

Experimental and numerical study of the heat transfer phenomena of extended surface heat sinks in steady state condition

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Abstract : For the validation of the software that performs the calculation of temperature profiles in cooling fins (ProfileFins) a rectangular fin heat sink was used, which was supplied with heat in order to be able to correctly measure the temperature variation in different air flow conditions and thus be able to realize our frontier conditions to obtain the temperature profile in a theoretical way and also to have the parameters to use in the software to be validated. Thus, the software presents a very acceptable margin of error concerning the theoretical calculation.

Keywords : Cooling fins, Heatsink, Profile Fins.

Introduction

Heat sinks or so-called cooling fins are widely used for cooling many of the electronic components such as computer processors, heat exchangers and radiators of many cooling systems. Therefore, we continuously seek to optimize the operation of these systems by increasing their heat dissipation capacities, such as the study of porous media dissipaters or microchannels with saturated liquids¹, up to the installation of thermal protection to the heatsinks, also applying variations to the inclination for heights of 15 mm onwards where for the first (15mm) the range where the thermal resistance was minimal was in order. $\theta=90^{\circ}-120^{\circ}$ and for higher heights $\theta=120^{\circ}-135^{\circ}$ (M. R. Shaalan., 2012)².

There are studies in the thermoelectric field to optimize the COP of heat sinks³. Up to an investigation with the help of software (NSGA-II) in which dual-sided microchannel dissipaters are optimized⁴. Numeric models have also been used to study the variation of the thermal resistance of the heatsinks for their geometry⁵. Research results have been obtained where the heat transfer coefficient is increased by up to 38.2%⁶. As fin geometry is well known, fins play an essential role in the operation of a heat sink⁷, with square fins being the most widely used.

This document presents the theoretical-experimental validation of the software for calculating temperature distributions on square geometry cooling fins.

Methodology

To validate the software Profilefins it was necessary to collect data experimentally. These data were later introduced in Profilefins and compared with a simulation carried out in Solidworks.

As it is a rectangular heat sink with two slots, the distribution of heat on the front differs from that on the sides, so it was necessary to compare these two distributions separately for each study carried out. The measurements of the same one were taken with a king's foot for the higher precision of the design at the time of simulating.

The experimental part was done using pre-calibrated equipment and environment. Then, in order to be able to carry out the analysis of the data taken, a numerical study was used to define the equations that represent the thermal behavior of the fins, in addition to which a simulation study was carried out in which the design of the fin in Solidworks was used to carry out the thermal studies of the fin with a complement to it.

Finally, the experimental results and the data produced by the simulations were compared with the results of the application study to validate whether the software used has the necessary efficiency for the assigned work.

Equipment Assembly

The equipment was assembled as shown in Figure 1 in order to take the necessary data for the validation of the software by experimental means, which consists of the heat sink ① on a metal plate with high thermal conductivity ② and this plate in turn on top of a mechanism that supplies heat ③, next to these three elements a draught-feeding mechanism ④ as shown in Figure 1.

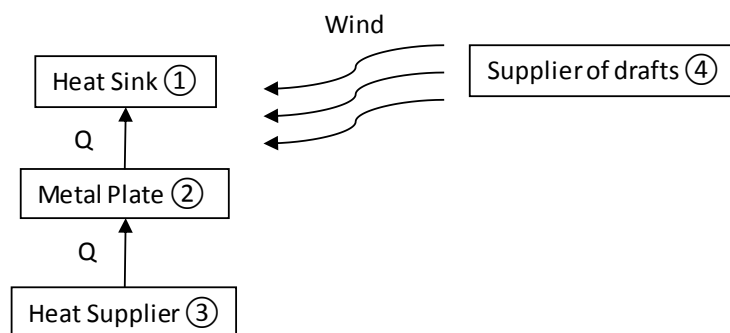


Figure 1. Work equipment

The mechanism that supplies heat does it by means of some resistances in the form of a spiral for this case, when placing the dissipator in it does not guarantee that the temperature transmitted to it is uniform, for this reason, the metallic plate is necessary as a means to secure the uniformity of this transfer in this case.

Additionally, the mechanism that supplies air currents has the peculiarity of regulating the air flow to which the heat sink is subjected to verify the behavior of the temperature along the same with a heat supply from the mechanism that supplies constant heat.

A thermometer was used to record the temperature data on the fin, and an anemometer was used to record the humidity and ambient temperature.

Experimental Study

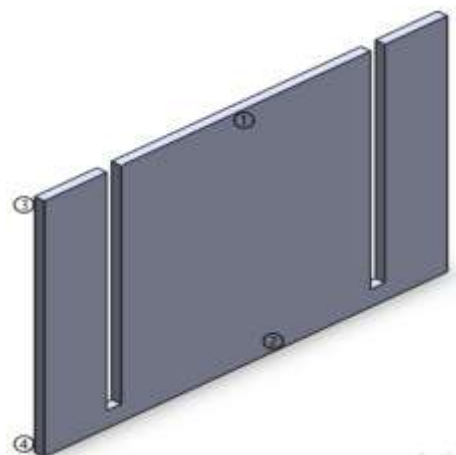


Figure 2.Representation of a heat sink fin

Based on three tests each with different airflow (Regimen) and each test by acquiring ten temperature data at the inlet and outlet of the fin heat supply (Figure 2) the results shown in *Error! Not a valid bookmark self-reference.* were obtained.

Table 1.Temperatures taken experimentally

Regimen	T1(°C)		T2(°C)		T3(°C)		T4(°C)	
4,7 m/s	36,3	35,2	52,6	52,8	32,4	29,5	45,6	46,1
	36,3	34,2	51,5	50,9	25,1	30,7	43,7	45,5
	35,3	33,5	53,0	53,2	29,5	30,9	44,5	44,8
	35,2	34,9	53,3	53,4	32,4	30,7	45,5	44,9
	36,3	35,6	52,7	53,5	29,5	30,2	47,1	49,9
4,3 m/s	36,6	38,3	63,2	58,3	36,1	32,9	46,3	56,1
	36,4	37,6	55,6	58,5	32,2	31,8	54,8	52,8
	35,6	37,3	57,7	68,1	33,8	35,8	46,5	54,7
	38,5	37,9	63,6	61,2	35,3	36,6	51,6	54,2
	37,8	35,8	62,7	60,5	33,0	35,5	50,6	47,8
3,6 m/s	40,7	41,7	63,2	73,1	73,8	37,1	60,0	53,5
	40,4	40,1	62,4	66,4	38,7	36,6	56,1	53,7
	41,5	39,9	62,8	9,9	40,0	40,6	56,4	53,2
	39,9	41,8	64,5	63,3	37,9	35,1	57,2	52,3
	39,6	39,7	61,6	61,7	38,9	37,6	52,8	53,8

The ambient temperature (22,1 °C) and relative humidity measurement (RH 49,3%) are data used in future calculations.

Fin Temperature Profile

It was initiated by general energy formula (Y. A. Cengel., 2011)

$$E_{in} + E_g = E_{out} + E_s, \quad (1)$$

where E_{in} is the energy that went into our system, E_g is the energy that generated the same, as it was a convection cooling system was assumed zero, E_{out} was the energy that came out of our system, finally, it was

the energy of the system itself which when considering that works in a stationary regime is zero what leaves us (1) such as

$$Q_{in} = Q_{out} + Q_{cv} \quad , \quad (2)$$

where Q_{out} was the heat that came out of the heatsink, Q_{in} the heat that came in by conduction through the metal plate and Q_{cv} was the heat that the convection heatsink received because of the surrounding environment; after performing a series of algebraic processes it was obtained that the expression for the calculation of temperature profiles of any of cooling fin is

$$\frac{\partial A_{(x)}}{\partial x} \frac{\partial \theta_{(x)}}{\partial x} + A_{(x)} \frac{\partial \theta}{\partial x} - \frac{h P_{(x)} \theta_{(x)}}{k} = 0 \quad , \quad (3)$$

where $A(x)$ is the area to which the heat was transferred, h is the convective capacity of the air, P is the perimeter, k is the thermal conductivity, and $\theta_{(x)}$

$$\theta_{(x)} = T_{(x)} - T_{\infty} \quad , \quad (4)$$

$T(x)$ the temperature at which the fin limit is set at a distance from the base, and T_{∞} the ambient temperature to which the system was subjected. Then continuing with a series of algebraic processes where he considered himself

$$A_{(x)} \frac{\partial \theta}{\partial x} = 0 \quad , \quad (5)$$

we would have that the general equation of the temperature profile is expressed as

$$\theta_{(x)} = C_1 e^{mx} + C_2 e^{-mx} \quad , \quad (6)$$

where C_1 and C_2 were constants which were defined boundary conditions to find their value and m is defined by the expression

$$m^2 = \frac{ph}{kA} \quad , \quad (7)$$

from which the area and perimeter are constant, now to determine the constants the first boundary condition (CF1) used in all fins is

$$T_{(x)}|_{x=0} = T_w \quad , \quad (8)$$

T_w being the temperature that came from the plate to which it is anchored, then it is necessary to consider two different equations and therefore two different boundary conditions (CF2) for the calculations performed, this to know which would be the most effective condition when comparing the results with the software.

The second boundary condition for the first equation was

$$T_{(x)}|_{x=l} = T_l \quad , \quad (9)$$

l is the maximum length of the heatsink and T_l the temperature at which that point was found.

With which the constants for the first equation were arrived at, which were

$$C_1 = \theta_w - \frac{\theta_l - \theta_w e^{ml}}{e^{-ml} - e^{ml}}, \quad (10)$$

$$C_2 = \frac{\theta_l - \theta_w e^{ml}}{e^{-ml} - e^{ml}}, \quad (11)$$

where C1 and C2 came from equation (6) evaluated in $x=l$ and $x=l$ and $x=0$ respectively, to end up replacing the constants in the general equation of the temperature profile (6) we had the first equation of the temperature profile.

$$\theta_{(x)} = \frac{\theta_l}{\sinh(ml)} \left\{ \sinh(mx) - \frac{\theta_w}{\theta_l} \sinh[m(x-l)] \right\}, \quad (12)$$

having the first equation of the defined temperature profile we proceed to the second equation of the temperature profile, then the second boundary condition for this case was

$$-k \frac{\partial T_{(x)}}{\partial x} \Big|_{x=l} = h [T_{(x)} \Big|_{x=l} - T_{\infty}], \quad (13)$$

and the new constants remained as

$$C_1 = \theta_w - \frac{h \theta_l + k m \theta_w e^{ml}}{k m (e^{ml} + e^{-ml})}, \quad (14)$$

$$C_2 = \frac{h \theta_l + k m \theta_w e^{ml}}{k m (e^{ml} + e^{-ml})}, \quad (15)$$

finally, the new constants were replaced in the general equation of the temperature profile (6) and we were able to define the second equation of the temperature profile.

$$\theta_{(x)} = \frac{\theta_w}{\cosh(ml)} \left\{ e^{m(l-x)} - \cosh(m(l-x)) - \frac{h \theta_l}{k m \theta_w} \sinh[mx] \right\}, \quad (16)$$

Convective Air Capacity

Because it was a forced convection process because the air supply mechanism used a variation of currents towards the heatsink, the calculation of the convective coefficient of air was performed under the expression⁸-

$$Nu = \frac{h l}{k} = C Re^n Pr^m, \quad (17)$$

where Nu is the number of Nusselt, Re the number of Reynolds, Pr the number of Prandtl and l the line length of a line at the same temperature. As when evaluating Reynolds the flow rate is laminar the constants C, n, m, take value de 0.664, 0.5 y $\frac{1}{3}$ respectively.

Study of the Software

The ProfileFins application⁹ allowed us to know the temperature distribution of a fin graphically, so it was necessary to provide data such as the material, length, thickness and width of the fin, the type of fin used, the temperature of the base and environment, the second border condition to use (the first condition is assumed as known temperature) and the convective coefficient (Figure 3).



Figure 3. Profilefins program user interface

This is how the data obtained from thermographic captures were entered in the application, which generated a temperature versus position graph (Figure 4)

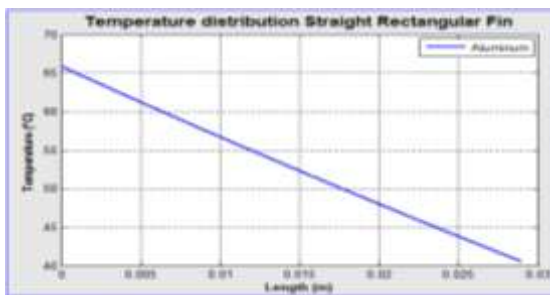


Figure 4. Graph temperature versus position graph for a 3.6 m/s speed given by Profilefins

The software then provided us with behavioral information (to be more precise, a point of the slope was located, and the temperature and position values were located) of the fin temperature which allows us to replicate it later.

Simulation Study

The simulation was carried out in the Solidworks software of the year 2016 in conjunction with its temperature estimation complement along surfaces.

A representation of the heatsink fins was constructed with the heatsink measurements resulting in (b)

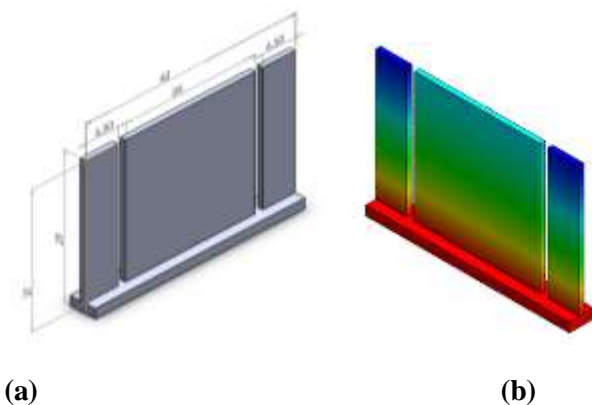


Figure 5. Representation of Heat Sink (a) in Solidworks and its thermal image (b)

The declaration of the types of thermal loads found in the heat sink was made using the data obtained experimentally, the first of these conditions were the temperatures encountered in points 1,2 and 4 of Figure 2, the next was the convection condition to which the fin of the heat sink was found.

Due to the nature of the study, three simulations were performed under the conditions of each airflow, defined as the aluminum of a K (230 W/m² K) which is similar to the K used by ProfileFins software. Giving us a thermal image as shown in (b)

The temperature distribution was then observed, and the temperature data was extracted at each point of the distribution for graphing purposes.

Results and Discussion

To use the formulas of the experimental study, it was necessary to take up again the table of temperatures and to make a table of averages of the temperatures taken for the different marked points. (*Error! Not a valid bookmark self-reference.*)

Table 2.Average temperatures of each regimen

Regimen	T1 (°C)	T2 (°C)	T3 (°C)	T4 (°C)
4,7 m/s	35,28	52,69	30,09	45,76
4,3 m/s	37,18	60,94	34,3	51,54
3,6 m/s	40,53	65,89	41,63	54,9

As the convective capacity of the air varies according to the velocities of the mechanism that supplies air, it was necessary to calculate several convective capacities which imply the calculation of several m which is shown in Table 3; this estimate is affected in turn by the calculations of the convective coefficient of air. However, only the calculations in points 1 and 3 of Figure 2 are needed.

Table 3.Convective coefficient of air and values of the variable m for each regimen

Regimen	h1	h3	m1	m3
3,6m/s	8,89	8,89	8,133362	16,26672
4,3m/s	9,72	9,68	8,50457	16,97411
4,7m/s	9,39	10,16	8,358956	17,38986

The thermal conductivity of aluminum is 209.3 W/mK, The density of the air relative to its relative humidity is 1174.3 Kg/m³ and the front and side area of the fin is 0,754 m y 0,1885 m respectively.

These data are necessary to know the temperature profile of the heatsink base and tip, the result of the temperature profile for the experimental study was plotted in

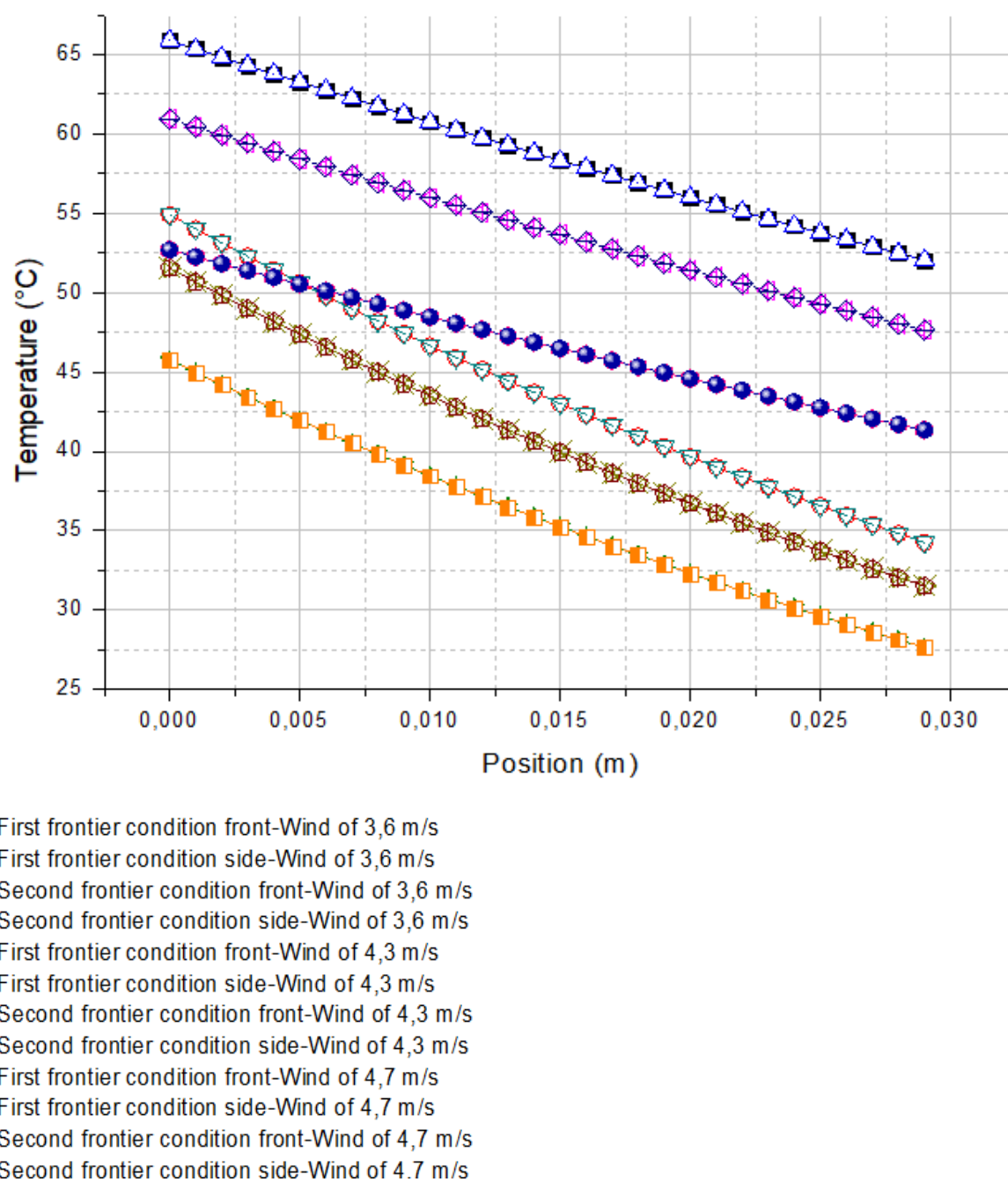


Figure 6.

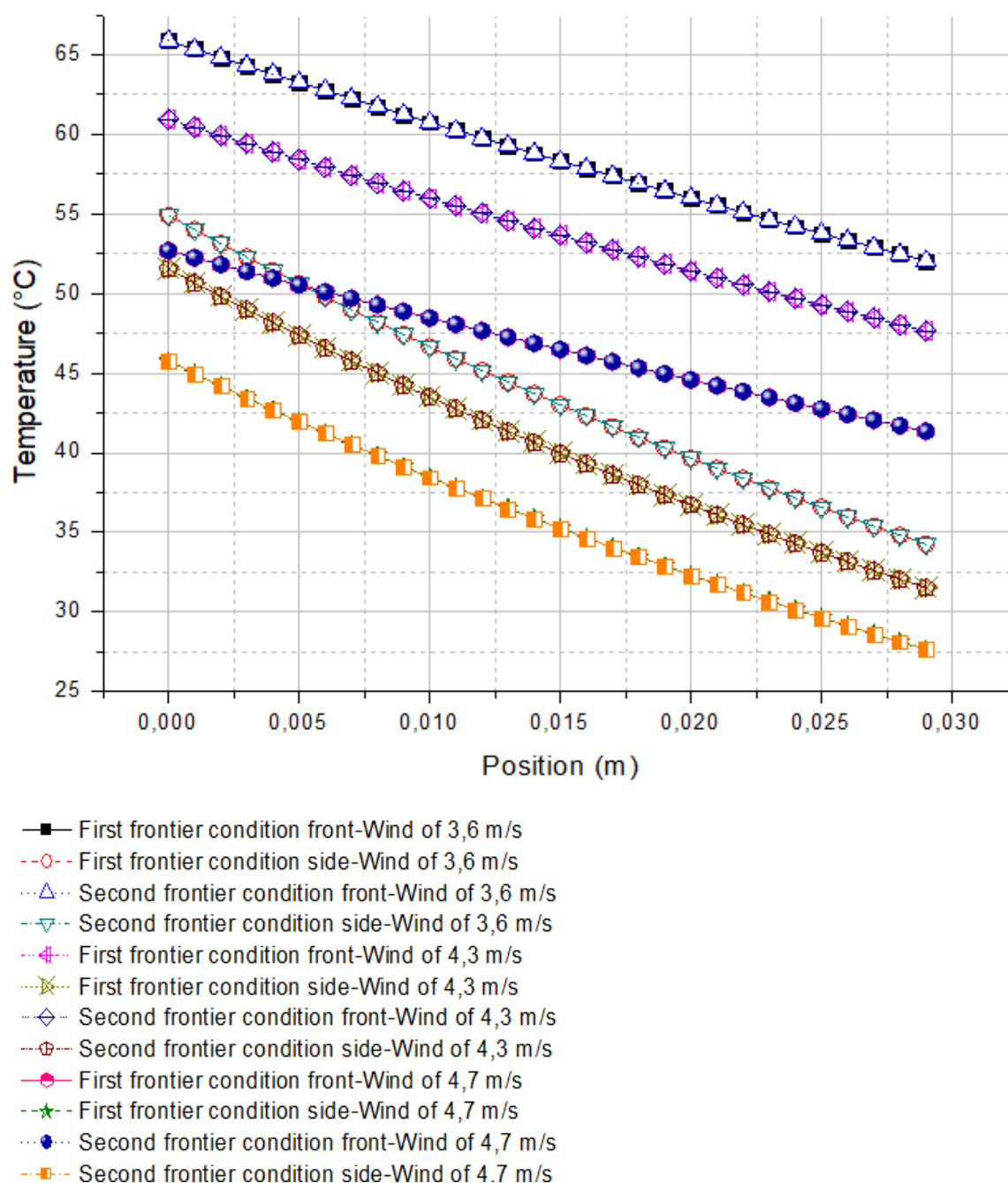


Figure 6.Comparative graph of the different border conditions

There is an equality in two of the four equations, this is due to the precision when using the two different border conditions and arrive at the same result which allows us to corroborate the calculations made.

Finally, we show the results of the thermographs made in Solidworks against the photographs taken by the thermographer; it can be seen that there is no significant difference between the scales which validates our simulation with the data received when comparing them with the software ProfileFins. (Figure 7)

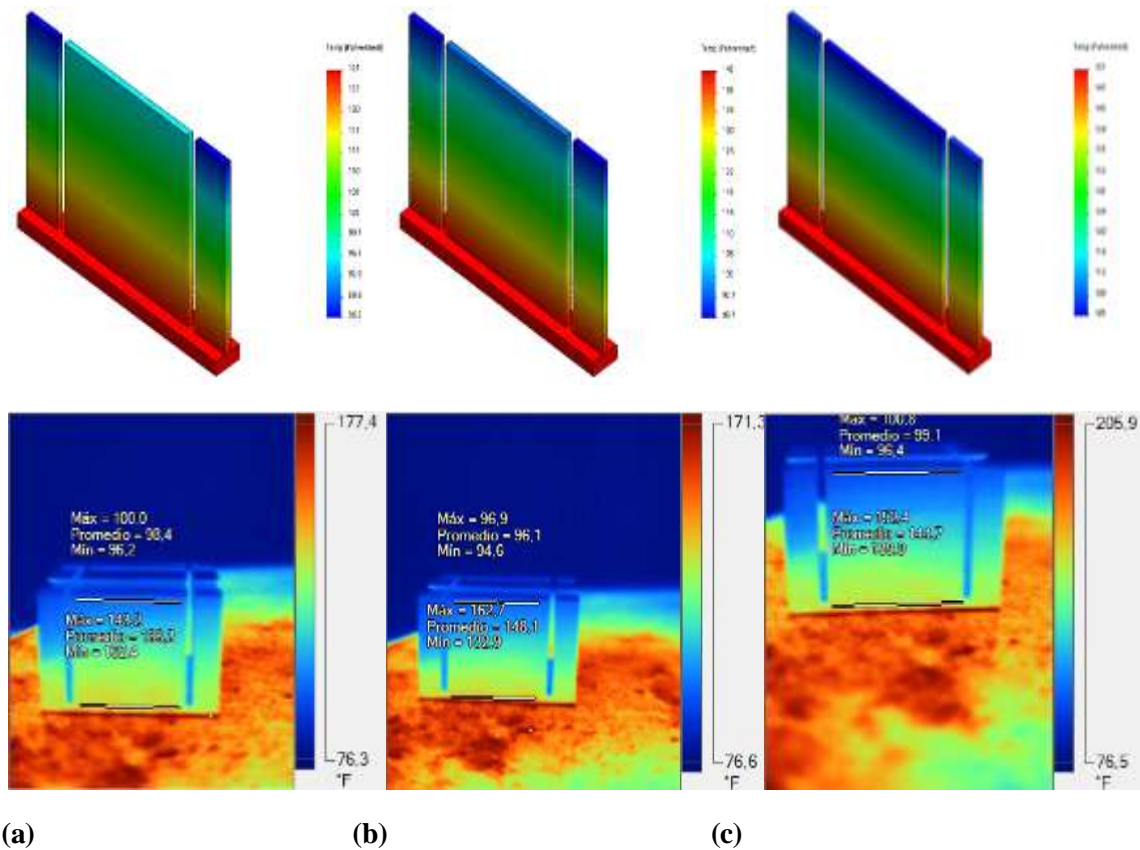


Figure 7. Simulation and thermographic capture for the speed of 4,7 m/s (a), 4,3 m/s (b) and 3,6 m/s (c)

Comparisons and Conclusions

Figure 8 compares the temperature profiles of the three studies carried out. Because Solidworks and ProfileFins are programs that work with exact boundary conditions in the iteration of their data, temperature vs. position curves begin to work. And end at the same point, furthermore, both programs worked under the same boundary conditions at the ends of the heatsink, a case that does not occur with theoretical analysis, moreover, the use of significant figures along the calculation line varied the temperature value along the length of the heatsink significantly.

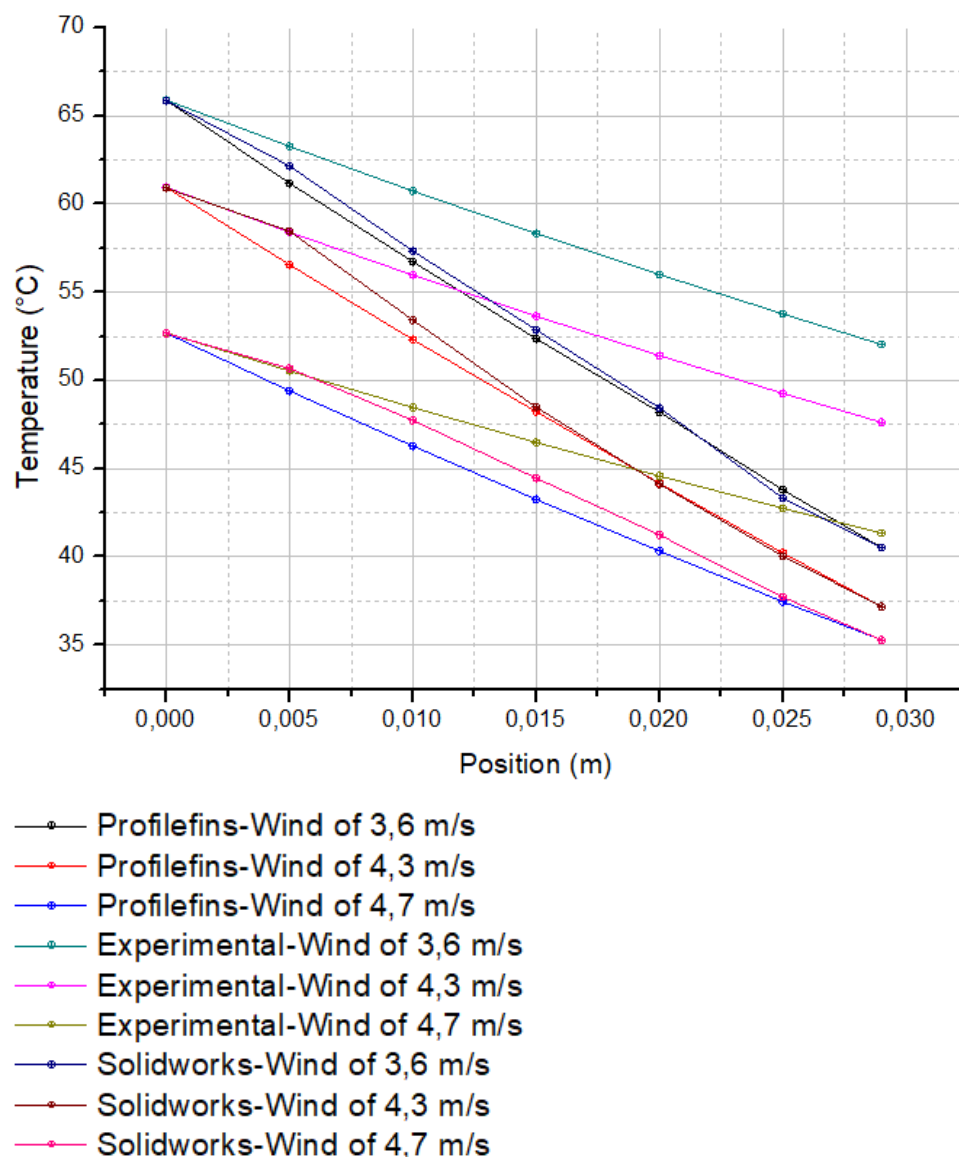


Figure 8. Comparative graph of temperature profiles

Under the need to implement digital tools for the study of temperature distribution in a heat sink, this software has been created capable of accomplishing this task; However, they need validation to check if they fulfill the assigned mission. The ProfileFins software provides data similar to experimental data, which confirms its reliability in the processing of cooling fins.

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