one. On the other hand, based on the limited data in the present study, the following equation is recommended to calculate the heat transfer coefficient for the group with large angles of the inclination.

$$h1/h2 = \sin \theta 1/ \text{sing } \theta 2 \qquad \text{for } 70^{\circ} \le \theta \le 90^{\circ} \qquad (2)$$

where h2 and θ2 are often referred to the values for the horizontal heat sink (θ = 90º)

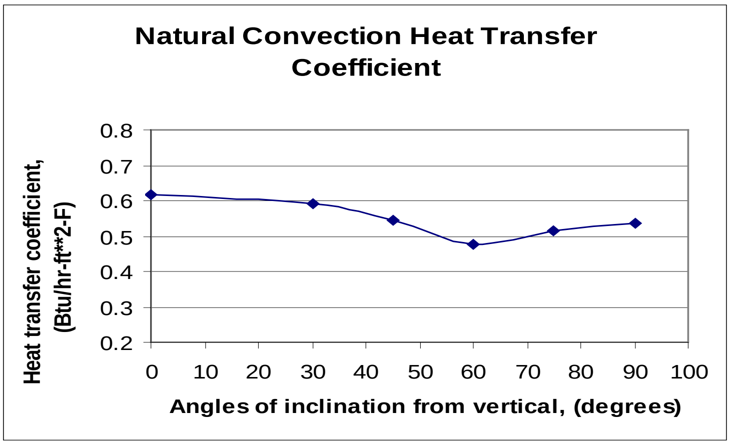


Figure 12 Heat Transfer Coefficients versus Heat Sink Orientations

Figure 11 shows the heat sink thermal resistance versus the angles of inclination. The trend of the thermal resistance is exactly inversed to that of the heat transfer coefficient. In other words, the thermal resistance increases with the angles of inclination from θ = 0º to 60º, and then, it decreases as increasing the angles of inclination from θ = 60º to 90º.

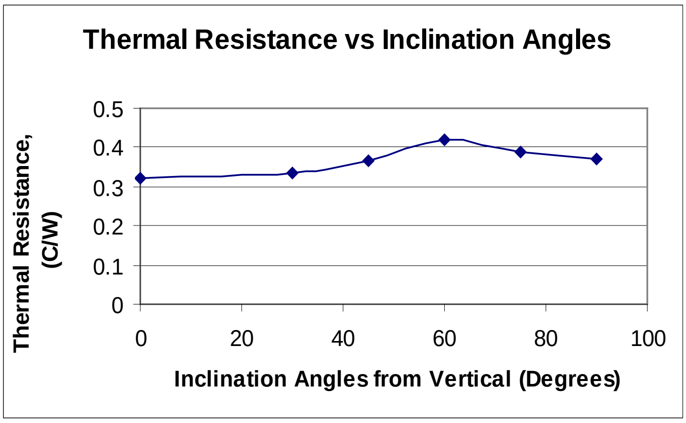


Figure 13 Thermal Resistance versus Angles of Inclination

4. Summary and Conclusion

The electronic equipment is often required to operate at various ambient temperatures and orientations in the real world applications. The latter is especially critical to the tower or pole or roof top mounted electronics under natural convection. However, it appears that no study has ever been made to examine the effects of orientations of the heat sink over full 90 degrees range from the vertical to the horizontal positions. Therefore, the present study is focusing at the analysis of natural convection heat transfer over the wide range of heat sink orientations to fulfill the needs for the industry.

The present analysis provides the insight view of the detailed flow fields over full range of angles of inclinations, including θ = 0º, 30º, 45º, 60º, 75º, and 90º which has never been made in the past. As illustrated in Figure 8, the flow field for those small angles of inclination (0º to 45º) is completely different from those of the large inclination angels (70º to 90º). The results from Figures 10 and 11 indicate that the heat loss as well as heat transfer coefficient is a function of $(\text{Cos}\ \theta)^{0.25}$ for the range of inclination angles between 0º and 60º as described by Equation (1). However, the percent errors between the actual CFD results and predicated values by using $(\text{Cos } \theta)^{0.25}$ relationship increase as the angle of inclination increasing. It is believed that the percent errors increased with increasing in the inclination angles is due to the changes of flow patterns from the vertical to horizontal positions. On the other hand, Q or h at any inclination angle can be determined approximately the value of the horizontal heat sink times (sin θ) for large inclination angels as proposed by Equation (2)

The natural convection heat transfer strongly depends on the system geometry shapes and orientations. For given heat load, ambient conditions and system vertical height, the system available pressure head governing the natural convection flow is fixed. Therefore, more complicated system configurations which generate higher system pressure drops, less available flow to system will be. This is the reason that the continuous fin heat sink performs better thermally than staggered fin and in-line fin heat sinks. It should also be noted that the present study is limited to the natural convection alone. The results from the present study can be directly applied to the indoor as well as outdoor equipment under the natural convection environments. However, for any outdoor equipment, the direct solar heating must be included in thesystem design and analysis.

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 air density is small. In addition, the solar heating must also be considered for cooling of the outdoor equipment.

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