A Student Research Project To Compare the Theoretical Pressure Drops Across Heat Sinks to Actual Measured Pressure Drops

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Abstract:

Heat sinks are an important component in many cooling schemes. In order to properly size a heat sink it is necessary to know how much air (or other fluid) is flowing between the fins. This involves comparing fan curves to the pressure drop curves for the heat sink.

This work focuses on standard finned heat sinks with uniform rectangular cross-section, evenly spaced fins. Theoretical information is available, and there are published theoretical curves. The purpose of this student's work was to determine how actual measured pressure drops compare with these theoretical curves.

This paper presents background information on the theoretical work. It describes the test method used and the test fixture that was designed to provide a wide variety of heat sink configurations. Finally, the results are presented and summarized.

Introduction:

Heat sinks are used extensively in the electronics industry for cooling various components throughout most designs. They are key components used to assure that the components do not overheat during operation. Some heat sinks are cooled using natural convection, but most are cooled with forced convection in air using a fan or blower to provide the air flow. It is not too difficult to determine the air flow requirements for a particular heat sink. In fact, that information can often be supplied by the manufacturer. However, selecting the right fan to supply the air flow is a little more difficult.

A typical data sheet for a fan includes a maximum flow value, maximum differential pressure value, and a performance curve. It is not uncommon for the maximum flow value from the data sheet to be used for fan selection. Unfortunately, this will result in an undersized fan because the maximum flow value is never attained in a real application. In fact, the fan usually operates significantly below the maximum due to back pressure caused by the system it is being attached to.

To determine what the actual flow will be for a particular fan a flow impedance curve for the system is superimposed over the flow characteristic curve for the fan. The point where the two curves intersect is the only operating point the system and fan have in common making it the actual operating point for the fan (Figure 1).

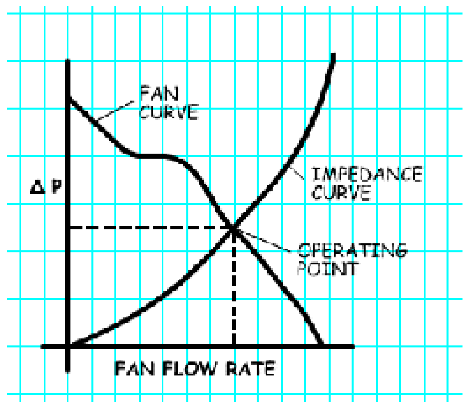


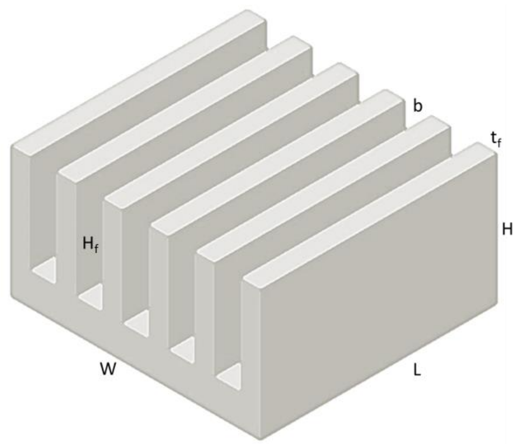
Figure 1 – Determining Flow Operating Point

Heat sinks have two factors related to their design that can reduce the actual air flow past the fins. One of them is the flow impedance of the channels. That is the focus of this project. The other is air escaping from outer edges of the fins which is called air flow bypass. This project does not deal with that factor. R.E. Simons has published an article in Electronics Cooling[1] which deals with this.

The purpose of this project was to compare the theoretical calculations presented by J.R. Colham and Y.S. Muzychka[2] and summarized by R.E Simons[3] with the actual results obtained by testing .

Theory:

The theoretical calculations were first published by J.R. Colham and Y.S. Muzychka[2] and subsequently summarized by R.E Simons[3]. The equations used for this projected are shown below for reference. No attempt is made here to derive these equations. See reference 2 for that information. Figure 2 shows the heat sink nomenclature used in the equations.



H = Total Height
H_f = Fin Height
L = Heat Sink Length
W = Total Width
t_f = Fin Thickness
N_f = Number of fins

Figure 2 – Heat Sink Nomenclature

It is assumed that all of the air that enters the channels remains in the channels. In other words, there is no flow by-pass. To model this with the test device the top edges of the fins are covered to enclose the channels.

The purpose of the testing is to produce a flow impedance curve for each of the fin configurations in the test matrix. Here are the applicable equations.

$$\Delta P = \left(K_c + 4 f_{app} \frac{L}{D_h} + K_e\right) \frac{\rho V^2}{2} \qquad \text{Eq 1}$$

ΔP = Pressure drop across the heat sink
K_c = A coef. representing the pressure drop due to a sudden contraction at the entrance
K_e = A coef. representing the pressure drop due to a sudden expansion at the exit
D_h = Hydraulic diameter
L = Length of the channel in the direction of flow
ρ = Air density
V = Average air velocity in the channels
f_{app} = Apparent friction factor for hydrodynamically developing flow in the channels

The coefficients for the sudden contraction and the sudden expansion are given by the following equations.

$$K_c = 0.42(1 - \sigma^2) \qquad \text{Eq 2}$$

$$K_e = (1 - \sigma^2)^2 \qquad \text{Eq 3}$$

Where $\sigma = 1 - \frac{N_f(t_f)}{W}$ Eq 4

The following equation can be used to find the apparent friction factor for hydrodynamically developing flow in the channels (f_{app})

$$f_{app} = \frac{\left[\left({3.44}/{\sqrt{L^*}}\right)^2 + (f \cdot R_e)^2\right]^{1/2}}{R_e} \qquad \text{Eq 5}$$

Where: $L^* = \frac{L/D_h}{R_e}$ Eq 6

And: The friction factor f is the friction factor for fully developed flow given by:

$$f = (24 - 32.527 \cdot \lambda + 46.721 \cdot \lambda^2 - 40.829 \cdot \lambda^3 + 22.954 \cdot \lambda^4 - 6.089 \cdot \lambda^5)/R_e \qquad \text{Eq 7}$$

Where λ is the aspect ratio: $\lambda = B/H_f$ Eq 8

Figure 3 shows a set of manual calculations using MathCAD to both demonstrate the process and to aid in troubleshooting the spreadsheet calculations.

$Nfin := 30 \quad Hf := 1.492$

$Length := 1.996 \quad \rho := 4.33374 \cdot 10^{-5}$

$Width := 2.025 \quad tf := 0.02$

$\mu := 3.82 \cdot 10^{-7}$

$G := 10$

$b := \frac{[Width - (Nfin \cdot tf)]}{Nfin - 1} \quad b = 0.049 \quad Dh := 2 \cdot b \quad Dh = 0.098$

$\sigma := 1 - \left[\frac{(Nfin \cdot tf)}{Width}\right] \quad \sigma = 0.704$

$Kc := 0.42 \cdot \left[1 - (\sigma^2)\right] \quad Kc = 0.212$

$Ke := \left[1 - (\sigma^2)\right]^2 \quad Ke = 0.255$

$V := \frac{G}{(Nfin \cdot b \cdot Hf)} \quad V = 4.547$

$Re := \frac{(\rho \cdot V \cdot Dh)}{\mu} \quad Re = 50.692$

$\lambda := \frac{b}{Hf} \quad \lambda = 0.033$

$$f := \left[\frac{\left[24 - (32.527 \cdot \lambda) + \left[46.721 \cdot (\lambda^2)\right] - \left[40.829 \cdot (\lambda^3)\right] + \left[22.954 \cdot (\lambda^4)\right] - \left[6.089 \cdot (\lambda^5)\right]\right]}{Re}\right]$$

$f = 0.453$

$L\alpha := \left[\frac{\left(\frac{L}{Dh}\right)}{Re}\right] \quad L\alpha := 0.401$

$$fapp := \frac{\left[\left(\frac{3.44}{L\alpha^{\frac{1}{2}}}\right)^2 + (f \cdot Re)^2\right]^{\frac{1}{2}}}{Re}$$

$fapp := 5.389$

$$\Delta P := \left[Kc + \left[4 \cdot fapp \cdot \left(\frac{L}{Dh}\right)\right] + Ke\right] \cdot \rho \cdot \frac{(V^2)}{2}$$

$\boxed{\Delta P := 0.196}$

Figure 3 – MathCAD Sample Calculations

Test Apparatus:

There are two primary pieces of equipment that are used for this project – an air flow bench and a heat sink mounting bracket.

The air flow bench is a device that is used for plotting flow impedance curves, such as those in this project. Figure 4 shows a schematic of the flow bench, and figure 5 shows a picture of the actual unit that was used for this project. Devices such as this are commercially available and can be fairly expensive. This unit was designed and built at the collage as a separate senior design project according to the ANSI/AMCA 210-99[4] standard for testing fans. Even though it was built for a relatively modest cost the results compare favorably with published fan curves produced using commercial equipment. The pressure differential across the nozzle plate is related to the air flow rate through the flow bench. A MatLab program is used to make the calculations converting differential pressure to flow rate. The differential pressure across the test specimen is measured directly. Plots are made using an Excel spreadsheet. Details of the data collection procedure were previously published[5], and will not be repeated here. (One change to the procedure was made for this project. The differential pressures were not taken using differential pressure gauges, but with differential pressure transducers. The output voltage from the transducers was recorded, then the data is converted to differential pressure in inches of water.) The heat sink mounting bracket is designed to allow a wide range of heat sinks to be built with different fin heights and fin spacing. Figure 6 shows a 3-D model of the heat sink mounting fixture.

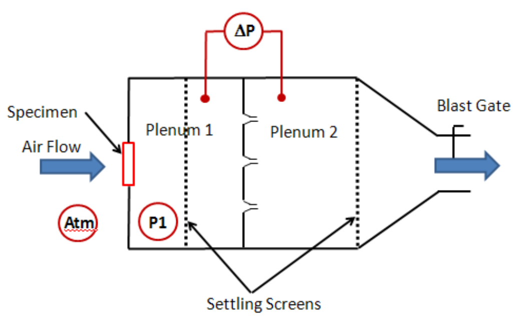


Figure 4 – Schematic of Air Flow Bench

Figure 5 – Picture of Air Flow Bench

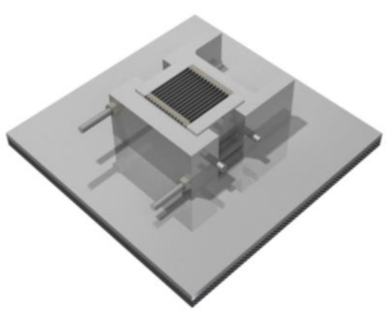


Figure 6 – Heat sink Mounting Fixture (Air Flow from Top to Bottom)

Procedure:

Twenty-five different configurations of the heat sink were tested. They varied by fin height and fin spacing. Figure 7 shows the test matrix that was used for this project.

Fin Height	Space Thicnkess					Number of Fins				
2.495	0.064	0.202	0.086	0.125	0.049	25	10	20	15	30.0
2.264	0.064	0.202	0.086	0.125	0.049	25	10	20	15	30.0
2.013	0.064	0.202	0.086	0.125	0.049	25	10	20	15	30.0
1.492	0.064	0.202	0.086	0.125	0.049	25	10	20	15	30.0
1.003	0.064	0.202	0.086	0.125	0.049	25	10	20	15	30.0

Figure 7 – Test Matrix

- Assemble the selected heat sink in the mounting fixture.
- Attach the mounting fixture to the inlet of the air flow bench.
- Adjust the flow rate on the air flow bench to obtain multiple data points.
    - Data included the differential pressure transducer voltage output for both the differential pressure across the flow measurement nozzle and across the heat sink.
    - Approximately 20-30 data points were collected for each heat sink. This is more than a sufficient number of data points to create an accurate impedance curve for the heat sink.
- Input the data into an Excel spreadsheet.
- Convert the output voltages into the corresponding differential pressures in inches of water using the spreadsheet.
- Use a custom MatLab program to convert the differential pressure across the nozzle into the air volumetric flow rate in CFM.
- Plot the flow impedance curve for the heat sink using Excel. Figure 8 shows what a typical flow impedance curve should look like.

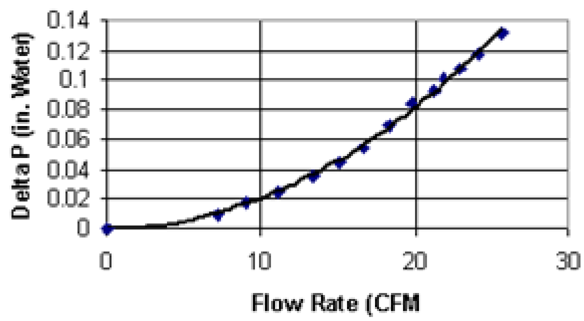


Figure 8 – Typical Flow Impedance Curve

Results:

The results shown below are examples of typical results from the testing. Three types of comparisons were made.

Flow impedance comparisons for an increasing number of fins at a fixed fin height: For this case the expectation is that a higher number of fins will cause a higher impedance due to an increased obstruction to the flow. Figure 9 shows a comparison of impedance curves for a fixed fin height of 1.492”. Note that the impedance does increase with fin numbers, as expected.

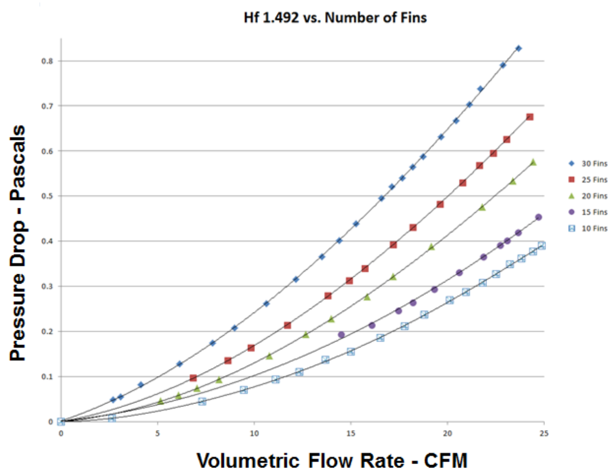


Figure 9 – Fixed Fin Height, Varied Number of Fins

Flow impedance comparisons for an fixed number of fins at a changing fin height: For this case the expectation is that a fins with the larger fin height will cause a lower impedance due to a decreased obstruction to the flow. Figure 10 shows a comparison of impedance curves for 25 fins with heights varying from 1.000” to 3.495”. Note that the impedance does decrease with fin height, as expected.

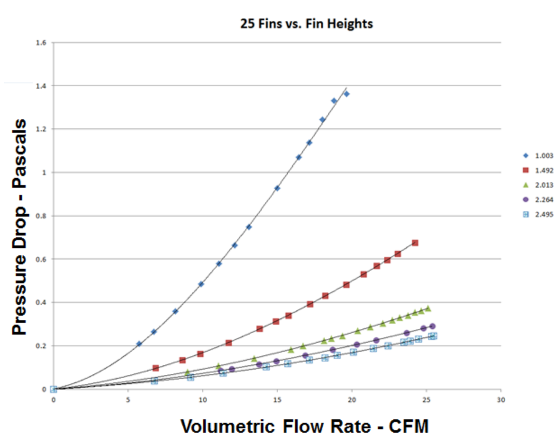


Figure 10 – Fixed Number of Fins, Varied Fin Height

The third comparison shows probably the most important result from the tests. Figure 11 shows a comparison of a theoretical impedance curve with an actual, measured impedance curve. The example shown below is for 30 fins with a height of 1.492". This curve is typical of what was found throughout the testing.

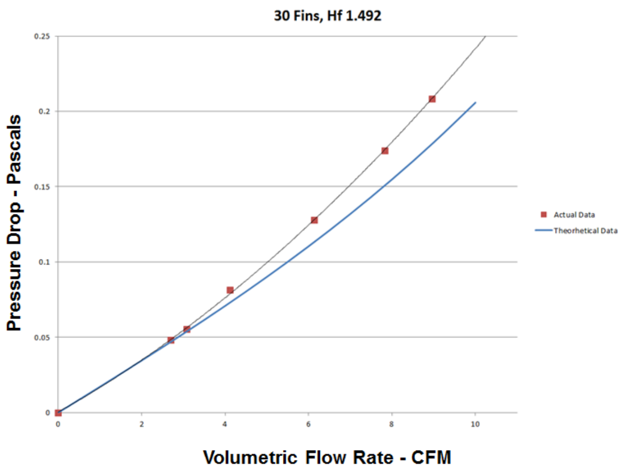


Figure 11 – Actual vs. Theoretical Impedance Curve

Application of this research in the classroom:

At Penn State Erie, The Behrend College the electrical engineering technology (EET) students are required to take a course in their senior year which is a thermal science course focused on electrical and electronic applications. Heat sinks are a very important part of this course. The students learn about the basic calculations for fins and heat sinks, the use of heat sinks in practice and the selection of a heat sink for a given application. They also learn about fans, including fan curves, impedance curves and the determination of the operating point in a given application. As part of this course there is a lab component where the students learn to use the flow bench that was used for this research. The topic of flow bypass, and flow pressure drop across a heat sink will be added to the course material, and a lab exercise duplicating some of the work done in this research will be added. This should help to enhance their understanding of how heat sinks work. This dovetails nicely with another lab exercise demonstrating that the actual flow rate that a fan delivers to a system can be significantly lower than the catalog flow rate for maximum flow[5]. They will learn that the principles behind that exercise also apply to heat sinks, and that they should not overestimate the air flow through a heat sink or risk overheating a component.

Conclusions:

Based on the work done on this project it appears that the theoretical work published by Colham and Muzychka[2] gives results very close to actual measurements. Actual measurements for all configurations generally ran slightly higher than the theoretical calculations. More work would have to be done to isolate the reason for that, but it is possible that the actual heat sink models do not quite match exactly with the assumptions for the theoretical work. Two areas that could be focused on for further study are the inlet conditions, and the fourth physical wall that is enclosing the channel. Either of those could add to the overall friction in the system. The effect of that is to increase the overall impedance of the device.

Bibliography:

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