27th ABCM International Congress of Mechanical Engineering
December 04-08, 2023, Florianópolis-SC, Brazil

COB-2023-0852

HEAT TRANSFER CORRELATION FOR THERMAL MANAGEMENT OF ELECTRONICS USING FINNED HEAT SINKS UNDER FORCED CONVECTION

Gustavo Paul Exel
Aron Martins Ferreira Milagres
Vitor de Melo Braga
Rafaela Garcia
Cesar J. Deschamps
POLO Research Laboratories
Universidade Federal de Santa Catarina
88040-900, Florianópolis, SC, Brazil
gustavo.exel@polo.ufsc.br
aron.milagres@polo.ufsc.br
vitor.braga@polo.ufsc.br
rafaela.garcia@polo.ufsc.br
deschamps@polo.ufsc.br

Abstract. *Effective thermal management and heat dissipation are crucial in the design of electronic systems, especially in high-performance applications. Heat sinks are passive cooling devices designed to increase the surface area of the component, which allows for more efficient heat transfer to the surrounding air. While Computational Fluid Dynamics (CFD) is widely used for thermal analysis in engineering applications, its use in design optimization routines is limited by its high computational cost. Therefore, thermal network models are frequently used as an alternative for thermal analysis, especially in the early design stages, but convective heat transfer coefficients are required as boundary conditions. This study reports the development of a correlation with the assistance of CFD for the thermal simulation of finned heat sinks of electronics under forced convection, considering wider ranges of Reynolds numbers and heat sink dimensions than available in current correlations. The proposed correlation was implemented in the thermal network model to predict the temperature of an electronic component, showing good agreement with the results of a full CFD model. By accurately predicting the temperature distribution on printed circuit boards (PCBs) using advanced lumped models, hardware engineers can optimize their designs for more efficient heat dissipation, resulting in significant improvements in both performance and reliability. This is especially crucial in high-performance systems, where heat dissipation can often be a limiting factor in achieving optimal performance.*

Keywords: *Thermal management, Printed Circuit Board, Heat transfer correlation, CFD*

1. INTRODUCTION

Predicting the temperature distribution in printed circuit boards (PCBs) is a challenge in electronics thermal management, to ensure that the operating temperature is within the safety limits. Although computational fluid dynamics (CFD) models can be used to analyze the temperature field, their high computational cost makes thermal network models a more cost-effective alternative. However, these simplified models require heat transfer correlations to estimate the temperatures of electronic components.

As indicated by Murshed and Nieto de Castro (2017), several methods are employed for cooling of electronic components, including heat pipes, microchannels, thermoelectric cooling, and heat pumps. Nevertheless, the conventional approaches primarily consist of heat sinks and fans. Heat sinks are fabricated using heat-conductive materials like aluminum or copper and are affixed to the components to increase the surface area for heat transfer, usually using forced convection to increase heat transfer. The analysis presented herein is focused on heat transfer in finned heat sinks under forced convection.

Significant studies have been directed at heat transfer correlations for heat sinks under forced convection. These studies have developed correlations using analytical approaches (Teertstra *et al*., 1999), experimental investigations (Naik *et al*., 1987; Jonsson *et al*., 2001; Kim *et al*., 2008), and numerical simulations (Sato *et al*., 2020). These works established correlations for Nusselt number as a function of Reynolds number for different heat sink geometries. However, there is still a need for correlations applicable to a wider range of geometries than those available in the literature in order to accommodate various applications of electronic cooling.

This article presents the development of a correlation to estimate the heat transfer coefficient in heat sinks under forced convection based on the results of a CFD simulation model. The developed correlation is then applied to a thermal network model of heat transfer in electronic components.

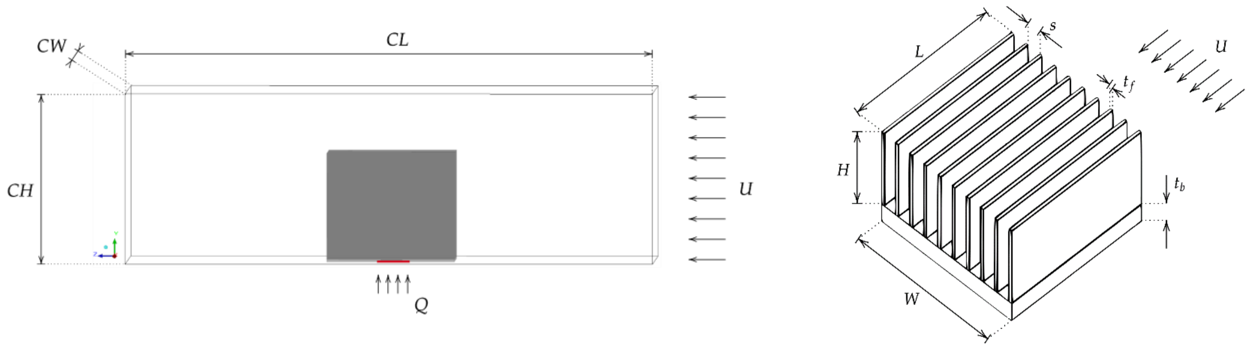
2. METODOLOGY

2.1 CFD model

The commercial code ANSYS Icepak was employed in the numerical simulations. A first-order upwind differencing scheme (UDS) was used due to its lower computational cost when compared with a second-order UDS. A sensitivity analysis regarding the results obtained with both schemes showed differences smaller than 0.6 °C for the surface temperature. It was also verified that the variation of the air thermophysical properties with temperature does not significantly affect the temperature distribution. The results obtained with and without considering variable thermophysical properties revealed differences within 10% for the Nusselt number. Therefore, the Boussinesq approximation was applied in all simulations. Thermal radiation heat transfer brought about a maximum difference of less than 1°C in the results and for this reason was neglected. The k-ω SST turbulence model was selected because its predictions were found in close agreement with correlations available in the literature.

The convergence and stop criteria were defined based on temperature and velocity probes located at the base of the heat sink and between channels, respectively. The simulation was stopped when the difference between the results of consecutive iterations was smaller than 0.004°C for the temperature probe and smaller than 0.001 m/s for the velocity probe. The final residuals for the worst-case scenario were 10^{-2} (continuity), 10^{-4} (velocity, free flow direction), 10^{-5} (velocity, vertical direction), 10^{-4} (velocity, transversal direction), 10^{-8} (energy), 10^{-3} (turbulent kinetic energy), and 10^{-3} (dissipation ω).

The physical domain of the simulations is illustrated in Figure 1a, where an aluminum heat sink is placed inside a wind tunnel with dimensions CL×CW×CH, ranging from 30 to 300mm. Velocity boundary conditions were applied at the inlet, pressure at the outlet, and no-slip condition at the walls. The air enters the wind tunnel at a temperature of 20°C. At the bottom of the heat sink, a heat flux of 0.1 W/mm^2 is prescribed over an area of 10 mm × 10 mm, resulting in a total heat generation of 10 W. This value was considered constant in all simulations. To investigate the influence of the heat sink geometry and velocity, the parameters illustrated in Figure 1b were varied in the simulations, following the range of values presented in Table 1.



(a) (b)

Figure 1 – Dimensions of (a) the simulation physical domain and (b) of the heat sink.

The domain dimensions (length, width and height) are represented by CL, CW and CH respectively, the inlet velocity by U, and the heat generation under the heat sink by Q. In turn, the heat sink dimensions (length, width and height) are denoted by L, W and H, the base height by t_b, the fin width by t_f, and the fin spacing by s.

Table 1 - Range of variation for velocity and for the heat sink geometric parameters.

U [m/s]	W [mm]	H / L	L / W	t_f [mm]	t_f / s
1	20	0,33	0,5	1	0,2
2	60	0,67	0,67	2	0,35
3	100	1	1	3	0,5
			1,5		
			2		

The heat transfer under forced convection is characterized by the Nusselt and Reynolds numbers, which can extend the applicability of the correlation to other fluids other than air used in our simulations. The Nusselt number is defined as

$$Nu_L = \frac{hL}{k} \tag{1}$$

where h is the convective heat transfer coefficient, L is the length of the heat sink, and k is the fluid thermal conductivity.
The Reynolds number is defined as

$$Re_{d_h} = \frac{Ud_h}{\nu} \tag{2}$$

where ν is the kinematic viscosity, U is the velocity, and d_h is the hydraulic diameter, defined by

$$d_h = \frac{4\,Hs}{2H + s} \tag{3}$$

where H is the height of the heat sink and s is the spacing between the fins.

2.2 Thermal Network Model

The thermal network model employed for simulation in this study closely follows the model introduced by Milagres (2021). The board with dimensions of 100 × 100 × 1.5 mm is discretized into smaller elements (20 × 20 mm), as depicted in Figure 2a. At the center of the board lies an electronic component with dimensions of 20 × 20 × 2 mm. The thermal conduction resistances of both the board and the electronic component were estimated as $R_c = l/kA$, where l represents the distance between two nodes, k is the thermal conductivity, and A is the cross-sectional area. For the specific region of the board containing the electronic component, heat conduction is modeled by a spreading thermal resistance denoted as $R_s = 0.3l/kA$. The thermal conductivity of the board is assumed to be 20 W/m·K.

The convective thermal resistances are shown in Figure 2b and are given by $R_h = 1/hA$, where h is the convective coefficient and A is the heat transfer area. The ambient temperature is 20°C. The convective coefficient for the board was estimated using the following correlation:

$$Nu_x = 0.68\,Re_L^{1/2}\,Pr^{1/3} \tag{4}$$

where Re_L is the Reynolds number and Pr is the Prandtl number. For the electronic component, the convective heat transfer coefficient was estimated using the correlation developed in the present article for the heat sink. The analysis using the thermal network model considered a heat sink with the following dimensions: width (W) and length (L) of 40 mm, height (H) of 23 mm, bottom thickness (t_b) of 2 mm, top thickness (t_f) of 1 mm, and spacing (s) of 3.33 mm. The simulations did not consider radiative heat transfer.

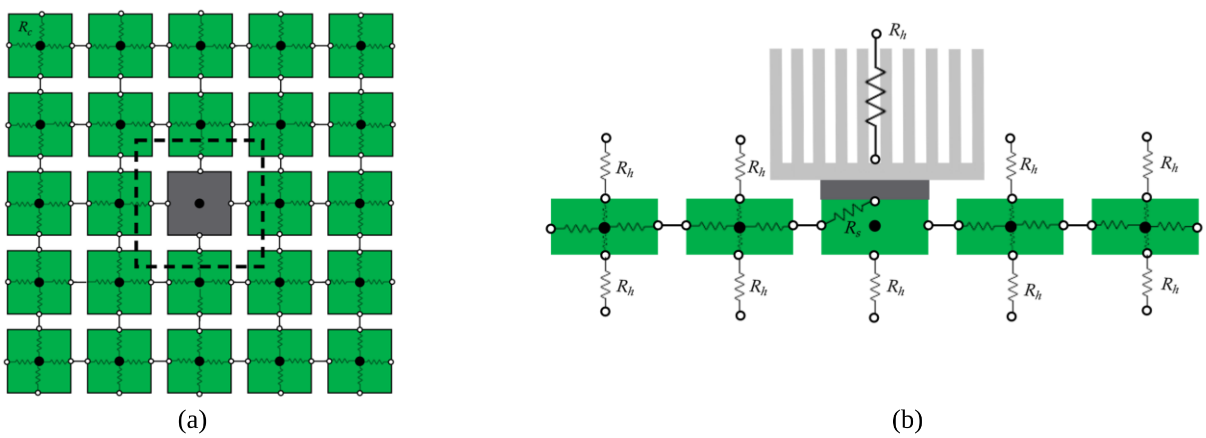


Figure 2 – (a) Discretization of the thermal network model for the board and (b) front view of the heat sink.

3. RESULTS

3.1 Mesh independence analysis

As shown in Figure 3, the discretization mesh used in the simulations had the highest refinement in the heat sink and its surroundings. A study was conducted to assess the result's independence regarding mesh refinement. In this respect, the sensitivity analysis considered the maximum element size in the free stream region (Δx_o, Δy_o and Δz_o), the maximum element size in the vicinity of the heat sink (Δy_i and Δz_i), as well as the mesh growth parameters in the transverse direction, such as the minimum number of elements on each fin (MEOE - Minimum Elements on Edge) and the size ratio increment from refinement on the fin (MSR - Maximum Size Ratio). The maximum size of the element near the heat sink and in the transverse direction (Δx_i) was not included in the analysis since the refinement in this direction is mainly controlled by the MEOE and MSR parameters.

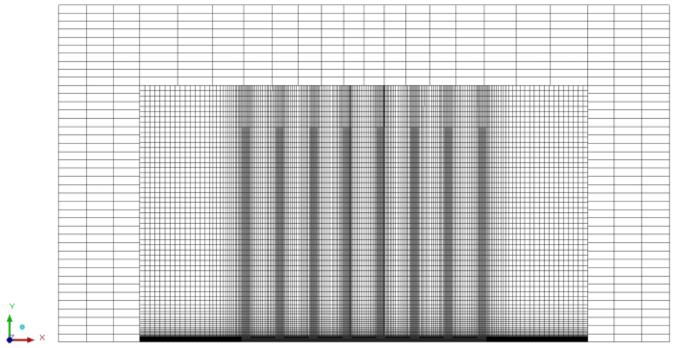


Figure 3 – Cross-sectional view of the generated cartesian computational mesh for the heat sink.

The influence of discretization on the average temperature of the heat sink is depicted in Figure 4. The mesh was refined until the surface temperature varied less than 0.1°C in comparison to the previous mesh. Consequently, the second-to-last configuration was the optimal choice, striking a balance between computational cost and accuracy. The parameters selected from this analysis were as follows: $\Delta x_o = 10$ mm, $\Delta y_o = 1$ mm, $\Delta z_o = 8$ mm, $MEOE = 9$, $MSR = 1.1$, $\Delta y_i = 1.25$ and $\Delta z_i = 4$ mm. For the heat sink used in this analysis, this configuration resulted in a mesh with a total of 578,116 elements.

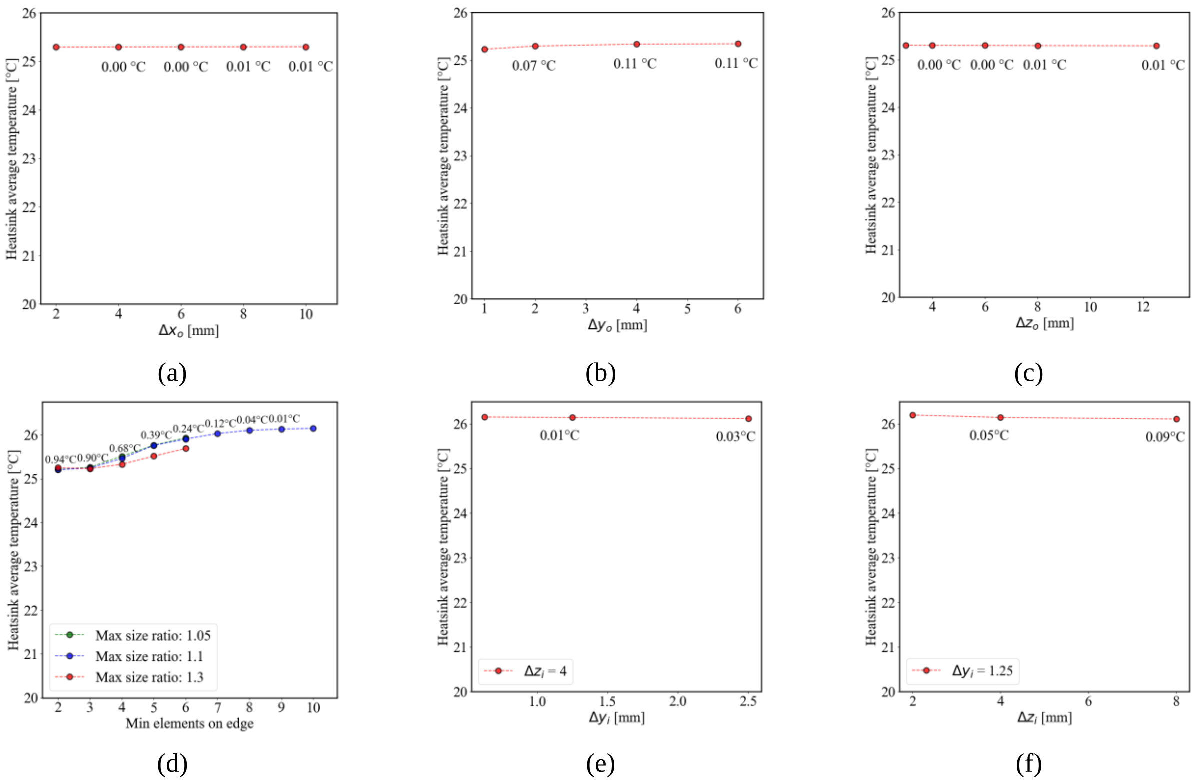
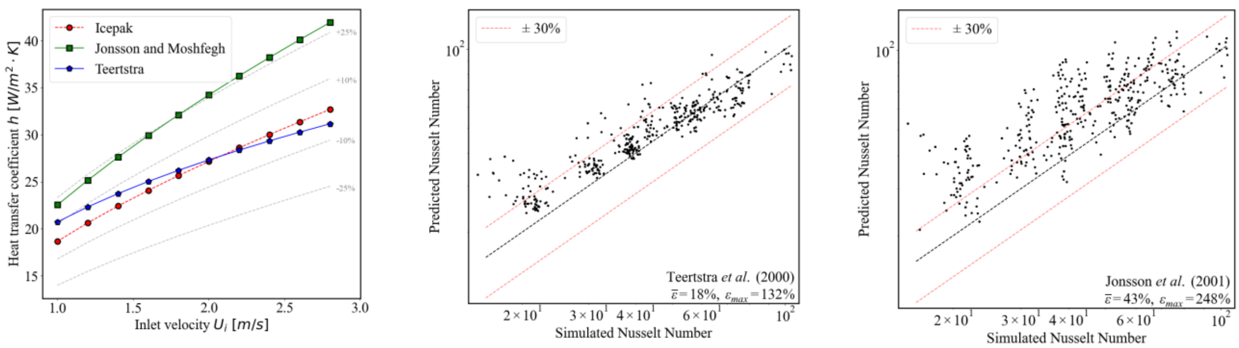


Figure 4 – Results of the mesh independence study, showing the variation of the results with respect to (a) Δx_o, (b) Δy_o, (c) Δz_o, (d) $MEOE$ e MSR, (e) Δy_i e (f) Δz_i.

3.2 Validation of the CFD model

With the mesh configuration described in Section 3.1, simulations were carried out for a heat sink geometry falling within the specified range defined in Table 1. The heat sink dimensions are $W = 5$ cm, $H / L = 0.63$, $L / W = 1.0$ $t_f = 1$ and $t_f / s = 0.23$. For this case, the heat transfer results are shown in Figure 5a, together with the predictions made by the correlations of Jonsson and Moshfegh (2001) and Teertstra *et al.* (2000). The best agreement is observed with the correlation proposed by Teertstra *et al.* (2000), with differences smaller than 10%. This can be attributed to the similarity between the heat sink geometries used by Teertstra *et al.* (2000) and the present study and because both studies adopted the same definition of hydraulic diameter for the Reynolds number. However, as indicated in Figure 5b, the correlation proposed by Teertstra *et al.* (2000) is not suitable to predict low Nusselt numbers, with differences exceeding 30% in relation to the results of the CFD model. Regarding the correlation of Jonsson and Moshfegh (2001), shown in Figure 5c, differences of up to 25% were observed across the entire range of Nusselt numbers. This great difference can be attributed to both the different geometry analyzed by Jonsson *et al.* (2001) and the different definition of hydraulic diameter adopted for the heat sink.



(a) (b) (c)

Figure 5 – Heat transfer results for a heat sink (W = 5 cm, H/L = 0.63, L/W = 1.0, t_f= 1 mm, t_f/s = 0.23): (a) present predictions compared with correlations in literature for a specific geometry, (b) comparison with the correlation of Teertstra *et al*. (2000) and (c) comparison with the correlation of Jonsson *et al*. (2001) for a wide range of geometries.

3.3 Proposed correlation

Different heat sink geometries were simulated following the dimensions shown in Table 1 to develop a new heat transfer correlation. Initially, the LAB Fit (Silva, 1999-2023) software was used to generate the following preliminary correlation for the Nusselt number (Nu_L) as a function of the Reynolds number (Re_{d_h}):

$$y = -7.61 + 0.27x_0^{2.71-0.47 ln\, x_0} \tag{5}$$

where $x_0 = ln\, Re_{d_h}$.

The next step was to use the Eureqa (version 1.24.0) software to improve this correlation, resulting in the following correlation:

$$Nu_L = 4.16 - 0.14x_1 - 3.58x_2 + 0.20y ln\, x_2 \tag{6}$$

where $x_1 = H / L$, $x_2 = s / L$ and y are given by Equation 5.

A comparison between the numerical results and the values given by the proposed correlation is presented in Figure 6. The correlation presented an average deviation of 8.7% in relation to the simulation results and a maximum deviation of 31.5% for a few flow conditions. This correlation is valid for heat sinks with t_f/s < 0.75 and $100 < Re_{d_h} < 1100$.

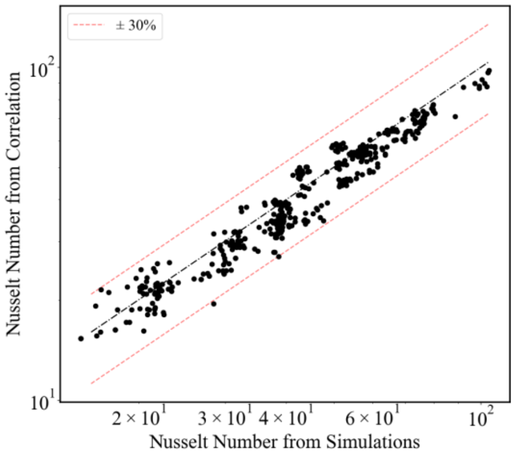


Figure 6 – Comparison between Nusselt number predicted by the correlation and by the CFD simulations.

3.4 The new correlation application to Thermal Network Model

The proposed correlation was introduced into the thermal network model described in Section 2.2. Table 2 shows the results of temperature for this PCB obtained with the Thermal Network Model and the new correlation compared with those predicted by the CFD. As can be seen, the differences between the results of both models are less than 1.1°C, indicating that the thermal network model combined with the new correlation is an effective tool to predict the temperature of the electronic component.

Table 2 – Results of temperature [°C] obtained with CFD and thermal network model for electronic component.

U [m/s]	CFD Model	Thermal Network Model	Difference
1	69.3	68.2	1.1
2	63.5	63.2	0.3
3	61.1	61.0	0.1

4. CONCLUSIONS

A correlation to estimate the convective heat transfer coefficient in heat sinks has been developed based on the predictions of a CFD model. After a mesh independence study, the CFD model was validated through comparisons with experimental data available in the literature. The CFD model was employed to simulate different heat sink geometries and free stream velocities, generating the data set used to develop the correlation with an average deviation of 8.7% and a maximum deviation of 31.5% for flow conditions. Finally, the correlation was introduced into a thermal network model to predict the temperature of an electronic component attached to a heat sink on a PCB. The results of this model for the temperature of the electronic component showed good agreement with CFD predictions, with a maximum difference of 1.1°C.

5. ACKNOWLEDGEMENTS

The present study was developed as part of a technical-scientific cooperation program between the Federal University of Santa Catarina and Whirlpool. Financial support was provided by the National Institutes of Science and Technology (INCT). Additional funding from EMBRAPII Unit POLO/UFSC is duly acknowledged.

6. REFERENCES

Eureqa Formulize. Nutonian. Available at: http://nutonian.wikidot.com/ (Accessed: 30 June 2023).
Jonsson, H.; Moshfegh, B, 2001. Modeling of the thermal and hydraulic performance of plate fin, strip fin, and pin fin heat sinks-influence of flow bypass. IEEE Transactions on Components and Packaging Technologies, v. 24, n. 2, p. 142-149.

Kim, S.J.; Kim, Dong-Kwon; OH, Hwan Hee, 2008. Comparison of fluid flow and thermal characteristics of plate-fin and pin-fin heat sinks subject to a parallel flow. Heat Transfer Engineering, v. 29, n. 2, p. 169-177.

Sato, A.I.; Altemani, C.A.C.; Scalon, V.L., 2020. Mean Nusselt Number Correlation for Tise Heatsink Thermal Design. Revista de Engenharia Térmica, v. 19, n. 1, p. 24-31.

Milagres, A.M.F., 2021. Um método de rede térmica para simulação de placas de circuito impresso regimes permanente e transiente. Master Dissertation. Universidade Federal de Santa Catarina 2021, Florianópolis, Brasil.

Murshed, S.M. S. and Nieto de Castro, C.A., 2017. A critical review of traditional and emerging techniques and fluids for electronics cooling. Renewable and Sustainable Energy Reviews, v. 78, p. 821-833.

Naik, S.; Probert, S.D.; Shilston, M.J., 1987. Forced-Convective Steady-State Heat Transfers from Shrouded Vertical Fin Arrays, Aligned Parallel to an Undisturbed Air-Stream. Applied Energy, v. 26, n. 2, p. 137-158, 1987.

Teertstra, P.; Yovanovich, M.M.; Culham, J.R., 2000. Analytical forced convection modeling of plate fin heat sinks. Journal of Electronics Manufacturing, v. 10, n. 04, p. 253-261, 2000.

7. RESPONSIBILITY NOTICE

The authors are the only responsible for the printed material included in this paper.