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Abstract

In this paper, we describe a novel high-performance heat sink based on screen-fin technology. The structural features of the heat sink are presented. The apparatus to measure both the pressure drop and heat transfer performance is described. Correlation equations of friction factor and Colburn-j factor as a function of the divergence half angle θ , coolant properties and flow rate through the mesh are formulated. A semi-empirical model is used to predict the heat exchanger pressure drop, thermal performance and to design prototype heat sinks. Prototypes are built and tested. By screen-fin technology, the best performance of this kind of heat sink with external dimensions: 76.2 mm wide, 63.5 mm deep with 38.1 mm high screen-laminate fins is 4.3 Watt/ $^{\circ}\text{C}$ at 62.3 Pa pressure drop of air flow through the heat sink.

Keywords

heat sink, screen-laminate

Nomenclature

A	area
c	fluid specific heat
d	wire diameter of mesh screen
D_h	hydraulic diameter of mesh screen
f	friction factor
F	modified friction factor
G	fluid mass velocity
h	convection heat transfer coefficient
H	height
j	Colburn j-factor
J	modified Colburn j-factor
k	thermal conductivity
ke	effective thermal conductivity
L	width
M	mesh number
n_{row}	number of rows of screen-fins
Pr	Prandtl number
q	heat transfer rate
Re	Reynolds number
St	Stanton number
t	thickness of screen
T	temperature
U	conductance of screen
W	depth
β	heat transfer surface area to volume ratio
ΔP	pressure drop
ε	volume porosity
ρ	fluid density
μ	fluid viscosity
θ	divergence half-angle

Subscripts

b	fin base
f	fluid
i	superficial
s	surface area or solid
n	n rows
o	approach

Introduction

The rapid advancement in microprocessor technology has led electronics thermal system designers to pay increased attention to the design of heat sinks. The increase in power density of micro-electronic packages has motivated the demand for much better thermal control devices and methods at both the silicon level and the system level. Challenges with respect to cost, performance, and weight of new design heat sinks need to be addressed. Based on the cooling mechanism employed to remove heat from the electronics, heat sinks can be largely divided into five categories:

- Single Phase System: this kind of heat sink includes air-cooled and single-phase liquid cooled heat sinks. Also, air-cooled heat sinks can be categorized as natural convection or forced convection heat sinks.
- Phase Change System: there is phase change occurring during the heating or cooling process. Examples include heat pipes, capillary pumped loops, loop heat pipes systems and thermal energy storage systems.
- Radiation Heat Sinks: generally this kind of heat sink is applied to spacecraft thermal control due to lack of air or other working fluid.
- Thermoelectric Heat Sink: use Peltier effect to remove heat.
- Hybrid Heat Sink: this type of heat sink integrates two or more of above technologies into a novel heat exchange system.

Air-cooled heat sinks are devices that enhance heat dissipation from a hot surface by a cooler ambient air. These devices have limited capacity due to the small thermal capacity of air. However, in the near term air cooling in conjunction with traditional heat-sink design will continue be the method of choice for heat dissipation due to the ease of application, reliability, and low cost of this technology.

The easiest way to increase heat dissipation is to increase the surface area. Finned surfaces achieve this. At present the market offers five archetypes of finned heat sinks involved in

mass production: stampings, extrusions, bonded/fabricated fins, castings and folded fins. Among these designs, stamped fins are the cheapest; extruded fins prevail for the good tradeoff in performance and cost; and folded fins have the best performance.

However, thermal performance not only depends on the presence of a finned surface but also on the fin shape and surface layout. Fin optimization is a very important way to enhance heat transfer performance and minimize the pressure drop across heat sinks. There are a number of reports on fin geometry design and array optimization. Lee [1] conducted a study on the optimum number of fins and fin length, and gave an analytical simulation model to compute performance of a parallel plate heat sink. Chapman et al [2] performed experiments with elliptical pin-fin heat sinks. Their results show that elliptical pin-fin heat sinks are able to minimize the pressure loss across the heat sink by reducing vortex effects, and to enhance thermal performance by maintaining a large exposed surface area available for heat transfer. Cimtalay and Fulton [3] developed a multi goal and multi constraint mathematical model to optimize and evaluate a heat sink on chip. A Compromise Decision Support Problem (DSP) [4] is used in their study. Their model provides a very flexible and interesting tool to make a good tradeoff among air velocity, fin geometry design, and cost of heat sinks. Heat transfer enhancement mechanisms in in-line and staggered parallel plate fin heat exchangers were also studied by Zhang et al [5] who examined effects of geometry.

Screen-Fin Technology

A screen laminate consists of bonded laminations of two-dimensional plain-weave conductive screens. Xu and Wirtz [6] show that plain-weave screen laminates can be fabricated into simple porous media structures with in-plane effective thermal conductivities up to about 78.5% of base material values.

Figure 1 shows a unit cell of an isotropic plain weave screen. The wire filament diameter is d , and the mesh pitch is $1/M$, where M is the “screen mesh number”. In this work, we consider only single layer isotropic screen laminates. The thickness of each screen is, $t = 2d$. Under these conditions, Xu and Wirtz [6] show that the porosity (ε) and specific surface area (β) are given for $0 < Md < 0.577$, as

$$(1 - \varepsilon) = -3.906 \times 10^{-4} \pi \cdot C \cdot Md \quad (1)$$

$$\beta = 4(1 - \varepsilon) / d \quad (2)$$

where $C = 123(M \cdot d)^4 - 384(M \cdot d)^2 - 640$ and $Md = M \cdot d$. It is noted that the specific surface area of a plate of thickness $2d$ is $\beta(\text{plate}) = 1/d$. Equations (1) and (2) show that a plain-weave with $Md = 0.318$ will have the same surface area. Therefore, screen laminates having $Md > 0.318$ will have a greater heat transfer surface area than equivalent sized plates.

Xu and Wirtz also show that, for cases where the thermal conductivity of the fluid contained within the screen is much smaller than the thermal conductivity of the screen wire filaments ($k_f \ll k_s$) the in-plane effective thermal conductivity of a plain-weave screen is

Fig. 1 Plain-Weave Unit Cell

$$ke_y = \frac{80 \cdot \pi \cdot k_s \cdot Md \cdot}{-C} \quad (3)$$

with k_f and k_s the thermal conductivity of the fluid and wire filament material, respectively.

Tong and London [7] reported measurements of friction factor and mesh heat transfer coefficient for inline plain weave laminates and staggered cross-rod matrices (no interweaving). Miyabe et al. [8] reported heat transfer coefficient correlations for plain-weave screen laminates that are in close agreement with those of Tong and London. Armour and Cannon [9] reported pressure drop correlations for plain-weave screens, but not laminations of screens. Park et al. [10] reported the thermal/fluid characteristics of isotropic plain-weave screen laminates and made a comparison between screen-laminate systems and unconsolidated packed bed matrices. They found that screen-laminate systems could generally be configured to offer thermal and pressure drop performance superior to unconsolidated packed bed matrices. Park et al also provide a semi-empirical model to compute U , the heat transfer coefficient based on fin contact area (A_b) of a woven laminate configured as a thin fin with fluid passing through the lamination.

$$\frac{U \cdot H}{ke_y} = mH \cdot \tanh(mH) \quad (4)$$

where $mH = \sqrt{\hat{G}_i (1 - e^{-ntu})}$ and dimensionless coolant superficial mass velocity $\hat{G}_i = \frac{cG_i H^2}{ke_y t}$; $ntu = St \cdot \frac{\beta \cdot t}{\varepsilon}$, and St ,

Stanton number, is defined as $St = \frac{h}{c(G_i/\varepsilon)}$, here G_i is the coolant superficial mass velocity.

Wirtz et al. [11] studied the thermal/fluid characteristics of 3-D, aluminum wire filament, bonded mesh deployed as a heat exchange surface. Their study shows that 3-D metallic weaves can be structured to have effective thermal conductivity that is two or more times greater than what can be achieved with other porous media. However, 3-D weave pressure drop is significantly higher than that of a screen laminate.

Screen Laminate Heat Sink Design Concepts

We fold the screen into a serpentine configuration, as shown in Fig. 2. In this kind of structure, we can gain a very big surface area and a reasonable pressure drop by controlling the divergence angle, 2θ .

Figure 2 shows top and side views of a screen-fin heat sink. The heat sink consists of a base plate (presumed to be in thermal contact with the electronics) and a fin structure. The fin structure consists of a screen-fin oriented perpendicular to the heat sink base, and laid out in a serpentine arrangement so that the divergence/convergence angle between segments of the screen is 2θ . The external envelope is L_0 wide, W_0 deep, and H_{tot} high. $H = H_{tot} - H_b$ is the fin height. ΔP is the pressure drop across the serpentine screen-fin heat sink. G_0 , $T_{f,0}$ denotes the inlet air mass velocity and temperature respectively.

As shown in the figure, the screen-fin is laid out in a serpentine arrangement so that the ambient air with uniform temperature T_f and flow rate G_0 flows down a converging section, through the screen and then out through a diverging section. This arrangement provides for an increase in heat transfer surface area, and it reduces the superficial mass velocity incident on the screen laminate, thereby effectively controlling coolant pressure drop.

The objective of the present work is to document thermal/fluid characteristics of plain-weave screen-fins that are laid out in a serpentine configuration, and to develop screen-fin heat exchanger technology with particular attention to air-cooled heat sink applications. In this work, we need to find the correlation between divergence angle of segments of the serpentine layout and the pressure drop and heat transfer. Based on this, a semi-empirical model is developed to predict the pressure drop and thermal performance of design-optimized woven mesh heat sinks.

Friction Factor and Stanton Number Correlations for Serpentine Screen-Fins

We postulate that the pressure drop across the mesh, ΔP , is functionally related to fluid, flow properties and θ as follows:

$$\Delta P = fn(\rho, G, \mu, d, \beta, t, \varepsilon, \theta) \quad (5)$$

where $G = \frac{G_i}{\varepsilon}$ is the internal mass velocity. Then dimensional analysis gives

Fig. 2 Schematic of Screen-Laminate Heat Sink

$$f = fn(Re, \theta, \varepsilon) \quad (6)$$

$$F = f \cdot \left(\frac{1 - \varepsilon}{\varepsilon} \right)^{0.4} \quad (7)$$

where $f = \frac{2 \cdot \rho \cdot \Delta P}{G^2 \cdot \beta \cdot t}$, is the friction factor. F is the modified friction factor as defined by Eq. (7), and $Re = \frac{GD_h}{\mu}$ is the mesh Reynolds number, with $D_h = \frac{4\varepsilon}{\beta} = \frac{\varepsilon d}{1 - \varepsilon}$ the mesh hydraulic diameter.

In a similar way, we postulate that the mesh heat transfer coefficient, h is functionally related to fluid, flow properties and θ as follow:

$$h = f(\rho, G, \mu, d, \beta, t, \varepsilon, k_f, \theta) \quad (8)$$

Dimensional analysis gives

$$St = fn(Re, Pr, d/t, \varepsilon, \theta) \quad (9)$$

The heat transfer can be expressed by a modified J-factor.

$$J = j \cdot \left(\frac{1 - \varepsilon}{\varepsilon} \right)^{-0.2} \cdot \left(\frac{t}{d} \right) \quad (10)$$

where $j = St \cdot Pr^{\frac{2}{3}}$ is the Colburn j-factor, Pr is the Prandtl number of the coolant. In the following, we describe experiments to determine the specific form of Eqs. (7) and (10).

Figure 3 shows a schematic of the experiment rig. Screen-fin heat sinks were prepared and mounted in a wind tunnel, then the pressure drop and heat transfer characteristics were studied.

Fig. 3 Schematic of Experiment Setup

In the experiments, the channel is of open loop, induced-draft design. Laboratory air passes through the test article, a plenum chamber and suitably long pipe to a laminar flow element, which measures flow rate; then to a variable speed exhauster. The pressures drop across heat sink and pressure drop across the laminar flow element, which is used to estimate flow rate G_i , are measured with an electronic manometer having $\pm 4\%$ accuracy. The laminar flow element has $\pm 3\%$ accuracy.

A flat-plate heater is used to simulate the power applied on the bottom of the screen-fin heat sink. Thermocouples are used to monitor the base temperature of the heat sink and inlet air temperature. In this case we measure the overall conductance, UA_b , and use Eq.(4) to back-calculate the wire-element heat transfer coefficient, h . We estimate that heat losses out of the bottom of the test rig are less than 2% of the input power (q), temperature measurements are accurate to $\pm 0.05^\circ\text{C}$, and q is measured to $\pm 4\%$.

Figure 4 shows a one-row test article consisting of 8 screen-segments ($\theta = 4.1^\circ$) that are 18mm high. Other test articles had up to six serpentine screens arranged in series ($1 \leq n_{row} \leq 6$, $3^\circ \leq \theta \leq 90^\circ$). The base plate of some test articles was covered with insulation in order to assess heat transfer from the exposed base plate surface. Test data show that the j -factor will increase about 10% if the base is insulated. This effect is considered in the data reduction.

In the present work, commercial grade isotropic copper screen ($k_s = 400 \text{ W/mK}$) is pre-coated with a solder bearing paste-flux (95% Sn/5% Pb, $k_s = 55\text{-}60 \text{ W/mK}$) and the screen/base plate assembly is re-flow soldered to form the test article. Test articles with different divergence angles ($6^\circ \leq 2\theta \leq 180^\circ$) and fin heights ($16.1 \text{ mm} \leq H \leq 28.4 \text{ mm}$) are investigated. All of our test articles are made of copper screen with $d = 0.48 \text{ mm}$ ($0.019''$), $M = 7.9 \text{ cm}^{-1}$ (20 in^{-1}) with $\varepsilon = 0.677$, $\beta = 2.68 \times 10^3 \text{ m}^{-1}$, and $key = 55.2 \frac{\text{watt}}{\text{m} \cdot \text{K}}$ as estimated by the Xu and Wirtz model [6].

A total of 18 samples were tested where the flow rate was varied over the following range: $1.76 \leq Re \leq 300$. All experiments were performed at least twice (on different days), to check the repeatability of the data. A Monte Carlo error propagation simulation indicates the following 95%-

Fig. 4 Screen-fin test article. $2\theta = 8.2$, $H = 18 \text{ mm}$, 8 segments, $n_{row} = 1$.

Fig. 5 F Factor of Screen Laminate Heat Sink

confidence level tolerance on computed results: Re less than $\pm 5\%$; f less than $\pm 15\%$; St less than $\pm 10\%$.

Both pressure drop and heat transfer are measured for a single row of isotropic screen laminates with $d = 0.483 \text{ mm}$, $M = 7.9 \text{ cm}^{-1}$. In Fig. 5, we present just several sets of pressure drop data ($\theta = 3^\circ, 7.1^\circ, 20^\circ, 34^\circ, 90^\circ$) in terms of modified friction factor, F . The data is best correlated with the following expression:

$$F = \left[(0.39 + \frac{24.5}{Re}) + \left(\frac{0.173}{\tan(\theta)^2} + \frac{26.33}{Re^{1.16} \cdot \tan(\theta)} \right) \right] \quad (11)$$

The expression contains both inertial loss terms and viscous terms. The first two terms (for $\theta = 90^\circ$) were developed by Park [10], and the second two terms reflect the effect of variations in screen-fin divergence angle. Eq. (11) reproduces the data that generated it with a standard error less than $\pm 10\%$.

Figure 6 plots the modified friction factor, Eq.(11) as a function of θ for three Reynolds numbers. As shown in the

Fig. 6 F factor versus divergence angle, Eq (11)

figure, F decreases very sharply when $0 < \theta < 20^\circ$ and the curve becomes flat when the θ is greater than about 40° .

Figure 7 summarizes heat transfer data in terms of modified Colburn j-factor (J) as a function of Reynolds number. We are correlating heat transfer data in terms of Colburn j-factor so that a direct comparison can be made with the data reported by Park, who experimented with water ($Pr = 9.2$). Data for $\theta = 3^\circ, 6.2^\circ, 9.5^\circ$ and $\theta \geq 20^\circ$ are plotted. The figure shows that J is sensitive to θ when it's less than about 14° ; and, variations in θ have no significant effect on J when θ is greater than 14° . We have chosen to correlate the data with a power-law. A regression analysis yielded the following results. Eq. (12) reproduces the data that generated it with a standard error of $\pm 7\%$.

$$J = \begin{cases} C(\theta) \cdot Re^{-0.722} & 0^\circ \leq \theta \leq 8^\circ \\ \{ [C(\theta - 4.3^\circ)]^{0.9} \cdot 4.608 - 2.718 [C(\theta - 4.3^\circ)]^{1.2} \} \cdot Re^{-0.722} & 8^\circ < \theta < 14^\circ \\ 2.362 Re^{-0.722} & \theta \geq 14^\circ \end{cases} \quad (12)$$

where $C(\theta) = 10^{[3.996 \sin(\theta) - 0.3]}$.

In Figure 8, we plot J (Eq.12) as a function of θ for three flow rates. The figure shows the rapid increase in J with increase in divergence angle up to $\theta \approx 20^\circ$, with J constant for $\theta > 20^\circ$.

Multiple-Row Heat Transfer & Pressure Drop A single row of screen laminate cannot provide enough surface area. In order to gain more surface area we need to pattern several rows of screen laminate in a series arrangement. Based on the test data for multiple rows, we develop friction factor and Colburn j-factor corrections that account for the decrease in row-by-row performance of a multiple row system. Here we define the overall pressure drop and conductance as ΔP_n and UAs_n as

$$\Delta P_n = \Delta P_1 \cdot n_{row}^{0.35} \quad 1 \leq n_{row} \leq 6 \quad (13)$$

$$UAs_n = UAs_1 \cdot n_{row}^{0.62} \quad 1 \leq n_{row} \leq 6 \quad (14)$$

From the above equations, it is evident that, with increases in the number of screen-fin rows, the overall conductance increases more rapidly than does the overall pressure drop.

Fig. 7. J Factor of Screen Laminate Heat Sink

Fig. 8. J Factor versus θ , Eq. (12)

So, a larger number of rows is favored with a design that seeks to optimize performance of a heat sink having a pressure drop constraint.

Benchmark Experiments Equations (11)-(14), together with Eqs. (1)-(4) can be used to design screen-fin heat sinks. In order to test the accuracy of the design algorithm, we constructed a prototype, measured its pressure drop and heat transfer performance, and compared these results with performance predictions embodied in above equations.

The prototype is 63.5 mm deep, 45.72 mm wide with 23.368 mm high screen-fins ($d = 0.483\text{mm}$, $M = 787.402\text{ m}^{-1}$), as shown as Figure 9. It consists 6 rows of serpentine-pattern screen-fins with divergence angle, $2\theta = 22^\circ$. We also studied how the row number affects performance by successively cutting the last row from the piece after each experiment so that we measured the performance with 4 rows, 5 rows and 6 rows. In Table 1 we present comparisons between the test

Table 1 Comparisons Between Test Data and Model, $2\theta = 22^\circ$.

		Conductance [watt/°C]					
		4 rows		5 rows		6 rows	
		Model	Test Data	Model	Test Data	Model	Test Data
ΔP [Pa]	24.907	1.15	1.01	1.22	1.105	1.205	1.17
	62.267	1.37	1.255	1.51	1.465	1.555	1.535
	99.628	1.47	1.37	1.66	1.645	1.695	1.745
Standard Error		10.5%		6.3%		2.5%

Fig. 9 Prototype Heat Sink

data and prototype. The table shows that our model agrees well with our test data. The standard error is less than 11%.

Discussion

Now that the correlations for divergence angle 2θ , heat transfer capacity and pressure drop are established, a semi-empirical model can be developed based on these correlations to predict the performance of this kind of heat sink and optimize the design. Because both heat transfer and pressure drop of this kind of heat sink are sensitive to the Md -product, divergence angle and number of rows, it is necessary to discuss the effect of variations of these parameters on overall heat transfer capacity and pressure drop.

Consider the one-row segment heat sink shown in Fig 10. The heat sink has depth $W_0 = 63.5\text{mm}$, width $L_0 = 45.72\text{mm}$, and fin height, $H = 38.1\text{mm}$.

Effect of Divergence Angle on Heat Transfer At fixed flow rate (Q), variations in θ have two effects on the pressure drop and overall conductance:

Fig 10 One-row Heat Sink Segment

- Increasing θ will result in a decrease in heat transfer surface area (A_s) and friction factor. However the mesh Stanton number will increase as per Eq. (12) and mesh Stanton number will vary with θ .
- The heat transfer surface area will increase with decreasing θ since more screen-fin segments can be placed on the base plate. However, at fixed flow rate (Q), an increase in surface area will result in a decrease in G_s , resulting in a decrease in Stanton number.

Figure 11 shows the overall effect of variation in θ on overall conductance of the segment (Fig 10) when $Q = 9.439$ liter/sec. The figure shows that $8^\circ < \theta < 13^\circ$ produces the highest unit surface conductance (depending on the number of screen-fin rows), with lower conductance indicated at higher and lower values of θ . This analysis also shows that the overall pressure drop is minimum when $11^\circ < \theta < 20^\circ$. Approximately the same result obtains for other flow rates.

Effect of Screen Laminate Row Number and Md -product

Figure 12 shows that the conductance of the screen-fin heat sink (Fig. 10) changes with row number with other conditions fixed. At small n_{row} , the surface area increases with the row number; so does the performance of the heat sink. But with further increase in n_{row} the flow rate reduces, leading to a decrease in overall performance, so for given Md -product and applied ΔP , there will be an optimum value of n_{row} . The figure shows that for the current overall configuration with $\theta = 11^\circ$, $\Delta P = 62 \text{ Pa}$ (0.25 inch H_2O), an optimum exists at n_{row}

Table 2 Screen-fin and plate fin characteristics. $L_o = 63.5\text{mm}$, $W_o = 76.2\text{mm}$, $H = 38.1\text{mm}$

Parameter	Screen-Fin Heat Sink	Plate-Fin Heat Sink
Material	Copper Plain-weave	Aluminum
Fin spec	$d = 0.56\text{mm}$ (0.022"), $M = 9.45\text{ cm}^{-1}$ (24in ⁻¹)	0.61 mm thick
θ	14°	
Thermal Conductivity	72 watt/(m·K)	167 watt/(m·K)
n_{row}	5	
n_{fin}		50
H.T. Surface Area	0.154 m ²	0.242 m ²
Weight of Fin Structure	193 gram	194 gram
Coolant flow @ $\Delta P = 62\text{ Pa}$	8.0 liter/sec	24.0 liter/sec

Fig. 11 Effect of Angle for Conductance

= 5, $Md = 0.528$. Other optimums obtain for other configurations.

Optimum Design of Screen Laminate Heat Sink Based on the above discussion, an optimum design does exist for a given size (L_o , W_o , H) and applied pressure drop, ΔP_n . If we consider a heat sink with a 76.2mm x 63.5mm (3" x 2.5") footprint having 38.1mm (1.5") tall screen-fins and operating at $\Delta P = 62\text{ Pa}$ (0.25 in H₂O), then systematic variation of design parameters results in the optimal design copper heat sink with properties summarized in the second column of Table 2. This heat sink will have an overall conductance of 4.3 watt/°C (0.23 °C/watt) when operated at $\Delta P_n = 62.27\text{ Pa}$.

Figure 13 compares the performance of the screen-fin optimal design with that of an aluminum plate-fin heat sink with geometric properties summarized in the third column of Table 2. Plate-fin performance is calculated as per reference [13]. The figure shows that the screen-fin heat sink performance is about 50% of that of the plate-fin sink when operated at the same coolant pressure drop. Under these conditions, the restrictive nature of the screen reduces the flow rate through the heat sink so that the mesh Stanton number is relatively small. Furthermore, reference to Table 2

Fig. 12 Effect of Number of Rows, $\Delta P = 62\text{ Pa}$.

Fig. 13 Optimized screen-fin and plate-fin heat sink performance.

shows that the heat transfer surface area of the screen-fin system is about 63% of that of the plate-fin system; and, the effective thermal conductivity of the screen-fins system are approximately 43% of that of the plate-fins. On the other

hand, the volume flow rate through the plate-fin sink is approximately 4 times that through the screen-fin sink.

Figure 14 compares the performance of the two sinks at the same volume flow rate of coolant. Under these conditions, the screen-fin sink has performance comparable to the plate-fin device up to a flow rate of approximately 10 liter/sec.

Fig. 14 Comparisons between Plate-Fin and screen-fin heat sinks

Conclusions

In this study, the performance attributes of an air-cooled heat sink based on screen-fin technology is investigated. Correlations of modified friction factor and modified Colburn j-factor are developed. They show that coolant pressure drop and Stanton number are sensitive to the angle of incidence of the superficial mass velocity. The modified friction factor is found to increase rapidly as $\theta \rightarrow 0$, while the modified Colburn j-factor is observed to $\rightarrow 0$. As a consequence, configurations with $\theta \rightarrow 90^\circ$ will maximize the heat transfer rate while they minimize coolant pressure drop. On the other hand, configurations having $\theta = 90^\circ$ generally present a minimum heat transfer surface area. Consequently, there is an optimum value of θ for a given heat sink configuration.

The design attributes of a air-cooled heat sink with a footprint of 76.2mm x 63.5mm (3" x 2.5") and 38.1mm (1.5") tall screen-fins deployed as multiple rows in a serpentine configuration is investigated. The design is optimized to operate at $\Delta P = 62$ Pa (0.25 in H₂O). The overall conductance is found to be 4.3 watts/°C with 8 liter per sec coolant consumption.

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