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Experimental comparison of micro-scaled plate-fins and pin-fins under natural convection☆

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ABSTRACT

The present work analyses, for the first time, the heat transfer from pin micro-fins. The scope of the present paper is comparing thermal performance of plate micro-fin and pin micro-fin arrays under natural convection conditions in air. Two fin geometries are considered: plate and pin fin arrays with the same thermal exchanging surface are tested. The investigation shows that the pin micro-fins can improve the thermal performance compared to plate micro-fin arrays. Indeed, pin micro-fins are found to have higher heat transfer coefficients and lower thermal resistances, as well as a better material usage. This makes pin micro-fins able to achieve both thermal enhancement and weight reduction than plate micro-fins. The radiative heat transfer is also calculated: a new model to determine the radiative view factors of pin fins is proposed and is used in the analysis. The effect of the orientation is considered as well.

1. Introduction

The term “fin” refers to an extended surface employed to improve the heat transfer from a solid to the surrounding fluid [1]. Due to their simplicity and reliability, finned heat sinks have been widely used in passive cooling applications [2]. The heat transfer is due to the natural convective motion of a fluid exposed to a thermal gradient and to the radiative heat transfer between bodies at different temperatures. Many fin designs have been proposed in literature [3] and the most commonly employed geometries so far have been the plate-fins and the pin-fins. Plate-fins are generally simpler to design and fabricate, whereas pin-fins are preferred where omnidirectional performance is required [4]. In 1986, Sparrow and Vemuri [5] showed that pin-finned arrays could achieve heat transfer coefficients 40% higher than those of plate finned array with the same area. Iyengar and Bar-Cohen [6] found that optimized pin-fins had better thermal performance per unit of mass than optimized plate-fins. Joo and Kim [4] recently demonstrated that, under fixed volume conditions, pin-fins have generally better heat dissipation per unit of mass than plate-fins, which, instead, have a better global heat dissipation. Despite its non-negligible contribution in the heat sink heat transfer [7], only Sparrow and Vemuri [5] took into account the radiation in the analysis.

Following the miniaturisation trend of electronic products, in the last years, micro-technologies have been considered as a solution to decrease the size and the cost of coolers [8,9]. In particular, micro-fins have been identified as one of the most promising technologies for the passive cooling of components [10]. Micro-fins under natural conditions have already shown the potential to achieve effectiveness as high as 10% [11] and have been found to positively benefit the heat transfer performance per unit of mass compared to a flat surface [12]. The correlations between the fin geometry and the heat transfer coefficients have been already presented [13,14], but the research so far has been mainly limited to plate-fins.

Recently, the pin micro-fins were found able to increase the combined convective/radiative heat transfer compared to plate micro-fins with the same area [14]. The present work aims to extend this investigation, analysing independently the contributions of radiation and convection and taking into account different heat sink metrics. For the first time, pin and plate micro-fins have been experimentally tested and their behaviours have been compared. The contributions of convection and radiation have been independently analysed, due to the non-negligible effect of radiation at micro-scale [14,15]. In this light, in lack of available pre-existing models, a new geometrical model to determine the view factors of the pin fins has been proposed and, for the first time, has been used. The aim of this paper is to report the first comparison between plate and pin micro-fins so far presented. Future directions have been identified and recommendations for future works have been reported, in order to extend the knowledge on micro-scaled natural convective fins and to contribute to the development of new, compact, low-cost heat sinks for industrial applications.

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Nomenclature

A	surface area
F	view factor
H	fin height
h	heat transfer coefficient
h_m	mass specific heat transfer coefficient
L	length of the array
Q	heat power
Q_{in}	power in input
s	fin spacing
T	temperature
t	fin thickness
tb	base thickness
V	volume
W	width of the array
x	geometric parameter
y	geometric parameter

Greek symbols

ε	emissivity
ε_{fins}	overall fin effectiveness
ρ	density
σ	Stephan–Boltzmann constant

Subscripts

amb	refers to the ambient
c	refers to convection
fins	refers to the finned array
flat	refers to the flat sample
i	refers to the i-wall of the fin
loss	refers to the losses happening on the case
r	refers to radiation

Abbreviations

HTC	heat transfer coefficient

2. Data collection and analysis

2.1. Experimental setup

In this study, two rectangular plate-fins and two square pin-fin geometries have been studied (Fig. 1). Square, 5 cm × 5 cm wide, 1.4 mm-thick silicon flat wafers have been used as baseplate. The fins have been obtained through a dicing machine, able to produce 2D profiles: the pin fins have been shaped by rotating the plate-fin arrays of 90°. All the dimensions are reported in Table 1. In each geometry, the fin thickness is equal to the fin spacing; therefore, there is no difference in surface between the plate fin and the pin fin arrays.

The experimental setup used for this investigation is shown in Fig. 2. It has already been presented and validated in [14]. The fin arrays have been placed in an insulating case made of a 1 cm-thick fibre thermal material, covered by an additional 1 cm-thick polystyrene layer. An electrical heater (Omega KHLV-202/2.5), bonded on the back surface of each fin array, has been used as heat source, operated through a DC power supply. The current and the voltage in input, measured by two digital multimeters (Fluke 115 and Fluke 8050A respectively), have been used to calculate the heat produced. The fin arrays have been placed inside a 25 cm × 25 cm × 25 cm box open on top, in order to limit any external interference.

The temperature of the fins has been measured by a thermal imaging camera (FLIR T425), considering a constant temperature across the fin array and the fin height. One K type thermocouple has been used to measure the temperature of the insulating case walls, whereas three more K type thermocouples have been employed to measure the temperature of the air nearby the fin array. The room temperature has been kept at 25 °C. The experiment has been conducted at steady-state: each test has been repeated three times and the temperature values have been recorded every two seconds by a data logger (Tesa 51408).

The fin arrays have been tested in horizontal position, both upward and downward orientated, under natural convection conditions. Power inputs of 5 W, 7.5 W and 10 W have been supplied to the heater, which corresponded to heating power densities of 2.0 kW/m^2, 3.0 kW/m^2 and 4.0 kW/m^2. The description of the validation and the calculation of the experimental uncertainties has been already reported in [12,14]. Out of a predicted experimental uncertainty of 8.25% on the measurement of the heat transfer coefficients, an average discrepancy of 6.07% was found between the experimental data and the results of a numerical model.

2.2. Data analysis

The power in input (Q_{in}) is dissipated by the experimental setup in three ways: convection from the fin array (Q_c), radiation from the fin array (Q_r) and combined convective/radiative losses through the insulating case (Q_{loss}) [13]:

$$Q_{in} = Q_c + Q_r + Q_{loss}. \tag{1}$$

The determination of Q_{loss} has been described in [14]. The value of Q_{loss} has been determined in each single test: an average of 26% of the input power has been found to be lost through the insulating case.

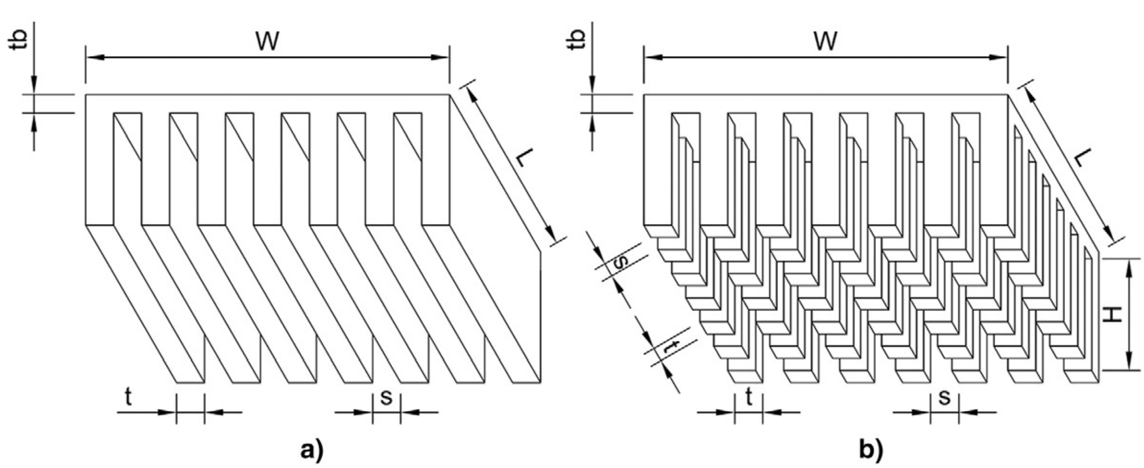

Fig. 1. Geometry of the fin arrays: (a) plate fin, and (b) pin fin. The parameters are labelled: s (fin spacing), t (fin thickness), H (fin height), tb (base thicknesses), W (width), L (length).

Table 1
Fin dimensions. Each geometry has been used to produce a plate fin and a pin fin array.

	W, L [mm]	t [μm]	s [μm]	H [μm]	tb [μm]	Number of fins (plate fins–pin fins)
Geometry #1	50	200	200	600	800	121–15,376
Geometry #2	50	400	400	600	800	61–3,844

The radiative heat transfer (Q_r) has been calculated by considering the same assumptions made by [5]: isothermal fins and not-radiating surrounding. The surrounding medium is assumed to be a large and black body, so that its temperature can be assumed equal to the ambient one [7]. The heat dissipated by radiation has then been calculated by using the Stefan–Boltzmann equation:

$$Q_r = \sum_i \varepsilon \cdot \sigma \cdot A_i \cdot F_{i,amb} \cdot \left(T_{fins}^4 - T_{amb}^4\right) \tag{2}$$

where ε is the experimentally determined emissivity of silicon (0.78), σ is the Stefan–Boltzmann constant ($5.67 \cdot 10^{-8}$ W m^{-2} K^{-4}), A_i is the area of each i-surface of the fins, $F_{i,amb}$ is the view factor between the i-surface and the ambient and T_{fins} and T_{amb} are the temperatures of the fins and the ambient respectively.

The view factors rely on the geometry of the fins and vary for each surface. The method to determine the view factors of the plate fins has been described in [14]. The pin fin arrays, represented in Figs. 3 and 4, have instead required a more complex method. Due to a lack of freely available methods in literature, a simplified method to calculate the view factors of the pin fin has been introduced. The view factors of the different fin walls have been summarized in Table 2. The main difference of pin fins compared to plate fins is that the view factor of the fin base cannot be considered uniform. For this reason, in the present work, the fin base has been classified in:

- Enclosed fin base: portion of the fin base that is limited on two sides by the fin walls. It is labelled as (b) in Figs. 3 and 4.
- Open fin base: portion of the fin based placed in the space among four adjacent pin fins. It is labelled as (f) in Figs. 3 and 4.

The view factor of the enclosed fin base can be calculated as described in [16], by using the equation for an enclosure with four

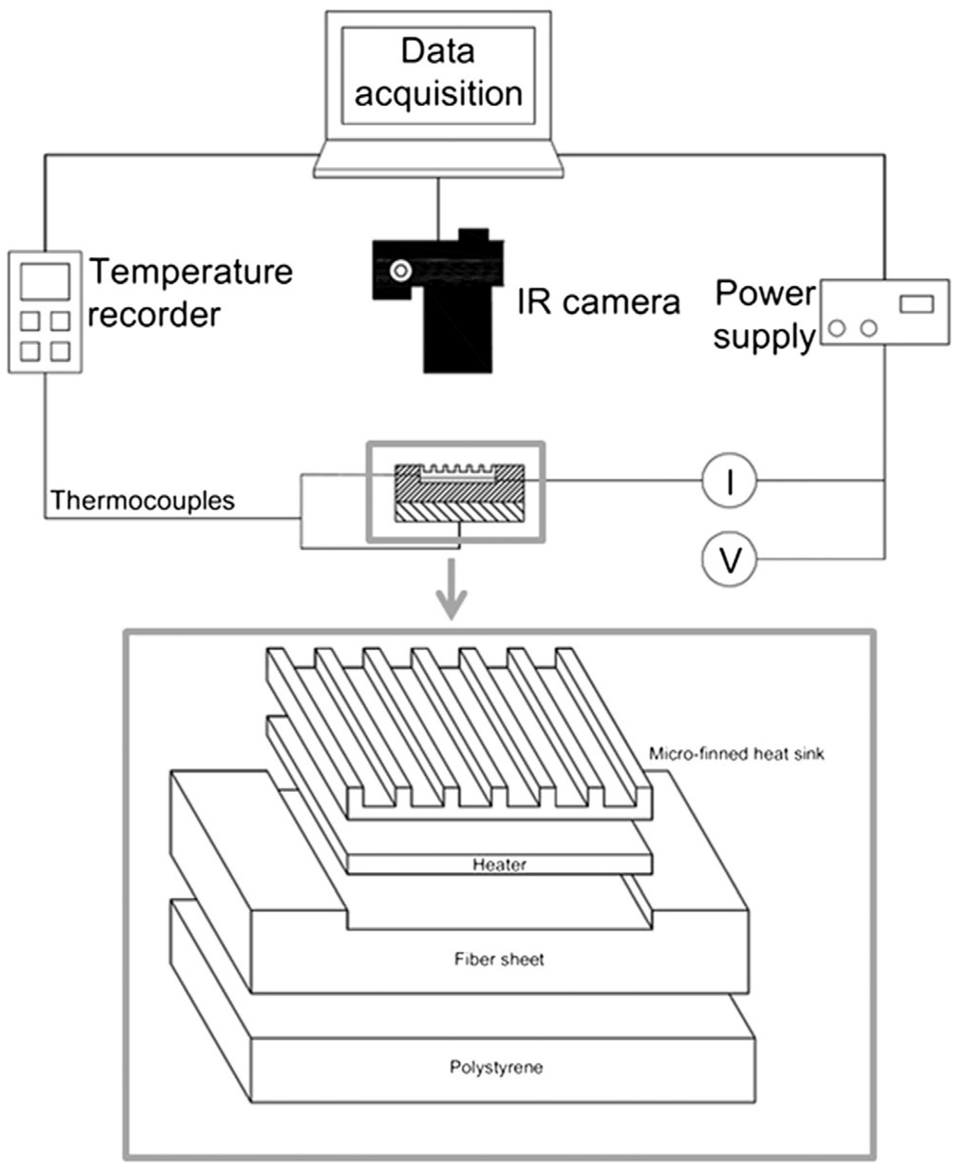

Fig. 2. Schematic of the experimental setup and of the exploded structure of the sample case.

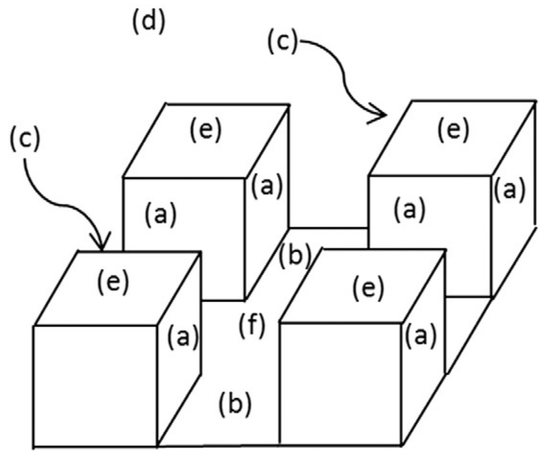

Fig. 3. 3D view of a 2 × 2 pin-fin array.

surfaces. In lack of referenced methods, instead, a new procedure for the determination of the view factor of the open fin base has been proposed. Indeed, by using the same equation for an enclosure with four surfaces, it can be seen that all the surfaces of the enclosure other than the open fin base consist of air. This means that the view factor obtained for the open fin base can be assumed to be 1. The view factors between the fin surfaces and the ambient ($F_{a,d}$, $F_{c,d}$, $F_{e,d}$) are similar to those of the plate fins and have been reported in Table 2 as well.

The heat transferred by convection has then been obtained through Eq. (1), by knowing the input power, the losses and the radiative contribution. The average convective heat transfer coefficient for a fin array has been expressed as follow:

$$h_{fins} = \frac{Q_c}{A_{fins} \cdot (T_{fins} - T_{amb})} \tag{3}$$

where A_{fins} is the area of the finned surface.

The heat sinks are generally categorized according to the thermal resistance. This parameter is analogous to the electrical resistance and measures the difficulty of the heat sinks to dissipate heat in air when a temperature difference is present. It is expressed as:

$$R_{fins} = \frac{(T_{fins} - T_{amb})}{Q_c} \tag{4}$$

In contrast with the heat transfer coefficient, the thermal resistance does not take into account the extension of the thermal exchanging surface. Indeed, it considers the overall performance of the heat sinks, instead of the performance per unit of surface. In order to quantify the effectiveness of fin material utilization, a different metric can be used. The mass specific heat transfer coefficient, proposed by [18,19], characterizes the effectiveness with which fin material is utilized in the promotion of heat transfer and has been express as:

$$h_m = \frac{Q_c}{V_{fins} \cdot \rho_{fins} \cdot (T_{fins} - T_{amb})} \tag{5}$$

where V_{fins} and ρ_{fins} are respectively the volume and the density of the finned array.

In order to measure the enhancement in performance due to the micro-fin arrays, the overall fin effectiveness (ε_{fins}) has been considered as well. This parameter expresses the ratio between the heat transferred by the fins and that transferred without fins over the same area of a flat surface of length L and width W. The fin effectiveness has been calculated as follow:

$$\epsilon_{fins} = \frac{Q_c}{Q_{flat}} \tag{6}$$

where Q_{flat} is the heat transferred by convection from the unfinned flat surface.

3. Results and discussions

3.1. Heat transfer coefficient

The heat transfer coefficient (HTC) is a coefficient of proportionality that determines, in presence of a difference of temperature between a surface and a fluid, the heat flux. It is not a property of the fluid, but depends on different factors, such as the velocity, the flow pattern and the properties of the fluids, as well as on the geometry of the surface [17,20]. In the present investigation, the heat transfer coefficient of the pin fins under natural convection in air has been found to be consistently higher than that of plate fins. Taking into account the same power input, upward-facing pin fin arrays have heat transfer coefficients 3% to 6% higher than plate fins (Fig. 5). It means that the larger volumes available for air facilitate the dissipation of heat. This improvement is enhanced when the arrays are downward orientated (Fig. 6): the HTC can be as high as 20%. Compared to the macro-scaled case [5], the enhancement is limited by the constraint air volumes between the fins, where the viscous force effects are increased.

Since the area of the finned surface is the same for each geometry, the higher heat transfer coefficients of the pin arrays lead to an enhanced convective heat transfer capacity. This is clearly shown by Fig. 7 and Fig. 8, which show the thermal resistance variations depending on the power in input, for upward and downward facing arrays

Fin top	Enclosed fin base	Fin top
Enclosed fin base	Open fin base	Enclosed fin base
Fin top	Enclosed fin base	Fin top

(e)	(b)	(e)
(b)	(f)	(b)
(e)	(b)	(e)

Fig. 4. Top view of the 2 × 2 pin-fin array shown in Fig. 3 (left) and nomenclature used for the determination of the view factors (right).

Table 2
Methods used for the determination of the view factor in the pin-fin arrays.

View factor	Description	Method	Notes
$F_{a,b}$	Fin wall to enclosed fin base	$F_{a,b}=\frac{1}{\pi\cdot y}\left\{y\cdot\tan^{-1}\left(\frac{1}{y}\right)+x\cdot\tan^{-1}\left(\frac{1}{x}\right)-\left(x^2+y^2\right)^{0.5}\cdot\tan^{-1}\left(\frac{1}{\left(x^2+y^2\right)^{0.5}}\right)+\frac{1}{4}\ln\left(\frac{(1+x^2)\cdot(1+y^2)}{1+x^2+y^2}\left(\frac{y^2\cdot(1+x^2+y^2)}{(x^2+y^2)\cdot(1+y^2)}\right)^{y^2}\left(\frac{x^2\cdot(1+x^2+y^2)}{(x^2+y^2)\cdot(1+x^2)}\right)^{x^2}\right)\right\}$	x = s/L and y = H/L [17]
$F_{a,c}$	Fin wall to opposite fin wall	$F_{a,c}=\frac{2}{\pi\cdot x\cdot y}\cdot\left\{\ln\left(\frac{(1+x^2)\cdot(1+y^2)}{1+x^2+y^2}\right)^{0.5}+x\cdot\left(1+y^2\right)^{0.5}\cdot\tan^{-1}\left(\frac{x1y}{\left(1+y^2\right)^{0.5}}\right)+y\cdot\left(1+x^2\right)^{0.5}\cdot\tan^{-1}\left(\frac{y}{\left(1+x^2\right)^{0.5}}\right)-x\cdot\tan^{-1}x-y\cdot\tan\right.$	x = H/s and y = L/s [17]
$F_{a,d}$	Fin wall to ambient	$F_{a,d}=1-F_{a,b}-F_{a,c}$	Enclosure with four surfaces [16]
$F_{c,d}$	Fin wall to ambient	$F_{c,d}=F_{a,d}$	
$F_{b,a}$	Enclosed fin base to fin wall	$A_a\cdot F_{a,b}=A_b\cdot F_{b,a}$	Reciprocity relation [16]
$F_{b,c}$	Enclosed fin base to fin wall	$F_{b,a}=F_{b,c}$	
$F_{b,d}$	Enclosed fin base to ambient	$F_{b,d}=1-F_{b,a}-F_{b,c}$	Enclosure with four surfaces [16]
$F_{e,d}$	Fin top to ambient	1	
$F_{f,d}$	Open fin base to ambient	1	Enclosure with four surfaces [16]

respectively. The thermal resistance of the pin fins has been found to be constantly lower than the plate fins. This means that the pin fins are able to transfer a larger amount of heat by convection under the same input conditions, compared to the plate fins.

The overall fin effectiveness is a figure of merit used to determine the impact of the fins on the heat transfer rate [17]. For the scope of the present investigation, the variation in overall fin effectiveness due to the pin fins compared to the plate fins has been considered. The use of pin micro-fin led to an enhancement in overall fin effectiveness that ranged between 3 and 7%, generally positively affected by the temperature. In particular, the less performing geometry (#1) is the one that had the most relative benefits from the introduction of micro-fins. The reduced number of fin geometries taken into account did not allow analysing the correlation between the geometry and the thermal performance. Despite that, the results shown in Figs. 5 and 6 are in agreement with the findings presented in [13,14]: the heat transfer coefficient is higher when the fin spacing and the fin thickness are increased. Further experimental studies on different fin geometries are recommended for a better characterization of the heat transfer in pin micro-finned arrays.

3.2. Mass specific heat transfer coefficient

It has already been demonstrated in literature [12] that the employment of micro-fins can lead to two benefits: an enhancement in thermal performance and a reduction in mass, compared to a flat surface. This is possible because the micro-fins are generally obtained by subtractive methods, so removing part of the material from the original surface. In particular, by using the mass specific heat transfer coefficient, it was demonstrated that dicing an array of plate micro-fins could enhance the effectiveness with which fin material is utilized by 50%.

The mass specific heat transfer coefficients of plate and pin micro-fin arrays have been investigated, both in upward (Fig. 9) and downward (Fig. 10) orientation. As expected, the pin fin arrays have been consistently found to perform better than the plate ones, with enhances ranging between 20% and 40%. These high values are due to the combination of the two advantages of pin fins: the increased heat transfer found out previously and the reduced volumes compared to the plate fins.

For each geometry, the pin and the plate fin arrays had the same surface extension. Despite that, the pin fin arrays were 13% lighter in volume than the plate ones. This makes pin fins particularly interesting for those application where a reduced weight is required. Along with that, taking into account the better heat transfer achieved the pin fins have been found as a more effective solution than plate fins for natural convective heat transfer at micro-scale. These results are in agreement with the conclusions of [5], where the use of pin fins in place of plate fins was encouraged.

Despite the negligible difference between the two plate micro-fin geometries, there is a discrepancy between the performances of the two pin micro-fin geometries. The geometry #1 is found to have higher values of the heat transfer coefficient, in both downward and upward orientation. This might be due to the more extended surface: the pin fin array #1 has a surface 38% larger than #2, which enables better convective heat dissipation among the pin fins, despite the lower heat transfer coefficients. More investigations are required to further analyse this parameter, in order to open the way the optimization of the micro-fin geometries.

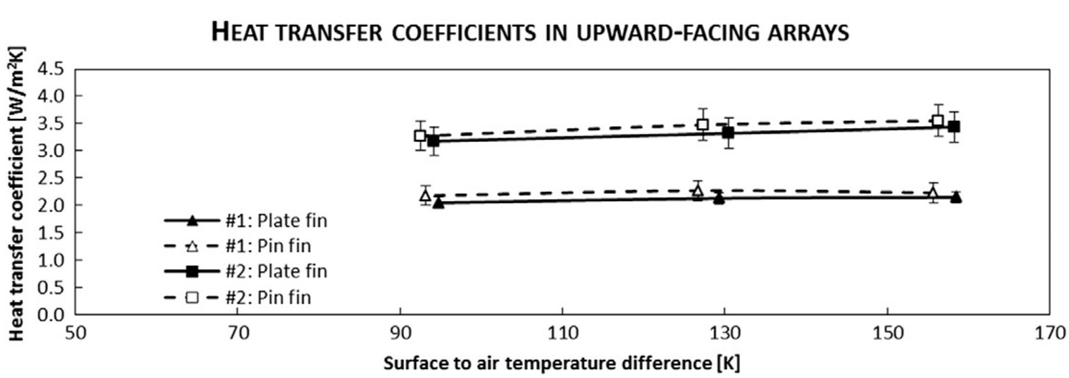

Fig. 5. Heat transfer coefficients of upward-facing plate and pin fins under natural convection.

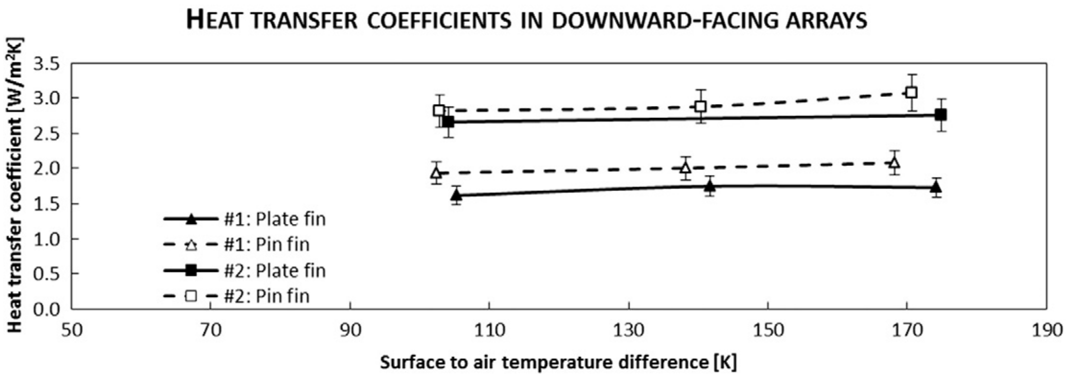

Fig. 6. Heat transfer coefficients of downward-facing plate and pin fins under natural convection.

3.3. The contribution of radiation

The radiative component of heat transfer of a heat sink under natural convection is generally considered negligible [7]. At micro-scale, instead, it has been already demonstrated that the contribution of radiation can represent a large percentage [14,15]. In this case, the heat transfer is found to contribute to the dissipation of 55 to 70% of the heat exchanged by the fins, because of the high temperature difference experiences in this work and to the high emissivity of silicon. As expected, this value is lower for the fins in upward than in downward orientation (Fig. 11). This is due to the fact that radiation is insensitive to the orientation, whereas the convective performance is reduced when the fins are facing downwards. This means that, under the same temperature difference, the contribution of radiation will be greater in downward orientation than in upward. The contribution of radiation increases with the temperature difference because of its dependence on the fourth power of the temperatures, whereas the convection on the first power only.

No remarkable difference has been found among the array geometries considered in terms of percentage contribution of radiation. This

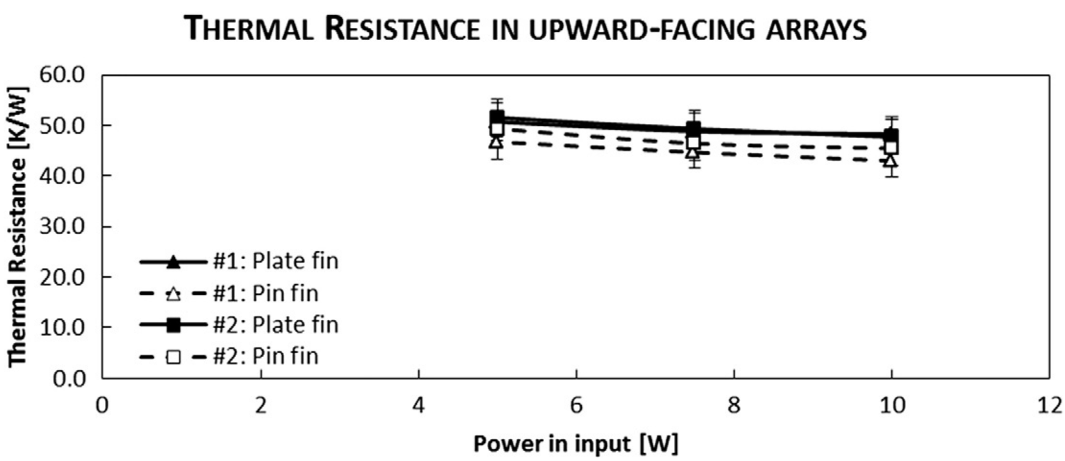

Fig. 7. Thermal resistance of upward-facing plate and pin fins under natural convection.

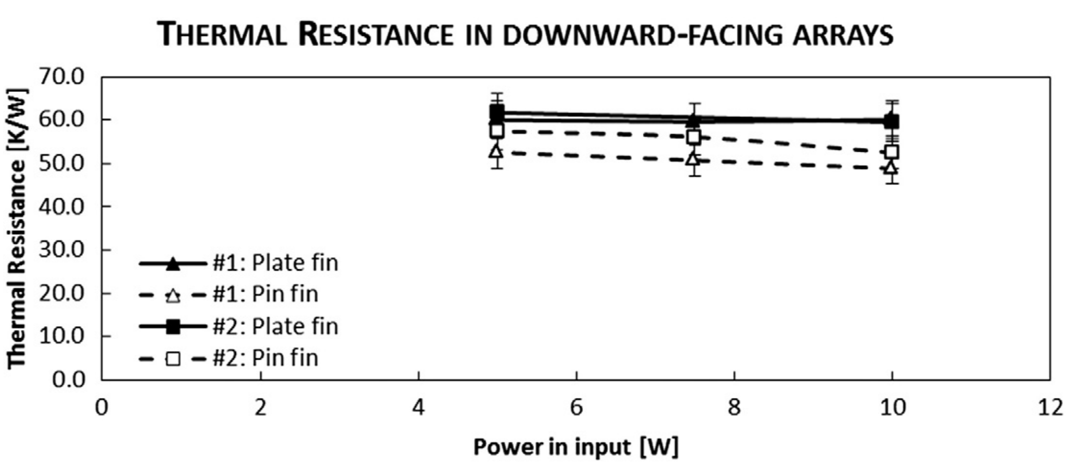

Fig. 8. Thermal resistance of downward-facing plate and pin fins under natural convection.

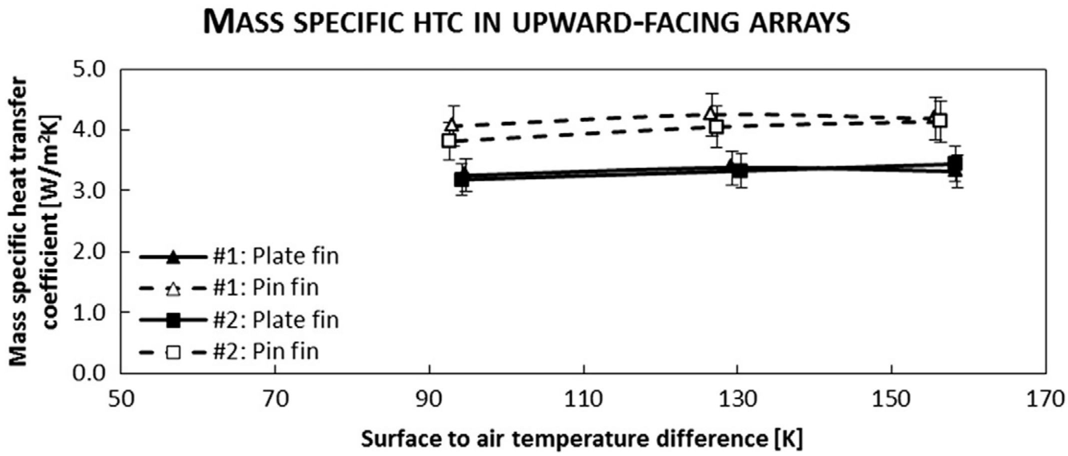

Fig. 9. Mass specific heat transfer coefficients (HTC) of upward-facing plate and pin fins under natural convection.

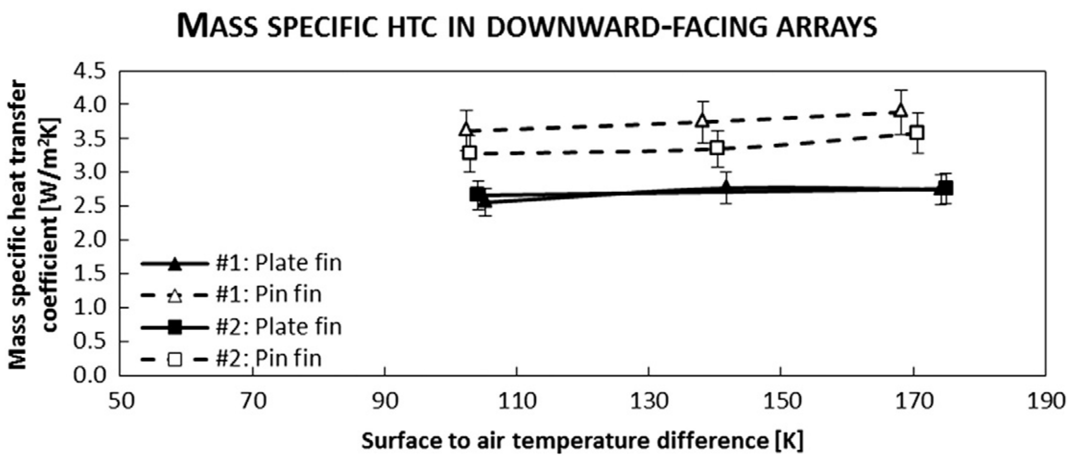

Fig. 10. Mass specific heat transfer coefficients (HTC) of downward-facing plate and pin fins under natural convection.

is due to the geometry of the fins that have been studied, whose fin spacings are equal to the fin thicknesses. Further studies are anyway required to develop a more accurate method to determine the view factors of the pin fin geometries. Moreover, more experimental investigations on different fin arrays would help identifying those geometries that optimize the radiative heat transfer and the combined convective/radiative heat transfer.

4. Conclusions

The present paper reported the preliminary results of an experimental investigation conducted on micro-fins array. It analysed the thermal performance of pin micro-fins compared to that of plate micro-fins with the same extensions. A novel simplified method to determine the view factor of the pin fins has been proposed and used for the investigation.

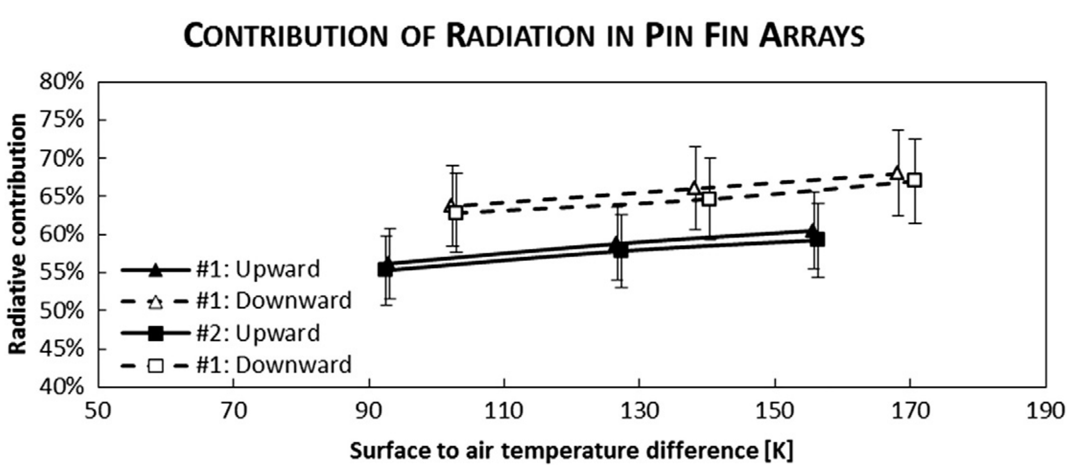

Fig. 11. Percentage of power dissipated by pin fins through radiation.

Pin fins have been found to perform better, due to the higher heat transfer coefficients, whose enhancement depends on the geometry and the orientation. An enhancement between of 3% to 7% has been registered in terms of overall fin effectiveness. Moreover, pin fins have shown the ability of optimizing the material usage, compared to the flat fins.

This paper has been intended to compare, for the first time, the performance of pin micro-fins and plate micro-fins. In order to do that, a novel method for the characterization of radiative heat transfer in pin fin arrays had to be introduced. It has been found that the use of micro-scaled pin fins should be encouraged, similarly to what happened at macro-scale. Despite that, further studies are recommended in order to refine this method and to widen the investigation on the contribution of radiation in micro-fins under natural convection. Moreover, more fin arrays have to be studied, in order to show the correlation between geometry and thermal performance in micro-fins. The effects of the materials, as well as the orientation, should not be neglected and will be presented in future works.

Acknowledgements

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