Optimum fin spacing for fan-cooled heat sinks

Keywords: optimum fin spacing fan-cooled heat sink heatsink optimal fin pitch parallel plate fin array optimization forced air cooling fan curve pressure drop maximum fin density baseplate backplane fin height thermal management improving performance

This paper discusses the optimum fin spacing for a fan-cooled heat sink. The heat sink geometry is shown in Figure 1. As shown in the figure, it is assumed that a shroud is present to confine air flow to the fin channels.

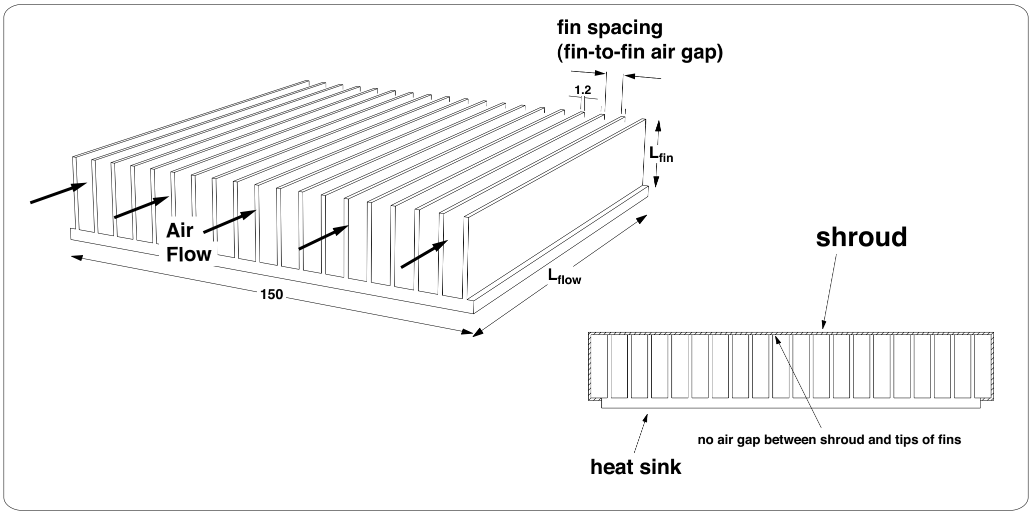

Figure 1: Heat sink and shroud configuration

Figure 2 is a schematic of the overall flow system:

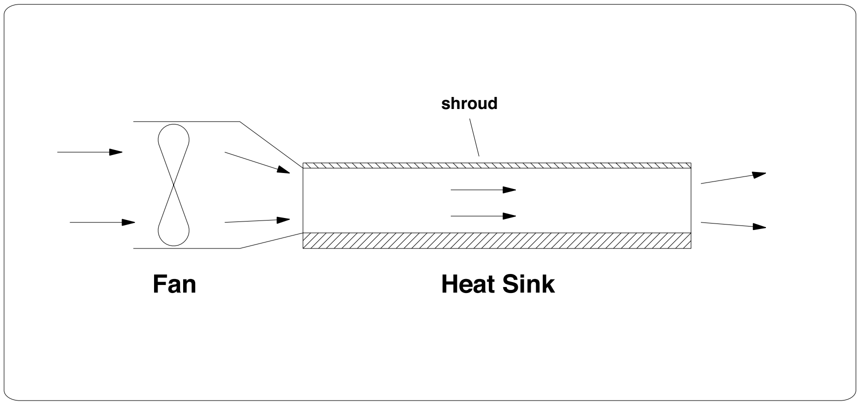

Figure 2: Overall flow system

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Flow, pressure and heat transfer calculations

When a fan or blower forces air through a heat sink, the flow obtained will be a function of both the fan capacity and the flow resistance of the heat sink.

Figure 3 shows the flow and pressure characteristics for a typical fan. As shown, the amount of air pressure that a fan can generate depends on the air flow. While the fan in the figure is rated at 50 CFM (85 m^3/hr), this flow rate will not be obtained when combined with a heat sink, because air pressure is needed to force the air through the fin channels.

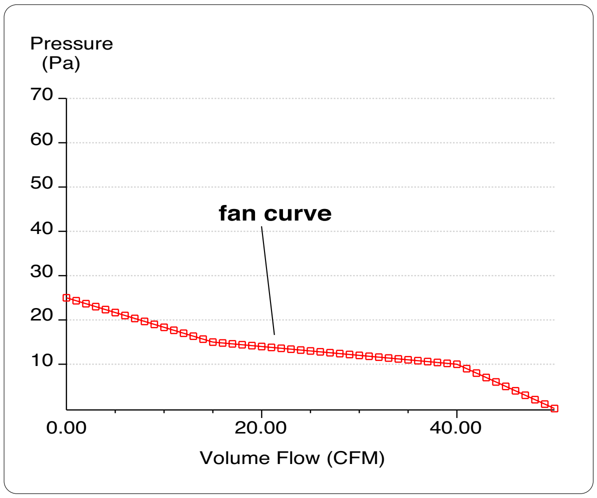

Figure 3: Flow and pressure characteristics of a typical axial fan

Figure 4 adds a flow resistance curve. The intersection between the fan curve and the flow resistance curve is the operating point of the system. This point represents the actual air volume which will pass through the heat sink fin channels. (Depending on the shape of the fan curve, it is actually possible to have two operating points, although this is a condition for the designer to avoid.)

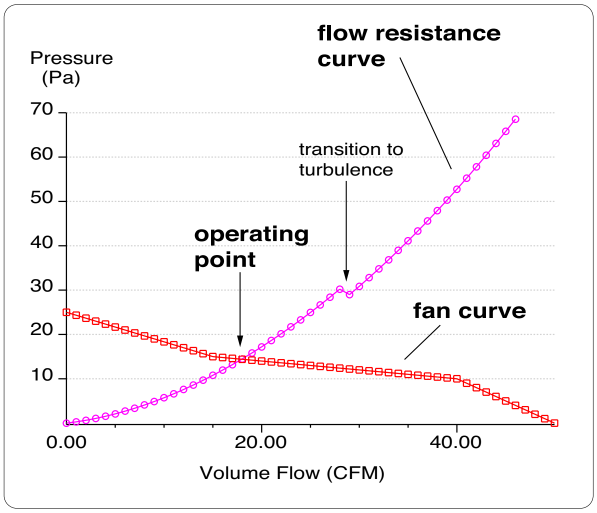

Figure 4: Flow and pressure characteristics of fan and heat sink combination

The fan curve for a particular fan model is obtained from the manufacturer of the fan. The system flow resistance can be calculated with a form of the Bernoulli equation:

$$\frac{\Delta p}{\rho} = \sum \left[\frac{V_{local}^2}{2} \left(K + \frac{4\, f\, L_{flow}}{D_{hyd}}\right) \right]$$

It is important to realize that the heat sink is only part of the system flow resistance. The fan must overcome all of the flow resistance elements, which includes the heat sink, inlet and outlet grilles, any filters, etc. If it can be assumed that: (a) inlet and outlet grilles are largely open area, (b) no air filters are present and (c) there is a straight-through flow path, the system will be largely the same as in Figure 2. In this case, the Bernoulli equation can be simplified to:

$$\Delta p_{overall} = \left(K_c + \frac{4\, f\, L_{flow}}{D_{hyd}} + K_e\right) \frac{\rho\, V_{chan}^2}{2}$$

where:

$\Delta p_{overall}$	- total pressure drop for the flow circuit
V_{chan}	- average air velocity within the fin channels
K_c	- loss coefficient for contraction into the fin channels
K_e	- loss coefficient for expansion as air exits the fin channels
f	- friction factor in the fin channels
D_{hyd}	- hydraulic diameter for the fin channels
ρ	- average air density within the fin channels

To calculate laminar flow volume and pressure drop, the method outlined by Robert Simons [1], based on equations from Culham and Muzychka [2], was used. Since the fan must push against the system flow resistance, it was appropriate to evaluate the K factors at $\sigma = 1.0$, so that $K_c = 0.42$ and $K_e = 1.0$.

For the turbulent flow regime, the base friction factor was obtained from the Petukhov equation [3], with an entry correction factor derived from Deissler [4]. The K factors were obtained from Kays and London [5], which yields $K_c = 0.5$ and $K_e = 1.0$, again for $\sigma = 1.0$.

To calculate heat transfer from the fins, a parallel plate model was used which takes into account entrance effects [6][7][8]. Flow can be either laminar or turbulent. There is a correction factor for the temperature rise of the air as it moves through the fin channels, as well as a fin efficiency correction factor. The heat sink baseplate was fixed at 75°C and the inlet air temperature was 25°C. The base-to-tip fin length (L_{fin}) was 25 mm, unless specified otherwise. Fin thickness was 1.2 mm with a thermal conductivity of 200.0 W/m-°C (typical for a conductive aluminum alloy).

Unless noted otherwise, the fan has a maximum air flow of 50 CFM (85 m^3/hr) and a static pressure rating of 25 pascals (0.1" H_2O). These fan specifications are typical of a 92 mm DC axial fan. The fan was modeled with a simple straight-line fan curve.

The actual calculations were performed with the Sauna V3.71 thermal program [9].

Fin spacing optimization at L_{flow} = 150 mm (6")

Before optimizing with a fan, it's quite instructive to obtain an optimization curve with a constant flow. Figure 5 on the next page shows a fin spacing optimization curve at a constant air flow volume of 25 CFM (42.5 m^3/hr) and L_{fin} = 25 mm. Note that, as shown in Figure 1, "fin spacing" refers to the fin-to-fin air gap. The figure shows that the optimum fin spacing is just 0.5 mm (.020"). However, the pressure drop at this fin spacing is a very elevated 1662 Pa (6.7" H_2O). It would take a massive fan to force this amount of air through the heat sink.

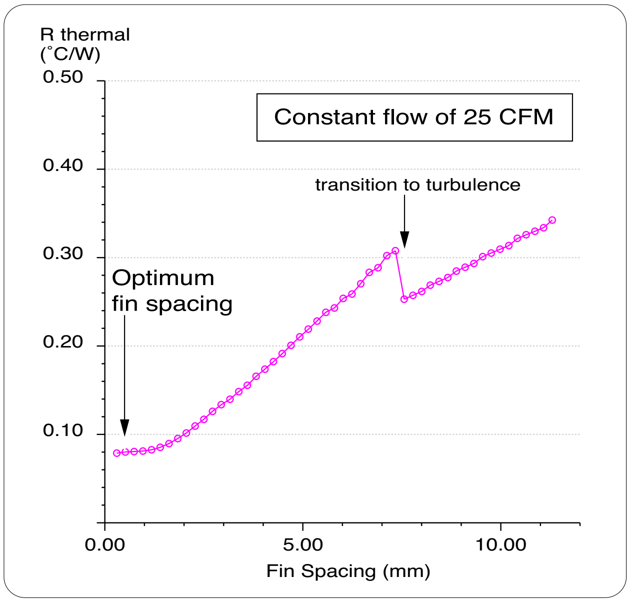

Figure 5: Fin spacing optimization at constant flow volume

The results are quite different when a fan is used, as shown in Figure 6. The optimum fin spacing is roughly 2.8 mm (0.11"), significantly larger than before. At the optimum point of the curve, there is the best possible balance between high heat transfer coefficients at close fin spacings and reduced flow due to the increased pressure drop associated with narrow fin channels.

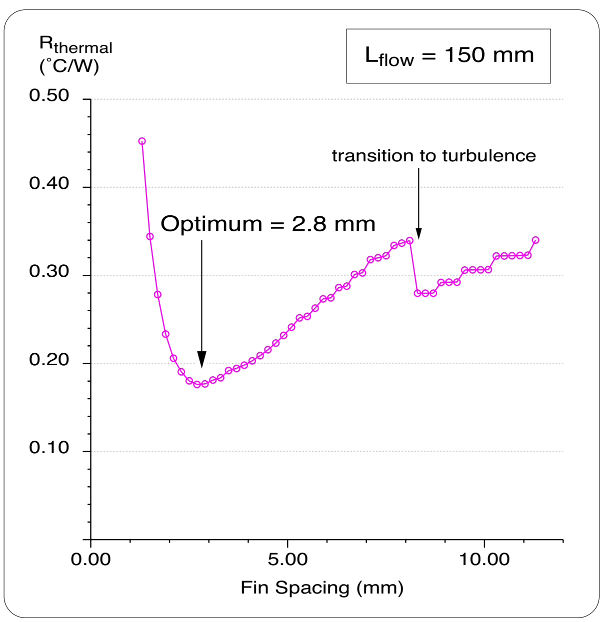

Figure 6: Optimum fin spacing with fan

There are several interesting points to mention about Figure 6. First, notice the optimum fin spacing occurs in the laminar flow regime, which is usually the case. However, as discussed later, there are situations where the optimum occurs within the turbulent flow regime.

Second, the optimum heat sink has fins which are relatively closely spaced. It will not be possible to achieve this optimum spacing with an inexpensive heat sink. A low cost extruded heat sink typically has a fin spacing of 6 mm, significantly greater than the optimum of 2.8 mm. It will be necessary to use bonded or swaged fins. Such a heat sink will cost more, but will provide a significant benefit.

Finally, there is a significant penalty for having a fin spacing which is less than the optimum. On the other hand, the penalty is not as severe for a fin spacing which is greater than optimum, so it is better to err in this direction.

Spacing optimization at L_{flow} = 300 mm (12")

The optimization curve for L_{flow} = 300 mm is shown in Figure 7. The shape of the curve is similar to L_{flow} = 150 mm. However, the optimum spacing occurs at 4.0 mm, which is 40% greater than before. There are two reasons for this. First, a longer heat sink has greater flow resistance and this will cause the fan to deliver less air volume. Second, the longer fin channels fill up with warm air, which causes a reduction in heat transfer.

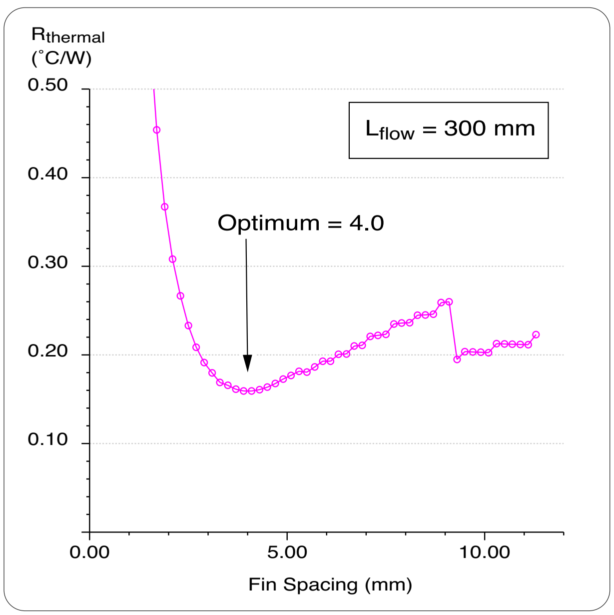

Figure 7: Optimum fin spacing for 300 mm flow length

It's also interesting to examine an optimization curve for a shorter flow length. Figure 8 on the next page shows curves for flow lengths of 75, 150 and 300 mm. For L_{flow} = 75 mm, the optimum fin spacing is reduced to 2.0 mm, just half the optimum value for L_{flow} = 300 mm.

Figure 8 clearly illustrate the danger of using a fin spacing which is less than the optimum. Note the performance of the L_{flow} = 300 mm heat sink with fin spacing = 2.0 mm. In this

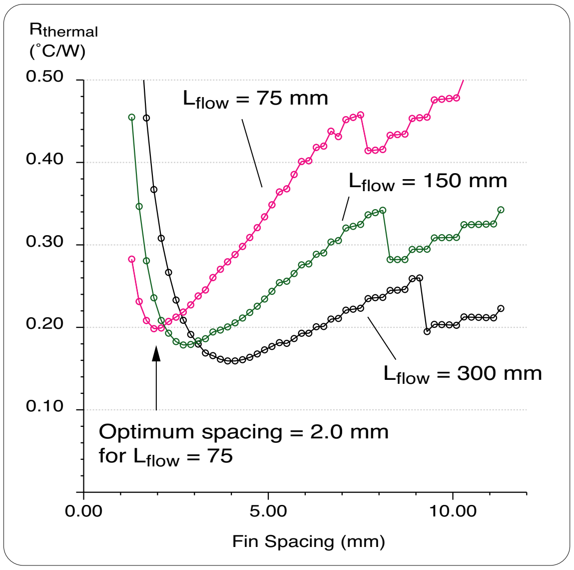

Figure 8: Optimum fin spacing for multiple flow lengths

case, the performance is degraded so sharply that it would actually be more effective to use a shorter heat sink, either L_{flow} = 75 or L_{flow} = 150, at the same spacing! Expensive sinks with closely spaced fins can be very effective in the proper application, but disastrous results will occur if the heat sink is misused.

Impact of varying fan capacity

In the real world, a designer can change either the heat sink or the fan (or both). So it is interesting to see how changes in fan capacity affect performance and optimum dimensions. Figure 8 shows optimum fin spacing at different fan capacities. All previous graphs have assumed a fan with maximum flow of 50 CFM and a maximum pressure of 25 pascals. So the curves in Figure 8 show the effects of doubling flow and pressure, as well as halving these parameters.

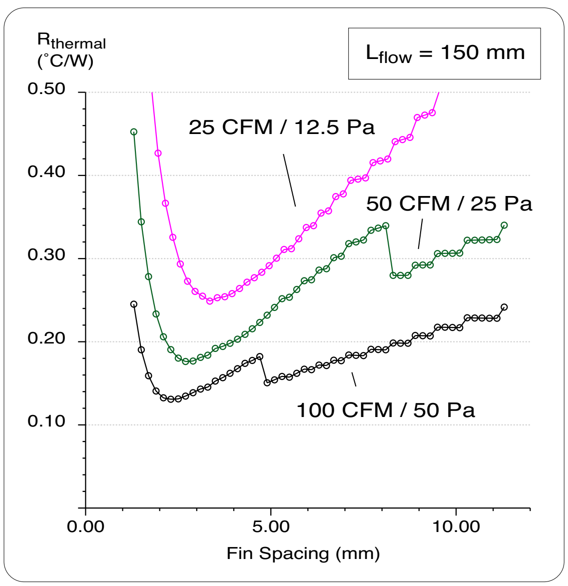

Figure 9: Effect of varying fan capacity

Not surprisingly, the lowest thermal resistance is obtained with the most powerful fan. The optimum spacing is also affected, with a smaller optimum spacing for the higher power fan. However, the fin spacing effect is much smaller as compared with changing the overall length of the heat sink. The optimum spacing for the 25 CFM / 12.5 Pa fan is 3.3 mm, which decreases to 2.4 mm for the 100 CFM / 50 Pa fan. This is a fairly modest shift considering that there is a 4:1 difference in both pressure and maximum flow. The flow length has a much stronger impact on optimum fin spacing.

Besides fin spacing, there is also a clear change in the transition point for the shift to turbulent flow. Since the most powerful fan is generating the most flow, turbulence occurs at a lesser fin spacing. Also, notice the shape of the curve for the 100 CFM / 50 Pa fan. Thermal resistance at the start of the turbulent regime is fairly close to the optimum thermal resistance located in the laminar regime. With an even more powerful fan, in fact, the optimum spacing would occur in the turbulent regime. (It should also be pointed out that it is difficult to predict heat transfer with great accuracy at the start of the turbulent zone.)

Impact of varying fin length

Finally, it is also interesting to look at the change in performance and optimum fin spacing as the base-to-tip fin length (L_{fin}) is modified. Figure 8 shows both shorter and longer L_{fin} as compared with the fin length of 25 mm used in the previous graphs. Figure 8 was generated with the default 50 CFM / 25 Pa fan.

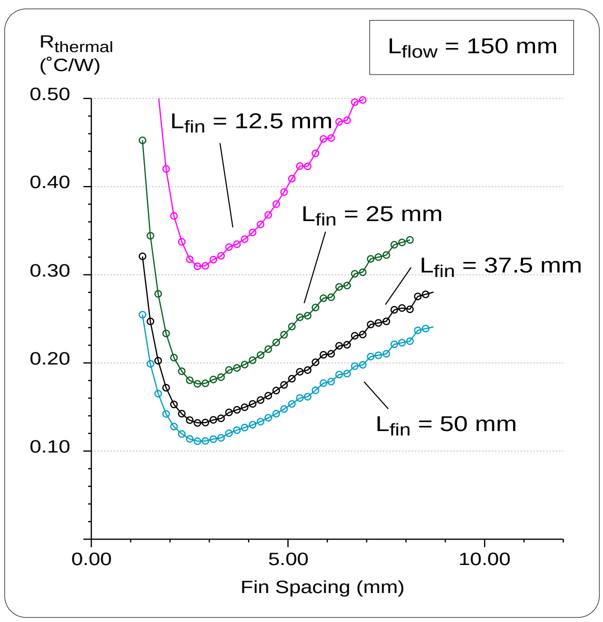

Figure 10: Effect of varying base-to-tip fin length

As far as fin spacing is concerned, the optimum spacing is nearly unchanged for the 4 different values of fin length.

Figure 8 shows very limited gains as the fin length starts to approach 50 mm (2"), which is surprising to some. The longer fins do provide larger fin channels, which reduces flow resistance. This, in turn, increases total volume flow through the heat sink. However, the linear air velocity is reduced because the increase in volume flow does not keep up with the increased cross-sectional area. The reduction in air velocity, along with fin efficiency effects, limit the gains from longer fins. Note that fan capacity will also have an impact. With a higher capacity fan, a somewhat longer fin length will be effective.

Conclusion

In summary, the optimum fin spacing for a fan-cooled heat sink was found to be primarily a function of flow length. With a 50 CFM / 25 Pa fan, the optimum spacing was 2.0 mm at L_{flow} = 75 mm, 2.8 mm at L_{flow} = 150 mm and 4.0 mm at L_{flow} = 300 mm. With a higher capacity fan, the optimum fin spacing will decrease, but not dramatically. Similarly, the use of a lower capacity fan will increase the optimum fin spacing by a moderate amount. Designers need to be aware that the use of a fin spacing which is less than the optimal value can result in very poor heat sink performance.

When the base-to-tip fin length was modified, there was virtually no impact on the optimum fin spacing. It was observed that there are diminishing benefits for fin lengths greater than 50 mm, although this will impacted by fan capacity.

Finally, it should be remembered that all optimizations were for steady state conditions with a uniform backplane temperature. For point source heat loading, there will be some impact on optimum dimensions, although it is expected that the impact will generally be slight. For transient and duty cycle operation, there will be a shift toward fin spacings which increase the overall heat sink mass.

If there are questions or comments about this article, please contact Thermal Solutions Technical Support (734-761-1956, support@thermalsoftware.com).

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