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Heat Transfer Enhancement with Perforated Pin Fins Subject to Impinging Flow

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Abstract: The rapid growth in compact multi-functional high speed miniaturized electronic devices demand highly stringent thermal management. The present work experimentally investigates the use of staggered perforated pin fins to enhance the rate of heat transfer while subject to a vertical impinging flow. In particular, the effect of vertical perforation diameters on each pin is studied. Results show that Nusselt number for pin fins with 5 number of horizontal perforation of 3mm vertical and horizontal perforation diameter is about 46% higher than the solid pins at Re_{Dh}=3.5×10^4. Further increasing the vertical perforation diameters leads to a significant drop in thermal dissipation. This is due to the reduction in axial heat conduction along the pins. More importantly, the pressure drop with vertical perforation diameter of 2mm, 3mm, 4mm and 5mm has recorded a reduction by 3%, 14%, 17% and 35%, respectively, compared with that in solid pins. Overall, perforated pin fin array is able to enhance thermal dissipation at a lower pressure drop.

Keywords: Forced Convection; Impinging Flow; Perforation; Pin Fins; Heat Sink

1 INTRODUCTION

The main cause of failure in electronics devices is due to the overheating of the respective high speed, multi-functional, and miniaturized components. Effective thermal management plays an important role to efficiently dissipate the unwanted thermal which ensure the reliability of the electronic components. In the recent years, the design of heat sink device is strongly dependent upon the need to balance thermal dissipation and pressure drop across the system such that the overall cost and efficiency may be optimized. An example of a familiar solution is to apply pin fin array onto a heat sink design.

Soodphakdee *et. al*. numerically investigated the rate of heat transfer for inline and staggered pin fins of different cross-sectional geometries [1]. They reported that staggered pins always outperform the inline arrangement at higher pressure coefficient. Their results supported the earlier experimental study carried out by Sparrow *et. al*. [2]. Tahat *et. al*. [3] optimized the lengthwise and spanwise arrangements of staggered and in-line pins. Moreover, at low Reynolds number, higher heat transfer coefficient is obtained with elliptical pin fins. However, at high Reynolds number circular pin fins is more effective. Yang *et. al*. [4] studied numerically the forced convective heat transfer in staggered aluminum porous pin fin arrays. They found higher heat transfer rate in porous pin fins than in solid pins. Wake formation behind pin is minimal leeward of porous pin consequently minimizing pressure losses across the heat sink. Their results supported the earlier experimental study carried out by Sahin and Demir [5]. Seyf and Layeghi numerically reported that metal foam inserted around the pins can redirect the flow behind pins, thus reduces wake formation [6]. The pressure coefficient for staggered pin fin array can be reduced effectively by either introducing perforated pin-fins or pin-fin-dimple arrays [7, 8]. Recently, Yong *et. al*. [8] experimentally and numerically studied the effects of number of horizontal perforation and perforation diameter on pin fin array. They reported that increasing the number of perforation is more important than increasing the perforation diameter.

Shankar *et. al*. [9] numerically investigated the effect of flow type on the performance of multi-material plate fin heat Sink. Their found that the temperature distributions for heat sink subject to impinging flow is lower than flooded flow. Kim *et. al*. [10] studied experimentally the heat transfer performance of plate fin and pin fin heat sinks when subject to vertical impinging flow. Their results showed that at small pumping power, the heat sink thermal resistance is lower for pin fins compared with plate fins. The inlet geometry of confined impinging flow on a smooth channel is found to be sensitive to heat transfer, where the inlet channel width and nozzle-to-plate spacing decrease with increasing Nusselt number and pressure drop [11]. Yang and Peng [12] numerically investigated the impinging heat transfer performance of heat sink with un-uniform fin width. They found that the effect of fin dimensions on forced convective

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heat transfer is more obvious at high Reynolds number. Foo *et. al*. numerically investigated the effect of number of perforation, horizontal and vertical perforation diameter on pin fin array subject to vertical impinging flow [13]. They reported that pin fins thermal dissipation is enhanced with vertical perforation. Nusselt number is the maximum for pins with 5 number of horizontal perforation (N) with 3mm in both horizontal (D_{hr}) and vertical perforation diameter (D_v).

The present paper focuses on the experimental study of forced convective heat transfer in staggered perforated pin fins array subject to single vertical impinging flow. The effect of vertical perforation diameter on the rate of thermal dissipation and pressure drop is investigated and compared with those for the solid pins.

2 EXPERIMENTAL METHODS

This study investigates the steady-state forced convective heat transfer in perforated aluminum pin fin array subject to vertical impinging flow. Figure 1 shows the schematic drawing of the experimental setup. The vertical inlet internal channel dimensions are 60×100×60mm in length, width and height, respectively. The horizontal internal channel is 500×113×65mm. The surrounding walls are mild steel of 3mm thick with the thermal conductivity of 50.2 $Wm^{-1}K^{-1}$. Air is supplied by a blower (ebmpapst, Germany) with a controllable flow velocity (maximum $12ms^{-1}$). The staggered pin fin arrays with and without perforations are installed in the test section that surrounded by acrylic walls. T-type thermocouples (Design Technology, US) are used to measure the thermofoil heated aluminum wall temperature. A 60W AC thermofoil heater (Minco, US) with stabilized heat flux is attached directly beneath an aluminum plate to mimic the electronic component thermal heating. The inlet and outlet flow velocities are recorded with hot wire anemometer (Lutron, TW). Pressure drop across the test section is measured using inclined well-type manometer (High Wycombe Bucks, UK). The inlet air temperature is measured with the hot wire anemometer and a T-type thermocouple records the outlet temperature.

Five sets of staggered pin arrays have been employed, namely, pin fins with 5 evenly spaced horizontal perforations each of D_{hr}=3mm, with vertical perforation diameter of (i) D_v=2mm [Fig.2(a)], (ii) D_v=3mm [Fig.2(b)], (iii) D_v=4mm [Fig.2(c)], and (iv) D_v=5mm [Fig.2(d)]. Thus, throughout the experiments the number and diameter of horizontal perforations are fixed. The control solid pin fin array is shown in Figure 2(e). In all cases, the pin fin array contains 14 pins of 8mm diameter, 50mm height with pitch of lengthwise and spanwise of 25mm.

Thermofoil supported electrical energy (E) is lost through convection heat loss, conduction heat loss, and radiation heat loss. The radiation energy only contributes to 0.5% of the total heat losses, due to the small temperature difference between the pin fin array and the ambient; 7.0% is contributed by the conduction heat losses to the surrounded channel [14]. The remaining thermal energy is transferred to the pin fin heat sink and being dissipated through forced convection,

$$q'' = \frac{0.925E}{A} \quad (1)$$

where A is the base plate surface area in contact with the thermofoil heater.

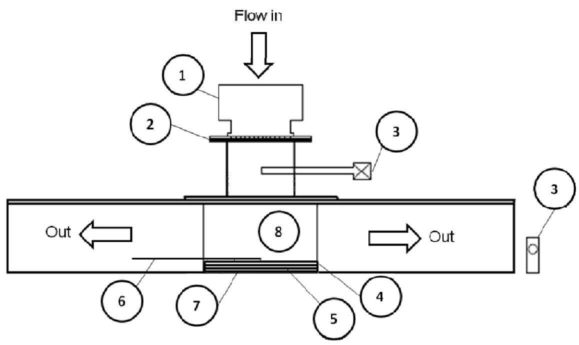

Figure 1: Schematic drawing of the experimental setup: 1. Blower, 2. air straightener, 3. hot wire anemometer, 4. aluminum heating plate, 5. thermofoil, 6. thermocouples, 7. Acrylic plate, and 8. test section

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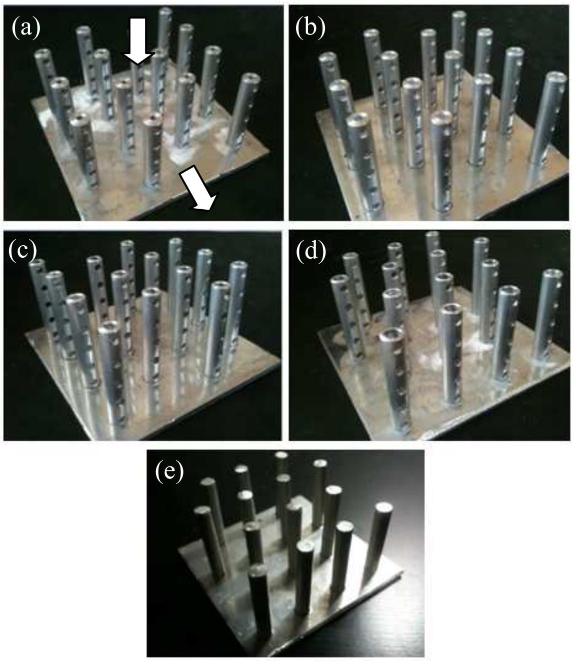

Figure 2: 5N, D_{hr}=3mm pin fin arrays with different vertical perforation diameters (D_v) - (a) D_v=2mm, (b) 3mm, (c) 4mm, (d) 5mm, and (e) control solid pin fin array; arrows in (a) represent the direction of impinging flow

The Nusselt number, $\overline{Nu}$, convective heat transfer coefficient, $\bar{h}$, thermal efficiency, η, pressure drop, ΔP, and Reynolds Number, Re_{Dh} are defined as follow,

$$\overline{Nu} = \frac{q'' D_h}{k_{air}\left(\overline{T_w} - \frac{T_{out} + T_{in}}{2}\right)} \tag{2}$$

$$\bar{h} = \frac{\overline{Nu} \cdot k_{air}}{D_h} \tag{3}$$

$$\eta = \frac{\overline{Nu}}{\Delta P} \tag{4}$$

$$\Delta P = P_{in} - P_{out} \tag{5}$$

$$Re_{Dh} = \frac{\rho u_o D_h}{\mu} \tag{6}$$

where $\overline{T_w}$ is the average base plate temperature, T_{in} the inlet temperature, T_{out} the outlet temperature, k_{air} the thermal conductivity of air, D_h the hydraulic diameter of the inlet rectangular channel, ΔP the pressure drop between inlet P_{in} and outlet pressure P_{out}, u_o the inlet velocity, ρ density of air, and μ the viscosity of air. It is important to note that all the experiments were repeated three times with maximum standard deviation of ±1^{o}C.

3 RESULTS AND DISCUSSION

3.1 Heat Transfer Performance

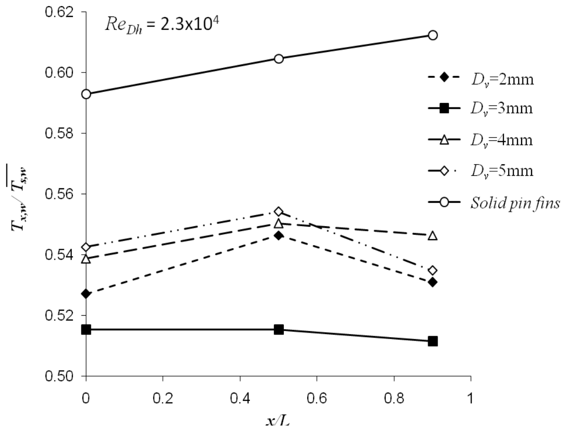

Figure 3: Local temperature distributions at different vertical perforation diameters with 5N and D_{hr}=3mm at Re_{Dh}=2.3×10^4

The present study focuses at the forced convective heat transfer enhancement with vertically and horizontally perforated pin fins subject to a vertical impinging flow. Solid pins and smooth channel are the controls to justify the thermal dissipative performance of the perforated pin fin arrays. The measured base plate surface temperature distributions at Re_{Dh}=2.3×10^4, *N=5* and *D_{hr}=3mm* are shown in Figure 3. All the temperatures are non-dimensionalised with the average smooth channel surface temperature $\overline{T_{s,w}}$. Results show that all the pin fins base plate surface temperatures are lower than the smooth channel and most importantly the solid pins. Minimum temperature is obtained for pin fin arrays with D_v=3mm. The temperature differences between the pin fins of 3mm vertical perforation diameter with the smooth channel and solid pins are about 42^{o}C and 8^{o}C, respectively. Interestingly, further increasing the pin fins vertical perforation diameter the surface temperature increases.

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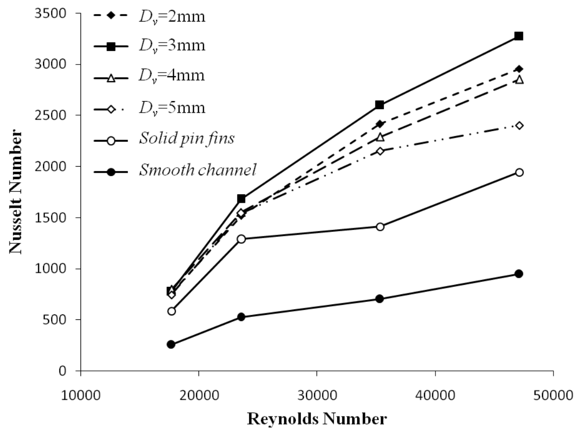

Figure 4: $\overline{Nu}$ *vs.* Re_{Dh} *– The effect of different vertical perforation diameters on pins*

Table 1: Empirical correlation equations of different pin fin arrays

D_v mm	Range of Velocity, ms^{-1}	Correlation Equation
2	3-8	$\overline{Nu}=0.121Re_{Dh}^{0.949}$
3	3-8	$\overline{Nu}=0.188Re_{Dh}^{0.899}$
4	3-8	$\overline{Nu}=0.426Re_{Dh}^{0.819}$
5	3-8	$\overline{Nu}=1.913Re_{Dh}^{0.666}$
Solid Pins	3-8	$\overline{Nu}=1.662Re_{Dh}^{0.653}$

Figure 4 shows the heat transfer performance of the different vertical perforation diameters with N=5 and D_{hr}=3mm at different Reynolds numbers. Clearly, Nusselt number increases with increasing Reynolds number. The solid and perforated pin fin heat sinks provide much higher thermal dissipation compared with the smooth channel. More importantly, thermal dissipation is higher with perforated pin fins than with solid pins where the dead flow zone leeward each pin can be effectively reduced [13]. It is found that increasing the vertical perforation diameter from 2mm to 3mm increases Nusselt number, but further increasing it to 4mm reduces Nusselt number. Thus, when the vertical perforation diameter is greater than 3mm, Nusselt number decreases. This is due to the decrease in the cross sectional area of the pin for thermal conduction along the pins. By coupling the effects of horizontal and vertical perforations, the heat transfer performance of the pins may be optimized. Maximum heat transfer is obtained for pin fins with N=5, D_{hr}=3mm and D_v=3mm. At Re_{Dh}= 3.5×10^4 the Nusselt number is 46% higher than that with solid pins. It is important to note that the current experimental observation is consistent with the previous numerical investigation reported by Foo *et. al*. [13].

Experimental results obtained for Nusselt number against Reynolds number of different pin fin configurations are fit into a nondimensional correlation equation,

$$\overline{Nu} = C\,Re^{m} \quad (7)$$

The empirical values of C and m are summarized in Table 1.

3.2 Pressure Drop

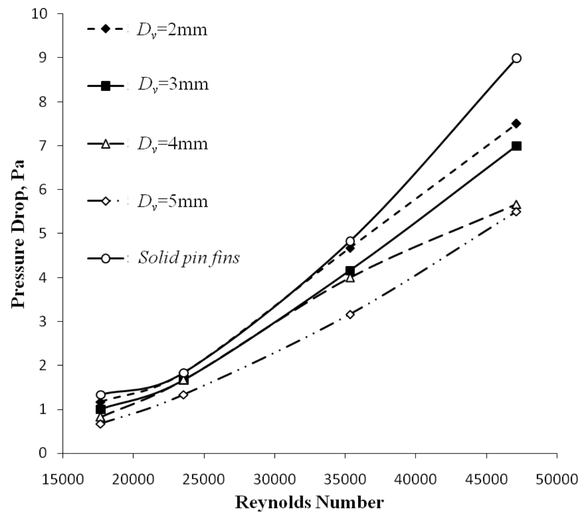

Figure 5: The effect of different vertical perforation diameters on pressure drop

The effect of the vertical perforation diameter on the measured pressure drop as a function of Reynolds number is shown in Figure 5. Pin fin arrays pressure drop increases significantly with increasing Reynolds number. It is found that pressure drop for solid pin fin array is the highest. At a given Reynolds number, pressure drop decreases with the increasing vertical perforation diameter. At Re_{Dh}=3.5×10^4, pin fins with 5 perforations of D_{hr}=3mm with different vertical perforation diameters of 2mm, 3mm, 4mm, and 5mm have recorded a reduction in pressure drop by 3%, 14%, 17% and 35%, respectively, compared to the solid pins. Solid pins present a larger blockage than perforated pins. As a result, it is evident that smaller pressure drop can be achieved using perforated pin fins.

3.3 Thermal Efficiency

Thermal efficiency (Pa^{-1}) is the ratio between Nusselt number and Pressure drop. It is the relative cost (pressure drop, hence pumping power) to achieve a certain rate of heat transfer. Figure 6 shows the dependence of thermal efficiency on different pin fin configurations against Reynolds number. Thermal efficiency decreases with the increase of Reynolds number. This effect is due to the increasing of pressure drop with increasing Reynolds number. As shown in Figure 6 perforated pin fins are always more efficient than solid pins. Although pin fin array with 5*N*, D_{hr}=3mm, and D_v=3mm possesses the optimum heat transfer performance, due the fact that the pressure drop for pins with D_v=5mm is lower than the former, the later outperform the rest in thermal efficiency at $Re_{Dh}<4.2\times10^4$. For Reynolds number greater than 4.2×10^4, the upstream fluid flow kinetic energy is higher enough to allow the flow to bend smoothly into the smaller perforation diameters of D_v=4 and 3mm, thus the thermal efficiency increases. Overall, Figure 6 suggests that by implementing vertical perforation onto pin fin heat sink design, this may maximize the rate of heat transfer at a minimal cost.

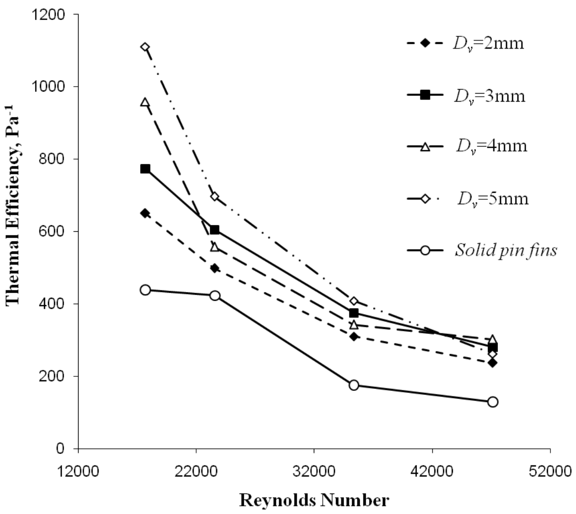

Figure 6: Dependence of thermal efficiency on vertical perforation diameter

4 CONCLUSION

Steady-state forced convective heat transfer enhancement of staggered and perforated pin fins subjected to an impinging flow has been studied experimentally. In particular, the effect of vertical perforation diameter along each pin is investigated. The conclusions of this study are:

1. Maximum Nusselt number is obtained for pin fin arrays with 5 perforations, 3mm of horizontal and vertical perforation diameter. This configuration is about 46% higher than the solid pins at $Re_{Dh}=3.5\times10^4$.

2. Significant pressure drop is achieved with perforated pin fins compared with the traditional solid pins. Pressure drop across the heat sink is smaller with increasing number of vertical perforation diameter.

3. Nusselt number increases with increasing vertical perforation diameter. Further increasing the perforation diameter will lead to a reduction in thermal dissipation. This is due to the decrease in heat conduction along the perforated pins. As a result, care should be taken while designing perforated pins in order to balance between the heat transfer performance and pressure drop.

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REFERENCES

[1] Soodphakdee, D., Behnia, M., and Copeland, D.W. 2001. A comparison of fin geometries for heatsinks in laminar forced convection: Part I - round, elliptical, and plate fins in staggered and in-line configurations. *The International Journal of Microcircuits and Electronic Packaging* 24, pp. 68-76.

[2] Sparrow, E.M., and Ramsey, J.W., and Altermani, C.A.C. 1980. Experiments on in-line pin-fins arrays and performance comparison with staggered arrays. *Journal of Heat Transfer – Transactions of the ASME* 102, pp. 44-50.

[3] Tahat, M., Kodah, Z.H., Jarrah, B.A., and Probert, S.D. 2000. Heat transfer from pin fin arrays experiencing forced convection. *Applied Energy* 67, pp. 419-442.

[4] Yang, J., Zeng, M., Wang, Q., and Nakayama, A. 2010. Forced convection heat transfer enhancement by porous pin fins in rectangular channels. *Journal of Heat Transfer – Transactions of the ASME* 132, pp. 051702-1-051702-8.

[5] Sahin, B. and Demir A. 2008. Thermal performance analysis and optimum design parameters of heat exchanger having perforated pin fins. *Energy conversion and management* 49, pp. 1684-1695.

Int. J. of Mechanical Computational and Manufacturing Research, Vol. 1. No. 2, (2012), 56-61, ISSN: 2301-4148

[6] Seyf, H.R. and Layeghi, M. 2010, Numerical analysis of convective heat transfer from an elliptic pin fin heat sink with and without metal foam insert. *Journal of Heat Transfer – Transactions of the ASME* 132, pp. 071401-1-071401-9.

[7] Rao, Y., Wan, C., Xu, Y., and Zang, Z.Z. 2011. Spatially-resolved heat transfer characteristics in channels with pin fin and pin fin dimple arrays. *International Journal of Thermal Sciences* 50, pp. 2277-2289.

[8] Yong, K.K.T., Lai, Y.L., Foo, J.J., and Chin S.B. 2011. Forced convective heat transfer enhancement with perforated pin fins. *Proceeding of the 2nd International Conference on Mechanical and Manufacturing Engineering*, Malaysia.

[9] Shankar, N.V.S., Rahul, D., VeerlaSrinivas, B.P., Vamsi, K., and Rao, M.M. 2012. Flow simulation to study the effect of flow type on the performance of multi-material plate fin heat Sink. *International Journal of Engineering Science & Advanced Technology* 2, pp. 233-240.

[10] Kim, D.K., Kim, S.J., and Bae, J.J. 2009. Comparison of thermal performance of plate-fin and pin-fin heat sinks subject to an impinging flow. *International Journal of Heat and Mass Transfer* 52, pp. 3510-3517.

[11] Lou, Z.Q., Mujumdar, A.S., and Yap, C. 2005. Effects of geometric parameters on confined impinging jet heat transfer. *Applied Thermal Engineering* 25, pp. 2687-2697.

[12] Yang, Y.T. and Peng, H.S. 2009. Numerical study of the heat sink with un-uniform fin width designs *International Journal of Heat and Mass Transfer* 52, pp. 3473-3480.

[13] Foo, J.J., Pui, S.Y., Lai, Y.L., and Chin, S.B. 2012. Forced convective heat transfer enhancement with perforated pin fins subject to an impinging flow. *SEGi Review* 5, pp. 34-45.

[14] Sara, O.N., Pekdemir, T., Yapici, S., and Yilmaz, M. 2001. Heat transfer enhancement in a channel flow with perforated rectangular blocks. *International Journal of Heat and Fluid Flow* 22, pp. 509-518.