Thermal Sciences

ME-495

Staggered Plate Heat Sink Experiment

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1. Introduction

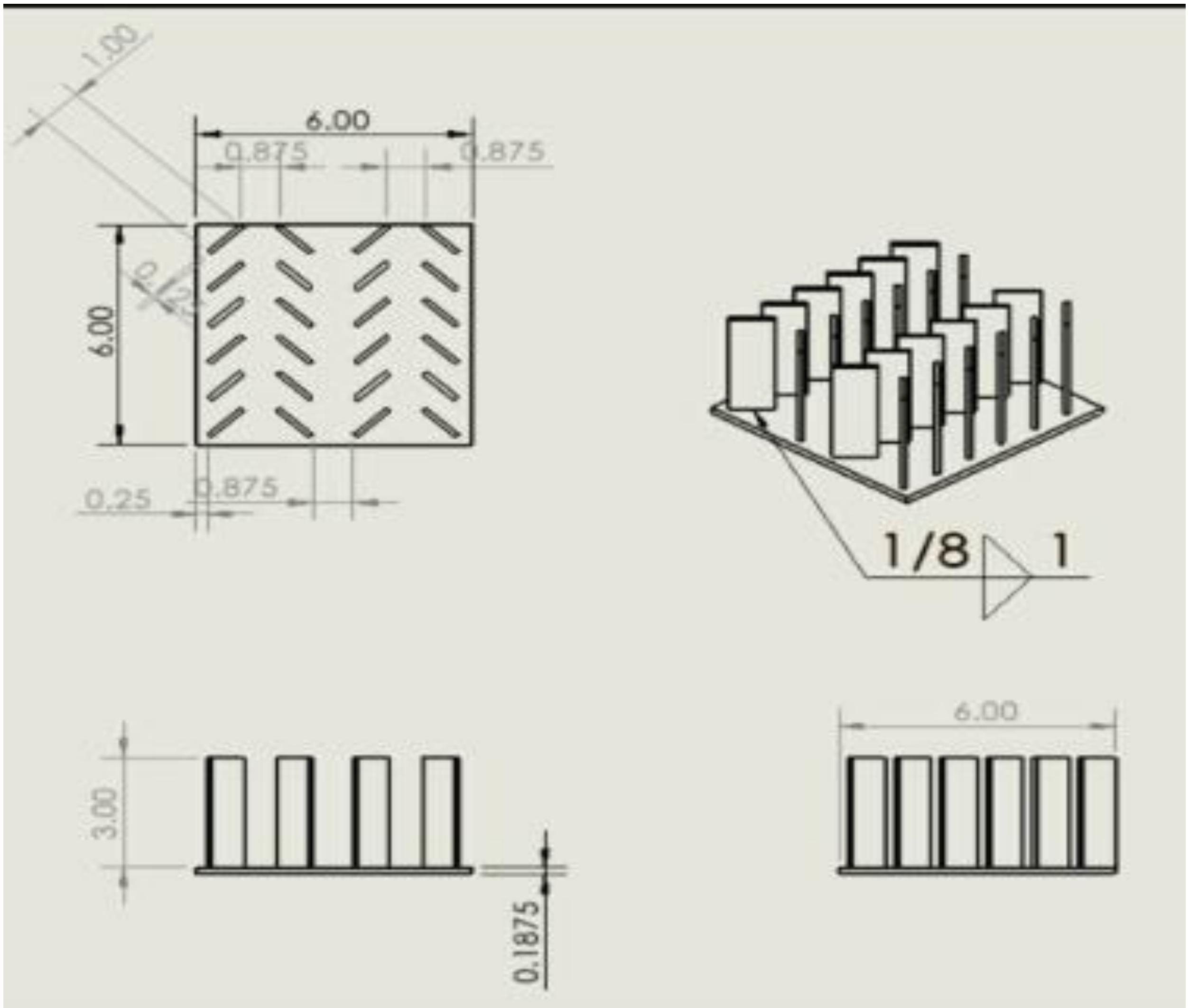
In recent decades, advancements in high-power electronic equipment have led to increased power density and substantial heat generation, which negatively affects device reliability and performance. To address these challenges, researchers have developed a range of thermal-management strategies aimed at improving the operational efficiency, dependability, and longevity of electronic systems. Among these strategies, heat sinks remain one of the most widely used solutions for dissipating excess thermal energy due to their simplicity, effectiveness, and cost-efficiency.

Fin integration is particularly effective for improving heat-sink performance, as fins significantly enhance convective heat transfer by increasing surface area and promoting stronger fluid interaction. Staggered plate-fin heat sinks, in particular, have demonstrated superior thermal performance compared with conventional in-line configurations. This improvement is primarily attributed to the angled or offset arrangement of the fins, which disrupts the boundary layer development and generates more turbulent and mixed airflow within the channels. The resulting flow instabilities enhance convective heat transfer by continuously renewing the fluid in contact with the fin surfaces, thereby reducing thermal resistance and improving overall heat-sink effectiveness.

We designed a square staggered plate heat sink with the following specifications:

	Imperial units	SI units
Material	Mild Steel (Low carbon steel)	
Length	6in	0.1524m
Thickness	0.1875in	0.004763m
Fin thickness	0.125in	0.003175m
Fin height	3in	0.0762m
Fin width	1in	0.0254m

Fin clearance	0.875in	0.022225m



2. Method

2.1 Convection Heat Transfer Coefficient

To calculate the heat transfer coefficient, we need to first use the Nusselt number formula [1, Eq.6.50] .

$$\overline{Nu} = \frac{\bar{h}L}{k_f} = f(Re_L, Pr)$$

Where $\overline{Nu}$ = average nusselt number

$\bar{h}$ = Average convection heat transfer coefficient

L = Length of the plate (Parallel to the air flow)

Re_L = Reynolds number

Pr =Prandtl number

And $f(Re_L, \mathrm{Pr})$ will depend on multiple factors. For laminar flow over a plate the next equation is provided [1, Eq 7.49]:

$$\overline{Nu}_L = 0.680\, Re_L^{1/2}\, Pr^{1/3}$$

To know if the system is laminar or turbulent, the Reynolds number must be found using the equation [3, eq 7-11]:

$$Re = \frac{\rho V L}{\mu} = \frac{VL}{\nu}$$

Where ρ = density

V = Velocity

μ = Dynamic viscosity

ν = Kinematic viscosity

For the flow to be turbulent on a plate, the Reynolds number must be greater than 5×10^5 [3, Ch. 2].

From here the Information we must get from tables is the thermal conductivity (k),Prandtl number, kinematic viscosity, and density. If working with air, these properties can be found in the table [1, tab. A.4]:

TABLE A.4 Thermophysical Properties of Gases at Atmospheric Pressure[a]

T (K)	ρ (kg/m^3)	c_p (kJ/kg · K)	$\mu \cdot 10^7$ (N · s/m^2)	$\nu \cdot 10^6$ (m^2/s)	$k \cdot 10^3$ (W/m · K)	$\alpha \cdot 10^6$ (m^2/s)	Pr
Air, $\mathcal{M}$ = 28.97 kg/kmol							
100	3.5562	1.032	71.1	2.00	9.34	2.54	0.786
150	2.3364	1.012	103.4	4.426	13.8	5.84	0.758
200	1.7458	1.007	132.5	7.590	18.1	10.3	0.737
250	1.3947	1.006	159.6	11.44	22.3	15.9	0.720
300	1.1614	1.007	184.6	15.89	26.3	22.5	0.707
350	0.9950	1.009	208.2	20.92	30.0	29.9	0.700
400	0.8711	1.014	230.1	26.41	33.8	38.3	0.690
450	0.7740	1.021	250.7	32.39	37.3	47.2	0.686
500	0.6964	1.030	270.1	38.79	40.7	56.7	0.684
550	0.6329	1.040	288.4	45.57	43.9	66.7	0.683

Using all this the average convection heat transfer coefficient can be found, and we can proceed to find the heat rate.

2.2 Heat Rate

2.2.1 Heat Rate Out from Heat SInk

Now that we have found the convection heat transfer coefficient, we first start by calculating the heat transfer from the base and ignoring the fins for now:

$$q = \overline{h} A_s (T_s - T_\infty)$$

Where A_s = Surface area of the plate excluding the are covered by fins

T_s = Surface temperature

T_∞ = Temperature of the air

We should now calculate the heat rate of the fins. Using case A from the following table will give the heat transfer for a single fin [1, tab. 3.4].

TABLE 3.4 Temperature distributions and heat rates for fins of uniform cross section

Case	Tip Condition ($x = L$)	Temperature Distribution θ/θ_b	Fin Heat Transfer Rate q_f
A	Convection: $h\theta(L) = -kd\theta/dx\vert_{x=L}$	$\dfrac{\cosh m(L-x) + (h/mk)\sinh m(L-x)}{\cosh mL + (h/mk)\sinh mL}$ (3.75)	$M\dfrac{\sinh mL + (h/mk)\cosh mL}{\cosh mL + (h/mk)\sinh mL}$ (3.77)
B	Adiabatic: $d\theta/dx\vert_{x=L} = 0$	$\dfrac{\cosh m(L-x)}{\cosh mL}$ (3.80)	$M \tanh mL$ (3.81)
C	Prescribed temperature: $\theta(L) = \theta_L$	$\dfrac{(\theta_L/\theta_b)\sinh mx + \sinh m(L-x)}{\sinh mL}$ (3.82)	$M\dfrac{(\cosh mL - \theta_L/\theta_b)}{\sinh mL}$ (3.83)
D	Infinite fin ($L \rightarrow \infty$): $\theta(L) = 0$	e^{-mx} (3.84)	M (3.85)

$\theta \equiv T - T_\infty$ $\quad m^2 \equiv hP/kA_c$

$\theta_b = \theta(0) = T_b - T_\infty$ $\quad M \equiv \sqrt{hPkA_c}\,\theta_b$

A table of hyperbolic functions is given in Appendix B.1.

Where T = Temperature

A_c = Cross sectional area of fin

P = Perimeter of the cross-sectional area

To get the total heat transfer by convection from the fins q_f is just multiplied by the total number of fins.

Total heat transfer by convection from the heat sink is then found adding the heat transfer from the fins and the plate.

To consider the infinite fin, it would mean that the tip would be adiabatic. This happens when $\tanh(mL) = 1$. This happens when $mL = \infty$, but if we consider $\tanh(mL) = 0.99$ to be good enough, this would happen when $mL = 2.65$. Therefore to consider infinite length we find the next equation [1, sec. 3.6].

$$L \geq L_\infty \equiv \frac{2.65}{m} = 2.65\left(\frac{kA_c}{hP}\right)^{1/2}$$

2.2.2 Heat Rate into the Air

The previous heat transfer found can then be compared to the heat transfer that is going into the air, as the thermal energy that leaves the heat sink by convection moves into the air. This heat transfer is calculated with [1, eq. 1.12e]:

$$q = \dot{m}c_p(T_{out} - T_{in})$$

Where $\dot{m}$ = mass flow

c_p = Specific heat capacity

The specific heat capacity can be found in Table A.4, previously referenced to in section 2.1. Mass flow can then be found with this simple equation [2, eq. 4.4a]:

$$\dot{m} = \rho AV$$

After measuring both temperatures, temperature in and temperature out, it is possible to calculate the heat transfer with experimental values.

3. Analysis

We assumed the velocity to be 9 meters per second, since that is a common air velocity for heats guns, which is what we are gonna use to force the air.

The cross-sectional area of the heat gun was measured to be 1.5in or 0.0381m.

We are gonna be using boiling water to keep the bottom surface of the heat sink at a constant temperature of 100°C, therefore this temperature is used for the max temperature of the film (air). The minimum temperature should be around room temperature, so we used 23°C for room temperature which was measured at the lab. The film temperature is then the average between the 2, which is calculated as $T_f = 61.5°C = 334.6K$.

We can now get air properties from table A.4 which was referenced before, and interpolating to find properties at T_f:

$$\nu = 1.92199 \times 10^5 \frac{m^2}{s}$$

$$Pr = 0.701485467$$

$$k = 0.0287494 \frac{W}{mK}$$

Now Reynolds number can be calculated:

$$Re_L = \frac{\left(\frac{9m}{s}\right)(0.1524m)}{1.92199 \times 10^5 \frac{m^2}{s}} = 36640.84$$

According to the Reynolds number, the flow is laminar over the whole length of the plate, therefore the equation for Nusselt number for laminar flow is used:

$$\overline{Nu_L} = 0.680(36640.84)^{\frac{1}{2}}(0.701485)^{1/3} = 115.6550204$$

Now the average convection heat transfer coefficient is:

$$\overline{h_x} = \frac{(115.6550204)\left(0.0287494 \frac{W}{mK}\right)}{0.1524m} = 21.817667 \frac{W}{m^2K}$$

For the next steps we also find the thermal conductivity of mild steel (1, Tab. A.1):

$$k = \frac{60W}{mK}$$

Then we find the hea transfer by convection, but to know which method to use we first need to find the length to consider infinite length:

$$L_\infty = 2.65\left(\frac{\left(\frac{60W}{mK}\right) * (0.003175m) * 0.0254m}{\left(21.817667 \frac{W}{m^2K}\right)(2m)(0.003175 + 0.0254)}\right)^{\frac{1}{2}} = 0.165m$$

Our fin length design is a lot lower than the length for infinite length. We also got convection at the fin tip, therefore we use case A from the table.

To start finding the heat transfer rate, we first calculate $\theta_b, m, and\ M$:

$$\theta_b = 100℃ - 23℃ = 77℃$$

$$m = \sqrt{\frac{(21.817667 \frac{W}{m^2K})(0.05715m)}{(\frac{60W}{mK})(8.0645 \times 10^5 m^2)}} = 0.40773m^{-1}$$

$$M = 77℃\sqrt{(21.817667 \frac{W}{m^2K})(0.05715m)(\frac{60W}{mK})(8.0645 \times 10^5 m^2)} = 5.98W$$

$$q_f = (5.98W)\frac{\sinh\left((0.40773m^{-1}) * 0.0762\right) + \left(\frac{21.817667\frac{W}{m^2K}}{(0.40773m^{-1})} * \frac{60\mathrm{W}}{\mathrm{kk}} / \mathrm{quatmemners}\right) / \left(\cosh\left((0.40773m^{-1}) * 0.0762\right)\right.}{\cosh\left((0.40773m^{-1}) * 0.0762\right) + \sinh\left((0.40773m^{-1}) * 0.0762\right)}$$

$$q_f = 5.37W$$

$$q_{fins} = 24 * 5.37W = 128.9W$$

For the heat transfer rate of the base, we first find the area of the base, excluding the are covered by the fins

$$A_b = 0.1524^2 m^2 - 24(0.0254m)(0.003175m) = 0.02129m^2$$

$$q_b = 21.817667\frac{W}{m^2K}(0.02129m^2)(77°\text{C})\left(\frac{K}{°\text{C}}\right) = 35.767W$$

$$q_{total} = 128.9W + 35.767W = q_{total} = 164.667W$$

$$\boxed{q_{total} = 164.667W}$$

We can find the temperature of the air after it passes through the heat sink, but specifications of the heat gun are needed. Air velocity of 9m/s is assumed, the diameter of the heat gun outlet is measured as d=1.5in or 0.0381m.

We then calculate outlet area and volumetric flow with it.

$$A = \frac{\pi(0.0381m)^2}{4} = 0.00114m^2$$

$$\dot{V} = (0.00114m^2)\left(\frac{9m}{s}\right) = \frac{0.01026m^3}{s}$$

The density is needed to calculate the air flow, and the heat capacity is needed to find temperature 2. From table A.4 we can find the density at T_f:

$$\rho = \frac{1.0462512kg}{m^3}$$

$$c_p =$$

$$\dot{m} = \left(\frac{1.0462512kg}{m^3}\right)\left(\frac{0.01026m^3}{s}\right) = \frac{0.0107354kg}{s}$$

$$T_2 = \frac{164.667W\left(\frac{°\text{C}}{K}\right)}{\left(\frac{0.0107354kg}{s}\right)\left(\frac{1008.324J}{kgK}\right)} + 23°\text{C} = 38.21°\text{C}$$

$$\boxed{T_2 = 38.21°\text{C}}$$

4. Experiment Setup

The equipment needed is stove, aluminum take-out pans, duct-tape, two thermocouples, staggered plated heat sink, computer, Arduino, alligator wires, jumper wires, bread board, water (H2O), heat gun, and anemometer.

To set up the experiment, we will place a mini stove on the table flowing this we will set an aluminum tray on the stove and fill it with water to act as a bowl to keep water boiled and give our heat sink a constant temperature. From here we will position the custom staggered-plate heat sink directly on top of the aluminum tray, ensuring contact between the water and the base of the heat sink. Next, we mount the heat gun on a holder in front of the anemometer at the inlet and move the stove with our heat sink next to it to ensure a better read on the airflow that passes through the fin array. From here we will place the thermocouples in front of the heat sink and in the back. After we ensure good placements to collect temperature, we route the wires away from the hot surface. After we connect the thermocouples and anemometer to the Arduino through circuitry and a breadboard to help transmit data to the computer. The last ste[would be to power the stove to boling point of water and then power the heat gun to then procede to record temperature and airflow data.

Bill of Materials	
Item	**Price**
Stove	$13
Arduino	$20
Aluminum take-out pans	$2
Steel (for heat sink)	$5
Bread board	$6
Wires (Bundle with variety)	$5
Anemometer (3d printed)	$2
Aluminum foil	$1
Total	$54.00

5. Calibration

The things that needed calibration were our ways of measuring temperature and velocity. For the temperature we are using thermocouples connected to an Arduino, and the code we created in Arduino can be edited to have more accurate readings. We used boiling water and put the thermocouple into the boiling water; the code was modified until it showed the desired temperature (100℃). This was repeated using melting ice with a desired temperature of 0℃. Multiple measurements were also compared with already calibrated thermocouples.

The next thing that needed calibration was the anemometer. We also made the anemometer from zero, this was made from a piece that would rotate when subjected to wind, and it was connected to an electric motor which would have voltage induced depending on the air velocity. A resistance was used to avoid overloading the Arduino and burning the components, and the voltage across this resistance was measured as the input. This input was directly proportional to the velocity of air, which means that we only needed to create a variable to go from the reading to velocity. The way to calibrate it was only through comparison with other readers, from our classmates and even from the lab.

6. Experiment Results

The anemometer outputs an average air velocity 9.5m/s, which is close to our assumptions. This makes sense as this is a common air velocity for heat guns.

Due to using the heat gun, even if it was set up to use the lowest heat possible, it was still ejecting air at a higher temperature than the room temperature. This means that our initial air temperature was also different from our assumed air temperature. This temperature had an average of 32.3 ℃.

The air temperature after passing through the heat sink had an average of 52.1℃. This is higher than our calculated, but this cannot be used as direct comparison as the air had an initial temperature higher than what was used in the calculations.

These temperatures and velocity can be used to calculate the heat transfer absorbed by the air while passing through the heat sink.

$$\dot{q} = \frac{9.5m}{s}(0.00114m^2)\left(\frac{1.046kg}{m^3}\right)\left(\frac{1008.324J}{kgK}\right)(52.1℃ - 32.3℃) = 226.24W$$

Experimental			
v	T1	T2	$\dot{q}$
9.5m/s	32.3 ℃	52.1℃	$226.24W$
Calculated			
v	T1	T2	$\dot{q}$
9m/s	23℃	38.21℃	164.667W

7. Conclusion

Overall, our experimental investigation successfully validated the expected thermal-performance benefits of the staggered-plated heat-sink configuration. The convection coefficient we measured was indeed higher compared with the predicted number, confirming that staggered enhanced configuration causes more increased turbulence and significantly improves heat transfer. These findings prove that staggered plated heat sinks are superior to conventional methods and demonstrate that strategic fin design can meaningfully improve cooling effectiveness in high-power systems.

References

[1] T. L. Bergman and A. Lavine, Fundamentals of heat and mass transfer. Hoboken, Nj: John Wiley & Sons, Inc, 2019.

[2] M. J. Moran, H. N. Shapiro, D. D. Boettner, and M. B. Bailey, Fundamentals of engineering thermodynamics, 9th ed. Hoboken, Nj: John Wiley & Sons, Inc, 2018.

[3] R. W. Fox, A. T. Mcdonald, P. J. Pritchard, and J. C. Leylegian, Fluid mechanics. Hoboken, Nj: John Wiley, 2012.

[4] "Thermal Conductivity of Solids and Metals," Nuclear Power. https://www.nuclear-power.com/nuclear-engineering/heat-transfer/thermal-conduction/thermal-conductivity/thermal-conductivity-of-solids-and-metals/