



# Thermal effectiveness and mass usage of horizontal micro-fins under natural convection



Leonardo Micheli <sup>a,\*</sup>, K.S. Reddy <sup>b</sup>, Tapas K. Mallick <sup>a</sup>

<sup>a</sup> Environment and Sustainability Institute, University of Exeter, Penryn Campus, Penryn, Cornwall TR10 9FE, UK

<sup>b</sup> Heat Transfer and Thermal Power Laboratory, Department of Mechanical Engineering, Indian Institute of Technology Madras, Chennai 600 036, India

## HIGHLIGHTS

- Individual and overall effectivenesses up to 3.3 and 1.1 are respectively found.
- The individual effectiveness increases by reducing the thickness to spacing ratio.
- Micro-fins can enhance the mass specific heat transfer coefficient up to 50%.
- The mass specific heat transfer coefficient increases with height and spacing.
- The correlations are found to be consistent for upward and downward orientations.

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## ABSTRACT

In recent times, the micro-technologies have gained prominence in various engineering applications. The micro-technologies are already in use for cooling purposes in several systems, but the information on the thermal performance of micro-fins under natural convective heat transfer conditions is yet limited. The correlations between heat transfer coefficients and fin geometry have already been investigated, but are not sufficient to optimize the design of the micro-finned arrays. For this reason, the present investigation gives an overview of micro-fins behavior taking into account, for the first time, different heat sink metrics: the fin effectiveness and the mass specific heat transfer coefficient. The results of an original experimental investigation are merged with the data available in literature. Natural convective micro-fins are able to achieve overall fin effectivenesses higher than 1.1. Even if not always beneficial in terms of heat transfer, micro-fins are found always positive in terms of the material usage. In this light, micro-fins can be considered advantageous in those applications that require a minimized weight of the heat sinks. Moreover, a limited effect due to the orientation is observed.

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## 1. Introduction

Extended surfaces/fins are commonly used to enhance the heat transfer from a system to the surrounding fluid. The application of fins has been investigated in numerous studies and is currently employed for many different purposes, such as electronics, industrial processes and energy generation. Fins in natural convection are considered as passive coolers, since they do not require input of mechanical or electrical power. Fins act by exploiting both the natural convective motion of a fluid due to a thermal gradient and the radiative heat transfer [1]. Passive coolers are generally considered more reliable and less exposed to cooling failures [2] than active coolers, which need instead external energy in input to perform.

Industries and consumers are always for the production of more efficient, more compact and low cost products. In this light, micro-technologies have gained much interest in the last decades, because of the better performance achieved and the limited space and material required compared to macro-scaled solutions. For this reason, micro-cooling technologies, including micro-finned arrays in forced convection conditions, have been extensively investigated [3–5]. Despite that, only a limited number of researches on naturally convective micro-fins can be found in literature. Kim et al. [6] investigated vertically orientated micro-fins and demonstrated that macro-fin heat transfer correlations could not be used to describe the behavior of micro-scaled systems. Mahmoud et al. [7] experimentally sorted out, for the first time, the correlations between micro-fin geometries and heat transfer coefficients. Even if lower than those of a flat surface, the heat transfer coefficients for horizontal upward facing fins were found to increase when the fin spacing was increased and the fin height was decreased. Recently,

\* Corresponding author. Tel.: +44 (0) 1326 259467; fax: +44 (0) 1326 370450.  
E-mail address: [l.micheli@exeter.ac.uk](mailto:l.micheli@exeter.ac.uk) (L. Micheli).

Micheli et al. [8] reported that the heat transfer coefficient is enhanced by the thickness of the fins. Shokouhmand and Ahmadpour [9] numerically demonstrated the non-negligible contribution of radiation, which accounted for more than 20% of the global micro-fins heat transfer.

Earlier researches mainly focused on the heat transfer coefficients. In real world applications, instead, engineers and system designers look for thermal resistance, compactness, weight and cost of the heat sinks. The present work aims to study, for the first time, the effective thermal enhancement due to the application of micro-fins on a horizontal flat surface by analyzing additional heat sink metrics, such as the individual and overall fin effectiveness and the mass specific heat transfer coefficient. The correlations between geometry and heat sink metrics are explored and commented in the present paper to contribute to optimize the design of micro-fins. The arrays are studied in upward and downward orientations: despite the upward conditions are known to be the best for natural cooling [10], different orientations might be required in some applications [11] and, for this reason, need to be investigated.

## 2. Heat sink metrics

The previous works [6,7] studied the heat transfer coefficient of micro-fins arrays ( $h_{fins}$ ), which is described as follows:

$$h_{fins} = \frac{Q_{fins}}{S_{fins} \cdot (T_{fins} - T_a)} \quad (1)$$

where  $Q_{fins}$  is the heat dissipated through the fins by convection,  $S_{fins}$  and  $T_{fins}$  are respectively the surface and the temperature of the finned surface, and  $T_a$  is the ambient temperature. In accordance with previous investigations on micro-fins,  $T_{fins}$  is considered constant along both the fin height and the length of the array [7,8]. Because of this assumption, no analysis of the fins efficiency has been carried out. Therefore, since no temperature gradient is considered across the finned surface,  $h_{fins}$  represents the average of the heat transfer coefficients of the various surfaces of the fins.

The aim of a fin is to increase the heat transfer from a surface to a fluid by increasing the thermal exchanging surface. In practical applications, it is required to understand the effective heat transfer enhancement introduced by the fins if compared to the original flat surface. The heat transfer coefficient measures the thermal property per unit of surface and it is not an indicator of the thermal performance of the heat sink, because it does not take into account the surface extension obtained when the fins are introduced. For this reason, a second parameter, the individual fin effectiveness ( $\varepsilon_f$ ), is generally used [12] to compare the behavior of a single fin and that of a flat surface:

$$\varepsilon_f = \frac{Q_f}{h_{flat} \cdot A_f \cdot (T_{flat} - T_a)} \quad (2)$$

where  $Q_f$  is the heat transferred by a single fin,  $A_f$  is its cross-sectional area, and  $h_{flat}$  and  $T_{flat}$  are respectively the heat transfer coefficient and the temperature of the flat plate.  $Q_f$  is obtained as:

$$Q_f = h_{fins} \cdot (2 \cdot L \cdot H + 2 \cdot t \cdot H + A_f) \cdot (T_{fins} - T_a) \quad (3)$$

where  $L$  is the fin length,  $H$  is fin height and  $t$  is the fin thickness (Fig. 1). Incropera et al. [13] defines the individual fin effectiveness as the ratio of the fin heat transfer rate to the heat transfer rate that would exist without the fin.  $\varepsilon_f$  is expected to be as large as possible. A fin array is made of a number of fins: for this reason the overall fin effectiveness ( $\varepsilon_{fins}$ ) has to be considered as well, in order to have a most accurate measure of the performance of the whole fin array [14–16]. The overall fin effectiveness ( $\varepsilon_{fins}$ ) is obtained as the ratio of heat transfer with fins to that without fins over

the same area of a flat surface of length  $L$  and width  $W$ :

$$\varepsilon_{fins} = \frac{Q_{fins}}{Q_{flat}} \quad (4)$$

where  $Q_{flat}$  is the heat transferred by the flat plane. The overall fin effectiveness directly compares the heat transferred by the fin array and by the unfinned surface: if  $\varepsilon_{fins} > 1$ , the fins have enhanced the thermal behavior of the surface.

Micro-fins are usually obtained through material subtractions: along with the effects on the heat transfer, they reduce the mass of the heat sink. This feature becomes particularly important in portable or tracked systems, such as concentrating photovoltaics, where a reduced weight means a reduced load for the tracker [17]. The mass specific heat transfer coefficient measures the effectiveness with which fin material is utilized in the promotion of heat transfer [18] and is expressed as:

$$h_m = \frac{Q_{fins}}{\rho \cdot V_{fins} \cdot (T_{fins} - T_a)} \quad (5)$$

where  $\rho$  is the density of the fin material and  $V_{fins}$  is the volume of the whole micro-finned heat sink.

## 3. Experimental investigation

### 3.1. Micro-fin geometries

A micro-fin is any extended surface with at least one micro-scaled dimension. In the present work, ten plate fin geometries have been considered: the fin dimensions are resumed in Table 1, according to the nomenclature shown in Fig. 1, and numbered from #1 to #10. The fin arrays are diced on 1.4 mm-thick, 50 mm × 50 mm-sized square silicon wafers and their thermal behaviors are compared with that of a flat unfinned silicon wafer.

### 3.2. Experimental setup

The experimental setup described in Ref. 8 is used, as shown in Fig. 2. The heat is generated by 5 cm × 5 cm-sized electrical heaters (Omega KHLV-202/2.5), regulated using a DC power supply (Weir 413D). The heaters are bonded to the back surface of the silicon wafers through a thin conductive adhesive (3M tape 966, 0.18 W/mK). The power in input ( $Q_m$ ) is calculated by multiplying the input voltage

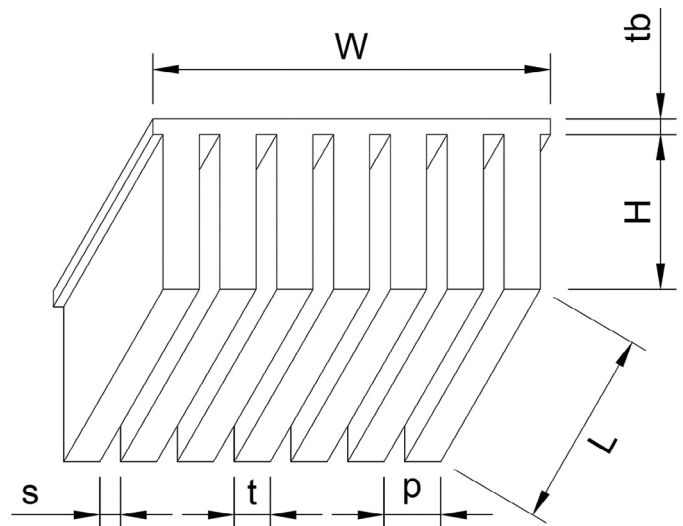


Fig. 1. Horizontal plate micro-fins geometry with typical dimensions.

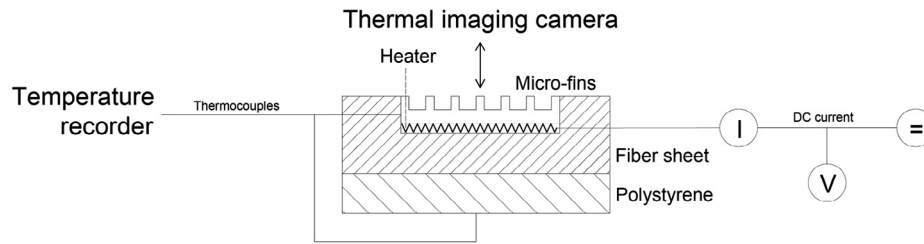


Fig. 2. Schematic of experimental setup.

Table 1

Dimensions of the fin arrays, in mm.  $L = W = 50$  mm.

Fin geometry no.	Height (H)	Pitch (p)	Thickness (t)	Spacing (s)	Base thickness (tb)
#1	0.6	0.4	0.2	0.2	0.8
#2	0.6	1.0	0.2	0.8	0.8
#3	0.6	0.6	0.4	0.2	0.8
#4	0.6	1.2	0.4	0.8	0.8
#5	0.6	0.8	0.4	0.4	0.8
#6	0.6	1.6	0.8	0.8	0.8
#7	0.8	0.4	0.2	0.2	0.6
#8	0.8	0.8	0.4	0.4	0.6
#9	1.0	0.6	0.2	0.4	0.4
#10	1.0	0.6	0.4	0.2	0.4

and the input current, measured using two digital multimeters (Fluke 115 and Fluke 8050).

In order to minimize the thermal losses from any surface other than the fins, each wafer is placed in a 1-cm thick case made of fiber thermal material (0.05 W/mK), covered on the back by a 1 cm-thick polystyrene block (0.03 W/mK). Measures are taken at steady state conditions, 20 minutes after the test has started, using some K type thermocouples placed on the back and the sides of the insulating case and a thermal imaging camera to determine the fin array temperature. The temperature of the fins is considered constant across the whole volume of each fin array, as well as the heat transfer coefficients [6–8]: indeed, due to the limited size of the fins, no gradient of temperature could be defined across the array. Therefore, the maximum temperature recorded by the thermal imaging camera at steady state has been used for the calculations, as shown in Fig. 3.

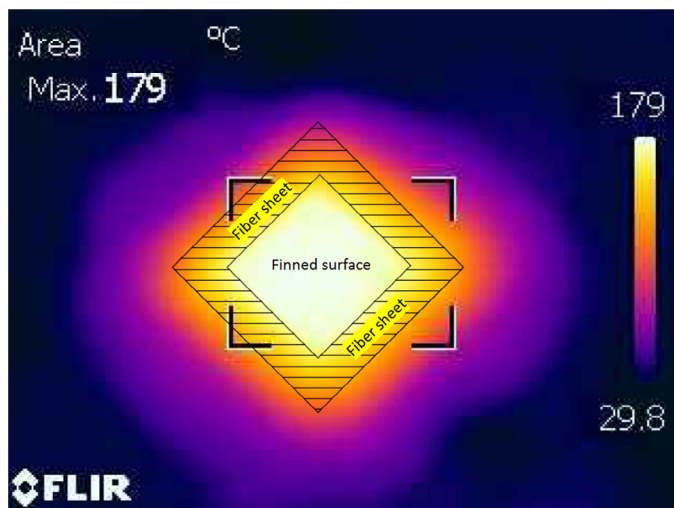


Fig. 3. Front shot of the fin array in upward position taken from the thermal imaging camera. The finned surface and the fiber sheet have been labeled.

The temperature of the surrounding air is measured averaging the outputs of three K type thermocouples located in proximity of the experimental setup.

Power inputs of 1.0 kW/m<sup>2</sup>, 2.0 kW/m<sup>2</sup>, 2.5 kW/m<sup>2</sup>, 3.0 kW/m<sup>2</sup>, 3.5 kW/m<sup>2</sup> and 4.0 kW/m<sup>2</sup> are supplied to the heater. Each test is repeated three times and the average value of the outputs is considered. The fin behavior is investigated both in upward facing and in downward facing conditions.

The power dissipated by convection from the fins array is calculated after the radiative heat transfer from the fins ( $Q_r$ ) and the losses that happen on the back and the sides of the samples ( $Q_{losses}$ ) [8]:

$$Q_{fins} = Q_{in} - Q_r - Q_{losses} \quad (6)$$

The radiative component is estimated by using the following equation [7]:

$$Q_r = \sum_i \varepsilon \cdot \sigma \cdot S_i \cdot F_{i,a} \cdot (T_s^4 - T_a^4) \quad (7)$$

where  $\varepsilon$  is the emissivity of fin material,  $\sigma$  is the Stefan–Boltzmann constant ( $5.67 \cdot 10^{-8}$  W/m<sup>2</sup> K<sup>4</sup>),  $S_i$  is the area of the correspondent  $i$ -surface of the fins,  $F_{i,a}$  is the view factor between the  $i$ -surface of the fins and the ambient, and  $T_s$  and  $T_a$  are respectively the temperatures of the  $i$ -surface and the ambient. The view factors of each surface are calculated according to the methodology suggested by Ref. 10.

The  $Q_{losses}$  are due to the heat dissipation happening on the unfinned surfaces of the experimental setup: these are the radiative and the convective thermal exchanges taking place on the sides and on the back of the insulating case. The losses have been calculated according to the procedures reported in Refs. 7 and 8: an average of 26% of the power in input is dissipated through the insulated case made of fiber thermal material and polystyrene.

### 3.3. Uncertainty and experimental validation

Taking into account the propagation of error for independent variables [7], the uncertainties on the calculation of the heat transfer coefficient and the heat transfer coefficients are found to be lower than  $\pm 8.3\%$  [8]. The uncertainties on the measurements of the fin surface, the fin and the ambient temperatures have been considered in the calculation. In particular, the thermocouple measurements depend on the emissivity of the material, which has been calculated from the reflectivity of the wafer, measured using a spectrometer (Perkimeter Lamda 1050) [8]. On the other hand, calculating the overall fin effectiveness raises the uncertainty up to  $\pm 10.3\%$ , since the uncertainty on the measurements made for the flat surface has to be considered as well, and it accounts for 6.1%.

Some of the overall fin effectivenesses presented in Section 4.2 are obtained by processing data available in graphical format in Refs. 7. In Ref. 7, the dimensions of the fins are clearly stated in the text, whereas the heat transfer coefficients and the temperatures are presented in figures only. These data, necessary to calculate the overall

fin effectiveness, have then been extracted using *Engauge Digitizer 4.1* (M. Mitchell, Engauge Digitizer, <http://digitizer.sourceforge.net>). This is open source software package able to convert graphs into numbers. The authors of Ref. 7 reported an experimental uncertainty of  $\pm 9.4\%$  on their data. Repeating the digitalization twice, a repeatability uncertainty of  $\pm 1.0\%$  is found for the present work. Moreover, the size of the markers in the graphs introduces an uncertainty that corresponds to a maximum  $\pm 3.1\%$ . Overall, the uncertainty on the digitalized heat transfer coefficients rises then to 9.9%.

The present experimental setup has been previously validated in Ref. 8. As a further proof of validation, the Nusselt numbers experimentally obtained in this investigation are compared with those predicted by referenced correlations for both upward and downward orientations. The experimental setup is firstly validated according to the correlation reported in Ref. 19 for upward-facing horizontal square plates in air. The mean heat transfer correlation is expressed as:

$$Nu = 1.05 \cdot Ra^{0.215} \quad (8)$$

where the Rayleigh number is calculated as [19]:

$$Ra = \frac{g \cdot \beta \cdot (T_s - T_a) \cdot W^3 \cdot Pr}{\nu^2} \quad (9)$$

where  $g$  is the gravitational acceleration,  $\beta$  is the volumetric thermal expansion of air,  $Pr$  is the Prandtl number, and  $\nu$  is the kinematic viscosity of air. All the properties of air are extracted from the tables presented by Ref. 16 and are evaluated for an air temperature of  $(T_s + T_a)/2$ , with the exception of the volumetric thermal expansion coefficient, evaluated instead at ambient temperature [7,10]. The numerical equation is found to consistently underestimate the experimental heat transfer coefficients, with an average discrepancy of 19%. This value falls within the  $\pm 20\%$  range reported by the author [19]. It has to be mentioned that considerable uncertainties can take place when natural convection correlations are adopted: a  $\pm 20\%$  uncertainty span is considered as acceptable [10], though in some previous investigations, the differences between experimental and numerical data have been found to amount up to 50% [19].

Secondly, the fin arrays in downward facing orientation are similarly validated using the equation proposed by Ref. 11 for downward-facing horizontal square plates in air:

$$Nu = [1 + 0.24 \cdot e^{(-0.0025L^*)}] \cdot 0.46 \cdot Ra_L^{0.20} \quad (10)$$

where  $Ra_L$  is the Rayleigh number expressed as function of the fin array half length, and  $L^*$  is the characteristic length, defined as:

$$L^* = \frac{L}{2 \cdot \sqrt[3]{\alpha \cdot \nu / g}} \quad (11)$$

where  $\alpha$  is the thermal diffusivity of air.

## 4. Results and discussion

### 4.1. Individual fin effectiveness

Micro-fins achieve lower heat transfer coefficients than flat plates [7]. This is due to the air gaps between the micro-fins which act similarly to an insulating layer [6], reducing the rate of heat transferred per unit of surface on the side walls and the base of the fins. The reduction in heat transfer coefficient can be balanced by the increase in thermal exchanging surface: the variation in heat transfer due to the fins can be measured through the individual and the overall fin effectivenesses.

In order to grant a real benefit, fins are generally required to achieve individual effectivenesses greater than 2 [13]. In this experimental investigation, the individual fin effectiveness ranges between 1.25 and 3.32. In particular,  $\epsilon_f$  is found to be enhanced by low fin thicknesses and high fin spacing. As shown in Fig. 4, the maximum individual fin effectiveness is achieved when the ratio  $t/s$  is minimized, whereas the individual fin effectiveness lowers if  $t/s$  increases. Increased fin spacings favor the thermal exchange because larger air volumes are present between the fins and, this way, air is more likely to exchange heat by convection than by conduction. Thus, in larger fin spacings, more heat is transferred through the walls of the fins, improving the amount of heat globally dissipated by the fins. The same behavior and a similar range of  $\epsilon_f$  are found when the micro-fins are downward orientated: the individual fin effectiveness ranges between 1.19 and 3.30. It means that the effect of the orientation on the individual fin effectiveness can be considered as negligible.

### 4.2. Overall fin effectiveness

The thermal performance of the fin arrays depends on the individual fin effectiveness and on the number of fins. For this reason, the overall fin effectiveness is generally considered a better index to measure the performance of a finned surface than the individual fin effectiveness [16]. Kim et al. [6] measured a heat transfer

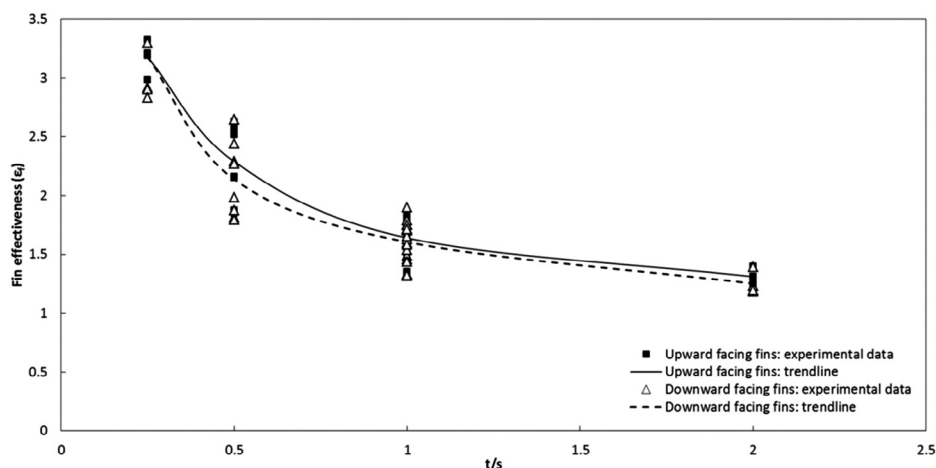


Fig. 4. The behavior of individual effectiveness of the fins depending on the  $t/s$  ratio for upward facing fins (■) and downward facing fins (△).

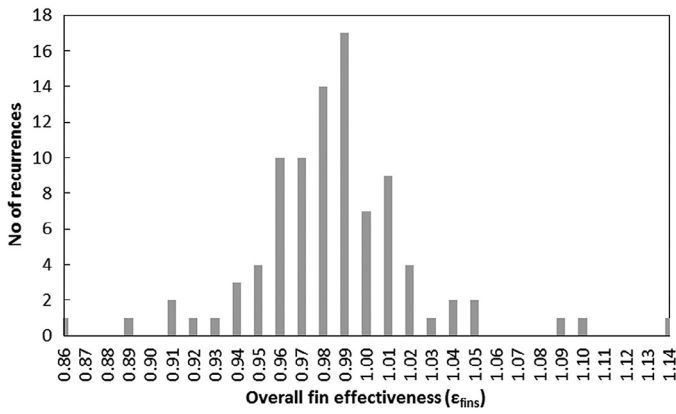


Fig. 5. Number of recurrence per overall fin effectiveness.

enhancement up to 10% after the installation of micro-fins, in agreement with the findings of previous researches on micro-fins in forced flow conditions. The overall fin effectiveness was found to increase when the fin spacing was decreased. This was explained because a limited spacing decreased the volume of air compared to that of the higher-conductive fin material, increasing the overall heat transfer. Within the range considered for that work (s spanned between 30 and 360  $\mu\text{m}$ ), enlarging the fin spacing did not benefit to the convective heat transfer, since air was still mostly acting as a solid conductive layer. For this reason, in those conditions, increasing the volume of the more conductive silicon benefitted to the heat transfer.

Despite the wider spacing considered, the limited enhancement is confirmed in the present experimental investigation, where the overall fin effectiveness is found to range between 0.98 and 1.02. This means that the introduction of micro-fins for natural convection is not necessarily positive for the overall heat transfer. The same results are obtained by analyzing the data reported by Ref. 7, where the overall fin effectiveness ranges between a maximum of 1.14 and a minimum of 0.86. In Fig. 5, a summary of the overall effectivenesses presented in this work and in Ref. 7 is reported. The average overall effectiveness is found as 0.985.

The results shown in Fig. 4 and Fig. 5 are not in contradiction. A good individual fin effectiveness does not necessarily lead to a good overall fin effectiveness. This is due to the spacing between the fins, which, at micro-scale, becomes a particularly critical parameter: a reduced spacing forces the air to thermally behave similarly to a solid, strongly affecting the heat transfer from the base and from the sides of the fins. Therefore, the big discrepancy between the individual fin effectiveness and the overall fin effectiveness remarks the complexity of heat transfer at micro-scale. The dimensions of fins and the spacings between them strongly affect the performance of an array of the fins, whereas it has only a limited effect on the heat transfer of a single fin. For this reason, the overall fin effectiveness is found to be lower than the individual fin effectiveness.

Due to the high viscous forces and to its low thermal conductivity, the air confined in the narrow volumes between two fins behaves as an insulating layer. For this reason, the heat transfer coefficient is found to decrease when micro-fins are added to a flat plane surface [7,8]. The drop in thermal convection can be balanced by the increase in surface: the overall fin effectiveness in natural convection can be as high as 1.14. Although an enhancement

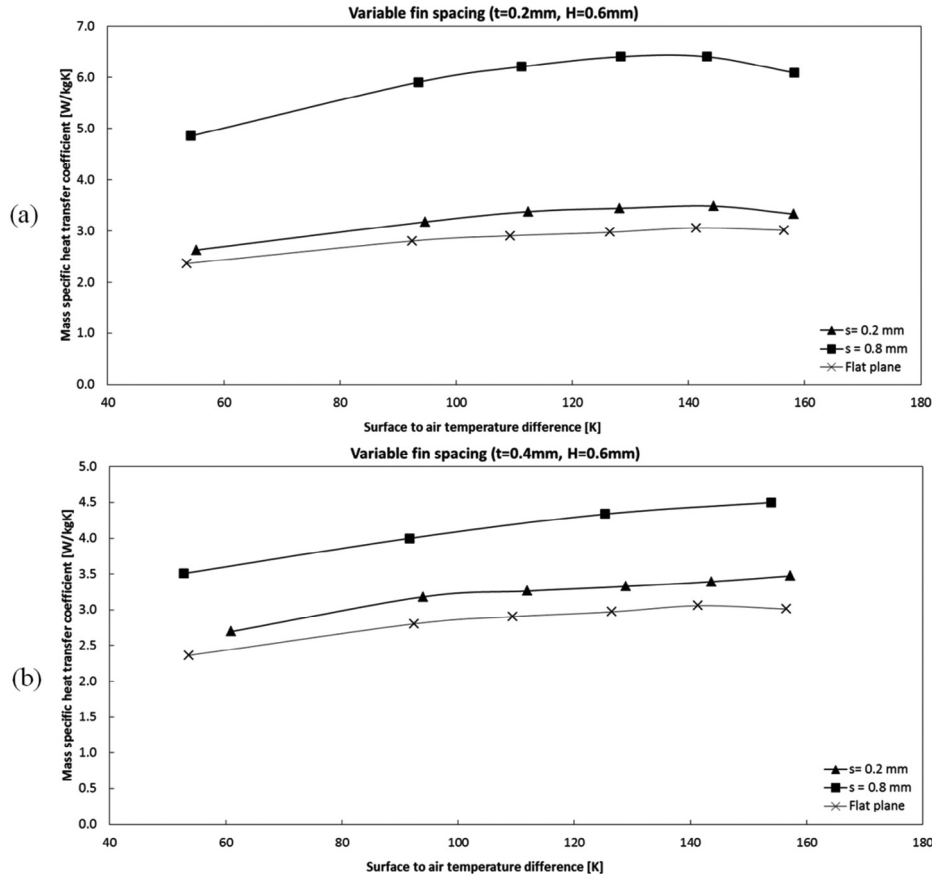


Fig. 6. The effects of fin spacing for upward-facing fin arrays on the mass specific heat transfer coefficient: (a)  $t=0.2\text{ mm}$  and  $H=0.6\text{ mm}$ ; (b)  $t=0.4\text{ mm}$  and  $H=0.6\text{ mm}$ .

in heat transfer coefficient has been proved when the base thickness is decreased and the spacing is increased, conflicting results have been obtained when the overall fin effectiveness is analyzed. The correlations between the overall fin effectiveness and the fin geometries have not been sorted out yet and are essential to design optimized micro-finned heat sinks. It has been clearly shown, instead, that the individual effectiveness of a single fin is enhanced when the fin thickness is lowered and the fin spacing is increased. The micro-fins tested in the present work shows individual effectiveness higher than 3, both in upward and downward facing conditions.

In real applications, a heat sink exchanges heat by both convection and radiation. The heat sink metrics used to investigate thermal performance of fin heat sink generally do not take into account the contribution of radiation [20]. It has already been shown that radiation should not be neglected when a naturally convective micro-finned array is designed [8,9]. Compared to a flat surface, the contribution of radiation in a finned array is influenced by an increased thermal exchanging surface and by some limiting view factors. For this reason, the combined contribution of radiation and convection in micro-fin arrays should be taken into account in future works.

### 4.3. Mass specific heat transfer

The mass specific heat transfer coefficient measures the thermal performances per unit of mass of the heat sink. Even if the traditional heat transfer coefficient is negatively affected, the mass specific heat transfer coefficient is found to be enhanced by the introduction

of fins. It means that the benefit in weight reduction due to the material subtraction is more effective than the change in the thermal performance.

The correlations between the mass specific heat transfer and the fin geometries are not coherent with those valid for the heat transfer coefficients. It is found that the specific mass heat transfer increases when increasing the spacing of the fins (Fig. 6), and/or increasing the height (Fig. 7). The increase in fin height means a drop in the heat sink weight, because the base thickness is decreased.

These results lead to the conclusion that the drop in weight, instead of an enhancement in heat transfer, is the most important benefit obtained by dicing micro-fins on a flat cooling surface. For the same reason, the mass specific heat transfer is found to increase when the fin thickness increases (Fig. 8).

The introduction of micro-fins leads to an improvement up to 50% of mass specific heat transfer coefficient compared to the flat surface. As expected, the maximum enhancements are found in conditions of maximum spacing and minimum thickness. This makes micro-finned heat sinks particularly preferable for moved systems, where the weight is a parameter to be limited, in those applications where the use of an active cooling system is not considered advantageous in terms of costs, volume and/or energy consumption.

The presented results are found to be consistent when the fins are downward oriented: the mass specific heat transfer coefficient increases when the fin spacing increases (Fig. 9), the fin height increases (Fig. 10), and the fin thickness decreases (Fig. 11). Compared to the upward conditions, the mass specific heat transfer coefficients are found to decrease by 13–27% for downward-facing micro-fin arrays.

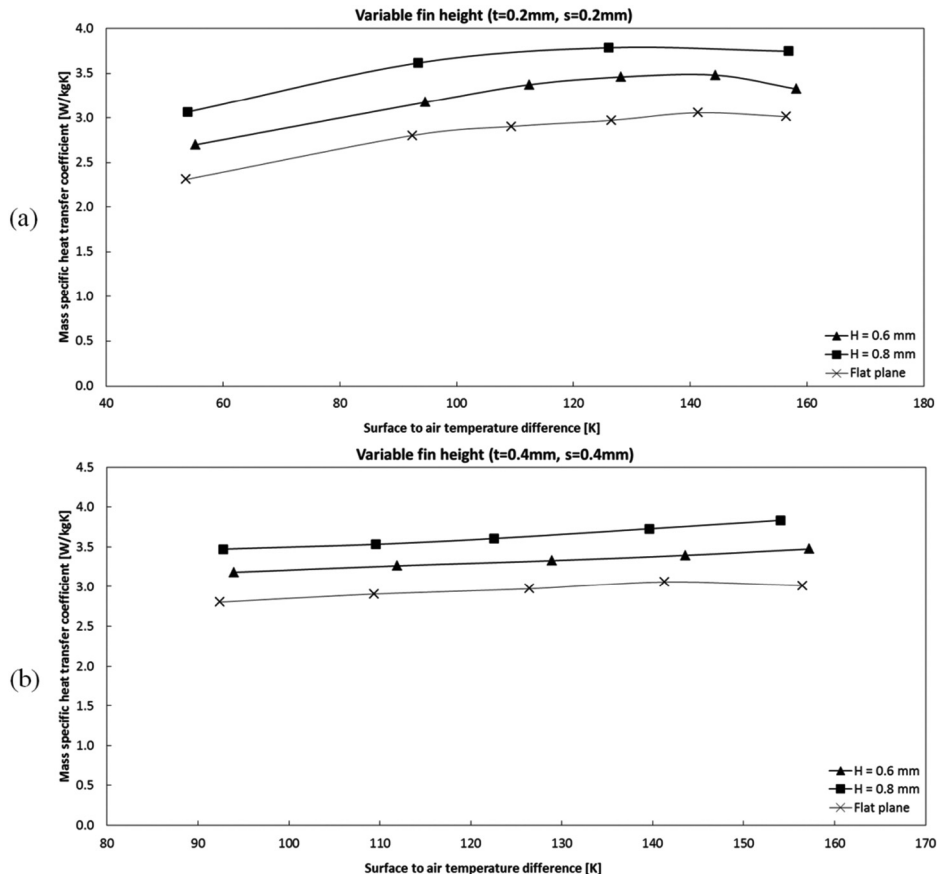


Fig. 7. The effects of fin height for upward-facing fin arrays on the mass specific heat transfer coefficient: (a) t=0.2 mm and s=0.2 mm; (b) t=0.4 mm and s=0.4 mm.

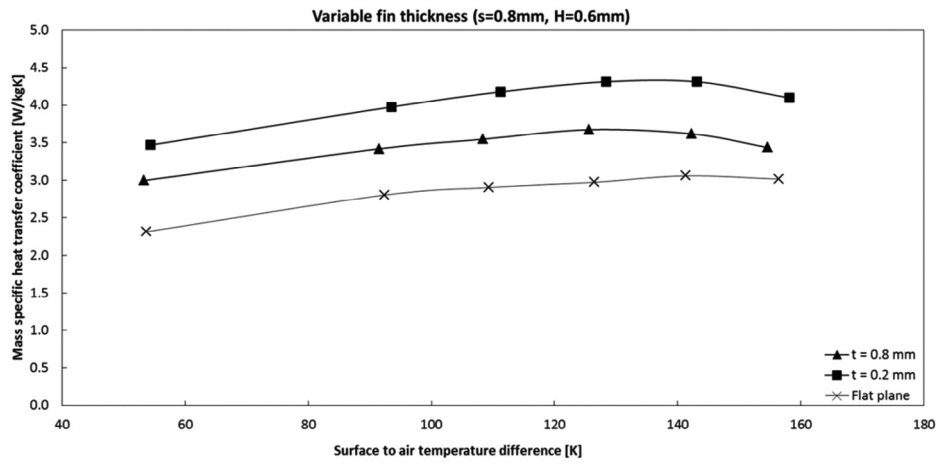


Fig. 8. The effects of fin thickness for upward-facing fin arrays on the mass specific heat transfer coefficient:  $s = 0.8$  mm and  $H = 0.6$  mm.

The results presented in the work can be good guidelines to identify the optimal design of micro-finned heat sinks. Natural convective micro-fins can find application in several fields: they are found to be a suitable passive cooling solution for those systems that require a limited heat sink's weight, such as concentrating photovoltaics.

In this light, more investigations on different geometries, materials and conditions are necessary to fully understand the benefit that can be obtained by naturally convective micro-fins. In particular, in order to make it possible to predict the micro-fin performance in a wider range of applications, a study on natural convective heat

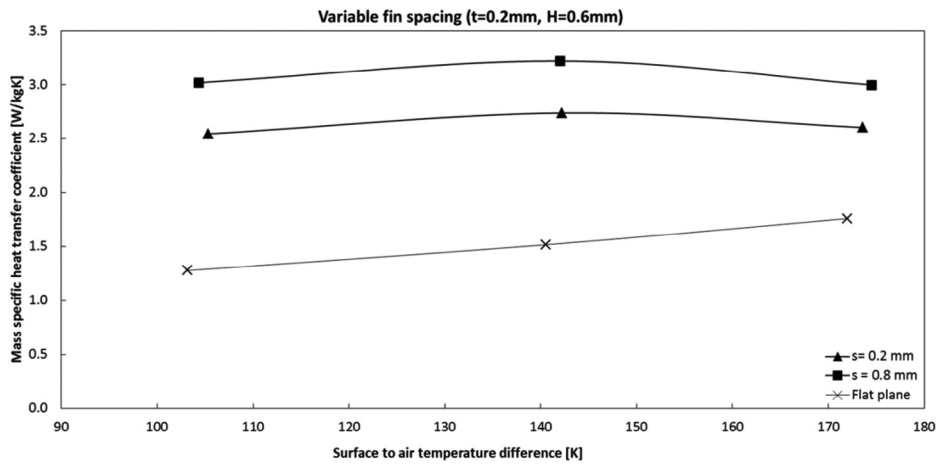


Fig. 9. The effects of fin spacing for downward-facing fin arrays on the mass specific heat transfer coefficient:  $t = 0.2$  mm and  $H = 0.6$  mm.

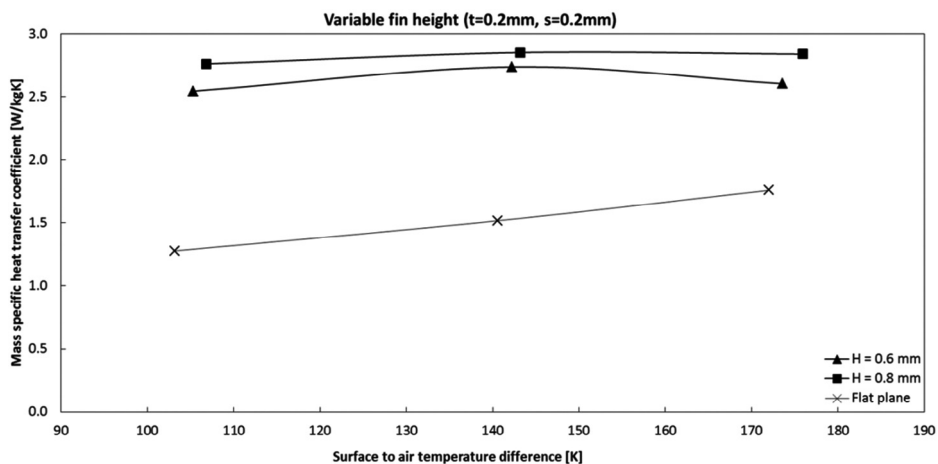


Fig. 10. - The effects of fin height for downward-facing fin arrays on the mass specific heat transfer coefficient:  $t = 0.2$  mm and  $s = 0.2$  mm.

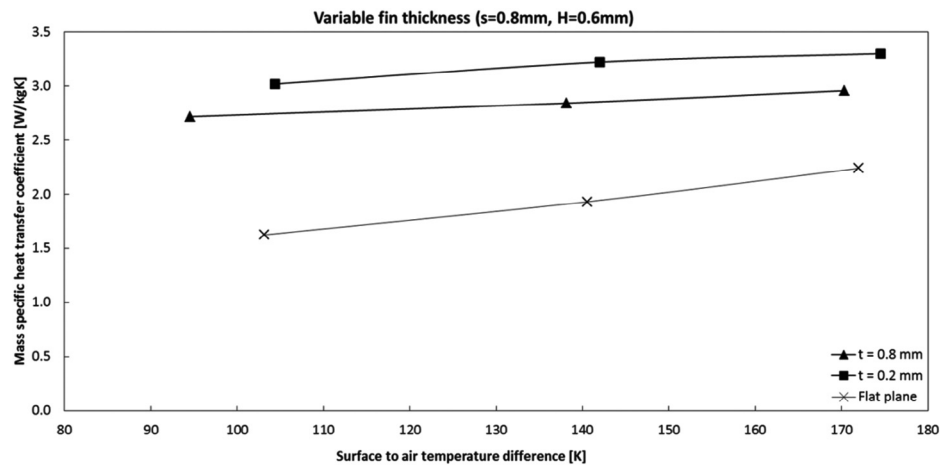


Fig. 11. - The effects of fin thickness for downward-facing fin arrays on the mass specific heat transfer coefficient:  $s = 0.8$  mm and  $H = 0.6$  mm.

transfer of micro-finned arrays for different tilt angles is recommended. The contribution of radiation should not be neglected in future works. Potential mass reductions and heat transfer enhancements are not the only benefits that a heat sink should achieve. Therefore, investigations on the costs, the applicability and the reliability of micro-finned array have to be carried out to make of this technology an effective solution for micro-scaled coolers.

## 5. Conclusions

Natural convective micro-fins are a high-potential, widely explored technology. Only a limited number of studies are available and these are mainly focused on the heat transfer coefficients, a metric that cannot be used for an exhaustive characterization of a heat sink. In this light, the present paper investigated the thermal performance of horizontal micro-fins under natural convective conditions using different heat sink metrics and showed that micro-fins can enhance the heat transfer of a flat surface. The experimental setup used in this work has been described and the results of the investigation have been presented. Micro-fins achieved individual effectiveness higher than 3, benefiting of high spacings and low thicknesses. The overall heat exchange of a surface has been found to be enhanced up to 14% by using an opportunely-designed micro-finned array, whereas in some cases, the micro-fins have been found to lower the thermal performance. Due to the limited variation of overall fin effectiveness and to the non-negligible experimental uncertainty, no clear correlations between geometry and overall fin effectiveness have been found. Further studies need to be carried out to understand how to optimize the design of a micro-fins array for natural convective applications. For the first time, the mass specific heat transfer coefficient has been investigated to understand the effects of the micro-fins on the weight of the heat sink. The introduction of micro-fins is found to be particularly beneficial to the material usage: a variation up to 50% in mass specific heat transfer has been observed when micro-fins are used. The orientation of the arrays did not show any qualitative effect on the correlations between the fin geometries and the thermal performance. Downward facing fins did have a lower heat transfer management, but showed thermal behaviors similar to those of upward facing fins.

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## Nomenclature

$A_f$	Cross-sectional area of a fin [ $m^2$ ]
$F_{i-a}$	View factors between the surfaces $i$ and the ambient
$g$	Standard gravitational acceleration [ $m/s^2$ ]
$H$	Height of a fin [ $m$ ]
$h_{fins}$	Heat transfer coefficient of the micro-finned surface [ $W/m^2 K$ ]
$h_{flat}$	Heat transfer coefficient of the unfinned surface [ $W/m^2 K$ ]
$h_m$	Mass specific heat transfer coefficient [ $W/kg K$ ]
$L$	Length of the fins [ $m$ ]
$L^*$	Characteristic length
$p$	Pitch of the fin array [ $m$ ]
$Q_f$	Heat dissipated by a single fin through convection [ $W$ ]
$Q_{fins}$	Heat dissipated by fins through convection [ $W$ ]
$Q_{flat}$	Heat dissipated by the unfinned surface through convection [ $W$ ]
$Q_{in}$	Heat supplied by the DC supply [ $W$ ]
$Q_{losses}$	Heat dissipated by the surfaces other than the fins [ $W$ ]
$Q_r$	Heat dissipated by fins through radiation [ $W$ ]
$s$	Spacing between adjacent fins [ $m$ ]
$S_{fins}$	Area of the finned surface [ $m^2$ ]
$S_i$	Area of the $i$ -surface of the fins [ $m^2$ ]
$t$	Thickness of a fin [ $m$ ]
$T_a$	Ambient temperature [ $K$ ]
$tb$	Thickness of the fin array's base [ $m$ ]
$T_{fins}$	Temperature of the finned surface [ $K$ ]
$T_{flat}$	Temperature of the unfinned surface [ $K$ ]
$T_s$	Surface's temperature [ $K$ ]
$V_{fins}$	Volume of the finned surface [ $m^3$ ]
$W$	Width of the fin array [ $m$ ]

## Greek symbols

$\alpha$	Thermal diffusivity [ $m^2/s$ ]
$\beta$	Coefficient of volumetric expansion [ $1/K$ ]
$\epsilon$	Emissivity
$\epsilon_f$	Individual fin effectiveness
$\epsilon_{fins}$	Overall fin effectiveness
$\nu$	Kinematic viscosity of air [ $m^2/s$ ]
$\rho$	Density [ $kg/m^3$ ]
$\sigma$	Stefan–Boltzmann constant [ $W/m^2 K^4$ ]

## Dimensionless numbers

$Nu$	Nusselt number
$Pr$	Prandtl number
$Ra$	Rayleigh number



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