The Effect of Fan Orientation on Heat Sink Performance

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Abstract

Heat dissipation is a critical factor in optimizing performance for thermoelectric assemblies. Most thermoelectric modules require some form of heat sinking to remove the waste heat during the transfer of thermal energy. These dissipaters come in many forms and sizes depending on the requirements of the application. One widely used method of heat sinking uses a finned plate with a tubeaxial fan to provide forced-air convection. However, the fan can be oriented to push air through the heat sink or to pull air into and out of the heat sink.

This paper presents test results on the performance of heat sinks with the fan in two different orientations. Different heat sink profiles were tested with each fan orientation to determine which orientation gave the best heat sink performance.

Introduction

Forced-air convection heat sinks are probably the most common method of removing heat energy away from thermoelectric modules today. Different types of these heat sinks include extruded fin, folded fin, bonded fin and staked fin, just to name a few. Though the heat sink profiles differ, forcing air through the finned area improves heat dissipation in almost all applications.

Several experiments were conducted to simulate common thermoelectric heat sink applications. Measurements of heat load and temperatures were taken to determine each heat sink's thermal resistance (HSR) for each fan orientation. Data is presented for the various heat sinks and the corresponding performance impact associated with one of two fan orientations: (1) air either being pushed, or (2) pulled through the finned area (see Figure 1).

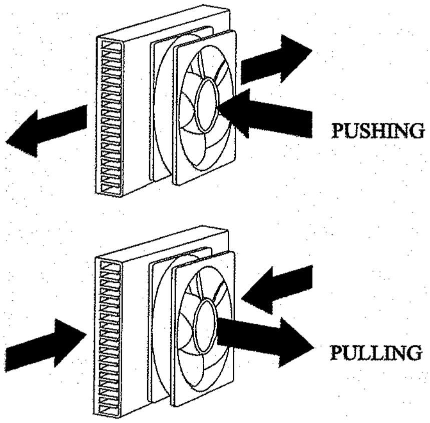


Figure 1. Fan Orientation

In addition to modeling, it is often necessary to actually test the heat sink to determine its HSR. Heat sink modeling can be done with computational fluid dynamics (CFD) and finite element analysis (FEA) software. However, that can become very expensive in consideration of the computer processing time. Empirically derived heat transfer equations are faster and more easily used than CFD and FEA codes, but they are not very accurate except in certain situations.

In any case of heat sink modeling, it must consider such effects as turbulence and non-uniform air flow. If the fan is pushing air, some turbulence could be generated at the base of the heat sink whereas pulling air might cause turbulence more towards the fin tip. Non-uniform air flow occurs because usually the fan is mounted very close to the fin tips of the heat sink. The air tends to flow along tangential velocity vectors (with respect to fan blade rotation) that are parallel to the heat sink fins. Furthermore, the airflow is both hydrodynamically and thermally developing. Models that do not account for these effects will yield inaccurate results.

Heat transfer correlations found in current literature lack the sophistication to incorporate these details [1]. However, they can be used to predict performance fairly well. Model results are presented for comparison with measured HSRs.

Test Set-Up and Procedure

Five different heat sink profiles were selected for this study. The overall dimension for these heat sinks measure 127 mm wide by 178 mm in length. The other dimensions for fin spacing, fin height, plate thickness and fin thickness varied from one heat sink to another (see Figure 2 and Table 1).

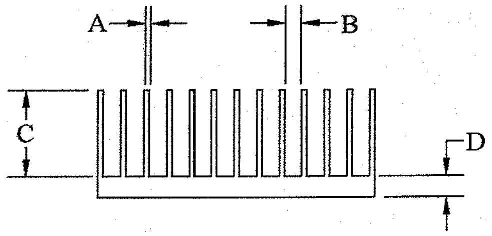


Figure 2. Heat Sink Profile

Table 1. Heat Sink Dimensions in mm

Heat Sink	A	B	C	D
4 FPI Extruded	2.00	4.80	36.25	10.00
5 FPI Extruded	2.10	3.40	41.00	7.80
8 FPI Bonded	0.62	2.45	36.00	10.00
10 FPI Bonded	0.62	1.80	36.00	11.00
8 FPI Hybrid	2.00 and 0.62	2.00	36.25 and 38.00	10.00

All experiments were set-up and performed in the same manner (see Figure 3). Two thermoelectric modules (TEMs) were wired in series and powered with approximately 15.5 VAC to provide heat load. The TEMs were assembled between the heat sink and a flat aluminum plate. Thermal grease was applied to both sides of the TEMs before compressing. The bolting configuration applied standard thermoelectric compressive forces. The entire aluminum plate and heat sink mounting surface was insulated using approximately 50 mm of polyurethane foam.

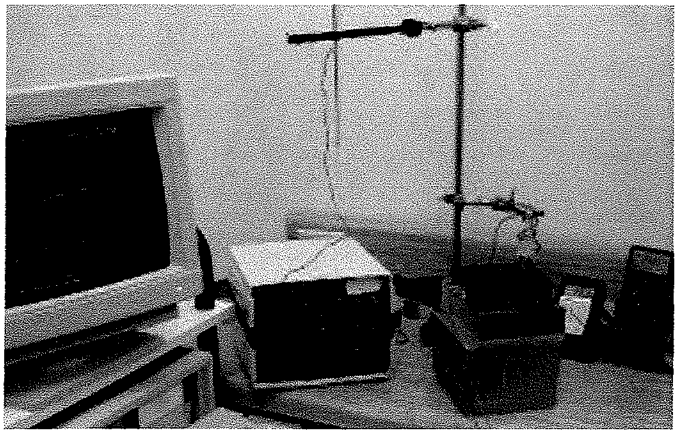


Figure 3. Photograph of Test Set-up

Each of the heat sinks was instrumented with T-type thermocouples in five locations. Four of the thermocouples were embedded into small holes in the heat sink plate using thermally conductive epoxy. Locations for these thermocouples were the center of the plate, left and right edges of the plate and one corner of the plate. Two additional thermocouples were used to measure ambient and top aluminum plate temperatures. Temperature measurements were made using TE Technology, Inc. model TS-1537 computerized test system. Voltage and current measurements were also gathered to determine the heat load applied to each heat sink.

A standard 24 VDC fan was selected for providing air flow. The fan was mounted directly to the heat sink fins and held in place with aluminum tape. Air flow and pressure drop measurements were also monitored to verify the fan performance based on the manufacture's specifications (see Figure 4).

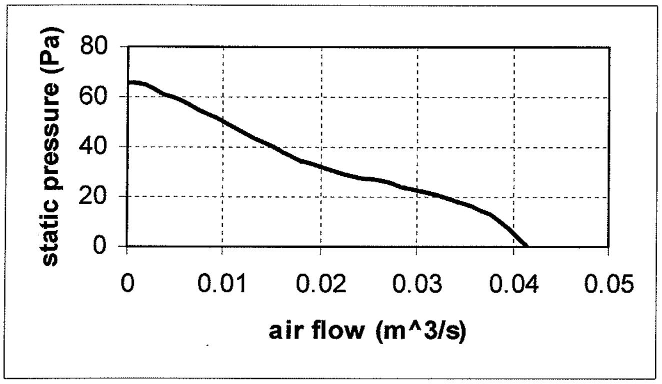


Figure 4. Fan Curve

Test Results

The thermal resistance of each heat sink was determined by first averaging the left and right edge temperatures. Then the center, corner, and average of the left and right edge temperatures were averaged together to derive an overall average heat sink temperature. The ambient temperature was subtracted from the overall average, and this difference was divided by the heat load put into the heat sink to determine the HSR.

The heat load was determined by multiplying the measured voltage by the measured current supplied to the TE modules. Since the top plate was heavily insulated, it was assumed that this surface was adiabatic. That is, all of the power to the TE modules was assumed to dissipate only through the heat sink.

The HSR of each heat sink is summarized in Table 2.

Table 2. Measured Heat Sink Thermal Resistance (HSR)

Heat Sink	Pushing (K/W)	Pulling (K/W)
4 FPI Extruded	0.0911	0.114
5 FPI Extruded	0.0860	0.102
8 FPI Hybrid	0.0743	0.0725
8 FPI Bonded	0.0806	0.0767
10 FPI Bonded	0.0757	0.0663

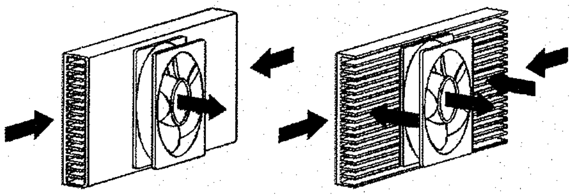
Each heat sink was also modeled using proprietary software that used empirically derived heat transfer equations for forced air convection. The software did not make a distinction as to fan orientation. The results are presented in Table 3.

Table 3. Software Calculated HSR

Heat Sink	HSR (K/W)
4 FPI Extruded	0.112
5 FPI Extruded	0.0867
8 FPI Hybrid	0.0721
8 FPI Bonded	0.0798
10 FPI Bonded	0.0679

Clearly, calculations provided a definite qualitative analysis. However, there was a significant difference between pushing and pulling in some heat sinks that calculations could not consistently and quantitatively predict.

An additional test was performed whereby the tops of the heat sink were taped off to simulate the use of a shroud which allowed air to be pulled in from the ends only (See Figure 5). The HSR measured for the 10 FPI bonded heat sink in this case was 0.0793 K/W. When the fan was pushing air, all of the air exited out the ends regardless of whether tape was applied to the tops or not.



10 FPI Bonded HSR

With Shroud		**Without Shroud**	
Pushing	Pulling	Pushing	Pulling
0.0757 k/w	0.0793 k/w	0.0757 k/w	0.0663 k/w

Figure 5. Air Flow Entrance

A possible explanation for the differences in performance between pushing or pulling air is the trade-off between turbulence at the expense of decreased air flow. It is thought that pushing air caused turbulence at the base of the fins whereas when the fan was pulling air little turbulence was created at the base of the heat sink. Turbulence tends to enhance heat transfer, but it also causes a significant pressure drop. Whether the heat sink performs better with air pushing or pulling depends on whether the heat sink benefits more from turbulence but at lesser air flow or benefits more from having more air flow at lesser turbulence.

Conclusions

For the heat sinks that were investigated, pulling air gave better performance when the fin density was high. Pushing air gave better performance when the fin density was low.

However, in many situations a fan shroud is used with the heat sink. The shroud (as simulated by taping the tops of the heat sink) can have a detrimental impact on performance as compared with the heat sink using no shroud. Again though, it must be emphasized that testing is paramount for each configuration. For example, pulling air yielded a lower HSR when the fan did not use a shroud while pushing air yielded a lower HSR when a shroud was used.

The use of empirical equations was effective at least in providing a good estimate of performance. Further research is needed to effectively predict what configuration would provide the best performance since it is not intuitively obvious.

In summary, fan orientation on various types of heat sinks can affect thermal transfer efficiency. In some cases, the performance of an assembly can be affected so much that the unit will not meet specified heat pumping requirements. Thorough examination of fan orientation in relationship to heat sink type should be performed before accepting or rejecting a heat sink on modeling alone.

References

1. Rohsenow, Warren M., Hartnett, James P., and Cho, Young I. *Handbook of Heat Transfer*. 3rd ed. New York: McGraw-Hill, 1998