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# Heat Transfer from Pin-Fin Heat Sinks under Multiple Impinging Jets

Hani A. El-Sheikh and Suresh V. Garimella

**Abstract**—The enhancement of heat transfer from a discrete heat source using multiple jet impingement of air in a confined arrangement was experimentally investigated. A variety of pin-fin heat sinks were mounted on the heat source and the resulting enhancement studied. Average heat transfer coefficients are presented for a range of jet Reynolds numbers ( $2000 < \text{Re} < 23000$ ). Two jet-to-jet spacings were investigated and the results were compared to single jets of both the same orifice diameter and orifice area. A total fin effectiveness was computed for the pinned heat sinks relative to the unpinned ones, and was in the range of 3 to 6; the highest value was obtained for the largest nozzle diameter. Results for the average heat transfer coefficient were correlated in terms of Reynolds number, fluid properties and geometric parameters of the heat sinks and the orifice plate.

**Index Terms**—Air jets, confined jets, electronics cooling, enhancement, heat sinks, heat transfer, jet impingement, multiple jets, pin fins.

## I. NOMENCLATURE

$A_{base}$	Heat sink base area (footprint), $\text{m}^2$ .
$A_{chip}$	Surface area of simulated chip, $\text{m}^2$ .
$A_d$	Area of orifice ( $\pi d^2/4$ ), $\text{m}^2$ .
$A_{HS}$	Total exposed surface area of heat sink (base and pins), $\text{m}^2$ .
$C$	Clearance above fin tips, m.
$C_p$	Specific heat of air, $\text{J/kg}\cdot\text{K}$ .
$d$	Orifice diameter, m.
$D_e$	Equivalent diameter of square heat sink base, m.
$d_p$	Pin fin diameter, m.
$H$	Nozzle-to-target spacing measured from heat sink base ( $H_p + C$ ), m.
$H_p$	Pin height, m.
$h_{base}$	Heat transfer coefficient based on $A_{base}$ , $\text{W/m}^2\cdot\text{C}$ .
$h_{HS}$	Heat transfer coefficient based on $A_{HS}$ , $\text{W/m}^2\cdot\text{C}$ .
$k$	Thermal conductivity of air, $\text{W/m}\cdot\text{C}$ .
$l$	Nozzle plate thickness, m.
$Nu_{base}$	Nusselt number ( $h_{base}d/k$ ).
$Nu_{HS}$	Nusselt number ( $h_{HS}d/k$ ).
$\text{Pr}$	Prandtl number ( $\mu C_p/k$ ).
$Q_{loss}$	Heat losses, $W$ .
$Q_{tot}$	Total heat input to heater assembly, $W$ .

$R_{cond}$	Conduction/constriction thermal resistance, $^{\circ}\text{C}/W$ .
$R$	Convective thermal resistance, ( $\equiv R_{bj}$ ), $^{\circ}\text{C}/W$ .
$R_{int}$	Thermal resistance of interface material, $^{\circ}\text{C}/W$ .
$\text{Re}$	Reynolds number ( $U_j d/\nu$ ).
$S$	Spacing between centers of adjacent jets, m.
$T_{base}$	Area-averaged temperature of heat sink base, $^{\circ}\text{C}$ .
$T_{jet}$	Jet exit temperature, $^{\circ}\text{C}$ .
$T_{chip}$	Temperature of heat source surface, $^{\circ}\text{C}$ .
$T_{riser}$	Temperature at the bottom of the riser, $^{\circ}\text{C}$ .
$T_{\infty}$	Ambient temperature, $^{\circ}\text{C}$ .
$\mu$	Dynamic viscosity of air, $\text{N}\cdot\text{s}/\text{m}^2$ .

## II. INTRODUCTION

IN ADDITION to high heat removal rates, electronic components also demand temperature uniformity along the surface. Air jet impingement, especially using multiple jets in conjunction with surface enhancement, is an attractive option since heat-removal levels similar to liquid cooling may be achieved by this means. In a previous study [1], the enhancement of heat transfer from a high-heat-flux component was studied using pin-fin heat sinks under a single impinging jet of air issuing from nozzles of different diameters. The present study investigates the enhancement of heat transfer using multiple air jets. As in [1], the radial spread of air after impingement is confined in a channel bounded by the target surface and the nozzle plate, representative of the channels formed by stacks of circuit boards carrying electronic components. The orifices are square-edged and short. The use of multiple jets is expected to provide greater options for trade-offs between air flow rate, pumping power and cooling rates, in addition to allowing greater temperature uniformity in the heat sink.

A review of the limited studies in the literature that have considered confined jet impingement is available in [1], and is not repeated here. The heat transfer coefficient under a single jet impinging on the target surface has a bell-shaped distribution with a maximum at the stagnation point, followed by a decrease with radial distance from the jet centerline. Secondary maxima have been reported at a radius of approximately  $1.9d$  for circular jets in several studies (e.g., [2]–[4]). This secondary peak is generally understood to coincide with the location where the boundary layer becomes turbulent in the wall-jet region [5].

When multiple jets are used, the local distribution of heat transfer coefficients changes depending on the number and spacing of jets in addition to the nozzle-to-target spacing, jet Reynolds number, and spent fluid exhaust. Huber and Viskanta [6] studied heat transfer in multiple air jet impingement on smooth surfaces. They showed that single jets resulted in lower

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local heat transfer coefficients than multiple jets, and attributed this degradation in heat transfer to jet interactions prior to impingement (in the free-jet region); inter-jet interactions became less pronounced as the nozzle-to-target spacing was decreased. A decrease in jet-to-jet spacing from  $S/d = 8$  to 4 was found to result in an increase in the average heat transfer coefficient; in fact,  $S/d = 4$  has been suggested as an optimum spacing in several studies [7]. Schroeder and Garimella [8] studied heat transfer to multiple air jets impinging on a smooth surface using three different nozzle arrays. The multiple jets were found to have higher local heat transfer coefficients than the corresponding single jets of the same diameter; Nusselt numbers averaged over a fixed heat-source area were 20% to 100% higher for the multiple jets at the same Reynolds number and nozzle-to-target spacing. Behbahani and Goldstein [9] studied local heat transfer from a flat plate to arrays of impinging circular air jets with two jet-to-jet spacings of 4 and 8. They showed that the smaller jet-to-jet spacing resulted in higher average heat transfer coefficients.

Although studies of jet impingement combined with surface modifications have shown good potential for obtaining significant enhancements in heat transfer, limited information is available in the literature on such configurations, especially with multiple jets. Copeland [10] studied multiple jet impingement on square-pin fin arrays using a perfluorinated dielectric liquid (FC-72). The effect of different heat sinks and nozzle configurations on heat transfer was investigated and the performance of multiple jets compared to single jets impinging on a smooth surface. The surface enhancement increased single-phase heat transfer by as much as 4.5 times. The study also reported that a reduction in total surface area of the heat sinks of 50% achieved by shortening the pins while maintaining the same aspect ratio (pin height-to-width ratio) decreased heat transfer coefficients by only 4%. Hansen and Webb [11] considered single-nozzle air jet impingement on six different types of modified surfaces including pin fin arrays of three different pin heights. They obtained enhancements in heat transfer relative to a smooth surface of 1.5 to 4.5 times. The tallest pin fins showed only slightly higher performance than the intermediate-height pins. Variation of nozzle-to-target spacing was found to have little effect on heat transfer in the range of  $H/d$  from 1 to 5. Heat transfer under single jets of FC-77 and water was studied with a variety of surface enhancements in a number of related studies [12]–[14] as discussed in El-Sheikh and Garimella [1].

The work presented here is part of an ongoing research program which addresses extending the limits of air cooling in the thermal management of high-heat-flux electronic components. The heat transfer from discrete (unenhanced) heat sources to single and multiple confined air jets [3], [8] and an optimization of the dimensions and layout of single and multiple nozzles to maximize the performance of impingement on extended surfaces [15] have been previously reported as part of this program. The present study is a continuation of previous work [1] in which the performance of a variety of pin-fin heat sinks was evaluated in conjunction with single air jet impingement. Two arrays of multiple nozzles are studied here and the extent of heat transfer enhancement obtained is quantified and compared to single jets. Predictive correlations are proposed for the average

heat transfer coefficients achieved in the multiple jet arrangement.

### III. EXPERIMENTS

The experiments for the present work were conducted using the same test facility as described in Schroeder and Garimella [3], [8] with modifications as laid out in El-Sheikh and Garimella [1].

Air at the required flow rate is delivered to a cylindrical plenum where the flow is conditioned before issuing from orifices machined in an orifice plate. A pressure tap located on the plenum wall is used to measure pressure drop across the orifice, while a thermocouple measures the air temperature just upstream of the orifice. The jet impinges on to a target plate held at a distance  $H$  away. Two arrays of multiple nozzles were investigated in the experiments with a nozzle diameter of 12.7 mm and two jet-to-jet spacings  $S/d$  of 2 and 3. The inter-jet spacing of 3 nozzle diameters was the maximum possible in these experiments while still maintaining the impingement to be on the heat sinks; a larger spacing would cause the jets to partially strike the area beyond the heat sinks. The orifices are square-edged with a thickness of 3.175 mm.

Details of construction of the  $20 \times 20$  mm square heat source simulating a heat-generating electronic component are described in [1]. The heat source is mounted flush with a target plate, and its surface temperature is determined as the arithmetic average of the readings from five 36-gage T-type thermocouples located just underneath the heat source surface.

Heat sinks are mounted on top of the heat source using screws (clamping force  $\approx 45$  lb), with a thermally conductive pad (T-gon 805) used as the interface material. Six copper pin-fin heat sinks, custom-made by PinFin, Inc., were studied in the experiments, as listed in Table I. The heat sink consists of three parts: riser, base, and pin fins, as shown in Fig. 1. The riser, which is  $20 \times 20$  mm square in cross section and 9.5 mm high, is brazed to the underside of the square heat sink base, and serves merely as a conduit for heat. Two sizes of heat sink footprint were considered: Small ( $A_{base} = 50.8 \times 50.8$  mm<sup>2</sup>) and Large ( $A_{base} = 76.2 \times 76.2$  mm<sup>2</sup>). With the small heat sinks, only one jet array ( $S/d = 2$ ) could be tested while for the large heat sinks, both jet arrays ( $S/d = 2$  and 3) were considered. The thickness of the heat sink base was held at 3.17 mm for all heat sinks. Pin fins of diameter  $d_p = 1.6$  mm were studied at pin heights of  $H_p = 25.4$  and 12.7 mm. Two unpinned heat sinks (smooth base) were studied to provide baseline results.

The riser temperature was measured at the center by a thermocouple positioned 1.52 mm from the bottom. The heat sink base temperature was measured at four different positions which are equally-spaced along the diagonal as shown in Fig. 1. The average heat sink base temperature,  $T_{base}$ , was obtained as an area-weighted average. The average heat transfer coefficient for the heat sinks was calculated according to

$$h_{base} = \frac{Q_{tot} - Q_{loss}}{A_{base}(T_{base} - T_{jet})} \quad (1)$$

in which  $Q_{loss}$  was experimentally determined to be  $Q_{loss} = 0.0645(T_{chip} - T_{\infty})$  as in [1]. An alternative definition of av-

TABLE I  
DIFFERENT HEAT SINKS USED IN THE EXPERIMENTS

	Description	Base footprint $A_{base}$ (mm x mm)	Effective Surface area <sup>§</sup> , $A_{HS}$ (cm <sup>2</sup> )	Pin height, $H_p$ (mm)	Pin diameter, $d_p$ (mm)
Large pinned heat sinks (76.2x76.2 mm <sup>2</sup> footprint)	Short	76.2x76.2	371	12.7	1.6
	Tall	76.2x76.2	747	25.4	1.6
Small pinned heat sinks (50.8x50.8 mm <sup>2</sup> footprint)	Short	50.8x50.8	179	12.7	1.6
	Tall	50.8x50.8	332	25.4	1.6 <sup>§§</sup>
Unpinned heat sinks (both footprints)	Large	76.2x76.2	58	0	-
	Small	50.8x50.8	26	0	-

<sup>§</sup> Includes surface area of pins and remaining footprint area of the base; i.e., side and bottom surface of the base not included.

<sup>§§</sup> A finer pin diameter,  $d_p = 0.96$  mm was also tested, results for which are included in the correlations in this paper.

verage heat transfer coefficient based on the total exposed heat sink area,  $A_{HS}$  (including the surface area of the pins and the base), is also used in the results presented in this paper:

$$h_{HS} = \frac{Q_{tot} - Q_{loss}}{A_{HS}(T_{base} - T_{jet})} \quad (2)$$

Thermal resistances (°C/W) may be calculated from these definitions using:

$$R = \frac{T_{base} - T_{jet}}{Q_{tot} - Q_{loss}} \quad (3)$$

It may be noted that the heat transfer coefficient in this study is always defined based on the temperature difference between the base of the heat sink and the jet. Thus, although the resistance of the interface material ( $R_{int}$ ) is determined (from measured values of  $T_{chip}$  and  $T_{riser}$ ), it is not factored into the discussion of results in this study. Uncertainties in the heat transfer coefficient ranged from 3.4% to 10.2% with the maximum value resulting for the large heat sink at the highest flow rate. The largest contribution to uncertainty (89%) comes from uncertainties in temperature measurement.

#### IV. RESULTS AND DISCUSSION

Average heat transfer coefficients are presented in this section for the two multiple-jet arrays considered. Multiple jets are compared to single jets of the same nozzle diameter ( $d = 12.7$  mm); comparisons are also made based on a fixed total orifice area ( $4 \times 12.7$  mm array compared to single 25.4 mm jet). The heat transfer coefficients obtained with the pinned heat sinks are

compared to the unpinned cases and the results are presented in the form of fin effectiveness values. The effect of changing jet-to-jet spacing on the heat transfer coefficient is explored. Correlations for heat transfer coefficients are then proposed for all heat sinks considered in this study. In view of the small effect of  $H/d$  on heat transfer [1], [10], [16] for the range of Reynolds number ( $2000 \leq Re \leq 23\,000$ ) and nozzle diameters considered, a single clearance height between the fin tips and the nozzle plate,  $C = 12.7$  mm, was used in experiments reported in this work.

The effect of jet-to-jet spacing ( $S$ ) on the heat transfer coefficient for the large heat sinks (open symbols in all figures) is shown in Fig. 2 as a function of Reynolds number. It is to be noted that the heat transfer coefficients in this and other graphs in the paper are presented on a logarithmic scale. For the unpinned heat sink, decreasing the jet-to-jet spacing from  $S/d = 3$  to 2 decreases the heat transfer coefficient by  $\sim 10\%$  at all the flow rates tested.

The effect of  $S/d$  on the heat transfer coefficient is reversed when the pinned heat sinks are considered. Fig. 2 indicates that both short and tall ( $H_p = 12.7$  and 25.4 mm) pinned heat sinks experience an increase in heat transfer coefficient of  $\sim 10\%$  when the jet-to-jet spacing decreases from  $S/d = 3$  to 2. In view of the unique design of the heat sinks used in this study, the heat source (and the riser) beneath the heat sink is significantly smaller in area ( $20 \times 20$  mm) than the heat sink base. The spreading resistance ( $R_{cond}$  in Fig. 1) results in a hot region at the center of the heat sink surrounded by a region which participates less effectively in heat transfer [1]. A smaller value of  $S/d$  concentrates the jets in the region of greater heat sink

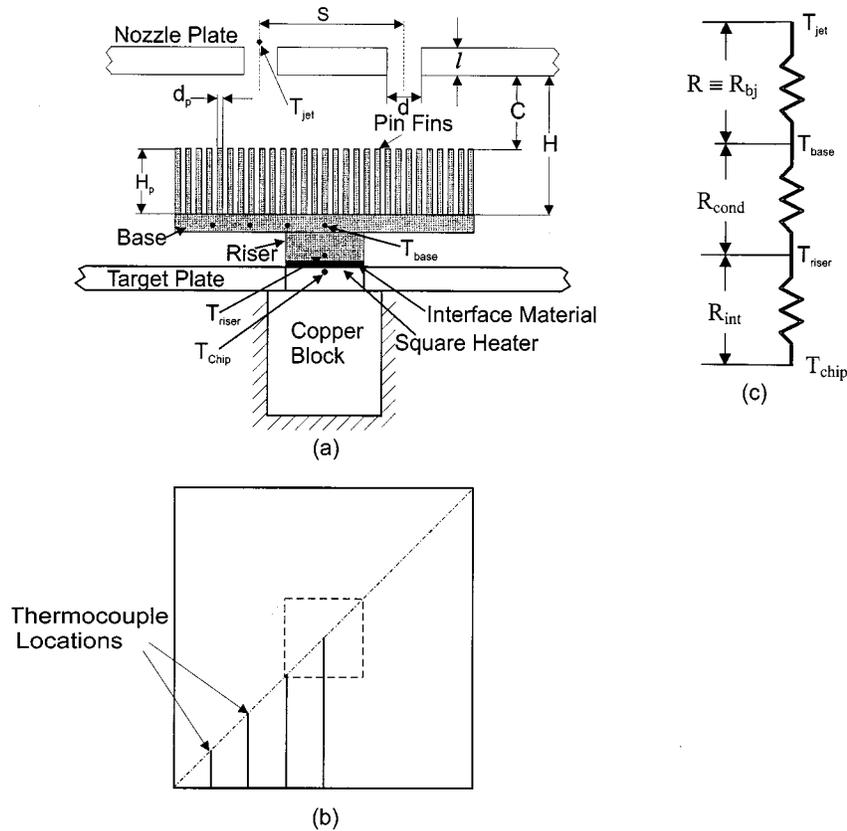


Fig. 1. Experimental details: (a) the pinned heat sink showing geometric parameters; (b) top view of the heat sink base showing thermocouple locations; (c) thermal resistance diagram.

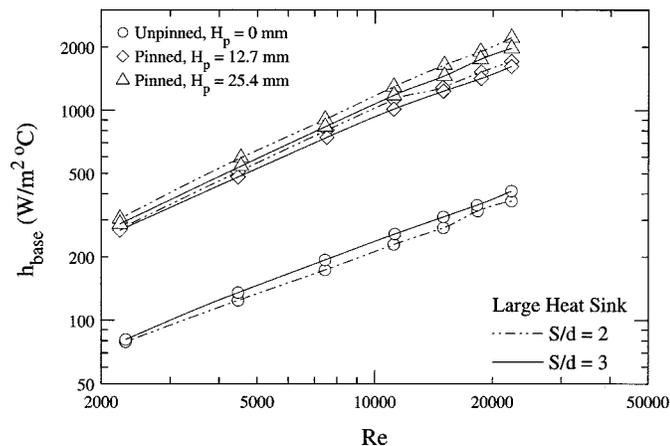


Fig. 2. Effect of inter-jet spacing on the heat transfer coefficient as a function of Reynolds number for the large heat sinks.

effectiveness. The presence of the pins also alters the flow field and may contribute to the different behavior of pinned heat sinks compared to smooth surfaces (unpinned) with respect to the influence of  $S/d$ .

A comparison between multiple jets ( $4 \times 12.7$  mm) and a single jet of the same nozzle diameter ( $d = 12.7$  mm) [1] is shown in Fig. 3, in terms of the variation of average heat transfer coefficients with Reynolds number for the unpinned and pinned small heat sinks (solid symbols in all figures) at the two different pin heights.

In all cases, the  $4 \times 12.7$  mm array yields higher heat transfer coefficients than the single jet at a given Reynolds number as expected. These increases in heat transfer coefficients (at fixed  $Re$ ) with multiple nozzles are obtained at the expense of air flow rates that are four times higher than for the single nozzle.

It is also seen from Fig. 3 that the difference between heat transfer coefficients from multiple and single jets is more pronounced for the taller pins than for the shorter ones. For instance, at  $Re = 18500$ , the short-pinned heat sink [Fig. 3(b)] shows an increase of 78% with the multiple jets, while the tall-pinned heat sink [Fig. 3(c)] shows an increase of 130%. In fact the heat transfer coefficient with single jets for the tall and short pins is quite comparable, whereas with the multiple jets, the taller pins result in a 30% higher  $h_{base}$  than the short pins. According to the mechanisms proposed in [1], increasing the pin height with a single jet adds heat transfer area primarily in a zone where air is stagnant (zone C [1]). However, with multiple jets the air flow is more uniformly distributed over the heat sink and thus a stagnant zone C no longer exists. Increasing the pin height under these conditions is thus more effective in enhancing heat transfer.

Similar results were also obtained for the pinned and unpinned large heat sinks, and for multiple jets with  $S/d = 2$  and 3. Higher heat transfer coefficients are also achieved when multiple jets are used for the large heat sinks, with the difference in heat transfer coefficients between single and multiple jets being more pronounced for the tall pins than for the short pins.

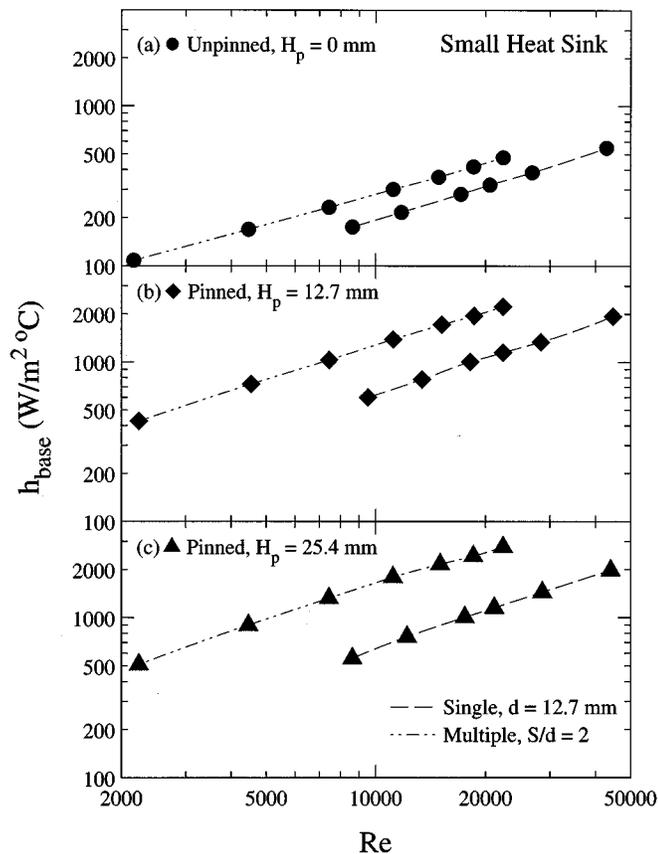


Fig. 3. Variation of heat transfer coefficient with Reynolds number for multiple and single jets with the same nozzle diameter ( $d = 12.7$  mm), for the small heat sinks.

In the results presented thus far, heat transfer coefficients are discussed as a function of Reynolds number. However, in practical applications, it is the flow rate of air which often determines design choices. A comparison of the performance of single and multiple jets based on total air flow rate is presented in Fig. 4 for the small heat sinks. Similar results (not shown) were obtained for the large heat sinks. The single jet is seen to result in higher heat transfer coefficients than multiple jets of the same diameter for all heat sinks considered. For the unpinned small heat sink [Fig. 4(a)], the single jet results in heat transfer coefficients that are 70% higher than those with multiple jets. However, the difference in heat transfer coefficient between single and multiple jets decreases to 40% and 10% for the short and tall pins, respectively, with an increase in pin height. It must be remembered that this superior performance of single jets compared to multiple jets at the same total flow rate comes at the expense of a significantly higher pressure drop.

As for the small heat sinks, the single jet outperforms the multiple jets in the case of the large heat sinks as well, with the difference being most pronounced for the unpinned surface (50%). For the tall pins, the difference between single and multiple jets almost completely vanishes in this case.

The measured pressure drop values for the single and multiple jets at all the flow rates tested are shown in Table II. The larger single nozzle has the lower pressure drops at a given flow rate as expected. It may also be seen from the table that the pres-

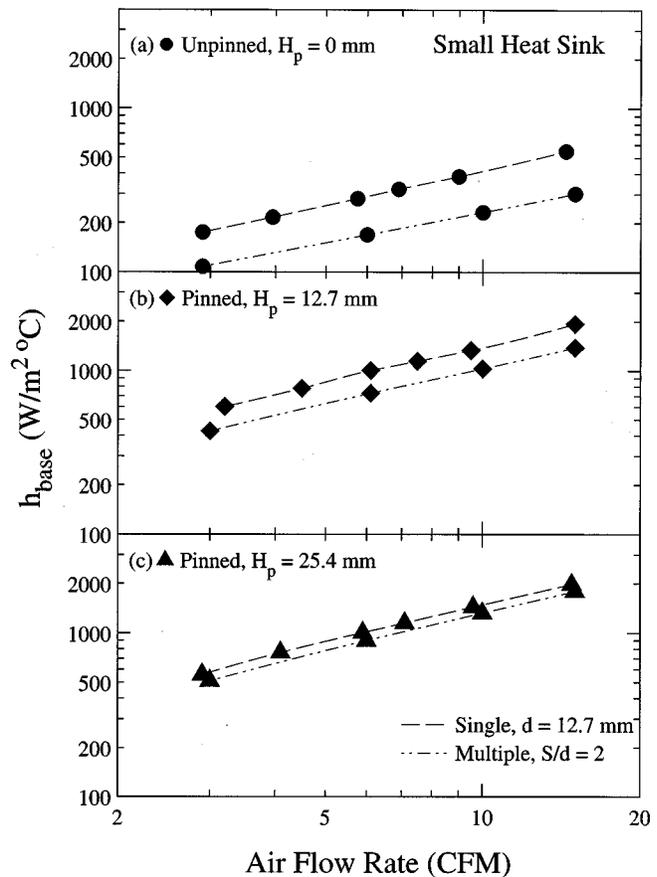


Fig. 4. Variation of heat transfer coefficient with air flow rate for multiple and single jets with the same nozzle diameter ( $d = 12.7$  mm), for the small heat sinks.

sure drop values are comparable for single and multiple nozzles having the same total orifice area (open area).

Fig. 5 shows the variation of the heat transfer coefficient with air flow rate for single ( $d = 25.4$  mm) and multiple ( $4 \times 12.7$  mm) nozzles having the same total orifice area; the results are for the unpinned and pinned small heat sinks. In view of the pressure drop being comparable between these two nozzle plates at a given flow rate (Table II), this graph also represents heat transfer coefficients roughly as a function of pressure drop. It is clear that the differences between single and multiple jets of the same open area in terms of heat transfer are more modest than seen in Fig. 4 for single and multiple jets of the same nozzle diameter. For the unpinned heat sink, the multiple jets yield higher convective coefficients than for the single jet by a factor of 1.2. However, the presence of the pins alters the results so that heat transfer coefficients become higher for the single jet with the pinned heat sinks. For the taller pins [Fig. 5(c)], almost no difference is seen between the two configurations. The large heat sinks (data not shown) yield results for the single and multiple jets that are less distinct, with the multiple jets having slightly higher heat transfer coefficients in all cases.

The total effectiveness of the pin fins is defined as  $\epsilon_f = (h_{base,pinned}/h_{base,unpinned})$ , and plotted in Fig. 6(a) as a function of air flow rate with different nozzle diameters for the two small heat sinks. The effectiveness ranges from 4 to 6,

TABLE II  
MEASURED PRESSURE DROP ACROSS DIFFERENT NOZZLES TESTED

	Flow rate (CFM)	Reynolds number, Re	Pressure drop (kPa)
Single Nozzle $d = 12.7$ mm	2.9	8639	0.11
	4.1	12213	0.24
	5.9	17575	0.5
	7.1	21150	0.75
	9.6	28597	1.4
	14.8	44088	3.2
Single Nozzle $d = 25.4$ mm	6	8937	0.05
	9.9	14746	0.14
	14	20852	0.28
	19.5	29044	0.54
	23.6	35151	0.79
	29.7	44237	1.28
Multiple Nozzles 4 x 12.7 mm	3	2234	0.01
	6	4468	0.043
	10	7447	0.13
	15	11171	0.29
	20.2	15043	0.52
	24.9	18544	0.8
	30	22342	1.18

with the tall pins being the more effective. The effectiveness values for the large heat sinks, ranging from 3.2 to 6, are shown in Fig. 6(b); at a given flow rate, the smaller  $S/d$  has the higher effectiveness.

Correlations were obtained for all the pinned heat sinks in this study in terms of Nusselt number ( $hd/k$ ) defined in two different ways: one for  $Nu_{base}$  based on the heat sink footprint area, and the other for  $Nu_{HS}$  based on the total heat sink surface area. The parameter  $(D_e/d)$  represents the ratio of the heat sink base area to the nozzle area in the correlations. The Prandtl number exponent was held constant at 0.4 [2], [4] since it was not an independently varied parameter. The first of these two correlations was

$$Nu_{base} = 3.361 Re^{0.724} Pr^{0.4} (D_e/d)^{-0.689} (S/d)^{-0.210}. \quad (4)$$

Equation (4) is valid for  $2000 \leq Re \leq 23\,000$  and  $S/d = 2$  and 3. Results from a third small heat sink with fine pins ( $d_p = 0.96$  mm) not discussed in the foregoing are also included in the correlation procedure. The average and maximum deviations of the experimental results from this equation are 10.4% and 34.8%. However, these deviations drop to 9.4% and 19.3%, respectively, if the two data points for the lowest flow rates (not fully turbulent, since  $Re \leq 2200$ ) with each of the two large heat sinks are not included. A plot of this equation is shown in Fig. 7. It may be noted that the correlations may be specific to the heat sink configuration used in this study, and best serve to illustrate the relative importance of the relevant parameters.

An alternative correlation, in terms of  $Nu_{HS}$ , was

$$Nu_{HS} = 1.92 Re^{0.716} Pr^{0.4} (A_{HS}/A_d)^{-0.698} \cdot (D_e/d)^{0.678} (S/d)^{-0.181}. \quad (5)$$

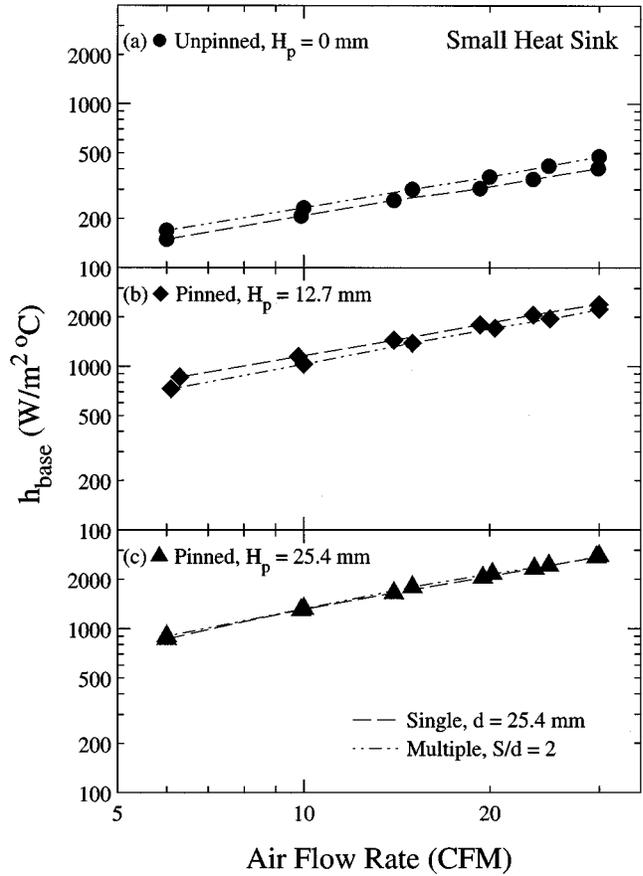


Fig. 5. Comparison of heat transfer coefficients for multiple and single jets with the same total orifice area, for the small heat sinks.

The average and maximum deviations for (5) are 6.2% and 35.6%, respectively. Again, if four data points for the large heat sinks at the lowest flow rates are excluded, the average and maximum deviations drop to 4.5% and 17.5%, respectively. A plot of Nusselt numbers calculated from (5) versus the experimental data is shown in Fig. 8. The agreement between the predicted and experimental results is considered quite satisfactory.

## V. CONCLUSION

Experimental results were obtained for the heat transfer in confined multiple air jet impingement on a square heat source to which heat sinks are attached. Four pin-finned heat sink assemblies with different pin heights and footprint areas were studied and compared to two baseline, unpinned heat sinks at two jet-to-jet spacings as a function of Reynolds number. The results for the multiple jets were compared to single jets, both at a fixed orifice diameter ( $d = 12.7$  mm) and for the same total orifice area (single jet with  $d = 25.4$  mm). At a fixed Reynolds number, the heat transfer coefficient decreased by 10% as the jet-to-jet spacing was decreased from  $S/d$  of 3 to 2 for the large, unpinned heat sink. However, an increase was observed for the pinned heat sinks when  $S/d$  was decreased from 3 to 2 largely due to the unique design of the heat sink in this study. Single jets yielded lower heat transfer coefficients than multiple jets of the same nozzle diameter for all heat sinks tested at a fixed

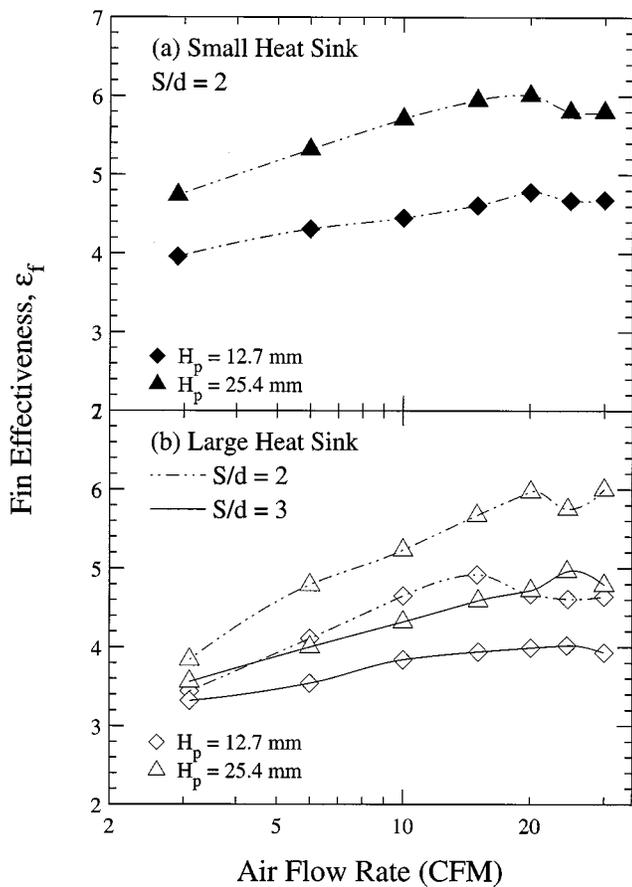


Fig. 6. Fin effectiveness as a function of air flow rate for the small and large heat sinks.

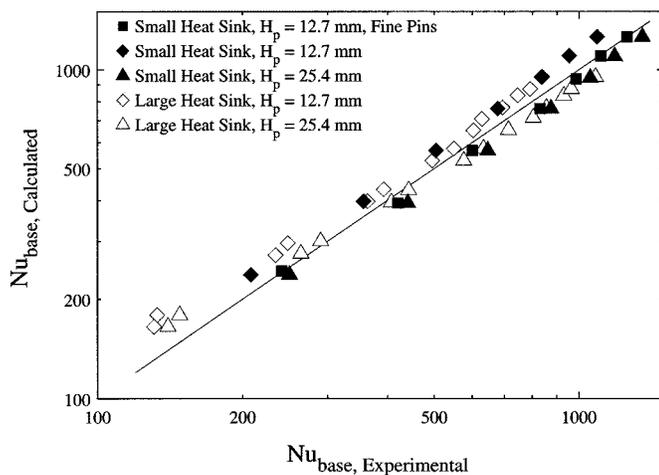


Fig. 7. Comparison between experimental and predicted (4) results for  $Nu_{base}$  for the pinned heat sinks.

Reynolds number. In contrast, the heat transfer coefficients with single jets were higher when compared on the basis of total air flow rate. Pressure drop measurements for single and multiple jets of the same total orifice open area were comparable for a given flow rate. The difference between heat transfer coefficients for these two nozzle configurations was also much less pronounced. The effectiveness of the pin-finned heat sinks, defined relative to the unpinned ones, was in the range of 3 to 6,

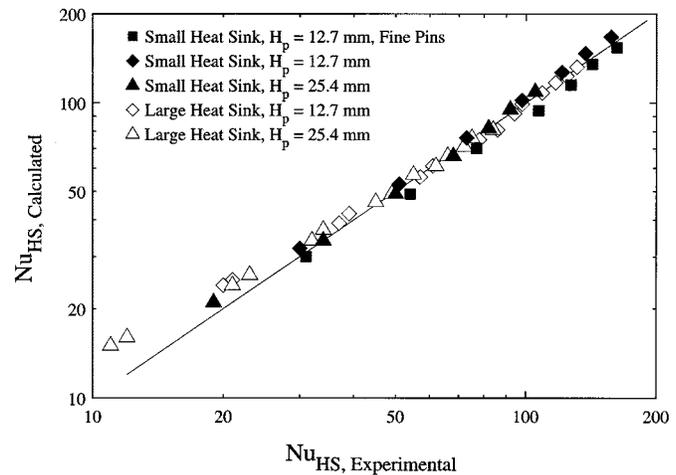


Fig. 8. Comparison between experimental and predicted (5) results for  $Nu_{HS}$  for the pinned heat sinks.

with the upper limit being achieved for the tall pins with the smaller jet-to-jet spacing. Predictive correlations are proposed for the heat transfer from the pinned heat sinks.

#### ACKNOWLEDGMENT

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