Thermal Design and Optimization of Natural Convection Polymer Pin Fin Heat Sinks

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***Abstract*—The design and optimization methodology of a thermally conductive polyphenylene sulphide (PPS) polymer staggered pin fin heat sink, for an advanced natural convection cooled microprocessor application, are described using existing analytical equations. The geometric dependence of heat dissipation and the relationships between the pin fin height, pin diameter, horizontal spacing, and pin fin density for a fixed base area and excess temperature are discussed. Experimental results of a pin finned thermally conductive PPS heat sink in natural convection indicate substantially high thermal performance. Numerical results substantiate analytical modeling results for heat sinks within the Aihara *et al.* fin density range. The cooling rates and coefficient of thermal performance, $\mathrm{COP_T}$, that relates cooling capability to the energy invested in the formation of the heat sink, has been determined for such heat sinks and compared with conventional aluminum heat sinks.**

***Index Terms*—Energy efficient, heat sinks, least material, polyphenylene sulphide (PPS) polymer, sustainability, thermally conductive.**

NOMENCLATURE

c_p	Specific heat of air [W/kg-K].
COP	Coefficient of performance [W/W].
d	Diameter of pin-fin [m].
E	Fabrication energy [J].
h	Heat transfer coefficient [W/m^2-K].
H	Height of the pin-fin [m].
k	Thermal conductivity [W/m-K].
L	Length of the array [m].
M	Mass of heat sink [kg].
n	Number.
Pr	Prandtl number [$\mu c_\mathrm{p}/\mathrm{k_f}$].
q	Heat sink cooling rate [W].
$\mathrm{Ra_{Sh}}$	$\mathrm{g}\beta\mathrm{Pr_b}\theta_\mathrm{b}S_h^4/\mathrm{L}\nu^2$, Raleigh's no. based on S_h.
$\mathrm{Ra_L}$	$\mathrm{g}\beta\mathrm{Pr_b}\theta_\mathrm{b}\mathrm{L}^3/\nu^2$, Raleigh's no. based on L.
s	Air gap between adjacent fins [m].
S	Center to center spacing [m].
V	Volume [m^3].
W	Width of the array [m].

GREEK SYMBOLS

β	Thermal coefficient of expansion [K^{-1}].
δ	Standard deviation.
η	Fin efficiency.
θ	Excess temperature [K].
μ	Mean dynamic viscosity of fluid [kg/μm-s].
ν	Mean kinematic viscosity of fluid [m^2/s].
ρ	Mean fluid density [kg/m^3].

SUBSCRIPTS

a	Array.
amb	Ambient.
avg	Average.
b	Fin array base.
bi	Bias error.
f	Fluid versus.
h	Horizontal.
lm	Least material.
m	Mass.
opt	Optimal.
p	Pin-fin.
pr	Precision error.
sc	Space claim.
T	Total.
v	Vertical.

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I. INTRODUCTION

A. Literature Review

PIN FIN arrays relying on natural convection can be effectively used as heat sinks for various electronic cooling applications. While, aluminum is the current material of choice for heat sinks, the availability of thermally conductive PPS polymers [1] raises the possibility of lighter, more energy efficient, moldable plastic heat sinks with thermal performance in the range needed for commercial applications. The present study involves the application of the Aihara *et al.* [2] correlation, together with the Sonn and Bar-Cohen least-material insulated tip pin fin relation [3], to the design and optimization of a polyphenylene sulphide (PPS) polymer, staggered pin fin array on a 10 cm by 10 cm vertical base, operating at 25 K above the ambient temperature and cooled by natural convection. This configuration is typical of advanced electronic cooling applications and facilitates a direct comparison between the present results and the reported thermal performance of natural convection heat sinks fabricated of aluminum [4]. The results of smaller scale heat sink experiments are presented and successfully compared to numerical natural convection simulations of PPS polymer heat sink performance (see Fig. 1).

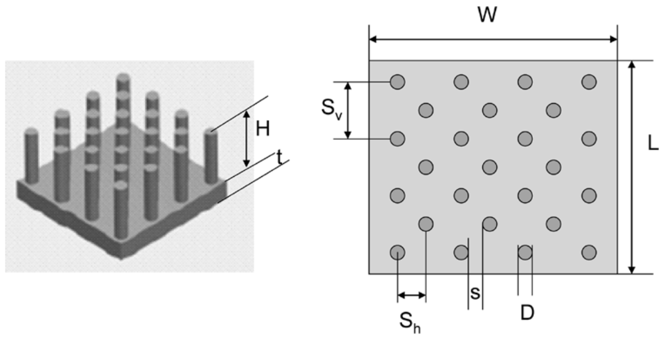

Fig. 1. Staggered pin fin heat sink.

While an extensive literature exists on convective heat sinks and fin arrays [5], [6], however the available literature devoted specifically to natural convection from pin fin arrays is relatively limited. Early, experimental results [7] were presented for a set of five staggered, widely spaced cylindrical pin fins (fin density of 0.31–1.17 fins/cm^2) on a vertical base with horizontal fins, horizontal upward facing base/vertical fins, and horizontal downward facing base/vertical fins that exchange heat by both natural convection and radiation. However, there is no available correlation for predicting natural convection pin fin array heat transfer, other than Aihara *et al.* [2], and it is obtained for the average heat transfer coefficient along a vertical surface of staggered pin fins, at a fin density of 2.42–9.90 fins/cm^2, one order of magnitude higher than in [7]. Consequently, this correlation [2], along with the work reported in [3], served as the basis for the current least-material optimization of vertical pin fin heat sinks in natural convection. The presented optimization methodology uses the Aihara *et al.* correlation for predicting cooling rates in a wider fin density ranges than reported in the original work [2]. This work focuses on presenting a design and optimization methodology for, and experimental validation of the thermal performance of PPS polymer pin fin heat sink. To date no study has addressed the use of low thermal conductivity materials, particularly thermally conductive polyphenylene sulfide (PPS) polymers, for heat sink applications.

Due to the low heat transfer coefficients encountered in natural convection cooled heat sinks for electronic applications, and the relatively weak dependence of fin performance on the thermal conductivity of the fin [6], and despite their relatively modest thermal conductivity, it may be expected that thermally conductive PPS polymers could be effectively used for commercial heat sinks. For example, the commercially available, thermally-enhanced polyphenylene sulphide (PPS) resin that possesses a thermal conductivity of 20 W/m-K, density of 1.7 g/cm^3, and a coefficient of thermal expansion (CTE) of 4–7 ppm/K [1], could constitute a lightweight, low interfacial stress alternative heat sink material. Moreover, the moldability and ease of fabrication of a PPS polymer heat sink may yield significant cost savings in the development of future commercial heat sinks. Interestingly, the energy required to produce a unit mass of this PPS polymer, at some 115 MJ/kg [8], is about half of the energy required to form the comparable mass of aluminum [9], making this a most attractive choice for sustainable development.

B. Pin Fin Equations

Heat transfer from an array of cylindrical fins is the sum of heat dissipation from the fins and the array base, and can be calculated as

$$q_T = \theta_b(h_p A_p \eta_p + h_b A_b) \tag{1}$$

where $\mathrm{A_b}, \mathrm{A_p}$ are the base area and pin fin area (with the fin tip contribution assumed to be negligible) available for heat transfer, respectively, and are given by

$$A_b = n_T(S_v S_h - \pi d^2/4) \tag{2}$$

$$A_p = n_T \pi d H. \tag{3}$$

The array dimensions L, W, and H, and the pin configuration d, S_v, and S_h affect the number of pins in the vertical and horizontal direction, as described as

$$n_v = (L - d)/S_v + 1 \tag{4}$$

$$n_h = (W/2 - d)/S_h + 1 \tag{5}$$

$$n_T = n_v n_h + (n_v - 1)(n_h - 1). \tag{6}$$

Following the form of the Elenbass correlation [10], the Aihara *et al.* correlation [2] predicts pin heat transfer coefficients to within $\pm 10\%$, in the range $d = 0.123$ cm, $W = 10$ cm, $L = 5$–20 cm. $h = 3.2$–6 cm, $S_\mathrm{v} = 0.209$–0.429 cm, $S_h = 0.212$–1.37 cm, $N = 2.25$–10.58 fins/cm^2

$$\begin{aligned}\mathrm{Nu_{Sh}} &= h_p S_h/k_b \\ &= [2S_v/\pi d]\left[(1/20)\left(\eta_p \mathrm{Ra}_{\mathrm{Sh}}^{3/4}\right)\right. \\ &\quad \left.\times\ (1 - 1/e^{120/\eta_p \mathrm{Ra_{Sh}}})^{1/2} + (1/200)\eta_p \mathrm{Ra}_{\mathrm{Sh}}^{1/4}\right].\end{aligned} \tag{7}$$

The parametric range of the Aihara *et al.* data falls within the following values [2]: In the test data reported by Aihara *et al.* [2], the vertical separation distance was set to 1.7 or 3.5 times the fin diameter. Heat transfer from the base of the array, using the classical correlation for laminar flow over a vertical flat plate [10], [11], is used for base plate heat loss

$$\mathrm{Nu}_b = h_b L/k_b = 0.59 \mathrm{Ra}_L^{1/4}. \tag{8}$$

Using (7), Iyengar and Bar-Cohen [12] determined that the optimum center-to-center spacing between staggered pin fins, i.e., the value that maximizes the volumetric heat dissipation rate, could be expressed in the form used for parallel plate heat sinks [6], [13], as in

$$S_{h,\mathrm{opt}} = 3.18P \tag{9}$$

where the parameter P, characterizing the thermal environment, is set equal

$$P = [L\nu^2/(g\beta\eta_p\theta_b\,\mathrm{Pr})]^{1/4}. \tag{10}$$

It is to be noted that the optimum spacing for a staggered array of pin fins is, thus, somewhat larger (by approximately 19%) than obtained for parallel plate arrays [4]. Following Sonn and Bar-Cohen [3], the relation between pin diameter and height, for a "least-material" fin, which maximizes heat transfer for a specified fin volume (or mass), is given by

$$d_{\mathrm{lm}} = 4.73 h_p H^2 / k_b. \tag{11}$$

The fin efficiency for a cylindrical pin fin is given by

$$\eta_p = \tanh(2Hd^2(h_p/(k_b d^5))^{1/2})/(2Hd^2(h_p/(k_b d^5))^{1/2}). \tag{12}$$

Using (11) and (12), the efficiency of the optimal pin fin is obtained as 0.789.

C. Heat Sink Thermal Design Metrics

In designing and optimizing air-cooled heat sinks, it is useful to consider several distinct thermal performance metrics, including the "array" heat transfer coefficient, the "space claim" heat transfer coefficient, and the "mass based" heat transfer coefficient. These are defined in succeeding paragraphs.

1) Array Heat Transfer Coefficient, h_a: The overall thermal capability of a convective heat sink can be represented by the array heat transfer coefficient, calculated using

$$h_a = q_T / \mathrm{LW}\theta_b. \tag{13}$$

2) Space Claim Heat Transfer Coefficient, h_{sc}: Following Iyengar and Bar-Cohen [4], the "space claim" heat transfer coefficient, h_{sc}, which represents the thermal utilization of the volume occupied by the heat sink ($L \times \mathrm{W} \times h$), can be determined as

$$h_{\mathrm{sc}} = q_T / (\mathrm{LWH}\theta_b). \tag{14}$$

3) Mass-Based Heat Transfer Coefficient, h_m: Due to the importance of minimizing the weight, as well as the cost, of a commercial heat sink, it is useful to evaluate the thermal utilization of the mass of the heat sink. This can be represented by the "mass based" heat transfer coefficient [4], as

$$h_m = q_T / (\theta_b V_p \rho_p). \tag{15}$$

In natural convection since no external pumping power is required, therefore it is expected that the least material fins will provide the least energy solution as well.

D. Array Heat Transfer Coefficients

Using the approach outlined above, it is possible to determine the array heat transfer coefficient for a range of pin fin geometries. Fig. 2 shows the variation of the array heat transfer coefficient, h_{a}, with fin height for a subset of polyphenylene sulphide and aluminum heat sinks configurations. Focusing first on the PPS fins, it may be seen that an "optimum" fin height exists for each fin diameter-fin spacing configuration. The array heat transfer coefficient appears to increase steeply, as this fin height is approached, and to decrease in a gradual fashion, as this value of H is exceeded. While the enlargement in fin area with height is responsible for the initial increase in h_{a}, the subsequent deterioration in fin efficiency as shown in Fig. 3 leads to a

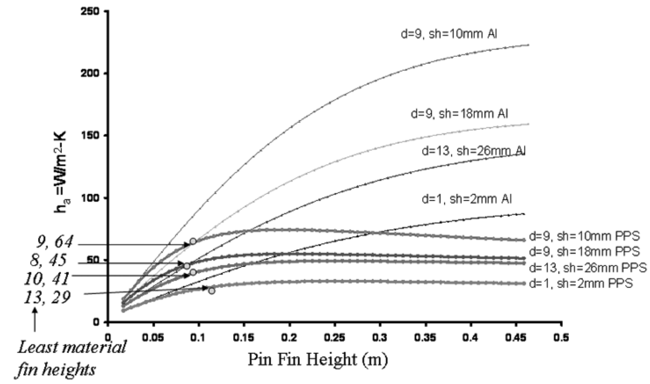

Fig. 2. Pin Fin h_{a} for enhanced PPS and aluminum.

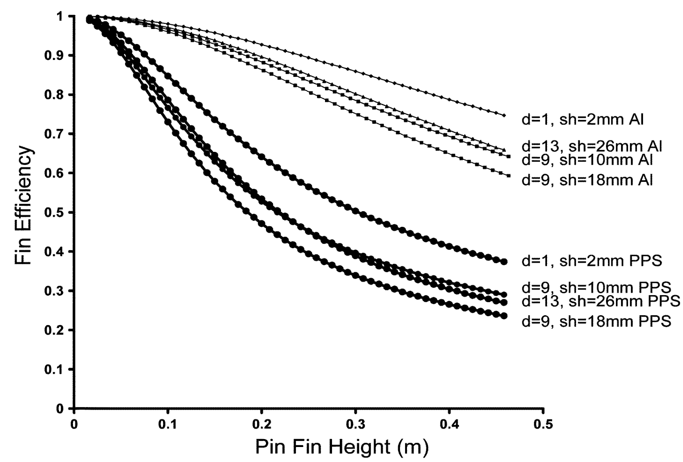

Fig. 3. Pin fin efficiency for enhanced PPS and aluminum.

reduction in the array heat transfer coefficient for fins of progressively greater height. Fig. 2 suggests that for PPS polymer pin fins this transition occurs around fin heights of 15 cm. and—that for the conditions stated—h_{a} values approaching ~ 73 W/m^{2}K, or some $\sim$15 times greater than natural convection from an unfinned surface (5 W/m^{2}K), can be achieved.

Alternatively, the h_{a} values for the aluminum array can be seen to reach and exceed $\sim$200 W/m^{2}K, or some 40 times the natural convection rates on an unfinned surface (5 W/m^{2}K), at relatively very high fin heights of some 45 cm. At this height the array heat transfer coefficient for the aluminum heat sink is still increasing, although at a slower rate than for shorter fins. This difference in thermal performance is associated with the much higher thermal conductivity of the aluminum fins. It may also be noted that in the most common range of fin heights, up to 5 cm, there is essentially no difference in the array heat transfer coefficients for these two materials and only relatively modest differences up to 10 cm fin heights.

The fin configuration that results in the highest possible heat transfer rate can be found by a comprehensive search of the solution space. In Fig. 2, showing just a few of the numerous configurations simulated, it may be seen that, for this PPS polymer material, an array of 0.9 cm fins, spaced 1 cm apart (center-to-

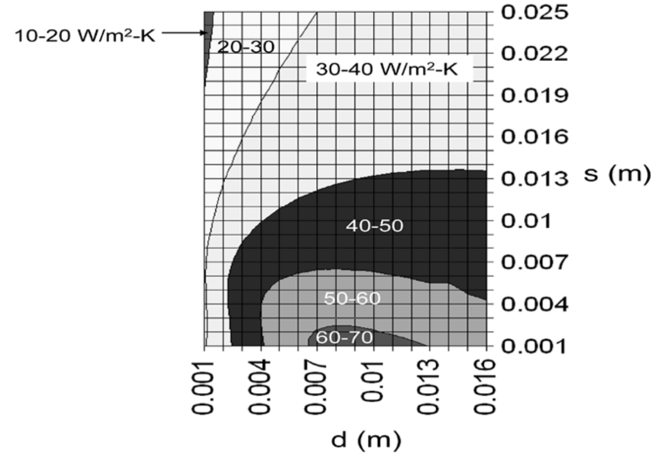

Fig. 4. Least material h_{a} for enhanced PPS fins.

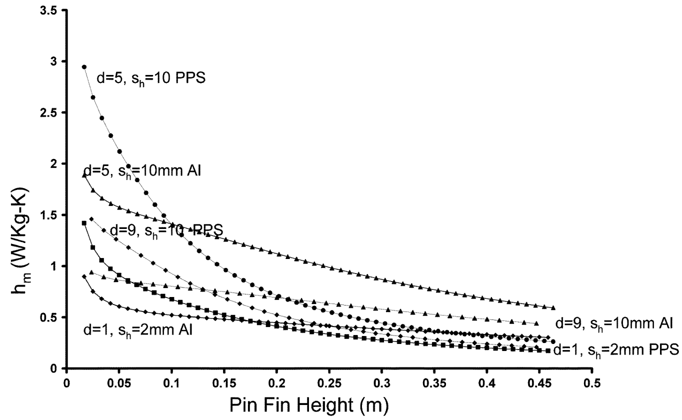

Fig. 5. Pin fin h_{m} for enhanced PPS and aluminum.

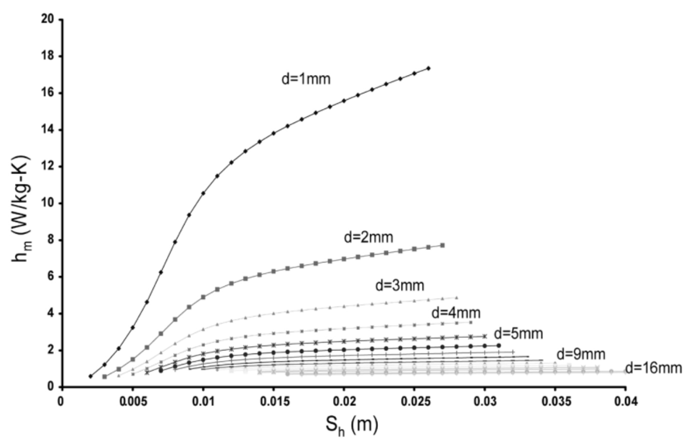

Fig. 6. Least material h_{m} for enhanced PPS.

center) in the horizontal direction and 1.6 cm apart in the vertical direction, consistently provides the highest heat transfer rates at all fin heights. The loci of the aluminum fins in Fig. 2 reveal that the 0.9 cm diameter fins with 1.0 cm spacing configuration also provides the peak thermal transport rates when aluminum is used. It is noteworthy that, for both materials, the thermal performance of the heat sink is maximized when the pin fin diameter is roughly equal to the horizontal fin spacing, as first encountered in optimum arrays of plate fins [13].

Use of (11) to determine the "least-material" fin height for a specified pin fin diameter can eliminate one degree of freedom and facilitate representation of the heat sink solution space in the form of Fig. 4. Thus, the array heat transfer coefficient is shown in Fig. 4, as a function of fin diameter and the "air gap" horizontal spacing between the fins. It reveals that the peak thermal performance for the described conditions, i.e., the "optimum array," are attained with fin diameters of approximately 0.9 cm and fin center-to-center spacing very close to 1 cm. The lateral spacing for the optimum pin fin array can be determined analytically by using (9). For an optimum pin fin array with a base length and width of 10 cm and fins of 0.789 efficiency, operating at a base temperature rise of $\theta_{\mathrm{b}} = 25\,^{\circ}\mathrm{C}$ in an ambient temperature of 45 °C, the calculated S_h value is 0.97 cm, very close to the spacing of 1 cm found by a extensive search of the solution space for this optimal design.

Using the horizontal spacing obtained from (9) to determine the optimum fin heat transfer coefficient from (7) and inserting the result in (11), with an assumed fin diameter of 0.9 cm, the "least material" fin height is found to equal 9.3 cm (Fig. 2) and to yield an array heat transfer coefficient of $\sim$ 64 W/m^2-K. From Fig. 2 it may be seen that increasing the height of these pin fins to 15 cm increases the array heat transfer coefficient to $\sim$73 W/m^2-K. However, this suggested 15% increase in heat transfer coefficient requires an increase in fin height and fin weight by some 67%, thus substantially lowering the mass-based heat transfer coefficient for this fin array. It may, thus, be argued that—as noted in [4]—an array consisting of such individually optimum, pin fins, placed at the optimum distance from each other, can closely approximate the optimum heat sink configuration.

E. Mass-Based Heat Transfer Coefficients

The heat transfer rate per unit mass of the aluminum and polyphenylene sulphide heat sinks is depicted in Fig. 5. The Fig. 5 indicates that for both materials and for each configuration, the h_{m} decreases monotonically with fin height, due to the incremental deterioration in the thermal transport capability of fin sections far removed from the base. However, the PPS polymer's reduced density (1700 kg/m^3) relative to aluminum (2700 kg/m^3) does provide the polyphenylene sulphide fins with superior mass-based performance, up to fin heights of approximately 10 cm. As the fin height is increased beyond 10 cm, the rapid decrease in fin efficiency (Fig. 3) of the low thermal conductivity PPS results in nearly constant heat transfer rates, despite the increase in physical surface area, and to a decrease in the mass-based heat transfer coefficient.

However, the high thermal conductivity aluminum heat sink experiences a more modest decrease in fin efficiency, resulting in slowly falling values of the mass-based heat transfer coefficient and h_{m}'s that are higher than the PPS fins for heights greater than 15 cm. The highest mass based heat transfer coefficient out of all the cases is obtained for PPS fin diameter of 0.5 cm with a horizontal spacing of 1 cm.

The mass based heat transfer coefficient of least-material fins increases with decreasing diameter and increasing horizontal spacing, as depicted in Fig. 6. The highest observed mass heat

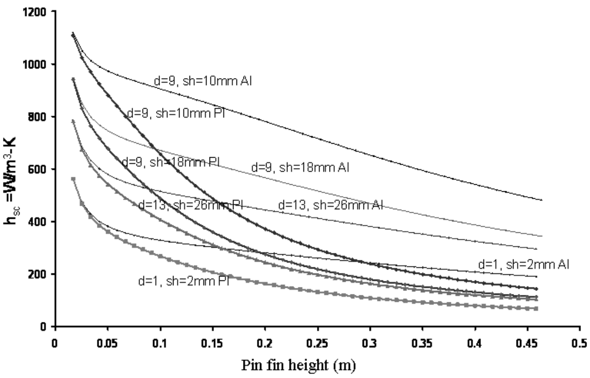

Fig. 7. Pin fin h_{sc} for enhanced PPS and aluminum.

transfer coefficient of the PPS polymer heat sink is obtained for an array that has the smallest considered pin fin diameter of 0.1 cm and center-to-center horizontal spacing of 2.6 cm yielding a value of 17 W/kg-K, nearly 18 times that for the optimum array. However, the array heat transfer coefficient for this configuration is just 18 W/m^2-K, or 28% of the optimum PPS polymer pin fin array value of 64 W/m^2-K.

F. Space-Claim Heat Transfer Coefficient

The "space claim" heat transfer coefficients, h_{sc}, are displayed in Fig. 7 for the polyphenylene sulfide and aluminum least material heat sinks. As previously noted for the mass-based heat transfer coefficients and due, again, to the deleterious effect of height on the fin efficiency, the h_{sc} values are seen to fall monotonically for both materials and all heat sink configurations.

However, as a direct consequence of the higher thermal efficiency of the aluminum fins, these heat sinks experience a less steep descent than the PPS heat sinks and provide significantly higher "space claim" heat transfer coefficient for fin heights greater than 10–15 cm. As may be observed in Fig. 7, for low aspect ratios the PPS pin fins have comparable "space claim" thermal performance to the aluminum fins.

G. Pin Fin Array Optimization

The maximum thermal performance for the specified PPS heat sink parameters, yields an h_{a} value of approximately 73 W/m^{2}K, for 0.9 cm fins, spaced 1 cm apart, while the highest mass based heat transfer coefficient of the PPS polymer material, at 17 W/kgK, is obtained for least-material fins of the smallest considered diameter (0.1 cm) and at a center-to-center horizontal spacing of 2.6 cm. Thus, the highest h_{a} configuration and highest h_{m} configuration, occur at substantially different fin dimensions. Depending on the requirements associated with specific applications, the "best" pin fin array arrangement can be expected to fall between the maxima of the array heat transfer coefficient (W/m^2-K) and the mass based heat transfer coefficient (W/Kg-K). However it is to be noted that an array consisting of individually optimum, pin fins, placed at the optimum distance from each other, can closely approximate the optimum heat sink configuration, providing a balance between

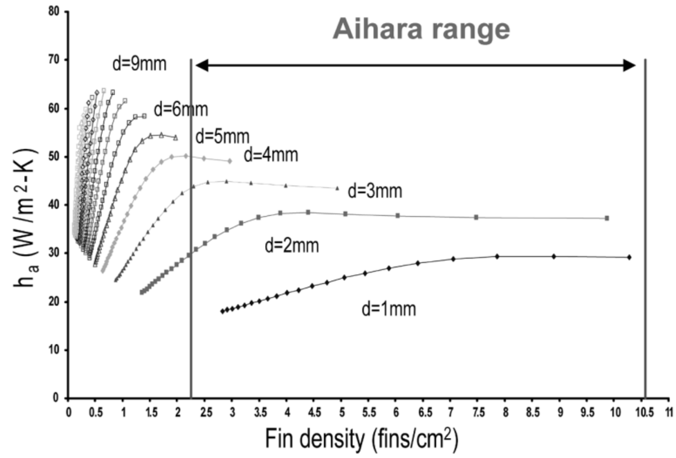

Fig. 8. Enhanced PPS pin fin h_{a} variation with fin density.

achieving the maximum thermal performance and minimizing the mass of a pin fin array.

H. Parametric Range

In basing the design and optimization of pin fin arrays on the Aihara *et al.* correlation, (7), some attention must be devoted to the parametric range of the data gathered and correlated in that pioneering study [2], particularly the fin density which reflects the combined impact of both pin fin diameter and fin spacing and varied from approximately 2.25 to 10.5 fin/cm^2. This fin density for the optimized pin fin array, with a pin diameter of 0.9 cm and spacing of 1 cm, achieves a value of 0.65 fins/cm$^{2\cdot}$, which is unfortunately below the range of the Aihara *et al.* correlation. Thus, it must be considered that, in certain of the situations described above, the heat transfer coefficients calculated using the Aihara *et al.* correlation may not be within the stated $\pm 10\%$ accuracy range To aid the reader, the array heat transfer coefficients have been re-plotted as a function of fin density in Fig. 8 with the domain of the Aihara *et al.* correlation clearly delineated in each figure.

In Fig. 8 array heat transfer coefficient as high as 50 W/m^2-K is achievable for a fin density of 2.3 fins/cm^2 within the Aihara *et al.* range using pin fin diameter of 4 mm. The least material optimized geometry suggests that an h_{a} as high as ~ 63 W/m^2-K can be achieved using thicker pin fin of 9 mm at much lower pin fin density of 0.65 fins/cm^2 but well outside Aihara *et al.* range. These predicted values are subjected to the accuracy of used Aihara *et al.* correlation. It is to be expected that within the Aihara *et al.* data range the predictions will be more accurate than outside this domain. Nevertheless, in the absence of other natural convection pin fin correlations, the described optimization methodology provides the best initial design for an optimum fin array and can be expected to yield an accuracy of $\pm 10\%$ for configurations that fall within the Aihara fin density range.

The thin, widely spaced PPS pin fins seem to provide the highest mass based heat transfer coefficients in Fig. 9, with a value of 17 W/kg-K achieved using a 1 mm diameter fin at a fin density of 3 fins/cm^2. This is well within the Aihara range, but provides an array heat transfer coefficient of just 18 W/m^2-K.

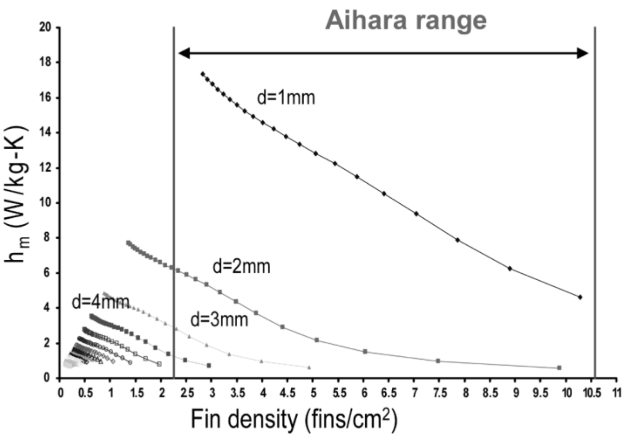

Fig. 9. Enhanced PPS h_{m} variation with fin density.

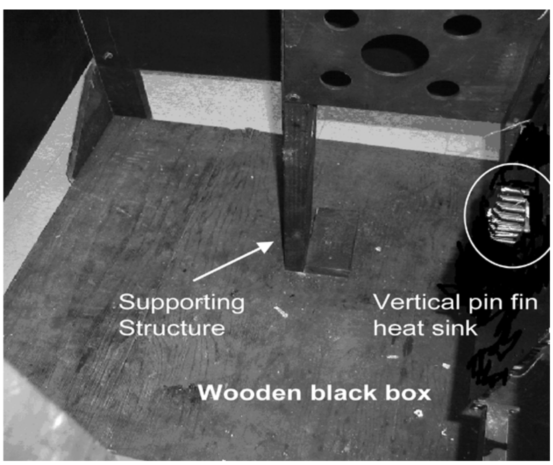

Fig. 10. Apparatus for natural convection heat sink experiments.

Least material optimized diameter of 9 mm and spacing 1 mm provides mass based heat transfer coefficient of $\sim$ 1 W/kg-K and array heat transfer coefficient as high as $\sim$ 64 W/m^2-K.

II. Experimental Study

A. Apparatus

Experimental verification of the thermal performance capability of the commercially available, thermally enhanced PPS polymer heat sinks [1] was performed. A 2 m $\times$ 2 m $\times$ 2 m wooden enclosure (Fig. 10) was used to isolate the experimental setup from the laboratory environment The inside walls of the wooden enclosure were painted black to establish a measured surface emissivity of 0.8 using infra red camera for the determination of the radiative heat transfer from the tested heat sinks. The room temperature ($\sim$25 °C) was used as a reference temperature. Commercial E type thermocouples, having an accuracy of $\pm$0.5 °C, were used with a copper thermocouple junction box kept at room temperature serving as a reference block. For data acquisition, a switching unit (Agilent 3499 A) and digital multimeter (Agilent 34 401 A) with Windows 2000-installed desktop computer were used. A Minco HR5590R11L12B thin film heater, having Nickel wire element of electrical resistance (11 ohms) with a silicone rubber covering, was used as the heat source in the experiments. An identical guard heater to minimize heat loss from the back of the heat sink was also used.

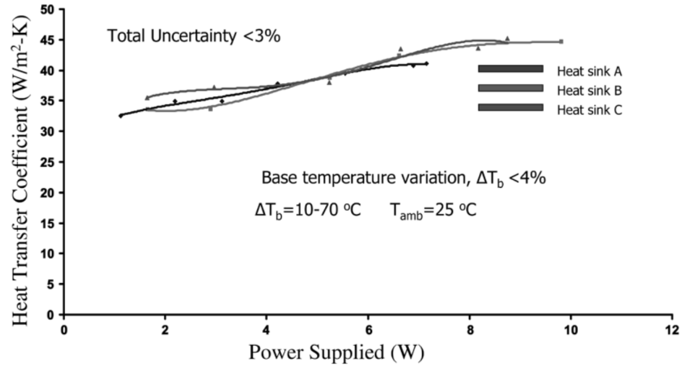

Fig. 11. Experimental h_{a} for enhanced PPS heat sink.

TABLE I
EXPERIMENTAL h_{a} FOR ENHANCED PPS HEAT SINK—VALUES AND UNCERTAINTY

Power (W)	h_a (W/m^2-K)	Deviation ($\pm$ W/m^2-K)	Bias error ($\pm$ W/m^2-K)	Total uncertainty ($\pm$ W/m^2-K)	% uncertainty
1.12	32.57	0.357	0.120	0.38	1.2
2.19	34.94	0.428	0.089	0.44	1.25
3.12	34.88	0.099	0.071	0.12	0.35
4.21	37.79	0.658	0.066	0.66	1.75
5.55	39.45	0.998	0.060	0.10	2.53
6.89	40.75	0.750	0.058	0.75	1.85
7.16	41.13	0.555	0.057	0.56	1.36

The resistance heaters covered about 51% of the heat sink base cross-sectional area.

The experimental plan was to test and verify the performance of the thermally conductive (k = 20 W/m-K) PPS heat sinks available from CoolPoly [1]. For this purpose, use was made of a heat sink configuration, which was relatively close to the optimum pin fin array configuration for base plate dimensions of 5.6 cm by 5.6 cm. The pin fin height is 18 mm and the diameter of this tapered fin is 0.398 cm at the base and 0.29 cm at the tip. It is to be noted that, unfortunately, at 1.39 fins/cm^2, this heat sink falls significantly below the parametric range of the Aihara *et al.* (7) correlation. Three distinct heat sinks of identical geometry, labeled A, B, C for identification and convenience, were individually tested inside the test chamber at different power levels ranging from 1.2–7.2 W.

B. Experimental Results

The individual heat sink results, expressed in terms of the array heat transfer coefficient, are shown and compared in Fig. 11. The heat transfer coefficient value as high as 41 W/m^2-K is obtained using the tested heat sink in natural and radiation heat transfer mode.

The overall average uncertainty for all experiments was calculated to be within 3% as shown in Table I. Fig. 12 shows the experimental convection-only heat transfer coefficient values obtained after removing the radiation contribution ($\sim$32%) obtained using numerical modeling (described in next section). The experimental results were found to follow the trend of the Aihara *et al.* correlation but to reach on an average only $\sim$95–70% of the analytically predicted values as shown in Fig. 12 as the base temperature rise above ambient (25 °C) increases from 10–70 °C. It is believed that this discrepancy is

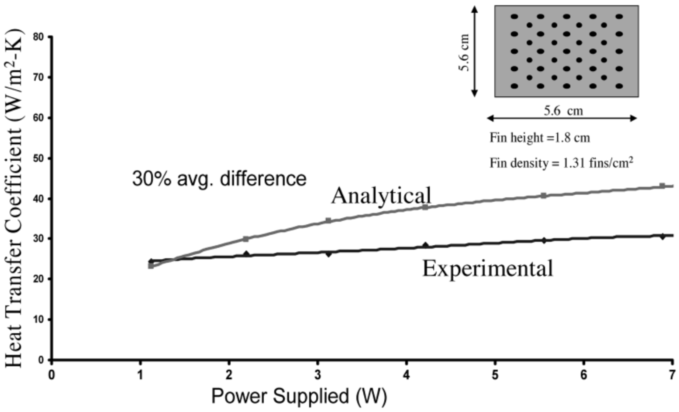

Fig. 12. Enhanced PPS h_a (convection only)—Experimental/Analytical.

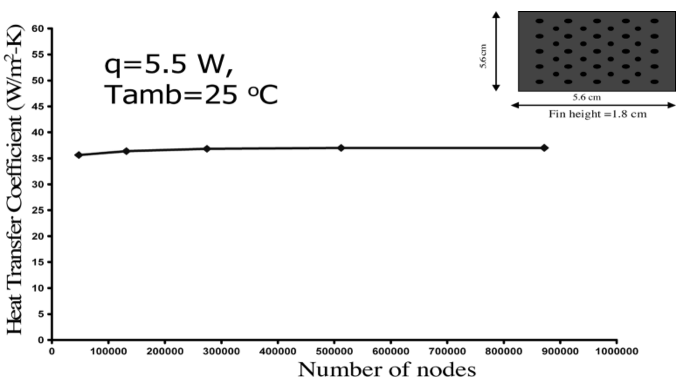

Fig. 13. Numerical convergence of h_a for enhanced PPS heat sink.

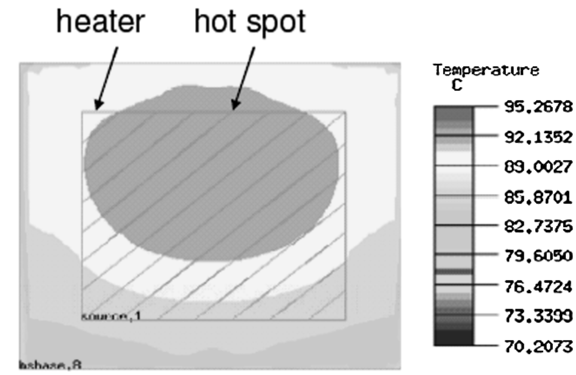

Fig. 14. Enhanced PPS heat sink base temperature distribution −5.5 W dissipation.

TABLE II
NUMERICAL h_a VALUES FOR ENHANCED PPS HEAT SINK

Power (W)	h_a (radiation and convection)	h_a (convection only)	% radiation loss
4.03	33.9	23.1	31.8
5.50	37	25.2	31.8
7.16	39.3	26.7	32.1
9.82	42.1	28.9	31.4

associated with the tested hest sink configuration falling well outside the fin density range of the Aihara *et al.* correlation described in (7).

III. NUMERICAL STUDY

The complete three-dimensional (3–D) geometry of the tested sample is generated, as shown in Fig. 14. The solution space was about ×5 times the heat sink size in order to capture the complete natural convection flow field. The identical heater size (Fig. 14) as used in experiments was modeled as a constant power source. An ambient temperature of 25 °C was used in the numerical calculations. A symmetry boundary condition was assumed on the back side of the heater, as was obtained in the experimental setup using guard heater. In the radiation calculations, the emissivity of the PPS polymer heat sink material was set equal to 0.8 as had been measured with an infrared camera in the laboratory.

The convergence criteria was set to be within 0.1% for the numerically obtained array heat transfer coefficient of the pin finned heat sink between two consecutive results during mesh refinement. The resulting plot in a combined convection and radiation mode is shown in Fig. 13 for an intermediate power value of 5.5 W. The plot indicates about 500 000 nodes are sufficient to obtain the desired accuracy.

The converged temperature distribution results are shown in Fig. 14. Symmetry along the vertical center line is observed in the base plate temperature distribution as expected. Since the heater occupies just 50% of the heat sink base, the temperature field displays a hot zone in the center. Due to the decreasing heat transfer coefficient in the vertical direction, as expected for buoyancy driven natural convection flow, this hot zone is shifted upwards from the geometric center of the base. The fins are hotter at the base side and cooler toward the tip end. Upper fins are hotter than bottom because of the rising hot air compared to incoming cool air at the bottom of the heat sink.

The magnitude of the air velocity is as high as 39.8 cm/s for 50 °C average excess temperature. The developed numerical models were solved with and without thermal radiation. As may be seen in Table II, based on the numerical simulation of the heat sink performance in the specified test chamber, thermal radiation contributed about 31–32% of the total heat transfer from the tested heat sink.

Comparison of the experimental values in Table I with the numerical results for the combined three-dimensional (3-D) convection and radiation heat transfer rates in Table II indicates that the CFD simulation provides very close agreement (4%) with the measured values.

Furthermore a complete 3-D CFD model was built in IcePak to verify the analytical modeling (convection only) results obtained using the Aihara *et al.* pin fin array correlation. Mesh refinement was performed to ensure a converged solution. As shown in Fig. 15, close agreement (analytical values just 1.2% higher than the numerical) between the CFD modeling results and the analytical modeling results was achieved for pin fin density of 8.62 fins/cm^2 which is within the Aihara correlation range (2.25–10.58 fins/cm^2).

The agreement of the CFD combined convection and radiation simulation with experimental results and the agreement of the analytical predictions with the CFD convection-only simulations serves to establish the accuracy of the analytical modeling and optimization methodology, when applied within the Aihara *et al.* domain. However for a PPS pin finned heat sinks with fin

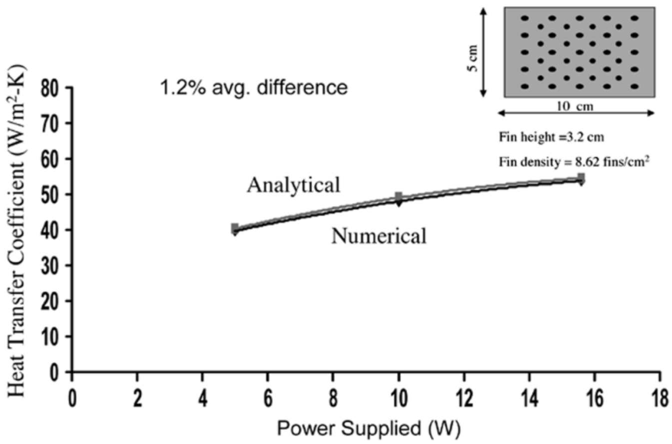

Fig. 15. Pin fin enhanced PPS heat sink h_{a} in Aihara domain.

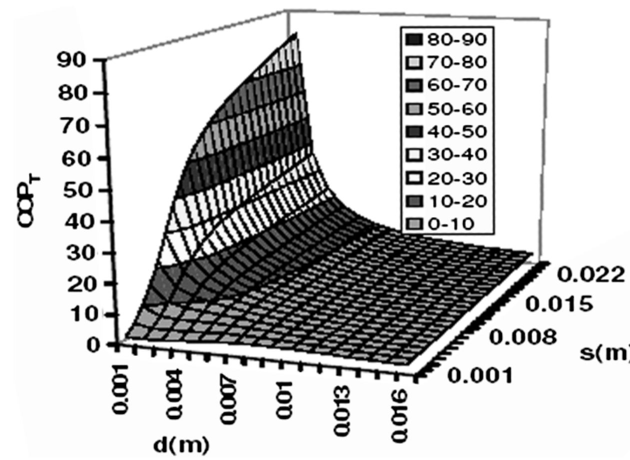

Fig. 16. $\mathrm{COP_T}$ for the enhanced PPS pin fin heat sink.

densities well below the Aihara *et al.* range, the analytical prediction may over predict the performance by as much as 30%. This was seen to be the case for a pin fin array with a fin density of 1.39 fins/cm^2.

IV. Energy Efficiency Analysis

A. Total Coefficient-of-Performance

Due to the large volume of electronic heat sink production in the world and the high-energy content of aluminum (estimated at 56 kW-hr/kg or 200 MJ/kg) [9], sustainability considerations may favor the use of the lower density and less energy intensive PPS polymers for this application. The energy efficiency analysis of PPS plastic heat sinks and comparison with conventional aluminum heat sinks can be carried out with the aid of a "total coefficient of performance" that relates the thermal energy that can be removed to the energy invested in the fabrication and operation of the specified heat sink, and defined as [12]

$$\mathrm{COP_T} = \frac{\mathrm{q_T t}}{\mathrm{E_T}}. \tag{16}$$

When applied to natural convection cooled heat sinks, E_{T} includes only the energy invested in the fabrication and delivery of the heat sink. This value is approximately $\sim$ 115 MJ/kg (32 kWh/kg) [8]), or about 58% the energy invested in the formation, extrusion, assembly and transport of aluminum heat sinks [9]. This fabrication energy includes the energy required to create the filler, at approximately 70% by weight, which is needed to create a high thermal conductivity (20 W/m-K) PPS polymer resin.

For an assumed 6000 hrs of operation (3 work years: 50 wk, 5 days/wk, 8 h/day), the $\mathrm{COP_T}$ for a PPS polymer heat sink can be related to the mass of the heat sink as

$$\mathrm{COP}_T = 0.188 \times \frac{q_T}{M}. \tag{17}$$

Similarly, the $\mathrm{COP_T}$ for an aluminum heat sink is given by

$$\mathrm{COP}_T = 0.108 \times \frac{q_T}{M}. \tag{18}$$

From Fig. 16 obtained using (17), the least material PPS polymer pin fin array has a maximum value of 82, some four times higher than the peak aluminum value of 21 obtained using (18). The highest observed $\mathrm{COP_T}$ pin fin array has the smallest considered pin diameter of 0.1 cm and air gap of 2.5 cm. Increase in the diameter and decrease in the spacing lead to lower $\mathrm{COP_T}$ for both the materials. However, for the maximum $\mathrm{COP_T}$ array, the array heat transfer coefficient is only 22 W/m^2-K (35% of optimum) for the PPS polymer and 41 W/m^2-K (21% of optimum) for the aluminum. For the least material optimum array design having pin fin diameter 0.9 cm and spacing 1 cm, the $\mathrm{COP_T}$ value for PPS polymer is approximately 4.5, or nearly three times greater than the identical aluminum array $\mathrm{COP_T}$ value of 1.6.

Since the invested energy in natural convection arrays depends exclusively on the required mass, the highest mass based heat transfer coefficient array will also be the highest $\mathrm{COP_T}$ design. As may be seen in Fig. 9, the highest values of the mass based heat transfer coefficients appear to lie within the fin density domain of the Aihara *et al.* [2] correlation. The mass based heat transfer coefficient vs. pin fin density plot for the optimum-height pin fins indicates that for smaller diameter and pin fin density the value is maximum and it decreases almost linearly with the increase in the pin fin density for a fixed diameter pin fin array (Fig. 9). As the pin fin spacing increases the pin fin density decreases for fixed diameter that results in increase of the mass based heat transfer coefficient (W/kg-K) and hence the $\mathrm{COP_T}$. The indicated highest mass based heat transfer coefficient values are within Aihara *et al.* limited $\pm$10% accuracy [14], [15].

V. Conclusion

The present study appears to provide the first reported systematic study of the thermal performance of an air-cooled heat sink fabricated of a relatively high thermal conductivity PPS polymer. The Aihara *et al.* correlation for a staggered pin fin array has been used for the design and optimization of such air-cooled, natural convection heat sinks. A PPS heat sink configuration in which the fin diameter was approximately equal to the optimum, center-to-center fin spacing was found to provide the highest natural convection thermal performance at every fin diameter. An optimum array, consisting of individual fins of the "least-material" height, placed at the optimum center-to-center distance was found to closely approximate the best achievable pin fin array performance. The PPS polymer heat sinks were found to provide comparable thermal performance to those achieved with aluminum heat sinks up to fin heights of approximately 5 cm, but yield $\mathrm{COP_T}$ value approximately three times greater than the identical aluminum array $\mathrm{COP_T}$ value for the least material optimum array design having pin fin diameter 0.9 cm and spacing 1 cm.

From the natural convection-radiation experimental results, the PPS polymer heat sink (20 W/m-K) provides array heat transfer coefficient as high as 41 W/m^2-K at a base of 5.6×5.6 cm at fin density of 1.39 fins/cm^2 having short fin height of 18 mm and the diameter of this tapered fin is 0.398 cm at the base and 0.29 cm at the tip. Results of the experimental testing and numerical thermofluid simulation of polyphenylene sulfide (PPS) heat sinks are within 4% agreement.

The optimization methodology followed herein appears to give accuracy within $\pm 10\%$ within Aihara *et al.* prescribed fin density range and seems to show as much as 30% over prediction for tested sparser pin fin heat sink. Therefore, it is recommended to use numerical or experimental verification outside Aihara *et al.* suggested fin density range of 2.25–10.58 fins/cm^2 in order to obtain more accurate cooling rate predictions.

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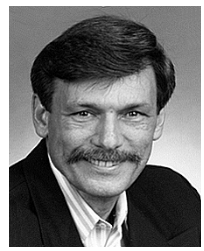

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