INTERNATIONAL CONFERENCE ON POLYGENERATION
TECHNOLOGIES for SMART CITIES
ICP 2015
Feb 18-20, 2015

Plate Micro-Fins in Natural Convection: Experimental Study on Thermal Effectiveness and Mass Usage

Leonardo Micheli[1], KS Reddy[2], Tapas K. Mallick[1]
[1]Environment and Sustainability Institute; University of Exeter, Penryn Campus, Penryn, Cornwall TR10 9FE, UK
[2]Heat Transfer and Thermal Power Laboratory, Department of Mechanical Engineering, Indian Institute of Technology Madras, Chennai 600 036, India

*Corresponding author: lm409@exeter.ac.uk

ABSTRACT: Every year, micro-technologies are gaining more attention among researchers and industries. Although they are already applied for cooling purposes in several installations, the researches on the thermal performance of micro-fins in natural convective conditions are yet limited. The correlations between heat transfer coefficients and geometry have already been investigated. The present study merges the results of an original experimental investigation with the data available in literature, in order to give an overview of the behavior of micro-fins in terms of different heat sink metrics: the fin effectiveness and the mass specific heat transfer coefficient. The introduction of micro-fins is found not to be always beneficial in terms of heat transfer, although always positive in terms of the material usage and can be considered advantageous in those applications that requires a minimized weight of the heat sinks.

Keywords: natural convection, micro fins, effectiveness, thermal resistance

Introduction

Fins are widely used to enhance the heat transfer from a surface 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 conditions act as passive coolers: they do not require input of mechanical or electrical power, because of the exploitation of natural laws [1]. Compared to active cooling, where an external energy is needed in input, passive cooling is considered more reliable and able to reduce the probability of cooling failures [2].

Industries and consumers are always after the production of more efficient, more compact and less expensive products. In this light, micro-technologies have gained much interest in the last decades, because of the faster performance achieved and the limited space and material required compared to macro-scale solutions. Micro-fins arrays in forced convection have been extensively investigated. Despite of that, the research in literature on naturally convective micro-fins is still limited. Kim et al. [3] investigated vertically orientated micro-fins and demonstrated the impossibility of using the macro-fin heat transfer correlations for micro-scale systems. Mahmoud et al. [4] firstly showed the correlation between the fin geometry and the heat transfer coefficients. Shokouhmand H. and Ahmadpour A. [5] numerically demonstrated that the contribution of the radiative exchange cannot be neglected in a micro-fins array.

All the previous researches on micro-fins in natural convection mainly focused on the heat transfer coefficients. In the real world applications, engineers and systems designers look for thermal resistance, compactness, weight and cost of the heat sinks. The present work uses the data of an experimental investigation to analyze additional heat sinks metrics, in order to effectively study for the first time the thermal enhancement due to the introduction of micro-fins. The thermal performances have been measured in terms of heat transfer coefficients, fin effectiveness and mass specific heat transfer. In order to contribute to the optimization of the design of micro-finned arrays, the effects of the geometry on the heat sinks metrics has been explored and reported.

Heat sink metrics

The previous papers [3,4] investigated the heat transfer coefficients of micro-fins arrays (h_{fins}), as follow:

$$h_{fins} = \frac{Q_{fins}}{A_{fins} \cdot (T_{fins} - T_{amb})} \qquad (1)$$

where Q_{fins} is the heat dissipated through the fins by convection, A_{fins} the area of the finned surface, T_{fins} the fins temperature, and T_{amb} is the ambient temperature. The aim of a fin is to increase the heat transfer from a surface to a fluid. 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 overall thermal performance of the whole 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 fin effectiveness, is generally used [6]:

$$\varepsilon_{fins} = \frac{Q_{fins}}{Q_{flat}} \qquad (2)$$

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

Micro-fins are usually obtained through material subtractions: along with the enhancing of the heat transfer, they reduce the mass of the heat sink. This feature becomes particularly important in portable or tracked systems, such as the concentrating photovoltaics (CPV), where a reduced weight means a reduced load for the tracker. The mass specific heat transfer coefficient measures the effectiveness with which fin material is utilized in the promotion of heat transfer [7] and is expressed as:

$$h_m = \frac{Q_{fins}}{\rho \cdot V_{fins} \cdot (T_{fins} - T_{amb})} \qquad (3)$$

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

Experimental investigation

A micro-fin is an extended surface, where at least one of the dimensions shown in Figure 1 falls in the range of the micro-scale. In the present investigations, the different fin arrays are obtained using a dicing machine and the range of dimensions studied so far is reported in Table 1.

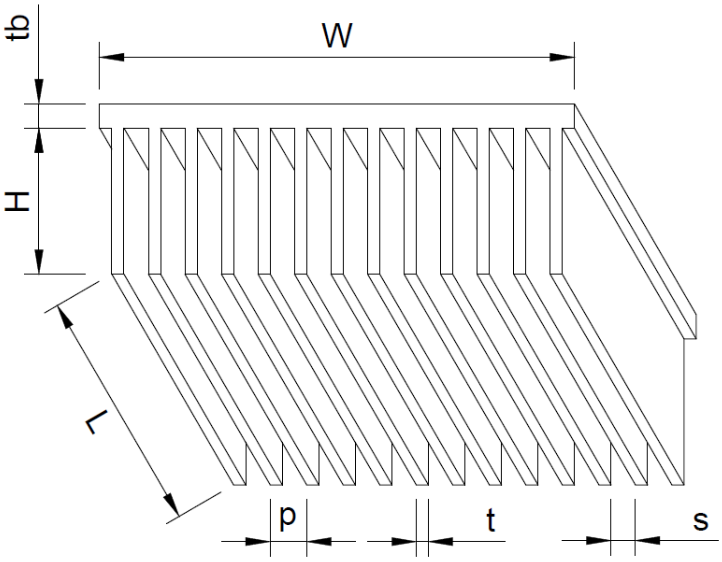

Figure 1 – Description of the fin dimensions.

Table 1 - Dimensions of the fin arrays

Materials	Width (W)	Length (L)	Height (h)	Pitch (p)	Thickness (t)	Spacing (s)	Base thickness (tb)
Silicon	50 mm	50 mm	0.6, 0.8, 1 mm	0.4-1.6 mm	0.2, 0.4, 0.8 mm	0.2, 0.4, 0.8 mm	0.4, 0.6, 0.8 mm

The heat is generated by 10W electrical heaters (Omega KHLV-202/2.5), regulated using a DC power supply (Weir 413D). The heaters are bonded to the silicon wafers bonded through a thin conductive adhesive (3M tape 966, 0.18 W/mK). The power in input is calculated by multiplying the input voltage (V_{DC}) and the current (I_{DC}), 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 fibre thermal material (0.05W/mK), covered on the back by a 1cm-thickpolystyrene block (0.03 W/mK). The power dissipated by convection from the fins array is calculated after the radiated heat transfer (Q_r) and the losses that happen on the back and the sides of the samples (Q_{losses}):

$$Q_{fins} = Q_{in} - Q_r - Q_{losses} \quad (4)$$

The radiative component is estimated by the Stefan-Boltzmann equation:

$$Q_r = \sum_i \varepsilon \cdot \sigma \cdot A_i \cdot F_{i,k} \cdot (T_w^4 - T_a^4) \quad (5)$$

where ε is the emissivity of fin material, σ is the Stefan-Boltzmann constant ($5.67 \cdot 10^{-8}$ W m^{-2} K^{-4}), A_i is the area of the correspondent i-surface of the fins, $F_{i,k}$ are the view factors between the surfaces i and k, and T_w and T_a are respectively the surface and air temperatures. The Q_{losses} are due to the heat dissipation happening on the unfinned surfaces of the assembly: these are the radiative and the convective thermal exchanges on the sides and on the back of the structure. All the calculations are usually conducted considering an air temperature of $(T_{surface}+T_{amb})/2$, with the exception of the thermal expansion, evaluated at ambient temperature [8]. The radiative exchange is found to contribute up to 56% to the overall heat dissipation. Measures are taken at steady state conditions, using thermocouples and IR cameras. Each Test has been conducted three times and the average value of the outputs is considered.

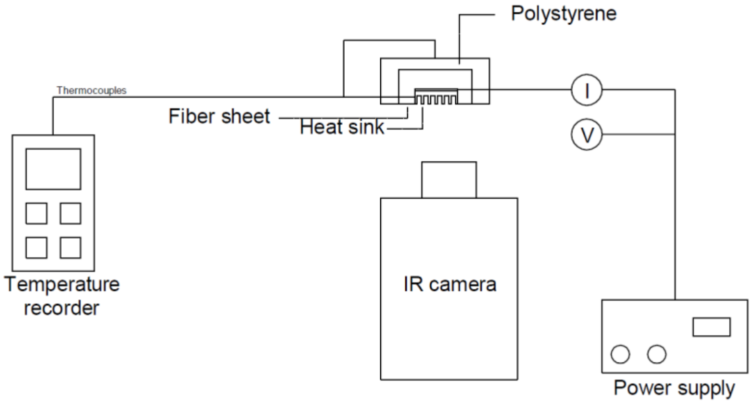

Figure 2 – Representation of the experimental apparatus

The uncertainties in the heat transfer coefficient measurement are usually calculated using the propagation of error for independent variables:

$$\frac{Uh_{fins}}{h_{fins}} = \sqrt{\left(\frac{UQ_{fins}}{Q_{fins}}\right)^2 + \left(\frac{UA_{fins}}{A_{fins}}\right)^2 + \left(\frac{UT_{fins}}{T_{fins}-T_{amb}}\right)^2 + \left(\frac{UT_{amb}}{T_{fins}-T_{amb}}\right)^2} \qquad (6)$$

where the uncertainties are indicated with the prefix "U". A maximum Uh_{fins} of ±8.25% was found. The uncertainties for the additional parameters reported in the present works have been calculated similarly.

Some of the results presented in this work are obtained by processing data available in graphical format in [4]. Whereas not explicit in the paper, the data have been extracted using *Engauge Digitizer 4.1* (M. Mitchell, Engauge Digitizer, http://digitizer.sourceforge.net). This is open source software that allows converting graphs into numbers. The authors of [4] reported an experimental uncertainty of ±9.4%. Repeating the digitalization twice, a repeatability uncertainty of ±1.0% has been found. Moreover, the size of the marker introduces an uncertainty that corresponds to a maximum ±3.1%. Overall, the uncertainty on the heat transfer coefficients rises then to 9.9%.

Results and discussion

a) Heat transfer coefficient

The heat transfer coefficient is directly dependent on the exchanging surface. For this reason, all the investigations reported a lower heat transfer coefficient for finned surfaces compared to the flat plates. As explained by [4], this is happening because heat transfer coefficients is inversely proportional to the area of the exchanging surface, which increases with the introduction of fins. Moreover, it has been demonstrated that the air in the micro channels between the fins act similarly to an insulating layer, reducing the rate of heat transferred per unit of surface on the side walls and the base of the fins. The heat transfer for horizontal fins in upward facing configuration is enhanced when the fin spacing increases (Figure 3), and the fin height decreases (Figure 4).

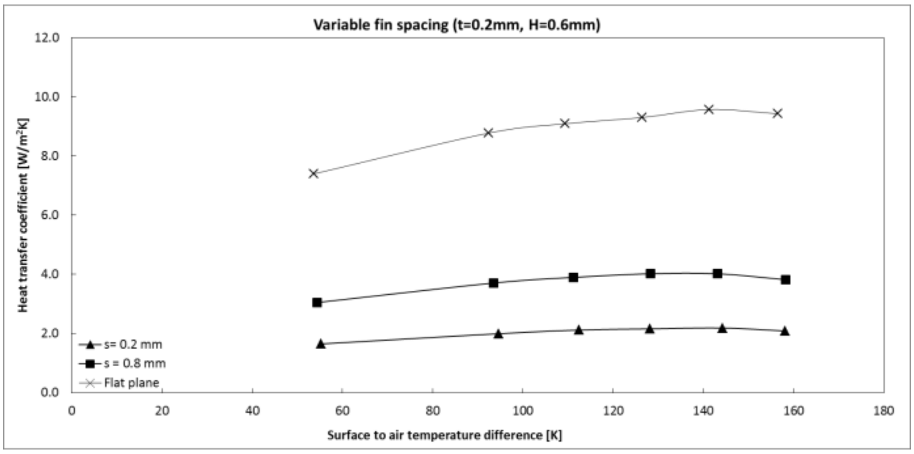

Figure 3 - The effects of the fin spacing for horizontal fin arrays on the heat transfer coefficient.

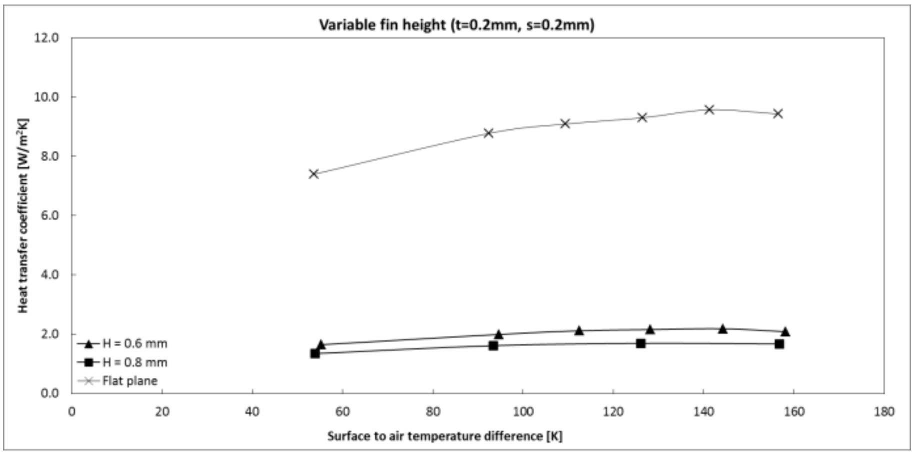

Figure 4 - The effects of the fin height for horizontal fin arrays on the heat transfer coefficient.

b) Fin effectiveness

Kim et al. [9] measured a heat transfer enhancement up to the 10% due to the installation of fins, in agreement with the findings of previous researches on micro-fins in forced flow conditions. The fin effectiveness is found to increase when the fin spacing is decreased. This is explained because a limited spacing decreases the volume of air compared to that of the higher-conductive fin material, increasing the overall thermal conductance.
The limited enhancement has been confirmed in the present experimental investigation, where the 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 have been obtained by analyzing the data reported by [4], where the fin effectiveness ranges between a maximum of 1.14 and a minimum of 0.86. In Figure 5, a summary of the effectivenesses presented in this work and in [4] is reported. The average effectiveness is 0.985.
No clear correlations between geometry and fin effectiveness has been found. Further studies need to be conducted to understand how to optimize the design of a micro-fins array for natural conductive applications.

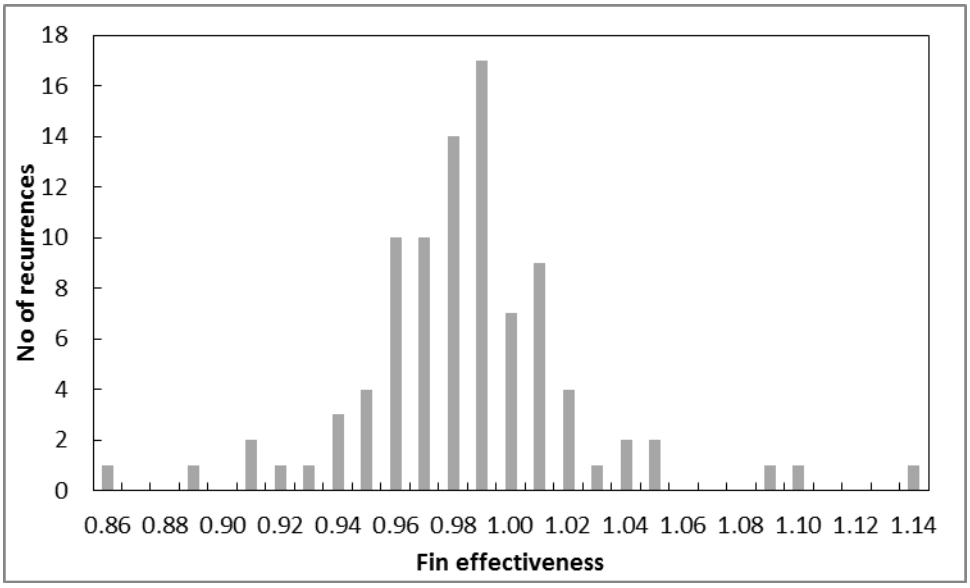

Figure 5 – Number of recurrence per fin effectiveness

c) *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 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 the one reported for the heat transfer coefficients. It was found that the specific mass heat transfer increases when increasing the spacing of the fins (Figure 6), and/or increasing the height (Figure 7). The increase in weight means a drop in the heat sink weight, because the base thickness is decreased. These results confirm 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. This makes micro-finned heat sink 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.

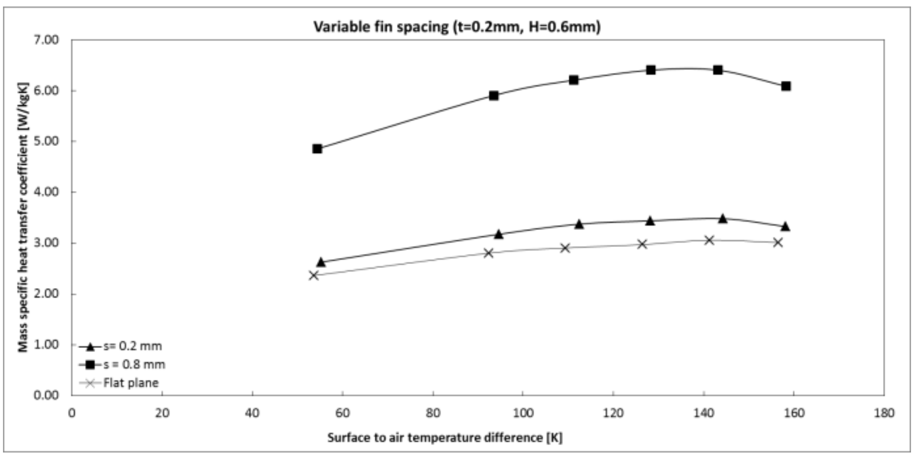

Figure 6 – The effects of the fin spacing for horizontal fin arrays on the mass specific heat transfer coefficient.

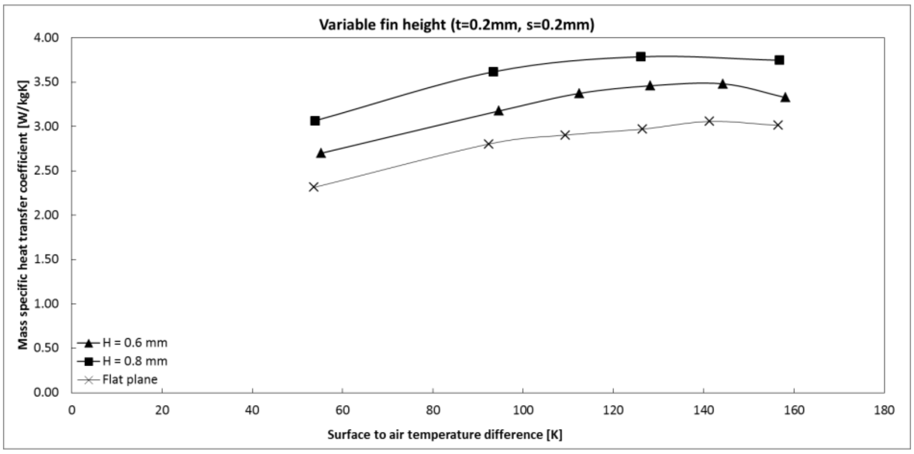

Figure 7 - The effects of the fin height for horizontal fin arrays on the mass specific heat transfer coefficient.

Conclusions

Heat transfer from extended surface is more complex at micro-scale than in macro-scale conditions. This is due to the fact that, in micro-fins, the conduction is dominant over the natural convection. When confined in the narrow space between two fins, the air tends to behave as an insulating layer, because of the low thermal conduction and the viscous forces higher than the buoyancy. For this reason, a reduction in heat transfer coefficient is registered when micro-fins are added to a flat plane surface. The drop in thermal convection can be balanced by the increase in surface: the effectiveness of fins in natural convection can be as high as 1.14, but the correlation between it and the fin geometry has yet to be sorted out. Although an enhancement 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 fin effectiveness is analyzed. The introduction of fins gives, in any case, a benefit related to the material usage: as resumed in Table 2, when referred to the unit of mass, a finned surface has better thermal performance than the flat plate. For this reason, they are particularly appealable for those applications in which the weight of the heat sink needs to be minimized, such as the tracked concentrating photovoltaic systems.

Table 2 – Summary of the experimental outcomes. The mass specific heat transfer coefficients are reported for different power inputs.

	H [mm]	p [mm]	t [mm]	s [mm]	t_b [mm]	Surface [$m^2 * 10^{-3}$]	Weight [$kg * 10^{-3}$]	h_m [W/kgK]			
								at 10W	at 7.5W	at 5W	at 2.5W
Flat surface						2.5	7.8	3.0	3.0	2.8	2.3
Fin #1	0.6	1.6	0.8	0.8	0.8	4.4	6.0	3.4	3.7	3.4	3.0
Fin #2	0.6	1.2	0.4	0.8	0.8	4.9	5.6	3.9	4.2	3.8	3.4
Fin #3	0.6	1.0	0.2	0.8	0.8	5.5	5.1	4.1	4.3	4.0	3.5
Fin #4	0.6	0.4	0.2	0.2	0.8	9.7	6.0	3.3	3.4	3.2	2.6
Fin #5	0.8	0.4	0.2	0.2	0.6	12.4	5.6	3.7	3.8	3.6	3.1

Acknowledgments

The financial support provided by the EPSRC-DST through the BioCPV project is duly acknowledged.

References

[1] L. Micheli, N. Sarmah, X. Luo, K.S. Reddy, T.K. Mallick, Opportunities and challenges in micro- and nano-technologies for concentrating photovoltaic cooling: A review, Renew. Sustain. Energy Rev. 20 (2013) 595–610. doi:10.1016/j.rser.2012.11.051.

[2] Y.-S. Tseng, H.-H. Fu, T.-C. Hung, B.-S. Pei, An optimal parametric design to improve chip cooling, Appl. Therm. Eng. 27 (2007) 1823–1831. doi:10.1016/j.applthermaleng.2007.01.012.

[3] J.S. Kim, B.K. Park, J.S. Lee, Natural Convection Heat Transfer Around Microfin Arrays, Exp. Heat Transf. 21 (2008) 55–72. doi:10.1080/08916150701647835.

[4] S. Mahmoud, R. Al-Dadah, D.K. Aspinwall, S.L. Soo, H. Hemida, Effect of micro fin geometry on natural convection heat transfer of horizontal microstructures, Appl. Therm. Eng. 31 (2011) 627–633. doi:10.1016/j.applthermaleng.2010.09.017.

[5] H. Shokouhmand, A. Ahmadpour, Heat Transfer from a Micro Fin Array Heat Sink by Natural Convection and Radiation under Slip Flow Regime, in: Proc. World Congr. Eng., 2010.

[6] P. Razelos, A Critical Review of Extended Surface Heat Transfer, Heat Transf. Eng. 24 (2010) 11–28. doi:10.1080/714044411.

[7] A. Bar-Cohen, M. Iyengar, A.D. Kraus, Design of Optimum Plate-Fin Natural Convective Heat Sinks, J. Electron. Packag. 125 (2003) 208. doi:10.1115/1.1568361.

[8] N.V. Suryanarayana, Engineering Heat Transfer, West Publishing Company, 1995.

[9] J. Go, S. Kim, G. Lim, H. Yun, J. Lee, Heat transfer enhancement using flow-induced vibration of a microfin array, Sensors Actuators A …. 90 (2001) 232–239.