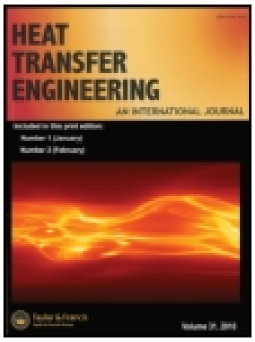

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An Assessment of Direct Chip Cooling Enhancement Using Pin Fins

JIANPING TU,[1] WALTER W. YUEN,[1] and YISHU GONG[2]

[1]Department of Mechanical Engineering, Hong Kong Polytechnic University, Hung Hom, Kowloon, Hong Kong
[2]An Hui Vocational College of Metallurgy and Technology, Maanshan City, An Hui, China

The use of pin fins to provide direct cooling of computer chip mounted on a printed circuit board (PCB) is studied experimentally. This concept is targeted toward improving the thermal design for the increasingly popular compact personal computer options such as the laptop notebook and web book, in which the use of a conventional heat sink might not be feasible or desirable due to the lack of space, as well as the light weight requirement. Experimental data show that the installation of a pin fin between a chip and the PCB can provide significant direct heat transfer enhancement. Data for pin heights of 4 mm, 7 mm, and 10 mm are presented. Results show that for typical chip size and power, pin fins can lead to an order of magnitude increase in the heat transfer coefficient compared to the flat plate forced convection results. While the existing heat transfer and pressure drop correlations yield good qualitative agreement with the data, the correlation tends to underpredict the heat transfer coefficient obtained in the present work.

INTRODUCTION

Current technologies on chip cooling are based mainly on soldering the chip on a high-conductivity heat sink and removing the heat by forced convection (e.g., air flow) in the back of the heat sink. As the chip power increases and forced convection is insufficient for the required heat dissipation, additional heat transfer is generally achieved by attaching heat transfer enhancement devices (e.g., heat pipe, thermoelectric cooler [TEC]) to the heat sink. While the heat pipe, TEC, and other heat transfer enhancement devices are useful to provide additional cooling for many computing equipments, they are not practical solutions for compact, portable, high-performance computing devices (e.g., notebook and web book) because of the general light weight requirement and the lack of space. For example, an external laptop (notebook) cooler is now commonly used to provide the needed additional cooling of notebook computers. While human comfort (the computer is "too hot" to touch) is an important motivational factor for the user's acceptance of a laptop (notebook) cooler, there are significant concerns about the durability and reliability of these equipment items when they are operating constantly at high temperature (313–323K [40–50°C] for current designs of notebooks and web books, and higher as these equipments become lighter, smaller, and thinner in future design). There is a need to reexamine the possibility of an integrated design in which direct cooling can be provided at the chip/printed circuit board (PCB) level to reduce its operating temperature.

A typical design with computer chips mounted on PCB is shown in Figure 1. It is interesting to note that the connecting pins are typically made from high-thermal-conductivity pure copper and coated with gold and planted inside the composite. They are thus already designed to provide a direct path for heat removal from the inside of the central processing unit (CPU). However, there has been no reported effort on further exploiting this geometric design (high-thermal-conductivity pin on a board) for direct heat transfer enhancement. The principal idea of the current work is to exploit this existing geometric arrangement (with the addition of more pins) to enhance thermal performance. In a design as shown in Figure 2, the present work proposes to add "dummy" pins (pins with no electrical function) to the existing connecting pins to form a pin array. Heat transfer enhancement can be achieved by providing air flow through the gap. The gap needs to be kept small (say, 1 cm or less) since this concept is expected to be utilized in compact, lightweight computing equipment (e.g., notebook or web book). However, even with the height restriction of less than 1 cm, some optimization

Address correspondence to Professor Walter W. Yuen, Department of Mechanical Engineering, The Hong Kong Polytechnic University, Hung Hom, Kowloon, Hong Kong. E-mail: ppyuen@polyu.edu.hk

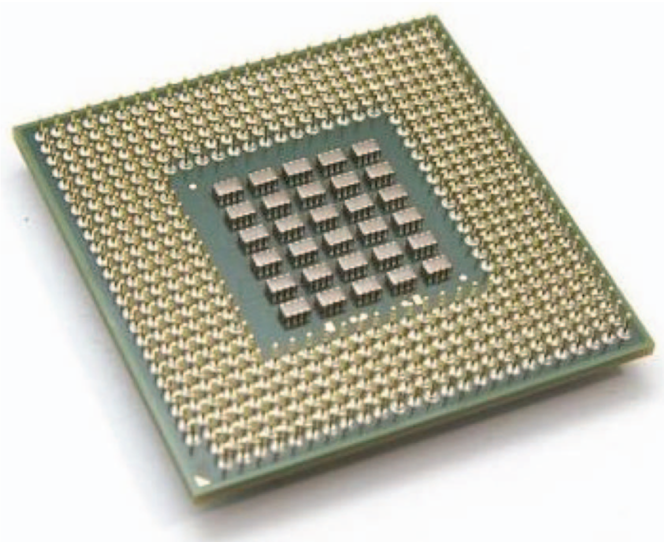


Figure 1 Typical computer chip design with connecting pins. (Color figure available online.)

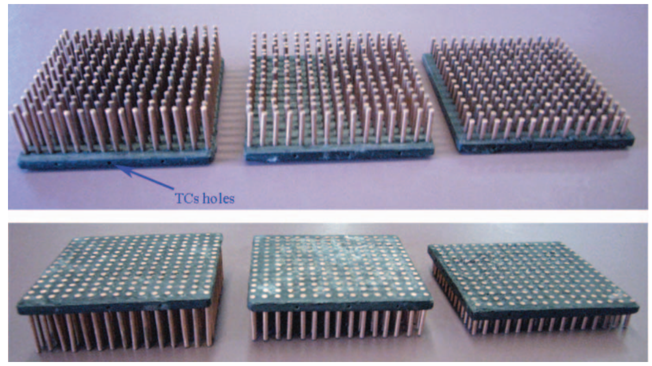


Figure 3 Three test articles. The height of pins are 10mm, 7mm, and 4mm, respectively. (Color figure available online.)

between heat transfer and pressure drop can be achieved with the pin height variation. Since the number of pins added is not large, there is also no "weight" issue associated with this design. The objective of this work is to provide some actual heat transfer data and to determine the heat transfer characteristics of this proposed pin structure design.

The idea of using pin fins as a heat sink has been studied extensively, particularly by researchers in the heat exchanger industries [1–6]. Data covering a broad range of design parameters (Reynolds number, pin diameter, pin density, gap height) have been presented together with empirical correlations [7, 8]. But none of the existing data have been generated at the geometric scale and dimension of the current chip/PCB design (e.g., gap of less than 1 cm, typical pin diameter of 1 mm or less). The current work thus provides the necessary experimental database for the relevant range of the design parameters, as well as an assessment of the applicability of the existing correlations [7, 8] for the proposed engineering design.

EXPERIMENTAL SETUP

To simulate thermal performance of a CPU, a composite was implanted wth a regular array of 1-mm pure copper pins as shown in Figure 3. The base surface (opposite side to the pin) was ground with 400-grade sandpaper to ensure that it was flat and smooth. A heat element was attached tightly to the base surface. The heater element is made from a Micon flex film heater and glued on a 3.81 cm × 3.81 cm × 0.635 cm (1.5 inches × 1.5 inches × 0.25 inches) pure copper block with high-thermal-conductivity Master bond. Thermal grease is employed between the copper heater element and composite test unit. The horizontal dimensions are chosen so that the heating area is on the scale of a computer chip. There are three holes on each side of the block (total 12 holes) for K-type thermocouples. The tops of the pins are then attached to a PCB board. A schematic of the complete test system is shown in Figure 4. A series of photographs showing the actual experimental set up is shown in Figure 5.

As shown in Figure 3, experiments were conducted with three test articles, with pin heights of 4 mm, 7 mm, and 10 mm, respectively. The pins were installed in a 15 × 15 array (the center to center distance between pin is thus 3.8 mm [0.1 inch]). Air at room temperature (∼20°C) was the coolant. As shown schematically in Figures 4a and b, the inlet pressure was set to a prescribed level by a regulator. A specific flow rate was then generated by the control of a needle valve. After passing through a particle and moisture filter and a flowmeter,

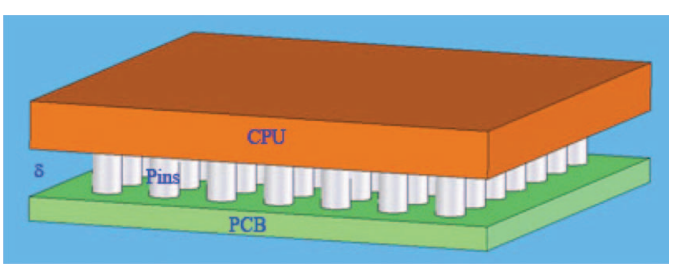


Figure 2 Schematic of the proposed CPU/PCB pin structure to enhance heat transfer. (Color figure available online.)

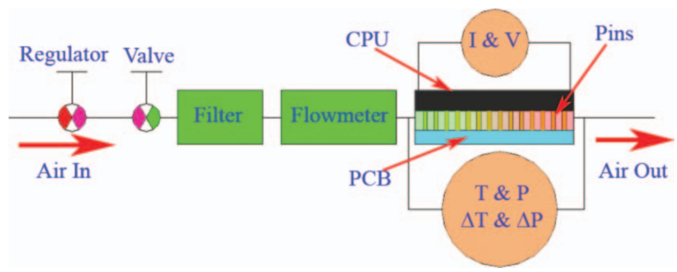
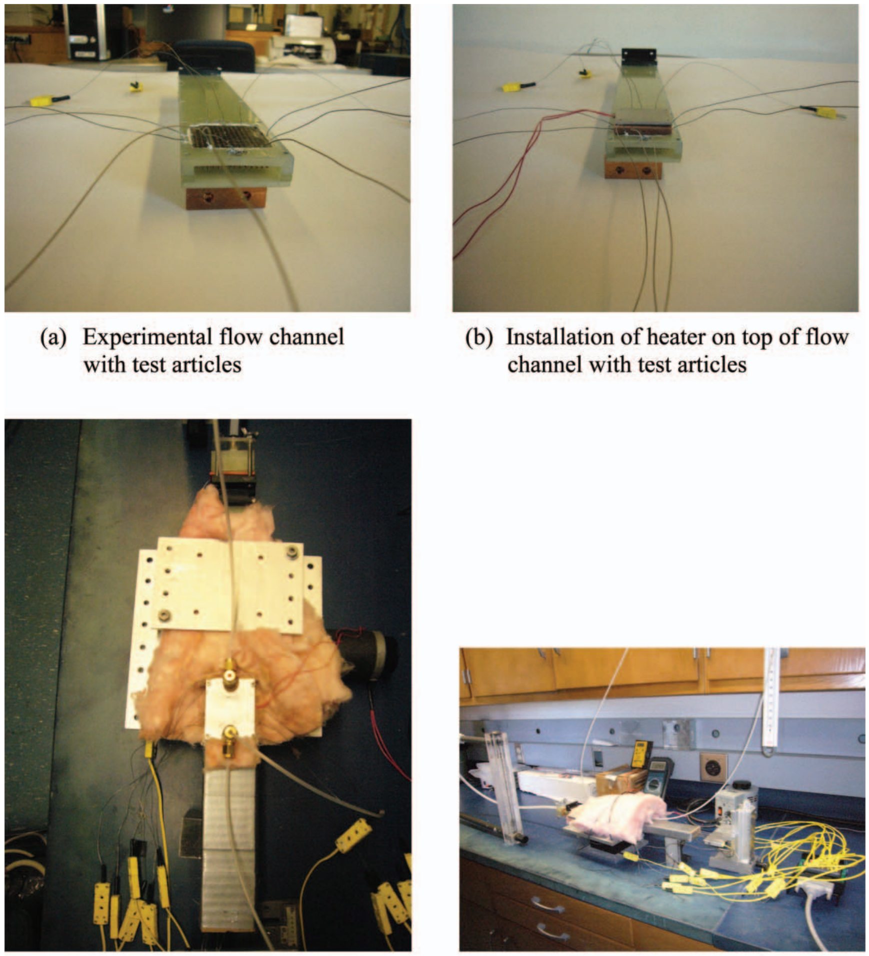


Figure 4 Thermal test system schematic. (Color figure available online.)



(a) Experimental flow channel with test articles

(b) Installation of heater on top of flow channel with test articles

(c) Wrapping of experimental set with ultra flexible fiberglass insulation

(d) View of complete experimental set

Figure 5 Selected photographs of experimental setup. (Color figure available online.)

the room-temperature air entered the test section. Voltage was directly loaded on the composite to generate power to simulate the CPU heating process. The accuracy of the instrumentations used in the experiment is summarized in Table 1. For all accepted runs, the overall energy balance between the energy provided by the heater and the measured heat loss was established to be within 10%.

Table 1 Accuracy of instrumentations used in the experiment

	Accuracy	Model	Supplier
Air flow meter	±2%	FL-3663C	Omega
Thermocouple K-type	±0.4%	KMQSS-020G-12 20 gauge K-type	Omega
Voltage and ampere meters	±0.15%	Fluke 175	Fluke
Pressure drop	Minor division of 2 mm water column	1221-M600-W/M	Dwyer
Pressure gauge	±0.5%	4007K21	McMaster

RESULTS

Data were generated over a range of air volumetric flow rate of 1000 to 4000 cm^3/s, which corresponds to an average flow velocity within the heat transfer area (with the pins installed) in the range of 10 to 40 cm/s, 7 to 20 cm/s, and 5 to 15 cm/s for the pin height of 4 mm, 7 mm, and 10 mm, respectively. The power provided for the heater was in the range of 10 to 30 W, corresponding to a heat flux range of 0.7 to 2.1 W/cm^2. These selected ranges of flow rate and power for the experiment are in the expected range of operation for a notebook and web book.

Pressure Drop

Pressure drop data are important because they indicate the power required to generate the enhanced heat transfer. In

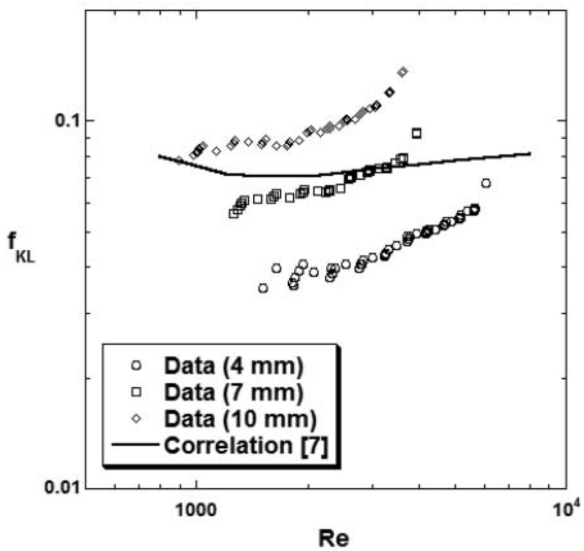


Figure 6 Comparison of the pressure drop data with the correlation developed by Kays and London [7].

Figure 6, the pressure drop data for the three pin heights are plotted in dimensionless form and compared with the correlation suggested by Kays and London [7]. For the presentation of data, the Reynolds number is defined by

$$\mathrm{Re} = \frac{U_m D_h}{\nu} \tag{1}$$

In the preceding expression, U_m is the characteristic velocity of coolant, given by

$$U_m = \frac{G}{A_c} \tag{2}$$

with G being the volumetric flow rate and A_c the cross-sectional flow area, which is defined by

$$A_c = (B - nd)\, H \tag{3}$$

where B and H are the width and height of the flow channel, n the number of pin in the horizontal direction facing the flow, and d the pin diameter. The hydraulic diameter D_h for the pin structure, based on a definition introduced by Kays and London [7], is

$$D_h = \frac{4A_c}{A_w} L \tag{4}$$

where A_w is the "wetted" area of the flow, given by

$$A_w = 2\,(B + H)\, L - 2n\,(n-1)\,\frac{\pi d^2}{4} + n\,(n-1)\,\pi d H \tag{5}$$

where L is the length of the heat transfer area. Note that one less pin is installed in the horizontal direction because of the presence of the side wall.

To account for the effect of properties variation due to the temperature increase (which can be significant for a coolant such as air) and the generally complex physics associated with the pressure loss in a pin structure, Kays and London [7] recommended the following expression for the dimensionless friction factor:

$$f_{KL} = \frac{A_c v_1}{A_w v_m}\left[\frac{2g\Delta p}{G^2 v_1} - \left(K_c + 1 - \sigma^2\right) - 2\left(\frac{v_2}{v_1} - 1\right) + (1 - \sigma^2 - K_e)\frac{v_2}{v_1}\right] \tag{6}$$

where v_1, v_2, and v_m are the specific volume of the coolant at the inlet temperature (T_1), outlet temperature (T_2), and the average temperature ($T_m = (T_1 + T_2)/2$), respectively. g is the gravitational constant, Δp is the pressure drop, and σ is an area ratio given by

$$\sigma = \frac{A_c}{A_f} \tag{7}$$

where A_f is the actual nonobstructed frontal flow area of the channel, given by

$$A_f = BH \tag{8}$$

In Eq. (6), two dimensionless parameters K_c and K_e are introduced to further account for the effect of Reynolds number (Re) on the friction factor. They are empirical expressions [7] given by

$$K_c = 2.3519 \times 10^{-8}\,\mathrm{Re}^2 - 1.9695 \times 10^{-4}\,\mathrm{Re} + 0.65565 \tag{9a}$$

$$K_e = -1.4854 \times 10^{-8}\,\mathrm{Re}^2 + 1.2403 \times 10^{-4}\,\mathrm{Re} - 0.18408 \tag{9b}$$

Results in Figure 6 show readily that while the single correlation proposed by Kays and London [7] is generally effective in predicting the range of the observed friction factor, it fails to account correctly for the effect of the flow (Re) and pin height (d) on the pressure drop. In general, the data show that the friction factor increases slightly with Re due to the increased flow velocity and the friction factor increases with pin height due to the increase fluid contact area. These behaviors are in agreement with physical expectation.

In a recent work [8], some data on pin fin heat sinks were generated and dimensionless correlations for both heat transfer and pressure drop are proposed. Even though the characteristic dimensions of the pin fin structure (pin height is 3 cm, heating area is 2–6 × 5 cm, pin diameter is 1 mm, and the distance between pin is 1 mm) and the geometric shape of the fin (square fin) are slightly different from those of the current work, this correlation gives another perspective on the relative accuracy of the current data. The comparison is shown in Figure 7. Note that the dimensionless variables used in the correlation in reference [8] are defined differently from those utilized by Kays and London [7] in Figure 6. Specifically, the friction factor used by Kim et al. [8] is given by

$$f_K = \frac{\Delta p}{N \rho U_m^2 / 2} \tag{10}$$

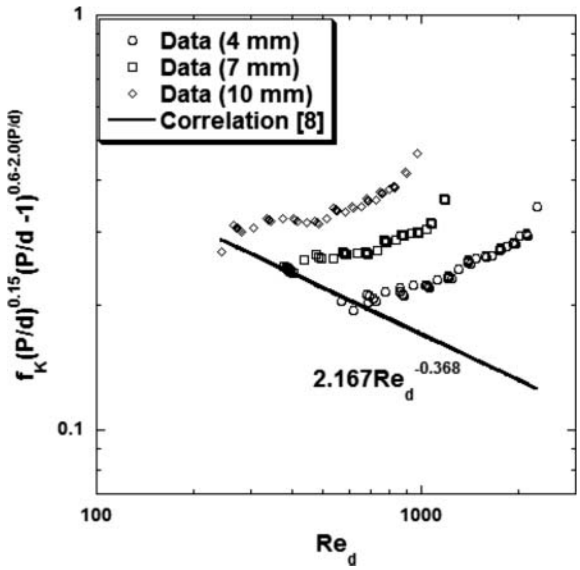


Figure 7 Comparison of the pressure drop data with the correlation developed by Kim et al. [8].

with N a geometric parameter defined as

$$N = \frac{L}{P}, \qquad P = d + W_c \tag{11}$$

where W_c is the center-to-center distance between pins. The Reynolds number is defined using pin width (dimension corresponds to diameter for the cylindrical pin) as the characteristic length, as

$$\mathrm{Re}_d = \frac{U_m d}{\nu} \tag{12}$$

The suggested friction factor correlation is [8]

$$f_K = 2.167\,\mathrm{Re}_d^{-0.368}\frac{(P/d-1)^{-0.6+2.0(P/d)}}{(P/d)^{0.15}} \tag{13}$$

with all properties evaluated at the free stream (inlet) conditions.

Results in Figure 7 show that while the current data agree with the Kim et al. [8] correlation in terms of the range of the friction factor, the data predict a different trend for the effect of the Reynolds number. The physical basis of this discrepancy is uncertain because of the lack of detail of the experimental data used as the basis for the correlation. For example, the correlation assumes that the flow is in the regime where the channel height (i.e., the pin height) does not have a strong effect on the friction factor and heat transfer. The current data, which are generated with smaller pin heights (4–10 mm, compared to 30 mm in ref. [6]), show a strong dependence on pin height. The effect of property variation is not accounted for in Eq. (13). Since the heating power for the data leading to the correlation is not known, the difference in the coolant temperature in the heating area can also lead to differences in the friction coefficient.

Heat Transfer

To determine the heat transfer coefficient from the measured data, the total heat transfer to the coolant is first estimated by

$$Q = \frac{G}{v_1}C_p\,(T_2 - T_1) \tag{14}$$

Accounting for the increase in heat transfer area by the fin, the heat transfer coefficient is determined by

$$h = \frac{Q}{A_b \eta_{0(c)}\,(T_w - T_m)} \tag{15}$$

where $A_b = BL$ is the total heating base area, T_w the heated wall temperature, and T_m the average coolant temperature in the heated region, given by

$$T_m = \frac{T_1 + T_2}{2} \tag{16}$$

The overall fin efficiency $\eta_{0(c)}$ is

$$\eta_{0(c)} = 1 - M\frac{\pi d^2}{4A_b}(1 - \eta_f) \tag{17}$$

where M is the total number of pins ($n(n-1)$ for the current arrangement) and η_f is the fin efficiency for a single fin, which, for the current geometry of a circular fin, is given by

$$\eta_f = \frac{\tanh(mH)}{mH} \tag{18}$$

with

$$m = \sqrt{\frac{4h}{k_p d}} \tag{19}$$

and k_p is the solid thermal conductivity of the fin. It is important to note that the overall heat transfer coefficient is a function of the fin efficiency, which is a function of the "local" heat transfer coefficient. In the present consideration, these two heat transfer coefficients are assumed to be approximately equal and its actual value is determined by iteration from Eq. (15). In terms of the Stanton number (St) and Prandtl number (Pr), which are defined by

$$St = \frac{h}{\rho G C_p} \tag{20}$$

$$\mathrm{Pr} = \frac{\nu}{\alpha} \tag{21}$$

the current data are presented in Figure 8, together with the correlation proposed by Kays and London [7].

In reference [8], the following heat transfer correlation is proposed:

$$Nu_d = \frac{0.142\,\mathrm{Re}_d^{0.561}}{(P/d)^{-0.65} + 0.35} \tag{22}$$

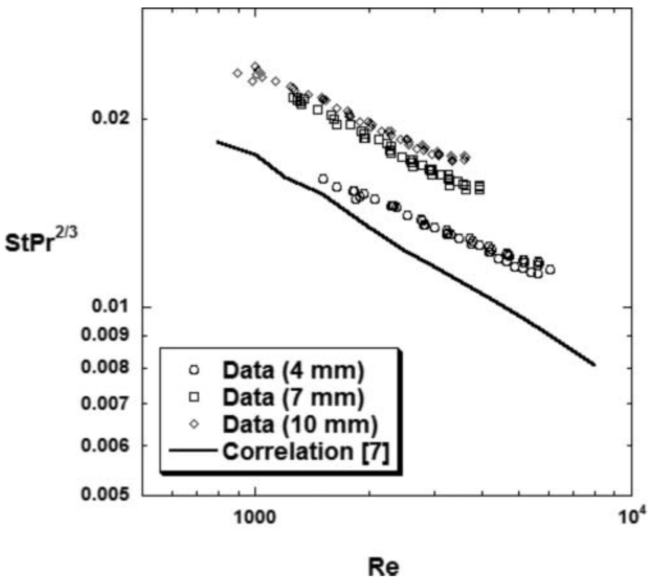


Figure 8 Comparison of the heat transfer data with correlation developed by Kays and London [7].

where Nu_d is defined by

$$Nu_d = \frac{hd}{k} \tag{23}$$

A comparison between the current data and Eq. (22) is presented in Figure 9.

Results in Figures 8 and 9 show that the current heat transfer data compare well with the correlations developed in references [7] and [8]. The magnitude of the dimensionless heat transfer is in the same range as predicted by the correlations. The qualitative behavior of the dimensionless heat transfer, as functions of Re and Re_d, is similar to those predicted by the two correlations. Pin height appears to have a relatively minor effect on the heat transfer. It is interesting to note that the two correlations underpredict the current heat transfer data. This suggests that for short pins, the heat transfer enhancement is higher than those predicted by the correlations. The two correlations are too "conservative" for the proposed application to compact high-performance computing devices.

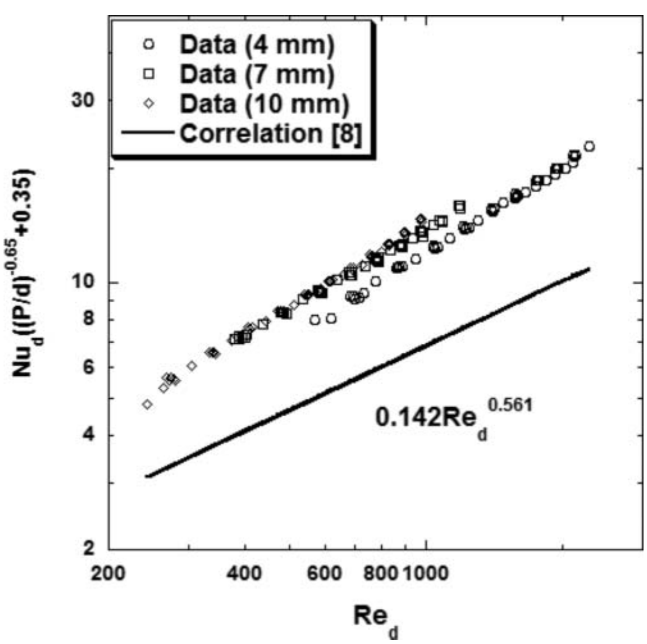


Figure 9 Comparison of the heat transfer data with correlation developed by Kim et al. [8].

Heat Transfer Enhancement

To provide a quantitative evaluation of the heat transfer enhancement by the installation of pin fins, a parallel set of experiments was conducted in the same setup with a heating plate without pin fins. Using a flat plate expression of the heat transfer coefficient given by

$$h = \frac{Q}{A_b\,(T_w - T_m)} \tag{24}$$

and a Reynolds number based on the channel height defined as

$$Re_H = \tfrac{U_m H}{\nu} \tag{25}$$

the two set of data are presented in Figure 10. In the electronic cooling industries, the parameter that is of interest to thermal designer is the thermal resistance, which is defined as

$$R = \frac{1}{hA_b} \tag{26}$$

A comparison of the two set of data, in terms of the thermal resistance, is shown in Figure 11. Results in Figures 10 and 11 show readily that the installation of pin fins in the chip/PCB gap

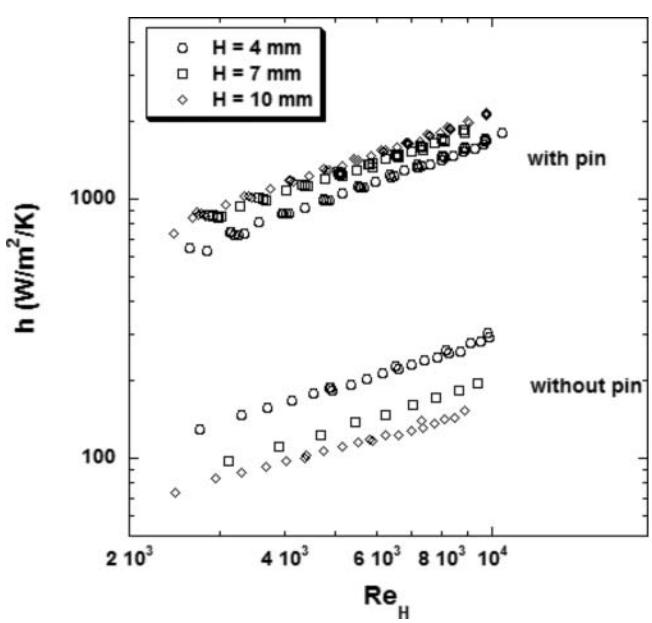


Figure 10 Comparison of the heat transfer coefficient generated with and without the pin fins.

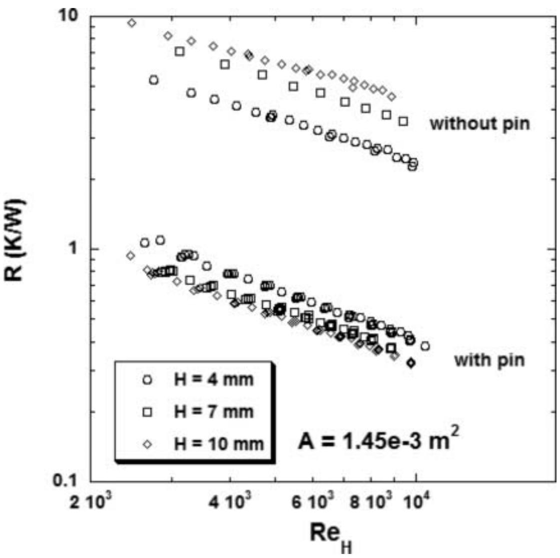


Figure 11 Comparison of the thermal resistance generated with and without the pin fins.

can lead to an increase of the heat transfer coefficient by a factor of 10 (or, similarly, a reduction in the thermal resistance by a factor of 10). This result demonstrates readily the effectiveness of using pin fins to enhance heat transfer directly from the CPU.

CONCLUSIONS

A concept to install a pin fin structure at the chip/PCB gap is introduced to provide direct heat transfer enhancement for the CPU. Experimental data are generated for various pin heights and power levels that are in the range of interest for the future generation of lightweight, compact computing equipments. Results show that the present data agree qualitatively with the existing correlations for heat transfer from pin fin structure. In general, the heat transfer enhancement by short fins (with fin height in the order of 4–10 mm) exceeds those predicted by the existing correlations. The installation of pin fins in the chip/PCB gap can lead to an increase of the heat transfer coefficient by a factor of 10. The approach to provide direct heat transfer enhancement from the CPU by installing pin fins in the chip/PCB board thus deserves serious consideration by thermal designer for compact high-performance computing devices.

NOMENCLATURE

A_b heated base area, Eq. (15) (m^2)
A_c cross sectional flow area, Eq. (3) (m^2)
A_f non-obstruct frontal flow area, Eq. (8) (m^2)
A_w wetted area, Eq. (5) (m^2)
B width of flow channel (m)
C_p specific heat of air (J/K-kg)
d pin diameter (m)
D_h hydraulic diameter, Eq. (1) (m)
f_K friction factor defined by Kim, et al. (8), Eq. (10)
f_{KL} friction factor defined by Kays and London (7), Eq. (6)
g gravitational constant (m^2/s)
G volumetric flow rate (m^3/s)
H height of flow channel (m)
I current (amp), Figure 4
h heat transfer coefficient (W/m-K)
k thermal conductivity (W-m/K)
K_c empirical constant, Eqn. (9a)
K_e empirical constant, Eqn. (9b)
k_p solid thermal conductivity of fin (W-m/K)
L length of the heating area (m)
M total number of fin
m dimensionless parameter, Eq. (19)
n number of pin in the horizontal direction
Nu_d Nuselt number, Eq. (23)
N dimensionless geometric parameter, Eq. (11)
p pressure (bar)
P pin distance parameter, Eq. (11) (m)
Pr Prandtl number, Eq. (21)
Q total heat transfer from the heated surface (W)
R thermal resistance, Eq. (26) (K/W)
Re Reynolds number used by Kays and London (7), Eq. (1)
Re_d Reynolds number using pin diameter as the characteristic length, Eq. (12)
Re_H Reynolds number using channel height as the characteristic length, Eq. (25)
St Stanton number, Eq. (20)
T temperature (K)
U_m mean velocity, Eq. (2) (m/s)
v specific volume (m^3/kg)
V voltage (volt), Figure 4
W_c center-to-center distance between pins (m)

Greek Symbols

α thermal diffusivity (m^2/s)
Δp pressure drop (bar/m)
ν kinematic viscosity (m^2/s)
η_f single fin efficiency, Eq. (18)
$\eta_{0(c)}$ overall fin efficiency, Eq. (17)
ρ density (kg/m^3)
σ area ratio, Eq. (7)

Subscripts

1 inlet condition
2 exit condition
m average condition between inlet and exit
w wall condition

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Jianping Tu is currently a research engineer at Allcomp, Inc, an engineering consulting company at the City of Industry, California. He was previously a reseach associate at the University of California at Santa Barbara, where much of the work of this paper was conducted. His research expertise is in the area of heat exchanger and heat transfer enhancement.

Walter W. Yuen is the Vice-President (Academic Development), as well as a Chair Professor at the Department of Mechanical Engineering and Building Service Engineering Department at the Hong Kong Polytechnic University. His research is in the area of thermal radiation heat transfer, two-phase flow, and electronic cooling. He was a professor of mechanical engineering at the University California at Santa Barbara (UCSB) until 2009. Much of the research reported in this paper was done while he was at UCSB.

Yishu Gong is a professor at the An Hui Vocational College of Meteallurgy and Technology at Maanshan City, An Hui, China. He conducted the reseach for this paper while he was a postdoctoral student at the University of California at Santa Barbara in 2009. His research interest is in metallurgy and heat transfer enhancement.