# EXPERIMENTAL DETERMINATION OF THE FORCED CONVECTION HEAT TRANSFER COEFFICIENT OF AN ALUMINUM COOLING PLATE WITH A CHANNEL SHAPE INSPIRED BY NATURE

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## ABSTRACT

Cooling is a crucial aspect in numerous applications where the optimal operation of electric, electronic, or electrochemical devices requires a controlled operating temperature. In this sense, metallic cooling plates are a suitable solution to dissipate heat from the surface of these equipment. The refrigeration capacity of cooling plates can be improved by circulating cold fluid along channels drilled in the metallic plate. The shape of these channels plays a critical role on the performance of the cooling plate since they affect both the distribution of temperature across the plate and the pressure drop required to pump the cooling fluid along the channel. The channel shape of a cooling plate can be optimized considering the Constructal law, which proposes the use of configurations found in nature to improve the performance in industrial applications. Following the Constructal law, a cooling plate made of aluminum, inside of which there is a channel with a shape resembling the outline of a flower, was built by a 3D printer. The performance of the plate was experimentally evaluated refrigerating the plate with various flow rates of cold water. To that end, an experimental facility was specifically designed and built to test the cooling capacity of the plate. The experimental setup consists of an enclosure inside of which the temperature of the atmosphere is controlled by a PID system connected to a thermoresistance and a heater, and a thermostatic bath to control the temperature of the cooling water at the inlet of the plate. The temperature of the plate was measured by an IR camera and the heat transfer coefficient by forced convection to the fluid were derived from the tests for both laminar and turbulent flow regimes of the fluid, obtaining values of 1703 and 3639 W/m$^2$K, respectively, with maximum variations of 1 % for three replicates of each test, proving the high repetitiveness of the experimental procedure proposed. The average characteristic cooling time of the plate was measured to be 34.9 and 16.5 s for Reynolds numbers of the cooling flow of 1249 and 4918, respectively. Thus, an increase on the flow rate by 4 times results in a reduction of the characteristic cooling time by approximately 50 %.

## INTRODUCTION

Flat plate cooling systems with channels along which a cold fluid refrigerates the plate are widely used in several industrial applications, such as cooling of electronic devices, temperature control in fuel cells or operating temperature reduction of solar photovoltaic modules. All these applications require a reduction in the peak operating temperature and the prevention of the appearance of hot spots. The operation of these systems in a specific temperature range results in an increase in the process efficiency and contributes to increase the equipment lifespan, limiting failures related to thermal issues [1].

Heat dissipation using a metallic flat plate with channels to circulate a cold liquid is an efficient refrigeration system due to the high heat transfer coefficients characteristic of forced convection to liquids. However, the use of small diameter channels and high velocity for the cooling liquid may lead to high pressure drops that implies a high pumping cost, counteracting the thermal improvement [2].

During the last decades, novel configurations for the cooling channels have been proposed to minimize the pressure drop of the fluid while improving the thermal efficiency of the refrigeration plates. Some of these configurations are inspired by nature, following the so-called Constructal law. In this regard, Almerbati et al. [3] run numerical simulations to determine the temperature distribution of a metallic plate using several shapes for the channel based on nature. They characterized the different configurations of the channel assuming that the cooling liquid circulating along the channel maintained a constant temperature. Samal et al. [4] complemented the previous work of Almerbati et al. [3] by including in the numerical simulation of the cooling plate the liquid circulating along the channel. They concluded that shapes of the channel inspired by nature patterns allow an improvement of the thermal performance of the cooling plates while limiting the pressure drop required to pump the fluid along the channel. In a more applied work in this line, Mosa et al. [5] used the Constructal law to evaluate the performance of radiant cooling panels by numerical simulations. The previous results published in the specific literature suggest a high potential of these cooling systems. However, most of the works available are purely numerical and experimental results related to metallic cooling plates with channels to circulate cold fluids are still scarce.

In this work, the refrigeration performance of an aluminum cooling plate with a channel to circulate cold water was evaluated experimentally. The shape of the channel was inspired by nature, specifically by the outline of flowers. An experimental technique based on an IR camera was proposed to characterize accurately the evolution of the distribution of temperature across the plate during transient cooling tests, which consisted in refrigerating the plate from a high initial temperature to the lower temperature of water, used as cooling fluid. The experimental measurements allow the derivation of the heat transfer coefficient by forced convection to the circulating water. The cooling tests were carried out for two different flow rates of water, corresponding to laminar and turbulent regimes, respectively, and the repetitiveness of the procedure was quantified by conducting three replicates of each refrigeration experiment.

## NOMENCLATURE

| | | |
|---|---|---|
| $A_{ch}$ | [m$^2$] | Lateral surface of the channel |
| $A_{ext}$ | [m$^2$] | External surface of the plate in direct contact to air |
| $c_p$ | [J/kgK] | Specific heat of aluminum |
| $h_w$ | [W/m$^2$K] | Heat transfer coefficient by forced convection |
| $k$ | [W/mK] | Thermal conductivity of aluminum |
| $t$ | [s] | Time |
| $t_c$ | [s] | Characteristic time |
| $T_0$ | [K] | Initial temperature of the plate |
| $T_a$ | [K] | Ambient temperature |
| $T_{av}$ | [K] | Average temperature of the plate |
| $T_{max}$ | [K] | Maximum temperature of the plate |
| $T_{min}$ | [K] | Minimum temperature of the plate |
| $T_s$ | [K] | Surrounding temperature |
| $T_w$ | [K] | Inlet temperature of water |
| $U_a$ | [W/m$^2$K] | Global heat transfer coefficient for heat dissipation to air |
| $V$ | [m$^3$] | Volume of solid plate |

Special characters

| | | |
|---|---|---|
| $\Theta$ | [-] | Dimensionless average plate temperature |
| $\rho$ | [kg/m$^3$] | Density of aluminum |
| $\sigma_T$ | [ºC] | Standard deviation of temperature across the plate |
| $\tau$ | [-] | Dimensionless time |

## MATERIALS AND METHODS

### Experimental setup

Fig. 1 shows a schematic of the experimental facility specifically designed and built to conduct the transient cooling experiments. The non-steady refrigeration tests were carried out under a controlled atmosphere. The cooling plate was introduced into an enclosure whose temperature is maintained constant by a PID system connected to an electric heater. The cooling plate was supported by a PVC insulator plate to minimize the heat losses through the bottom of the plate, which is considered adiabatic. The initial temperature of the plate was set by a flat plate electric resistor connected to a PID system, whose input was given by a thermoresistance located at the bottom of the cooling plate. The power of the flat plate is 200 W and the dimensions are 200x300 mm$^2$, resulting in an average heat flux of 3.3 kW/m$^2$. The temperature of cooling water was controlled by a thermostatic bath. The required flow rate of cooling water was supplied to the cooling plate by a gear pump and measured by a flow meter. The temperature distribution of the top surface of the cooling plate was monitored during the cooling experiment by an IR camera and stored in a computer.

### Cooling plate

The cooling plate tested is square in shape, with a side of 15 cm and a thickness of 1 cm. The plate was printed in 3D, using aluminum as printing material, and leaving a free channel of circular cross section with 6 mm in diameter on the middle plane of the plate thickness to circulate cooling water. The properties of the solid aluminum used to print the plate are: density $\rho$ = 2650 kg/m$^3$, specific heat $c_p$ = 890 J/(kg·K), and thermal conductivity $k$ = 113 W/(m·K). The shape of the channel was selected to resemble the outline of a flower. It is shown in Fig. 2, where a top view of the channel is represented. The roughness of the channel is lower than 80 µm.

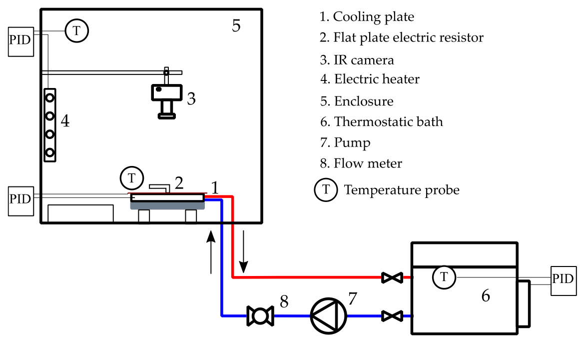


**Figure 1** Schematic of the experimental facility.

### Experimental procedure

Prior to the refrigeration experiments, the temperature of the ambient air of the enclosure was set to $T_a$ = 23 ºC by the PID system connected to the electric heater. The temperature of cooling water was also fixed to $T_w$ = 23 ºC in the thermostatic bath, which recirculated water to approach to the setting point. The flat plate electric resistor was placed over the cooling plate to increase its temperature to $T_0$ = 55 ºC. Once the temperature of ambient air, cooling water and cooling plate were stabilized at the desired values, the refrigeration test started by suddenly removing the flat plate electric resistor from the aluminum plate and pumping cold water along the channel at the selected flow rate, while monitoring the evolution of the temperature distribution on the top surface of the cooling plate with the IR camera. Considering the small thickness of the plate and the high thermal conductivity of aluminum, the Biot number is less than 0.1, so the temperature gradient along the plate thickness can be considered negligible. The uncertainty of the ambient temperature is ±0.8 ºC, whereas the uncertainty of the IR camera is ±2 ºC.

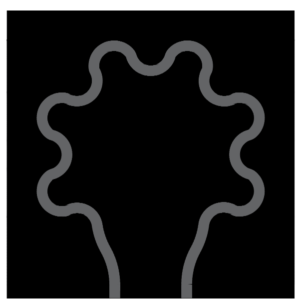

**Figure 2** Top view of half of the cooling plate. The channel is represented in grey and the solid aluminum in black.

Two kinds of experiments were performed to characterize the cooling capacity of the system. First, the heat dissipation from the cooling plate to ambient air was studied by increasing the temperature of the plate to the initial temperature and monitoring the temperature of the top surface of the plate while the plate released heat by combined natural convection and radiation to ambient air, i.e., with no circulation of cooling water. This first test allowed the determination of the global heat transfer coefficient for heat dissipation to ambient air of the plate. Then, refrigeration tests circulating water along the channel of the plate were conducted to determine the heat transfer coefficient by forced convection to cooling water. These experiments were carried out for two different flow rates, corresponding to laminar and turbulent flow regimes of water in the channel. Each case was replicated thrice to prove the repetitiveness of the experimental procedure proposed.

## THEORY

Considering the whole solid plate as a thermodynamic system and assuming that cold water circulating along the channel is kept at the inlet temperature $T_w$, which coincides to the temperature of ambient air $T_a$, i.e. $T_w = T_a = T_s$, the energy conservation equation for the plate can be written as follows:

$$\rho V c_p \frac{\mathrm{d}T_{av}}{\mathrm{d}t} = -U_a A_{ext}(T_{av} - T_s) - h_w A_{ch}(T_{av} - T_s) \quad (1)$$

where $\rho$ is the density of aluminum, $V$ the volume of the solid plate, $c_p$ the specific heat of aluminum, $T_{av}$ the average temperature of the plate, $t$ the time, $U_a$ the global heat transfer coefficient by combined natural convection and radiation to ambient air, $A_{ext}$ the external surface of the plate in direct contact to ambient air, $T_s$ the temperature of the surroundings (either ambient air or cooling water), $h_w$ the heat transfer coefficient by forced convection to cooling water, and $A_{ch}$ the lateral surface of the channel inside the plate.

The energy conservation equation, Eq. (1), can be expressed in dimensionless form as a function of the dimensionless average temperature $\Theta$, defined as:

$$\Theta = \frac{T_{av} - T_s}{T_0 - T_s} \quad (2)$$

and the dimensionless time $\tau$:

$$\tau = \frac{t}{t_c} = t\left(\frac{\rho V c_p}{U_a A_{ext} + h_w A_{ch}}\right)^{-1} \quad (3)$$

Using these dimensionless parameters, the solution of the energy conservation equation of the plate reads:

$$\Theta = \exp(-\tau) \quad (4)$$

This equation allows the calculation of the global heat transfer coefficient by combined natural convection and radiation $U_a$ by an inverse exponential fitting of the experimentally measured average temperature of the plate $T_{av}$ with time $t$ during a cooling experiment to ambient air, i.e., with no cooling water flow, $h_w = 0$ W/m$^2$K. After determining the value of $U_a$, the fitting of the average temperature of the plate with time to an inverse exponential in the form of Eq. (4) can lead to derive the forced convection heat transfer coefficient $h_w$ as a fitting parameter for a cooling experiment circulating cold water along the channel of the plate.

## RESULTS AND DISCUSSION

The measurements carried out by the IR camera during the experimental campaign were postprocessed to determine the distribution of temperature across the plate and the time evolution of the maximum, minimum and average temperature of the plate, from which the forced convection coefficient was determined. The analysis was conducted for two different water flow rates, resulting in Reynolds numbers of Re = 1249 and 4918, corresponding to laminar and turbulent flow, respectively.

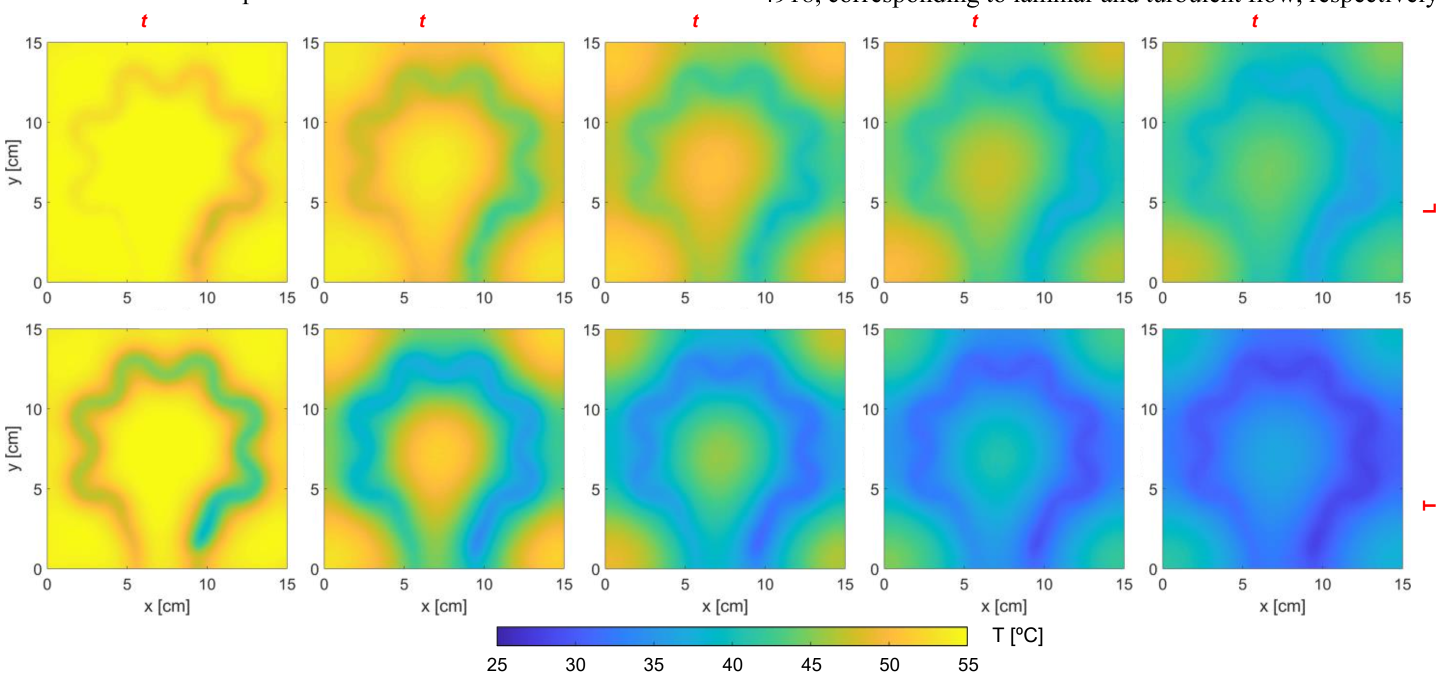


**Figure 3** Evolution of the distribution of temperature across the plate as a function of time for laminar and turbulent flow.

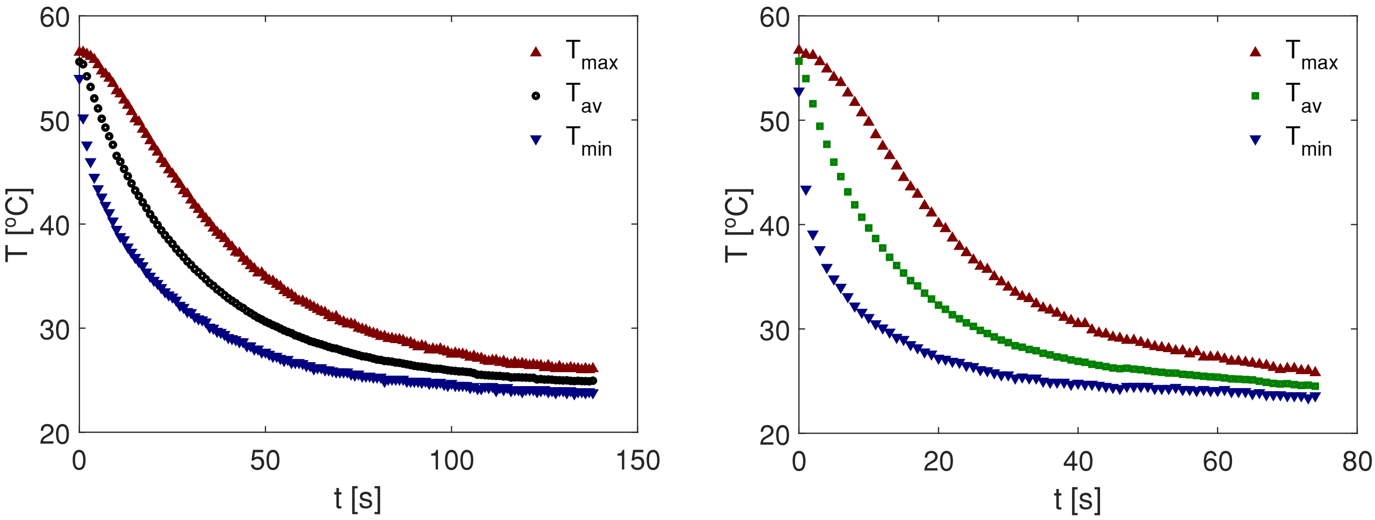


**Figure 4** Time evolution of the maximum, average and minimum temperature of the plate for (left) laminar and (right) turbulent flow.

**Distribution and evolution of temperature across the plate**

The IR camera allowed to measure the distribution of temperature across the top surface of the cooling plate. The high resolution of the IR camera permits the measurement of temperature on 412x412 points uniformly distributed on the plate surface. Therefore, the distribution of temperature across the plate surface could be accurately determined from the IR camera measurement, which was found to be a useful non-intrusive measurement technique. The distribution of temperature across the plate is illustrated in Fig. 3 for specific instants of the cooling process, from $t = 2$ s to $t = 22$ s with a time interval of 4 s, for both the experiments with a laminar and a turbulent flow of cooling water.

The capacity of cold water flowing along the channel to cool down the plate can be observed in Fig. 3. As time progresses, flowing water refrigerates the plate, whose temperature gradually decreases. At the first stages of the cooling process, most of the plate is still at the initial temperature and the channel along which cold water flows is visible in the images as a zone of lower temperature. The zone of the channel where the temperature is lower corresponds to the inlet of the cold water flow, whereas the heating of water in its path along the channel results in a higher temperature of the fluid close to the end of the channel. Heat is dissipated by conduction across the aluminum plate to the channel as the non-steady cooling experiment evolves. A faster cooling process can be attained by increasing the flow rate of cold water along the channel, due to the higher cooling capacity and the increasing values of the forced convection heat transfer coefficient for a higher fluid velocity and when evolving from a laminar to a turbulent regime [6].

The evolution of the aluminum plate temperature during the cooling process was analyzed focusing on the characteristic temperatures of the plate, namely the minimum temperature $T_{min}$, the maximum temperature $T_{max}$, and the average temperature $T_{av}$. The time evolution of the characteristic temperatures of the plate is depicted in Fig. 4 for both the laminar and the turbulent flow cooling experiments. Initially, $t = 0$ s, the whole plate is at the initial temperature $T_0$, and the maximum, minimum and average temperature of the plate coincide at around $T_0 = 55$ ºC in both cases, when cooling water is still not flowing along the channel. Then, as time progresses, the minimum temperature decreases suddenly when cold water enters in the channel, obtaining the minimum temperature of the plate over the inlet of cooling water to the channel. Fig. 3 shows that the maximum plate temperature is located in all cases in the corner closest to the outlet of the liquid flow. At this point, the distance to the channel is maximum, and compared to the closest corner to the inlet of cooling water, the liquid is slightly warmer at the outlet resulting in a decreased of the heat flux from this point to the channel. The average temperature of the plate follows an inverse exponential decrease with time, approaching gradually to the inlet temperature of water, $T_w = 23$ ºC. When comparing with the average temperature, the evolution of the maximum temperature of the plate is delayed due to the time required to dissipate heat by conduction across the aluminum plate from points distant to the channel. A clear effect of the flow rate of cooling water can be also observed in Fig. 4. A higher flow rate of cold water results in a faster cooling of the plate (notice the different scale in the x-axis of Fig. 4). However, the higher capability of water to remove heat from the channel of the plate when increasing the flow rate leads to a higher variability of temperature across the plate, i.e., a larger difference between the maximum and minimum temperature, due to the slower heat dissipation by conduction across the plate compared to the higher forced convection.

The higher variability of temperature in the plate for the higher water flow rate can be observed better by determining the evolution of the standard deviation of the temperature across the plate $\sigma_T$ during the cooling processes with a turbulent and laminar flow. The results are shown in Fig. 5, where the evolution of the standard deviation of temperature across the plate with time is represented. In both cases, the standard deviation of temperature in the plate increases sharply at the beginning of the refrigeration experiment, when cold water starts to flow along the channel and the regions of the plate far from the channel are still at a high temperature. After the sudden increase, the standard deviation of the plate temperature decreases progressively as the cooling process evolves. The effect of the cold water flow rate on the standard deviation of temperature in the plate is evident in Fig. 5. The higher flow rate corresponding to turbulent flow induces a higher maximum value of the standard deviation of the plate temperature, of around 4.5 ºC, whereas for laminar flow, the lower flow rate of cooling water results in a lower maximum value of $\sigma_T$, of approximately 3 ºC. The higher refrigeration capacity achieved with larger cooling fluid flow rates can also be observed in Fig. 5, as the decrease of the standard deviation of the plate temperature for the turbulent flow is faster than that of the laminar flow cooling experiment.

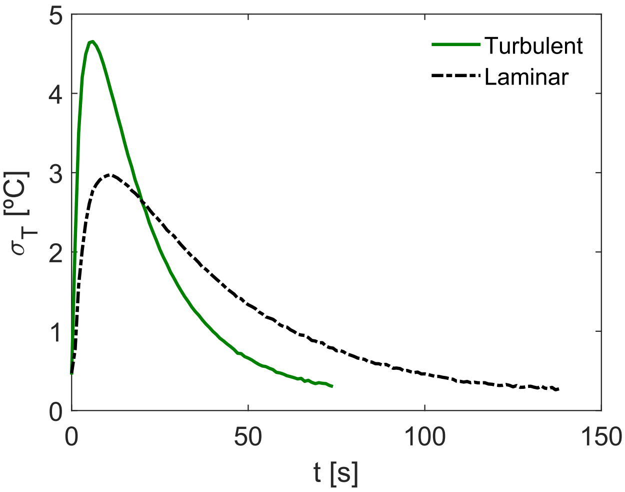


**Figure 5** Time evolution of the standard deviation of temperature across the plate for laminar and turbulent flow cooling processes.

**Determination of the heat transfer coefficient due to forced convection to the cooling fluid**

The results of the heat dissipation to ambient air by combined natural convection and radiation experiment are reported in Fig. 6. Obviously, heat dissipation to ambient air is much slower than refrigeration by forced convection to a cold liquid, which can be clearly observed by comparison of the time evolution of the average temperature of the plate during the dissipation to ambient air experiment depicted in Fig. 6 a) with the average plate temperature during forced convection refrigeration tests, represented in Fig. 4. The experimental evolution of the dimensionless temperature Θ with time is plotted in Fig. 6 b), together with the fitting of this parameter to an inverse exponential decrease as that of Eq. (4). A fairly good agreement between the experimental data and the fitting can be observed in the figure. The global heat transfer coefficient by combined natural convection and radiation from the plate to ambient air, $U_a$, is determined as the fitting parameter from Eq. (4), fixing $h_w = 0$ W/m$^2$K. The value obtained was $U_a = 10$ W/m$^2$K, which is similar to typical values for heat dissipation to ambient air obtained from widely used correlations such as those of McAdams [7] and Churchil and Chu [8]. This value is the average result of two replicates of the ambient dissipation test for which the deviation of each experiment from the average value is below 0.4 W/m$^2$K, corresponding to a maximum deviation of 4 %. Therefore, the facility built and the experimental procedure proposed was found to be reliable to obtain repetitive results for the transient cooling of the plate by dissipation of heat to the ambient.

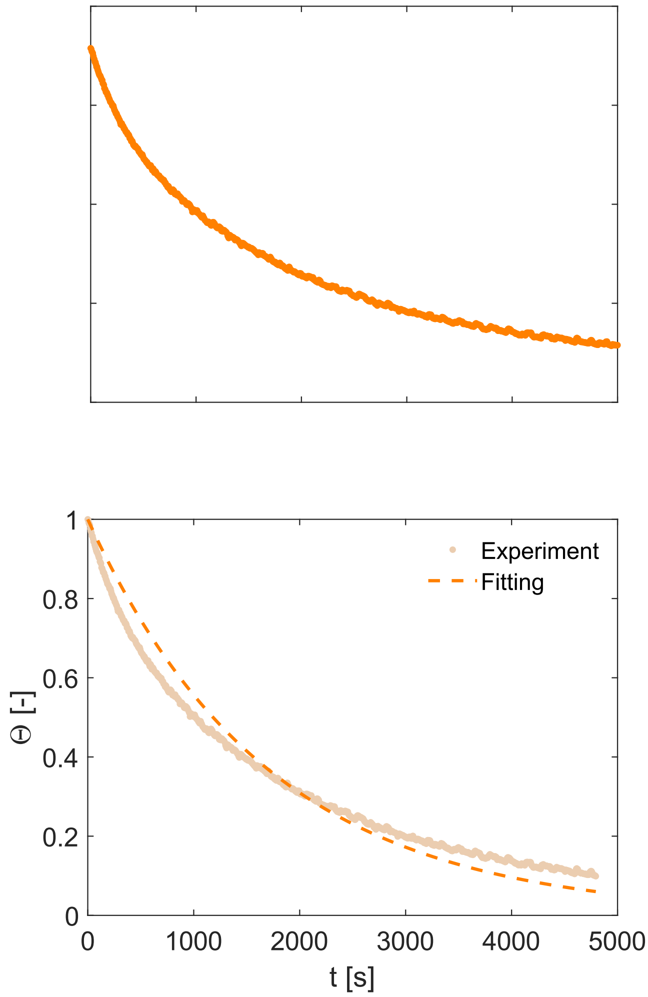


**Figure 6** Heat dissipation to ambient air. Time evolution of (top) average temperature and (bottom) dimensionless temperature.

The time evolution of the average temperature of the plate during the transient cooling tests circulating cold water along the channel of the plate, illustrated in Fig. 4, was used to calculate the evolution of the dimensionless temperature with time for both laminar and turbulent flows. The experimental values of the dimensionless temperature for both flow regimes are included in Fig. 7, together with the inverse exponential fitting in the form of Eq. (4). Again, the match between the experimental curves and the fitting was satisfactory, informing of the reliability of the process here proposed to determine the forced convection coefficient. From the fitting, the heat transfer coefficient by forced convection to the cold water flow, $h_w$, can be derived from Eq. (4) as a fitting parameter, provided that the global heat transfer coefficient for heat dissipation to ambient air, $U_a$, is known from the dissipation to ambient air test.

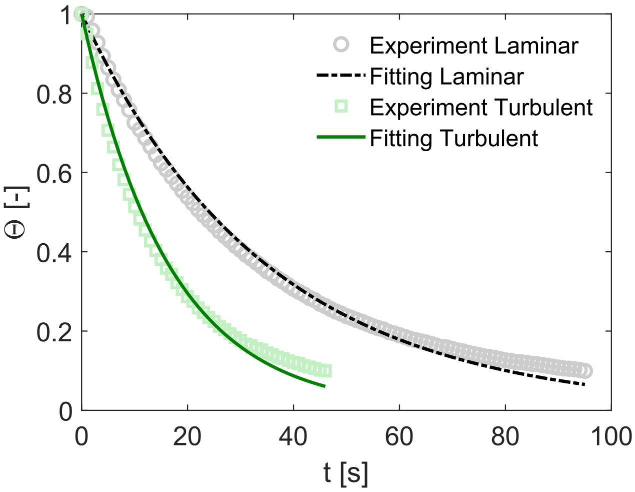


**Figure 7** Experimental measurement and fitting of the time evolution of the dimensionless temperature.

The transient refrigeration tests circulating water along the channel were conducted for two different flow rates, corresponding to Reynold numbers of 1249 and 4918. Thus, the flow is expected to be laminar in the former case and turbulent in the later. For each case, three replicates of the experiment were carried out to check the repetitiveness of the procedure. For each case, the evolution of the dimensionless temperature measured experimentally was fitted to an inverse exponential trend, as indicated by Eq. (4), obtaining the characteristic time $t_c$ and the forced convection coefficient to cold water $h_w$ as fitting parameters. The values obtained for the characteristic time $t_c$ and the forced convection coefficient to cold water $h_w$ for each replicate are reported in Table 1.

**Table 1** Characteristic time and forced convection coefficient to cold water obtained for each replicate of the test.

| Re [-] | $t_c$ [s] | $h_w$ [W/m$^2$K] |
|---|---|---|
| 1249 | 34.6 | 1718 |
| 1249 | 35.2 | 1686 |
| 1249 | 34.8 | 1705 |
| 4918 | 16.4 | 3661 |
| 4918 | 16.6 | 3610 |
| 4918 | 16.5 | 3646 |

The effect of increasing the flow rate on the refrigeration performance of the plate can also be observed in Table 1, where an increase in the flow rate by around 4 times results in a reduction in the characteristic time of cooling by approximately 50 %. The average values of the forced convection heat transfer coefficient obtained for Reynolds numbers of 1249 and 4918 were 1703 and 3639 W/m$^2$K, respectively. These values are in good agreement with those reported in specific literature for forced convection in tubes, proving the accuracy of the experimental method proposed to derive the coefficients. In addition, the maximum deviation of all replicates respect to the average convection coefficient is below 1 %, confirming the high repetitiveness of the experimental procedure proposed.

## CONCLUSION

The refrigeration performance of a metallic cooling plate with a channel along which cold water circulated was evaluated experimentally. An experimental technique based on monitoring the evolution of the temperature distribution across the plate by an IR camera was proposed. The shape of the channel inside the plate was inspired by the outline of flowers, following the concept of Constructal law. The cooling plate was built by a 3D printer, using aluminum as support material.

The cooling tests allow the derivation of both the global heat transfer coefficient for dissipation to ambient air and the heat transfer coefficient by forced convection to cold water flowing along the channel. The global heat transfer coefficient for heat dissipation to ambient air was derived from the time evolution of the average temperature of the plate measured experimentally when no water was circulated along the channel, obtaining a value of 10 W/m$^2$K, in agreement with typical values found in the specific literature. The refrigeration experiment flowing cold water along the channel were performed for two values of the flow rate, obtaining values of the heat transfer coefficient of 1703 and 3639 W/m$^2$K, similar to those reported in the literature for forced convection to liquids. The tests were replicated thrice, obtaining deviations of the heat transfer coefficient by forced convection below 1 % for all replicates.


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