Economic Optimization of Heat Sink Design

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INTRODUCTION

This paper describes the analysis and derivation of an optimum heat sink design for maximizing the thermoelectric cooling performance of a laboratory liquid chiller. The methods employed consisted of certain key changes in the design of the heat sink in order to improve its thermal performance. Parametric studies were performed in order to determine the optimized cooling system design per dollar.

The objective of this project was to analyze the thermal performance of an initial simple heat sink design and improve cooling performance while reducing the cost and overall size of the cooling system. Several changes were examined in an effort to improve the thermal performance and/or to reduce overall cost. The result obtained has provided some guidelines for the selection/design of the most effective and economical heat sink configuration. These results were somewhat surprising since they are contrary to what one might instinctively expect without the benefit of the detailed analysis presented in this paper.

APPLICATION PARAMETERS

The objective of this project was to re-design this system as necessary to meet the following specifications:

- The ambient temperature shall be 26.7ºC.
- The cold plate temperature shall be -1.1ºC.
- The operating voltage shall be < 12 VDC.
- Liquid volume = 1 liter.
- Cool down from 12.7ºC to 0.55ºC < 45 minutes.
- Heat sink configurations to consider:
  - Natural convection extruded
  - Forced convection extruded
  - Forced convention staked
- Costs estimated for each case.
- Heat pumping maximized per unit dollar.
- Exchanger size minimized per unit dollar.
- Simplify assembly to assure good quality and producibility.

HEAT LOAD ANALYSIS

The gross heat pumped by the TE module was designed to match the sum of active heat loading plus passive heat loading. The active heat load was the transient heat load during cool-down of the water. The passive heat load was due mostly to insulation losses from ambient and the plate-to-plate heat load internally in the TE cooling sub-system. Actual passive heat loads depended on the instantaneous thermal conditions of the system. However, thermal conductances were constant for a given mechanical configuration. Therefore, thermal conductances were calculated to instantaneously account for the dynamically changing passive heat loads. These conductances were inserted into the thermal model and used to calculate the heat load for each conductance. Calculations were performed for each instant in time in order to determine the overall dynamic system performance.

Transient Heat Load

It was a design goal that the time required to cool one liter of liquid from 12.78ºC to 0.55ºC would be a maximum of 45 minutes. The thermal properties of water were used to calculate an average of 55 Watts for the transient heat load of the cooling system during the cool-down period.

Ambient Heat Load

No significant changes were made in the size and shape of the liquid reservoir. Therefore, it was only necessary to model the original configuration in order to determine the ambient heat load. Since the interior volume was liquid, the inside wall temperature was set equal to the liquid temperature. The exterior wall temperature was not set to ambient but modeled together with the polyurethane wall insulation and an effective convection coefficient for still air. The result of this model was a value of 0.09 Watts/°C for the total thermal conductance due to ambient losses (KTA).

Plate-to-Plate Heat Load

The plate-to-plate heat load was defined as the heat that was conducted from the heat sink back into the cold plate. The total plate-to-plate thermal conductance (KTH) was the sum of the KTH of the insulation between the plates plus the KTH due to the bolts that secured the cold plate to the heat sink. The original design had a total KTH of 0.13 Watts/°C. The final design used a reduced number and size of bolts, yielding a total KTH of 0.09 Watts/°C. This change in total KTH was due to the reduction on the KTH for the bolts from 0.06 Watts/°C to 0.03 Watts/°C.

HEAT PUMPING CAPACITY

The gross heat pumped included passive heat loads and active heat loads. The temperature of the cold plate was held constant at -1.1°C. However, the actual cold plate temperature would likely be less than -1.1°C if the fluid were to be cooled within 45 minutes. The reason for this was because the fluid would add additional thermal resistance to heat transfer.

Net Heat Pumped

The net heat pumped was the result of subtracting passive heat loads from the gross heat pumped. The temperature of the cold plate was held constant at -1.1°C. However, the actual cold plate temperature would likely be lower, as indicated above.

Estimated Cool Down Time

The cool down times presented in Figure 1 were determined by the thermal model based on a starting cold plate temperature of 12°C and an ending cold plate temperature of -1.1°C. It is clear from this graph that the design changes were, indeed, sufficient to meet the desired specifications. In addition, the graph shows the relative effect of increasing HSR on the cool down times.

HEAT SINK OPTIONS

The performance of a TE cooling system is highly dependent upon the heat sink resistance (HSR) as described by Nagy, et.al (1). It was desirable to obtain the lowest HSR in order to have the most effective heat pumping from the TE module. However, the type of heat sink chosen depended also on system costs and overall size constraints. There were several types of heat sinks to consider: (1) a natural convection heat sink; (2) a low fin density heat sink and fan; and (3) a high fin density heat sink and fan. Each HSR's, however, included an additional 0.03°C/Watt resulting from the thermal grease joint used to ensure good thermal contact between the TE module and heat sink.

Natural Convection Heat Sink

The natural convection extruded heat sink design selected for this application measured 381mm long by 274mm wide with 96mm tall fins and had a HSR of approximately 0.2°C/Watt. This value was based on its published HSR from the extrusion manufacturer. It should also be noted that when using this large of a natural convection heat sink the use of two thermoelectric modules was required for better spreading of the thermal load

Low Fin Density Forced Convection Heat Sink

This heat sink consisted of a popular, low-priced, finned aluminum extrusion and a fan. There were many extrusions to choose from, but their widths were limited to the fan sizes available. That is, an undersized fan would not be able to distribute air flow properly over an oversized heat sink. Therefore, money spent on an oversized heat sink would not be well spent.

The heat sink dimensions for this extrusion design were 122mm wide, 140mm long and having 14 fins measuring 58mm tall. The HSR was lowered to 0.2°C/Watt. when utilizing a small 12VDC fan. The particular fan used in this case measured 120mm square by 25mm deep.

High Fin Density Forced Convection Heat Sink

High fin density heat sinks consisted of an extruded base with densely packed slots into which thin sheets of aluminum were bonded. These types of heat sinks are referred to as "staked-fin" heat sinks. High fin densities of this nature could not be extruded since the extrusion die would not be able to sustain the forces developed in the process of making such a thin fin. However, this type of heat sink typically has a significantly lower HSR than an extruded heat sink for the same overall dimensions.

A HSR of 0.2°C/Watt was again achieved with a high density heat sink having dimensions 82mm wide by 82mm long by 38mm tall fins. This heat sink used a 80mm square by 25mm deep fan.

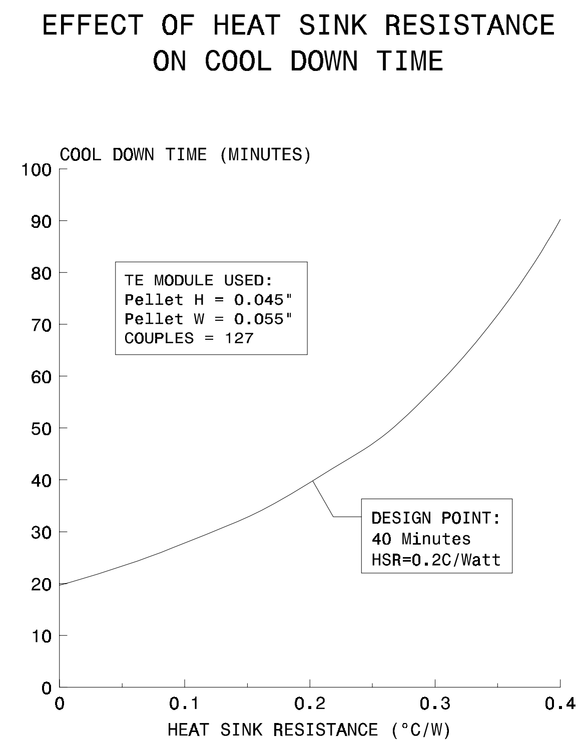
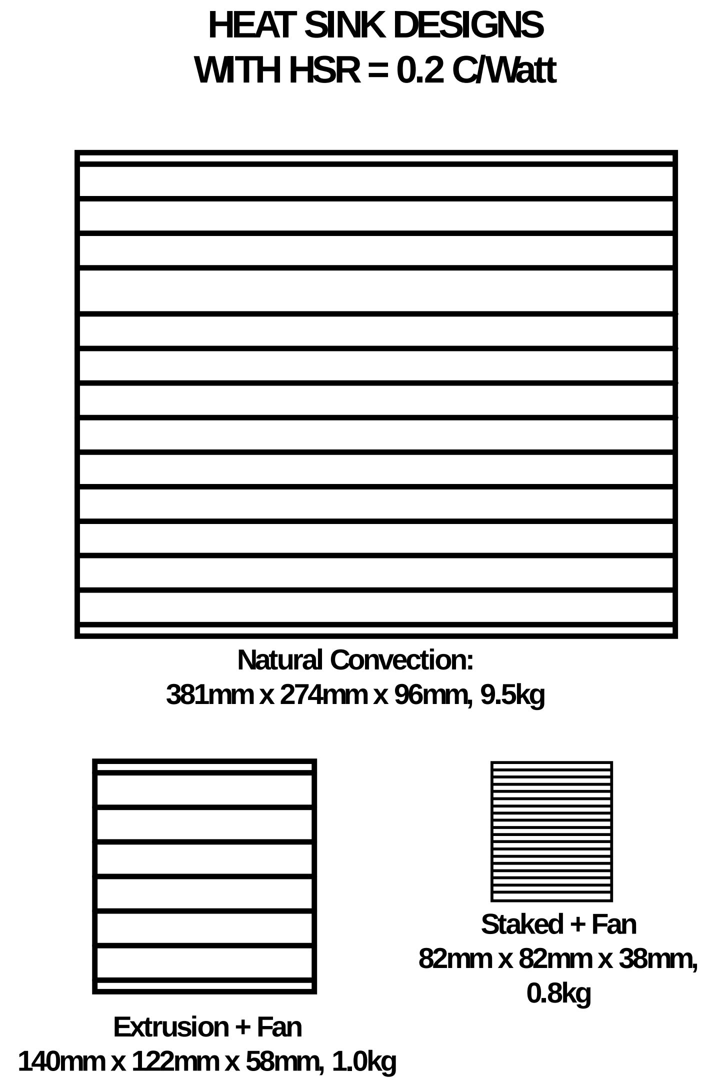


FIGURE 1. A curve plotting the effects that a heat sink resistance (HSR) has on the cool down time for this system.



HEAT SINK PRICING

The chief variable in determining sub-assembly costs was in the selection of the optimum heat sink. The prices of the three heat sinks were based on actual quotations in both the 500 and 10,000 piece quantity. As the table in Figure 2 illustrates, the price differences between the heat sink designs was very significant. These price estimates also included costs for machining, and fan mounting as required. It is evident that the high density heat sink is the cost effective choice for the application. In addition, the smaller size and lighter weight of the heat sink, as illustrated in Figure 3, will reduce shipping and packaging cost as well.

FIGURE 3. A scaled illustration depicting the dramatic size differences of the three selected heat sink designs.

	Natural Convection Original Heat Sink @ 0.20°C/W HSR		Forced Convection Extruded Heat Sink @ 0.20°C/W HSR		Forced Convection Staked Heat Sink @ 0.20°C/W HSR	
	500 pc	10,000 pc	500 pc	10,000 pc	500 pc	10,000 pc
Heat Sink	$77.50	$68.20	$14.25	$11.50	$15.75	$12.90
Fan	N/A	N/A	$11.25	$9.49	$8.30	$6.16
Shroud	N/A	N/A	$6.75	$5.50	$4.50	$3.70
Total Price	$77.50	$68.20	$32.25	$26.49	$28.55	$22.76

FIGURE 2. A table showing the cost comparisons of the varying heat sink designs.

DISCUSSION

The major change in this application was the redesign of the natural convection extruded heat sink. Heat sink comparisons were performed using a natural convection extrusion, a popular off-the-shelf extrusion utilizing a fan for forced convection heat transfer and a ten fin per inch high density heat sink using a fan.

Another major change to the exchanger was the elimination of two TE modules having 127 couples each of 1.0mm square by 1.27mm tall that were wired together in series. These modules were replaced by a single TE module having 127 couples of 1.4mm square by 1.14mm tall pellets.

The mounting hardware from the aluminum cold plate to the heat sink was also modified. The number of screws was reduced from six #6-32 screws to four #4-40 screws. This change did not significantly improve the thermal performance. It did, however, simplify the mechanical design and, therefore, reduce cost.

The heat loads were carefully examined in this project. Analysis and management of each component of heat load was the key to design optimization. The transient heat load to cool one liter of water in 45 minutes was calculated to be an average of 55 Watts during the cool-down period. This was from a starting temperature of 13°C to an ending temperature of 1°C.

Passive heat loads were modeled as thermal conductances. This was done because the key temperature differences continually change during the transient cooling period. The conductances were then used in calculating heat pumping rates using TE Technology's modeling software. The analysis of the passive heat loads further revealed that the thickness of the insulation needed no modification as it already represented a good cost/performance trade-off.

Heat sink options for natural convection and forced convection were examined in order to quantify what would be required to achieve a given HSR. It was determined that high fin density heat sinks were cost competitive with low fin density heat sinks. High fin density heat sinks also provided better thermal performance. Natural convection heat sinks, big enough to meet the system requirements, were ruled out as cost prohibitive.

CONCLUSION

The design presented herein represents the best possible fluid cooling system subject to the imposed design constraints.

The heat sink options were analyzed on a price basis. Natural convection heat sinks would have to be very large in order to achieve the same HSR as obtainable in a much smaller forced air heat sink. The large size required for a natural convection heat sink made this option cost prohibitive.

The most surprising result obtained via this study was that high fin density heat sinks are price competitive with low fin density heat sinks (as shown in Figure 3). This is contrary to what one's instinct may tell him. At first, it would seem that the cheapest heat sink is an off-the-shelf aluminum extrusion with a fan. This case study has revealed that a similar performing, properly designed staked fin heat sink with the same performance can actually be cheaper! Furthermore, the use of a high fin density heat sink would provide better heat pumping performance and, consequently a shorter cool down time all within a smaller overall heat sink size.

REFERENCES

1. Nagy, M. J. and Buist, R. J., "Effect of Heat Sink Design on Thermoelectric Cooling Performance," in *Proceedings of the XIII International Conference on Thermoelectrics*, Kansas City, Missouri, USA, 1994.