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Augmentation Improves Thermal Performance of Air Cooled Heat Sinks

Augmentation of extended surfaces used to dissipate heat increases the overall effectiveness of a heat sink and increases the heat removed per unit volume. This amount of increase depends on the number of augmentations, air flow velocity and overall length of the cooling channels.

Many of today's high power, high speed electronic power components are capable of power outputs and heat flux concentrations that exceed the thermal capabilities of conventional air cooled heat sinks. This paper offers the user of these components a new alternative in the ongoing problem of removing the excess heat generated by power electronics. This latest thermal advance modifies the heat dissipating fins in a bonded fin heat sink to provide multiple, short, cooling fins of the same total length as the original one piece, single surface fin. The effect of this modification increases the overall thermal performance of the forced air cooled heat sink by 15 to 25% depending on the speed of the cooling air. Static pressure drop is little affected, allowing same size air moving devices.

Demand for most electronic products requires smaller packages with higher performance and more functions than the preceding generation. The increased thermal performance required of these smaller parts is mandatory to producing high efficiency, reduced sized systems. Thermal management of these smaller, higher performance systems is as important a design requirement as the electronic design and should be considered as one of the critical items in project development.

Most heat sinks used in cooling of electronics are extruded aluminum heat sinks that use flat fins or extended heat transfer surfaces to dissipate waste heat to the ambient air. Flat fins are simple to manufacture but can only remove a given amount of heat per square inch of surface. This performance decreases dramatically as the length of the fins increases. Staggered and augmented fins are shown to increase performance by 15 to 25% or more depending on air speed. Mathematical models will be used to show the physical basis for this thermal performance increase. These calculations are based on classic heat transfer equations. Additional pressure drop due to these fin augmentation techniques shows little increase in both modeling and laboratory testing.

The nature of air flow on extended cooling surfaces is to form a layer of stagnant, insulating air over any surface exposed to the air stream. This insulating layer results in a heat sink temperature rise and is caused by the viscous nature of air, or any other fluid, used to transfer heat in a convective mode. The build-up of this "film" or "boundary layer" decreases the amount of heat energy that can be dissipated from each square centimeter of cooling area by acting as an insulation layer between the heated surface and the cooling air. The thickness, and its ability to transfer heat, of this insulating layer is dependent on many parameters, some are:

- The velocity of the air over the extended surface
- The turbulent intensity of the incoming air
- The length of the cooling surface in the direction of air travel
- The roughness of the heat transfer surface

Many methods of increasing the heat transfer characteristics of a flat surface are available to the design engineer wishing to promote turbulence to help decrease temperature rise. Any increase in heat transfer at the air to metal (usually aluminum) interface will increase the heat sink's overall ability to remove overall heat from the semiconductor and thus reduce temperature rise at a given heat load. This desirable characteristic will help reduce the junction temperatures of the semiconductor devices being cooled or offer the designer a potential reduction in the volumetric requirements of the heat sink. This will allow smaller system package sizes as well as decreased weight and, if correctly designed, reduce the cost of the entire assembly.

Empirical tests of various fin augmentation techniques will be shown to confirm the analytical modeling. The various extended surface configurations studied are as follows:

- Fins partially sheared perpendicular to air flow (tuning fork)
- Partial bending of the fin to direct the air in a more effective path (bent fin)
- Flat fins (control point)
- Increased length of flat fins by 20%

The insulating boundary layer near the leading edge of a flat surface is generally thin and the airflow within the layer is relatively fast. This provides an increase in the heat energy the film can remove. As the flow continues, uninterrupted, down the length of the fin the boundary layer thickness or air grows and consequently reduces the amount of heat that can be removed from a given surface area. The thickness of this layer continuously grows as the length of the fin increases and only comes back to the original state when the fin breaks and allows mixing of the cooler duct flow air with the heated molecules from the boundary layer. The boundary layer thickness, and hence its inability to remove heat from a solid surface, increases continuously in the direction of air flow. Depending on air speed, thickness of the boundary layer can increase to the point of merging layers between closely spaced fins, further reducing the heat transfer efficiency.

It should be noted that extruded fins with minute serrations offer little or no increase in the additional heat transfer when compared to a flat fin of equal height. The thickness of the boundary layer is, in all but the fastest air speeds, thicker than the height of the amplitude of the serrations. This closes off the "valleys" between the serration's peaks to any additional heat removal, resulting in a stagnant recirculation zone over the valleys and effectively limits any added surface area increase. The result is the effective surface area of the fin is essentially equal to the straight line height of the fin without the added length of the serrations.

Heat Transfer and Boundary Layer Thickness

Calculating the heat transfer coefficient based on the length of the heated surface is a fairly straightforward process. The calculation shown here demonstrates that the thickness of the boundary layer and the resulting heat transfer coefficient from that surface are dependent on the length of the surface in the direction of air travel. This formula from [1] is shown as:

h = (J//t)0.66 x (0.323) x (D x V x L/Dv)0.5 x D x Cp x J
(1)

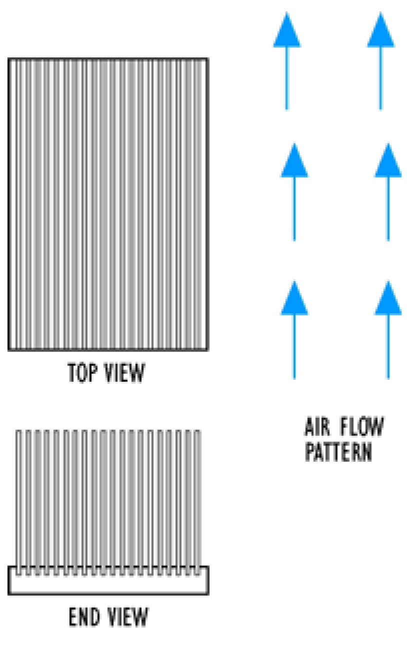

Figure 1.Standard Case

Where:
h = Heat transfer coefficient
J = Velocity of the air past the fin
D = Air density
L = Length of the cooled surface in the direction of air flow
Dv= Dynamic viscosity
t = Thickness of the boundary layer
Cp= specific heat of the cooling air

This expression applies to both laminar airflow as well as turbulent airflow over flat plate surfaces typical of those found on an aluminum heat sink extrusion. The expression has a strong dependence on the air velocity, physical properties of the cooling air as well as the length of the surface in the direction of air travel.

Equation (1) shows that many shorter cooled surfaces will remove more heat than one single surface of the total equivalent combined length. This is due to the heat transfer being directly proportional to the thickness of the boundary layer and to the thinning or reattachment of this layer at every leading edge in the flow pattern. To restate this: if the boundary layer of stagnant air along the heat dissipation surface is relatively thin the heat transfer is proportionately greater than if the boundary layer is thick. The longer the uninterrupted cooled surface the thicker the boundary layer will be.
From [1] also comes the following formula where the thickness of the boundary layer, t, is predicted by:

$$t = \frac{4.64 \times L}{(D \times J \times L/Dv)}$$

(2)

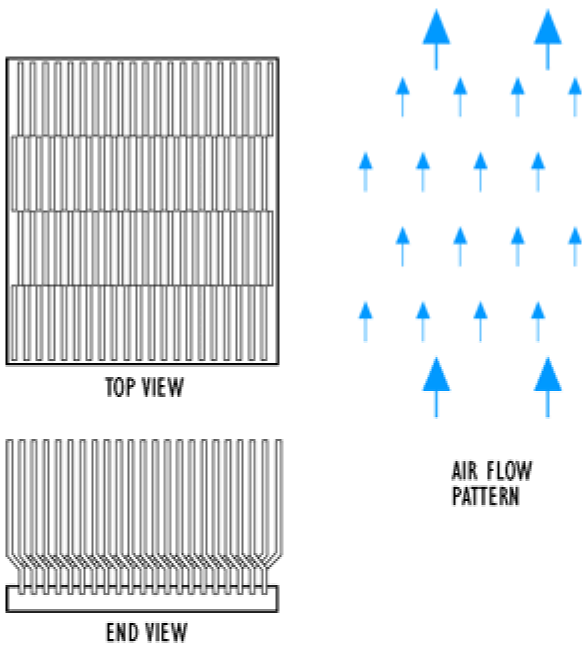

Figure 2. Tuning Fork Case

As an additional and simpler indicator of this direct relationship, in laminar airflow the amount of heat transfer is shown as being equal to just two variables. These are:

$$h = \frac{3 \times K}{2 \times t}$$

(3)

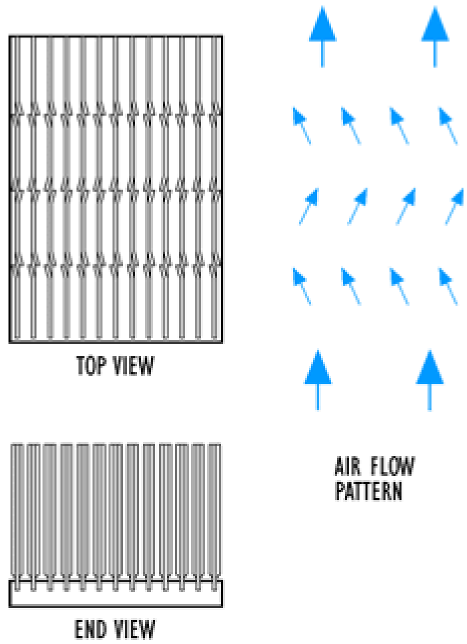

Figure 3.Bent Fin Case

Where:
K = Conductivity of the air

Heat transfer is improved by reattachment of the boundary layer at every leading edge. Whenever a cooling fin offers a leading edge or new projection into an air stream or when the heat transfer surface is interrupted the film layer becomes thinner and heat transfer is increased.

In both natural and forced convection heat removal increases with more surface area exposed to the cooling air stream. This is expressed by the basic formula of convection cooling:
Q = hA DT

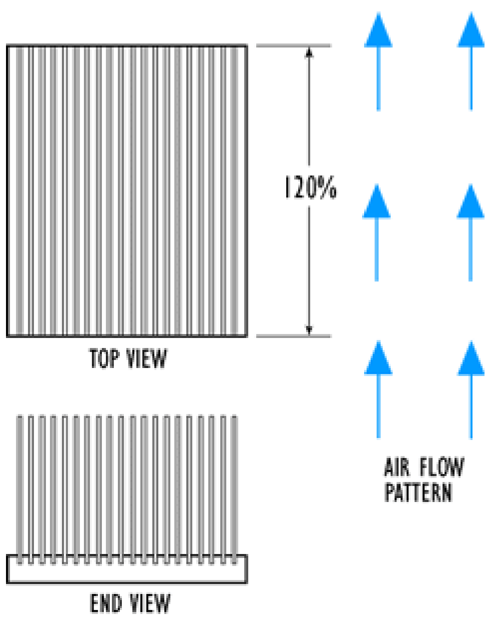

Figure 4. 120% Length Case

Where the heat input is equal to the heat transfer coefficient (h) times the surface area exposed to the cooling air flow (A) times the differential temperature (DT).

However, after a certain channel length the addition of more length adds no cooling effect. In effect this tells us that in the direction of airflow the efficiency of heat removal decreases as the length of the heat sink increases. This is due primarily to the stagnant layer of air molecules which, through the inherent viscosity of the air "stick" to the cooling surfaces and act as insulation resisting the flow of heat from the surface to the cooling air stream. This is depicted mathematically as the "h" value. When the heated surface is interrupted in the direction of air flow the insulation layer is also interrupted and causes the thickness to decrease. These interruptions cause the boundary layer to become thinner or disappear altogether, allowing increased heat transfer rates to the cooler air passing the surface. The properly designed heat sink limits the length between flow interruptions sufficient to maximize heat removal but still minimize the effect of additional pressure drop.

Effect on Pressure Drop

The cost of this additional heat removing capacity is at the price of increased resistance to air flow. To optimize the number of leading edges in a design pressure drop must also be considered. These must not present the cooling air with too much resistance to the air flow, but be enough to be effective. A large number of leading edge points could decrease velocity and the subsequent mass flow of the cooling air to the point that the augmentation techniques would reduce the total thermal resistance of a heat sink to less than the original single piece fin design. This would choke air flow and slow heat removal.

In classic fluid flow analysis the total pressure drop through a heat sink can be broken up into four segments. For the purpose of analyzing a heat sink or other flat plate heat exchanger these four sections are:
1. The entrance effect which analyses the air as it goes from free steam velocity to channel flow.
2. The flow acceleration, which is based on Bernoulli principles.
3. The core friction effect or the drag that the extended fin surface has on the passing air.
4. The exit effect as the air again reduces velocity. This section actually decreases the total pressure drop.

As a complete, workable equation Kayes and London[2] wrote these separate parts of the equations as follows:
DP = G2 / 2gc x J/P
((Kc + 1- f2) + Entrance Effect
2(J1/J2 - 1) + Flow Acceleration
fDA x Vm/J1 - Core Friction
(1 - f2 - Ke) J2 / J1) Exit Effect

Where:
P = Pressure
G = Free flow area
Gc = Gravitational constant
Kc = Entrance Coefficient
Ke = Exit Coefficient
J = Velocity at specific points noted as subscripts
DA = Change in area from channel to total area between fin
f = Ratio of free-flow area to frontal area

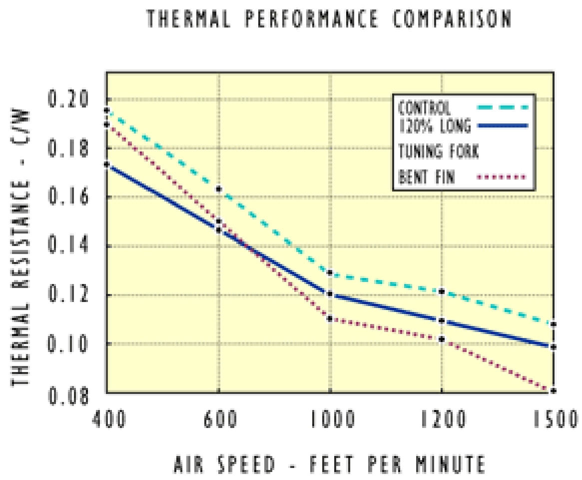

Figure 5.Thermal Performance Comparison

In use of this equation it can be predicted that the friction factor of air flowing in constricted channels between fins will account for a large portion of the total overall system pressure drop. Any effect of inlet and outlet pressure drops and flow acceleration will be small (typically less than 10%) relative to the overall. In using these equations for pressure drop prediction the core friction segment of the equation dominates the resultant pressure drop figures and in tests contributes a majority of the actual observed pressure loss.

This basic formula confirms what was found in testing; the added pressure drop due to the augmentation techniques described here had very little measurable effect on pressure drop and a resulting unmeasurable effect on forced air mass flow rate.

Fin Augmentation Techniques to Improve Heat Transfer

To help improve the overall thermal performance of finned heat sink manufacturing methods have been developed that will modify a flat finned part and create several smaller, more efficient finned sections. This modification can be done using one piece of fin to reduce manufacturing costs and increase "in fin" heat transfer.

For laboratory testing, four separate and distinct models were built, as depicted in Figures 1-4.

Flat fin heat sink, cut to the control length desired (Figure 1).

Fins that are split into 4 equal sections over the control length and bent from the base perpendicular to the air flow; tuning fork. (Figure 2).

Fins that are split into 4 equal sections but the leading and trailing edges are twisted sideways to divert the air flow; bent fin. (Figure 3).

Flat fin heat sink, cut to 120% of the control length (Figure 4).

These test samples were all constructed of similar material with an extruded aluminum base and fins epoxy-bonded into formed slots. The overall dimensions were identical (except for case 4 where the length is 20% greater) thus giving the first three heat sinks all the same volume and total surface area as well as the same number of fins or extended surfaces. Forced air cooling was used with the velocity controlled using a variable speed fan.

Table 1. HEAT SINK COMPARISONS				
VELOCITY FT/MIN.	STANDARD CASE #1 (° C/W)	TUNING FORK CASE #2 (° C/W)	BENT FIN CASE #3 (° C/W)	120% LENGTH CASE #4 (° C/W)
400	0.195	0.187	0.194	0.163
600	0.163	0.149	0.150	0.146
1000	0.129	0.111	0.111	0.120
1200	0.121	0.100	0.102	0.110
1500	0.109	0.092	0.080	0.099

Test Results

The forming of the fins was done to exacting dimensions and designs. The augmentation takes care to optimize the fin positions from one fin section to the next downstream section. This is done by placing the leading edge of each new fin section in the exact center of the airflow channel from the preceding center. This technique takes advantage of the maximum heat transfer and maximum air velocity being in the exact center of the previous channel. At this point maximum heat removal occurs, minimizing the boundary layer thickness and its subsequent resistance to heat flow.

Laboratory test results show that the amount of improvement attributable to the augmented fin design varies as a direct result of the air flow velocity past the heat sink fins. The additional cooling is also dependent on the type of fin augmentation that is used. Improvements range from 10% at 400 feet per minute to 25% or more at 1500 feet per minute.

Table 1 shows actual test comparisons.

Pressure drop across the parts tested is shown to be very similar and almost undetectable in the velocity range tested (Figure 5).

Acknowledgement

Credit for conducting laboratory experiments goes to James Coffee and Mark Pellilo.

References

1. Kraus, Allan and Avram Bar-Cohen, "Thermal Analysis and Control of Electronic Equipment," McGraw-Hill, 1983.
2. Kayes, W.M. and London A.L., "Compact Heat Exchangers," McGraw-Hill, 1955.
3. Sathyamurthy, P., P.W. Rundstadler and Dr. Seri Lee, "Numerical and Experimental Evaluation of Planar and Staggered Heat Sinks," Fifth Annual ITHERM Conference May 29, 1996.