THERMAL PERFORMANCE EVALUATION OF A THIN DOMESTIC DIRECT EVAPORATIVE COOLING AIR CONDITIONING SYSTEM

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Abstract. *Acclimatization devices by evaporative cooling increasingly conquer costumers, whereas they involve, generally, lower costs with energy and maintenance, in comparison with traditional wall air conditioners. Brazil is typical tropical country, where high annual average temperatures occur, leading to the need of the usage of an air conditioner, which is provided, almost completely, by mechanical compression refrigeration systems, instead of acclimatization devices by evaporative cooling. In many countries, this means of acclimatization is already employed in large scale. In this work, a study of an acclimatization system by evaporative cooling for domestic air conditioning is presented. The main motivation for this work was the search for a device with appropriate geometries for an environment esthetic integration; in other words, it's concentrated upon the performance evaluation of evaporative panels, manufactured in cellulose, with "harmonious looking". By "harmonious looking", it's understood, in this work, a device with reduced depth, in comparison with its height and width, resembling a blackboard shape. A mathematical model for the convective heat and mass transfer quantification between air and water in the "filling" of the panel is developed and plotted by computer. Through the computational simulations of operational configurations in counter-flow and crossed flow, from several conditions of air input, the thermal performances were calculated. The analysis of technical viability is focused upon the prediction of thermal efficiency (ratio between air temperature reduction and the maximum temperature reduction possible) of the panels as a function, among others, of its thickness, searching the minimal possible values for this dimension. Data about possible head losses through the filling of the panels were also calculated. The results showed a real possibility for the design of efficient evaporative coolers with the characteristics or geometrical features desired, in other words, comprised by thin panels.*

Keywords*: air conditioning, evaporative cooling, evaporative cooling pads, mathematical modeling, esthetics shapes.*

1. INTRODUCTION

Brazil, as a typically tropical country, presents high annual average values of environment temperature, fact that makes often necessary the environment conditioning. Among the several acclimatization systems, there are those operating by *evaporative cooling*. These equipments involve, at first, lower costs of energy and maintenance than common wall conditioners. Unfortunately, in Brazil, the acclimatizators by evaporative cooling aren't widely employed yet, remaining unknown by the majority of population; on the other hand, they increasingly win space, whereas they present lower energy consume and maintenance costs than the conventional wall conditioners. Lower costs of acquisition and maintenance, together with demands of domestic acclimatization, could represent, for Brazilian population, a stimulus to the usage of evaporative coolers, particularly, in warm and dry zones of the country. Another interesting point concerns about humidification of environments, given the common weather conditions of low humidity (typical autumn/winter situation in Belo Horizonte), which can be, at first, done through the correct usage of the evaporative cooling, ensuring an appropriate range for humidity, accordingly with thermal comfort criteria.

Designs beholding esthetic features and low levels of noise would be basic in order to ensure a market of domestic acclimatization equipments. Models of evaporative coolers, available today, involve physical features that remind to the conventional wall conditioners. The main motivation of this work is the quest for an alternative of air conditioning, capable of reconcile energy economy and interesting esthetic looking for domestic usage. This study concentrates upon the thermal performance evaluation of evaporative panels manufactured in cellulose, for domestic usage, operating in counter-flow and crossed flow configurations, aiming for the maximum thickness reduction, in comparison with its width and height. Computational simulations, from a mathematical model for the quantification of convective heat and mass transfers between air and water in the "filling", make possible a preview of the operational and geometrical limits for an evaporative cooler as a function of the filling thickness.

2. MATHEMATICAL MODELING

The total heat transfer when environmental air gets in touch with water involve convective latent and sensitive exchanges, of which the measuring can be done through *enthalpy potential* (Stoecker and Jones, 1985). A thermal analysis of the process in steady-state condition for a generic control volume (Fig. 1) yields:

$$q_t \;=\; q_s + q_l \;=\; L\, c_{Pag}(T_{ag,s} - T_{ag,e}) = G.(h_{ar,s} - h_{ar,e}) = (h_c A / c_{pu})(h_I - h_a) \tag{1}$$

being q_t the total heat transfer rate, q_s the sensitive heat transfer rate, q_l the latent heat transfer rate, h_c the coefficient of convective heat transfer, A the area of the water surface, c_{pu} ($= c_{ps} + \omega_a.c_{pI}$) the specific heat of humid air, c_{ps} the specific heat of the dry air, c_{pI} the specific heat of the water steam, h_D ($= h_c / c_{pu}$.) the coefficient of convective mass transfer, h_i the enthalpy of saturated air at the water temperature, h_a the enthalpy of the air.

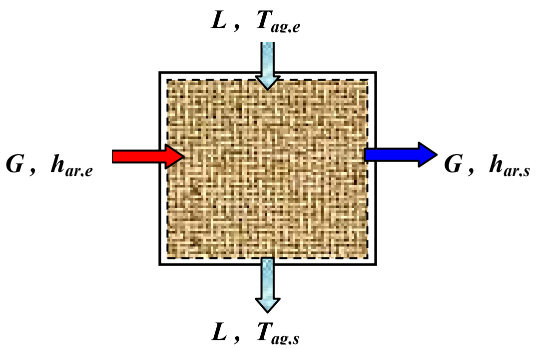

Figure 1. Control volume for an evaporative cooling system in cross-flow and counter-flow.

The value of h_cA/c_{pu} is hard to determine, from the convective heat and mass transfer coefficients point of view (of which evaluation is complex and, yet today, involve the study of many subjects not totally explored), as from the thermal exchange surface. Accordingly to Camargo et al (2003), evaporative panels is cellulose, manufactured by *Glacier-Cor* (Camargo, 2003), have a thermal exchange area by volume unit ratio of *400 m^2/m^3*, value adopted in this work. For the calculation of heat and mass convective coefficients, it's adopted the Dowdy and Karabash correlation (1987), which yields the Nusselt number for an air flow through a cellulose evaporative panel:

$$Nu = 0{,}10\left(\frac{l_e}{l}\right)^{0{,}12} . Re^{0{,}8} . Pr^{1/3} \tag{2}$$

where l is the thickness of the evaporative panel and l_e is defined as the inverse of the ratio of the thermal exchange area by unit of volume.

Accordingly to Camargo et al. (2003), the velocities through the cellulose panel above are limited to *3,5m/s*, which represents $Re \approx 550$. To respect these limits is fundamentally important, for not to provoke carrying of liquid particles of water by the air stream (*carry over*), for the environment to be acclimatized.

With concerns about the head loss through the evaporative panel, were adopted the data given by the manufacturer *Glacier-Cor*, for several velocities of the air and panel thickness.

The efficiency of an evaporative cooler is defined as the reduction of air temperature by the maximum reduction possible (Johnson et al., 2003), in other words, it's the difference between the input and output temperatures of dry bulb of the air in the cooler divided by the difference between the input temperatures of dry bulb and wet bulb of the air:

$$\eta = \frac{TBSe - TBSs}{TBSe - TBU} \tag{3}$$

The values of η are situated between *70* e *90 %*.(ASHRAE, 1997). Camargo et al. (2003) developed the following formulation for the efficiency of an evaporative cooler:

$$\eta = 1 - exp\left(-\frac{h_c A / c_{Pu}}{\dot{m}_{ar}}\right) \tag{4}$$

where m_{ar} is the mass flow rate of air.

3. METHODOLOGY

First of all, simulations were made to evaluate the mathematical model and the computational algorithm, using, as reference, the work of Camargo et al. (2003). In the mathematical simulation, the same experimental values for the parameter h_cA/c_{pu} were adopted, obtained by Camargo et al. (2003). The resultant efficiency values are compared with the data from the above authors.

Following this step, computational simulations were conducted, in order to make a comparison between the evaporative coolers operating in crossed flow and counter-flow configurations. For the operational counter-flow configuration, the cellulose panel was divided in 30 cells, which led to a one-dimensional problem. In the operational configuration of crossed flow, the mesh had 50 cells, being the analysis two-dimensional. From several psychrometric conditions of the environment air and different volume flow rates of water, values of h_cA/c_{pu} that could make possible efficiencies of 75%, 85% e 90% were calculated.

The ulterior analyses were concentrated upon the determination of the dimensions of the evaporative panel, using as reference the efficiency of 85% and the necessary values of h_cA/c_{pu}. The present study was limited to the operational configuration of crossed flow, since the counter-flow configuration, due the disposition between the flows of water and air, would demand larger equipment and, thus, was ignored.

4. RESULTS AND DISCUSSION

For the validation of the analysis presented in this work, mathematical simulations were conducted, adopting the same experimental values to the parameter h_cA/c_{pu}, obtained by Camargo et al. (2003). The resulting efficiency values are compared with the data from the above authors, to which the results of this work showed good concordance (Table 1).

Table 1 – Efficiencies obtained by Camargo et al. (2003) and from the present work

Mass flow rate [kg/s]	**Re**	h_cA/c_{pu}	**Camargo et al. (2003)**	**This work**
0,203	153	0,424	**87**	**88**
0,233	178	0,427	**86**	**84**
0,297	226	0,518	**80**	**83**
0,419	322	0,687	**75**	**81**
0,458	353	0,738	**74**	**80**
0,480	370	0,767	**73**	**79**

In the comparison of the configurations of crossed flow and counter-flow, the flow rates of input air and recirculation water were kept constant at *1000* m^3/h e *0,05 kg/*s, respectively, for two psychrometric conditions of input air $TBS=36^oC/UR=60\%$ e $TBS=42^oC/UR=30\%$. For the same values of h_cA/c_{pu}, very similar thermal behaviors were observed between the two operational conditions (Table 2).

Table 2 – Comparative study between crossed flow and counter-flow configurations

CROSSED FLOW				**COUNTER-FLOW**		
Air at 36 ºC e Relative Humidity of 60% - flow rate of 1000 m³/h						
h_cA/c_{pu} kg/s	T output ºC	η (%)		h_cA/c_{pu} kg/s	T output ºC	η (%)
0,275	32.1	56		0,275	32,1	56
0,550	30,4	81		0,550	30,4	81
0,602	30,2	84		0,602	30,2	83
1,203	29,2	98		1,203	29,2	97
Air at 42 ºC e Relative Humidity of 30% - flow rate of 1000 m³/h						
h_cA/c_{pu} kg/s	T output ºC	η (%)		h_cA/c_{pu} kg/s	T output ºC	η (%)
0,200	32,9	45		0,200	32,9	45
0,400	31,2	70		0,400	31,2	70
0,800	29,7	91		0,800	29,7	91
1,000	29,4	95		1,000	29,4	95

Similar thermal performance means different groups of geometry, filling and velocities, with its interactions, generating the same response for air acclimatization. Thus, all the following results refer only to coolers in crossed flow configutration.

Posterior computational essays were made to evaluation of recirculation water flow rate influence upon the cooler performance. It was considered, for the input air, $TBS=30^oC$ e 40^oC, $UR=25\%$ e 70%, flow rate of *1000* m^3/h, and, for the recirculation water, *0,05 kg/s* e *0,5 kg/s*. Thermal efficiency values were kept at *75%, 85%, 90%*. It wasn't observed a significant influence of the recirculation water flow rate upon the thermal performance of the cooler. It's important to emphasize that, in the mathematical model proposed, thermal interactions of the water with external elements were not considered.

For the definition of the "esthetically" desired dimensions, panels of reduced thickness – objective of the present work, the following strategies were considered for the air flow in a crossed flow cooler (Fig. 2).

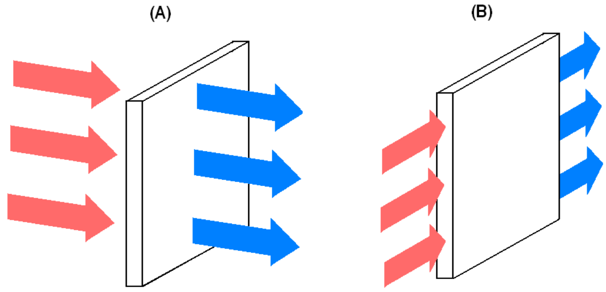

Figure 2. Different air array for a cross-flow direct evaporative cooler.

First, it was necessary to determine the values of h_cA/c_{pu}, for an efficiency of *85%* and air flow rates of $1000\ m^3/h$, $2000\ m^3/h$ e $3000\ m^3/h$, kept a recirculation water mass flow rate of *0,05 kg/s* (Table 3).

Table 3 – h_cA/c_{pu} required for an efficiency of 85%

Efficiency 85%	
Flow rate (m^3/h)	h_cA/c_{pu} (kg/s)
1000	0,632
2000	1,264
3000	1,896

Determined the value for parameter h_cA/c_{pu} corresponding to the kept flow rate, it's possible to establish a geometrical configuration for the panel, considering 3 widths, *10, 15* e *20 cm*. The thickness of *20 cm* was defined, by the purpose of the study, as maximum limit value. Results for the three input air flow rates are found in Table 4.

Table 4 – Geometrical dimensions for panels – efficiency of 85%. Configuration Fig. 2B.

Air flow rate: 1000 m^3/h						
Crossed Flow						
Width	Height	Length	Volume	Reynolds	hD (kg/m2.s)	h_cA/c_{pu}
20 cm	42,1 cm	23,8 cm	20000 cm^3	525	0,079	0,632 kg/s
15 cm	56,1 cm	23,8 cm	20000 cm^3	525	0,079	0,632 kg/s
10 cm	84,1 cm	23,8 cm	20000 cm^3	525	0,079	0,632 kg/s

Air flow rate: 2000 m^3/h						
Crossed Flow						
Width	Height	Length	Volume	Reynolds	hD (kg/m2.s)	h_cA/c_{pu}
20 cm	84,1 cm	23,8 cm	40000 cm^3	525	0,079	1,264 kg/s
15 cm	112,2 cm	23,8 cm	40000 cm^3	525	0,079	1,264 kg/s
10 cm	168,2 cm	23,8 cm	40000 cm^3	525	0,079	1,264 kg/s

Air flow rate: 3000 m^3/h						
Crossed Flow						
Width	Height	Length	Volume	Reynolds	hD (kg/m2.s)	h_cA/c_{pu}
20 cm	126,2 cm	23,8 cm	60000 cm^3	525	0,079	1,896 kg/s
15 cm	168,2 cm	23,8 cm	60000 cm3	525	0,079	1,896 kg/s
10 cm	252,3 cm	23,8 cm	60000 cm3	525	0,079	1,896 kg/s

For $Re = 525$, the air velocity is, approximately, *3,30 m/s*, value that is close to the appropriate superior limit for the filling of cellulose, without bringing about the "carry over" of liquid particles of water. The number of Reynolds is influenced by the relation *width x height*, since these dimensions define the transversal section, through which the air flow passes, and, then, the velocity of the air.

With the widths and heights configured for $Re = 525$, the length of the panel was fitted for the appropriation to the value of h_cA/c_{pu} that would represent an efficiency of 85% (Table 4). It's observed that the values of length and volume of the evaporative panel are constant, despite the air flow rate or the width. This occurs, because, for the same Reynolds number, the ratio between the straight section at the panel entry and the flow rate keep the same value. The velocity of the air for each configuration is also the same, causing the same value for the heat transfer coefficient and demanding the same length to reach a given efficiency.

For a given number of Reynolds and same efficiency, the height and the volume of the evaporative panel, just like the value of h_cA/c_{pu} tend to grow linearly with the air flow rate. The dependence of the efficiency with the value of h_cA/c_{pu} , for different flow rates, can be seen in Fig 3.

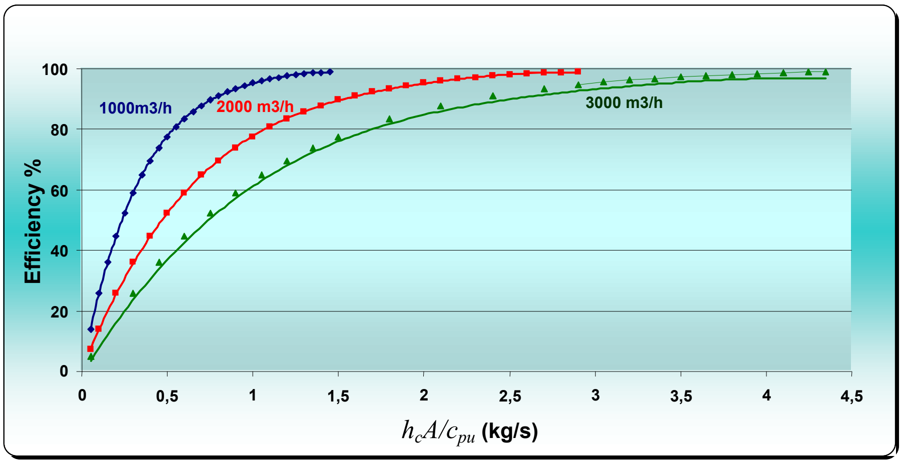

Figure 3 – Variation of efficiency with the value of h_cA/c_{pu} , for three air flow rates

It's important to emphasize that the effect of height variation is different from the influence of length variation upon the parameter h_cA/c_{pu}. The height and the width of the panel define the number of Reynolds, which is related, for its turn, to the value of h_cA/c_{pu} . Results for $Re = 400$, kept the efficiency of *85%*, are found in Table 5.

Table 5 – Geometrical dimensions for $Re = 400$, efficiency of *85%* - Configuration Fig. 2B.

Air flow rate: 1000 m^3/h						
Crossed Flow						
Width	Height	Length	Volume	Reynolds	hD (kg/m^2.s)	h_cA/c_{pu}
20 cm	55,2 cm	22,4 cm	24674 cm3	400	0,064	0,632 kg/s
15 cm	73,6 cm	22,4 cm	24674 cm3	400	0,064	0,632 kg/s
10 cm	110,4 cm	22,4 cm	24674 cm3	400	0,064	0,632 kg/s

Air flow rate: 2000 m^3/h						
Crossed Flow						
Width	Height	Length	Volume	Reynolds	hD (kg/m^2.s)	h_cA/c_{pu}
20 cm	110,4 cm	22,4 cm	49349 cm3	400	0,064	1,264 kg/s
15 cm	147,2 cm	22,4 cm	49349 cm3	400	0,064	1,264 kg/s
10 cm	220,8 cm	22,4 cm	49349 cm3	400	0,064	1,264 kg/s

From Table 5, it can be observed that, for smaller numbers of Reynolds, the heights are bigger, in order to ensure an efficiency of 85%. Furthermore, with the reduction of the number of Reynolds, the necessary length to ensure an efficiency of 85% decline. The reduction of length, with the reduction of the number of Reynolds, shows that, in spite of the time in which water and air interact in the evaporative panel grows, the velocity has more significant influence upon the response of the cooler. A reduction of around *24 %* for the velocity (reduction of the number of Reynolds from *525* to *400*) results in a reduction of just *6,2 %* for the length (approximately *0,9 cm*).

Different values for the height and the length can be put together, with the objective of dimming the evaporative panel volume, without lose efficiency, in order to reduce financial costs of the equipment. The financial value of the evaporative panel is directly related with its volume. The volume of a panel with a number of Reynolds *400* presents a superior value by *23,3%* than the volume for a Reynolds of *525*.

Considering, now, the strategy for an air flow in a cooler of crossed flow represented in part *A* of Fig. 2, new results are found in Table 6.

Air flow rate: 1000 m^3/h						
Crossed Flow						
Width	Height	Length	Volume	Reynolds	h_D (kg/m^2.s)	h_cA/c_{pu}
42,4 cm	42,4 cm	20 cm	35955 cm^3	245	0,044	0,632 kg/s
80 cm	80,0 cm	15 cm	96000 cm^3	69	0,016	0,632 kg/s
195,0 cm	195,0 cm	10 cm	380250 cm^3	12	0,004	0,632 kg/s

Air flow rate: 2000 m^3/h						
Crossed Flow						
Width	Height	Length	Volume	Reynolds	h_D (kg/m^2.s)	h_cA/c_{pu}
60,0 cm	60,0 cm	20 cm	72000 cm^3	245	0,044	1,264 kg/s
113,0 cm	113,0 cm	15 cm	191535 cm^3	69	0,016	1,264 kg/s
275,5 cm	275,5 cm	10 cm	1759002 cm^3	12	0,004	1,264 kg/s

Air flow rate: 3000 m^3/h						
Crossed Flow						
Width	Height	Length	Volume	Reynolds	h_D (kg/m^2.s)	h_cA/c_{pu}
73,5 cm	73,5 cm	20 cm	108045 cm^3	245	0,044	1,896 kg/s
138,4 cm	138,4 cm	15 cm	287318 cm^3	69	0,016	1,896 kg/s
337,5 cm	337,5 cm	10 cm	11139062 cm^3	12	0,004	1,896 kg/s

Table 6 – Geometrical dimensions for panels – efficiency of 85%. Configuration Fig. 2A

The values for the length of the panel were limited between *10* e *20 cm*, with the intention of making a comparison of the new results, based on an equivalent geometry, with the results obtained for the Fig. 2B configuration. A square air entry section was considered, since either composition between height and width values that produce the same area leads to similar responses.

Kept constant the length of the evaporative panel in *10 cm*, for an efficiency of *85 %*, the number of Reynolds results in a value lower than *550*. For the length of *15 cm*, the number of Reynolds becomes bigger than the established limit. However, for this case, if the width and height are increased in *10%*, the number of Reynolds is situated inside the range considerated appropriate; yet the value of h_cA/c_{pu} is modified and the efficiency lightly increases.

The geometrical dimensions of the panels, obtained for operational strategies (A) or (B) from Fig. 2, are quite different. Configuration (A) has a similar shape of a square blackboard and with thicknesses of *10, 15* or *20 cm*. Reminding that the relation between *width x height* of configuration (A) can be any, since the value of the straight section area is kept the same Configuration (B) has its physical feature seemed with a tower of constant straight section.

The decision for the choice between both configurations, (A) e (B), must consider, also, other factors, not beheld in this analysis: fans for air supply, water reservoir, pump, etc. The head loss through the panel has to be determined or measured, in order to define the ventilation system. Dimensions and distribution of the ventilators can be crucial for the definition of the final looking of the evaporative cooler.

Thus, it's presented, next, an estimate of the head loss in the configurations previously discussed.

For the calculations of the head loss through the evaporative panel, data yielded by the manufacturer *Glacier-Cor* (Fig. 4) were used and the results were generated for the operational configurations of crossed flow (Fig. 2). Simulations of head loss are presented in Tables 7 and 8, below.

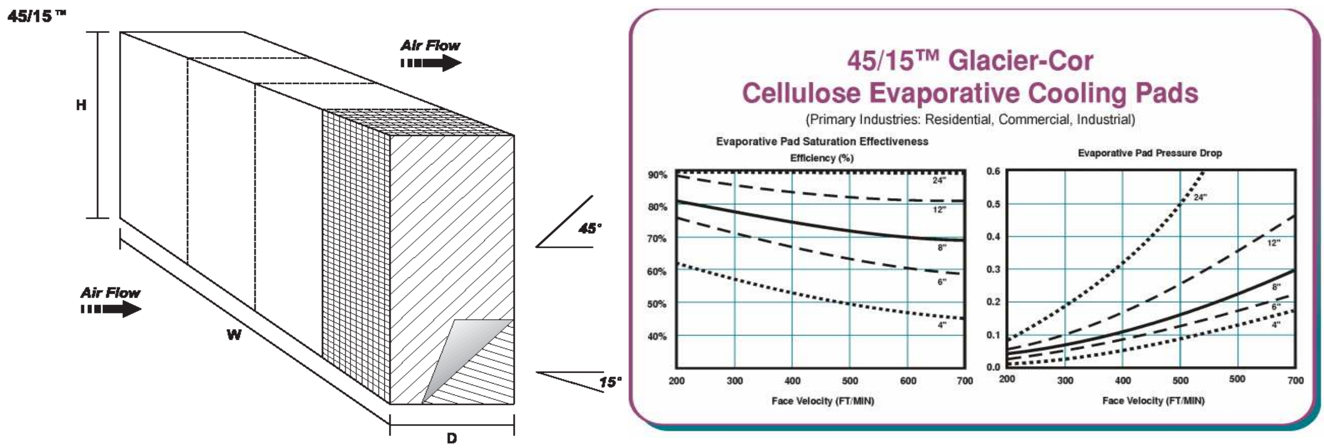

Figure 4 – Evaporative panel in cellulose 45/15^{TM} from Glacier-Cor. Source: www.glacier-cor.com

Table 7 – Head losses through the panel – Efficiency of 85%. Configuration of Figure 2B.

Air flow rate: 1000 m^3/h				
Crossed Flow				
Width	Height	Length	Area	ΔP
20 cm	42,1 cm	23,8 cm	841 cm^2	74,0 Pa
15 cm	56,1 cm	23,8 cm	841 cm^2	74,0 Pa
10 cm	84,1 cm	23,8 cm	841 cm^2	74,0 Pa

Air flow rate: 2000 m^3/h				
Crossed Flow				
Width	Height	Length	Area	ΔP
20 cm	84,1 cm	23,8 cm	1682 cm^2	74,0 Pa
15 cm	112,2 cm	23,8 cm	1682 cm^2	74,0 Pa
10 cm	168,2 cm	23,8 cm	1682 cm^2	74,0 Pa

Air flow rate: 3000 m^3/h				
Crossed Flow				
Width	Height	Length	Area	ΔP
20 cm	126,2 cm	23,8 cm	2523 cm^2	74,0 Pa
15 cm	168,2 cm	23,8 cm	2523 cm^2	74,0 Pa
10 cm	252,3 cm	23,8 cm	2523 cm^2	74,0 Pa

Table 8 – Head losses through the panel – Efficiency of 85%. Configuration of Figure 2A.

Air flow rate: 1000 m3/h				
Crossed Flow				
Width	Height	Length	Area	ΔP
42,4m	42,4 cm	20 cm	1798 cm^2	15,4 Pa
80,0 cm	80,0 cm	15 cm	6400 cm^2	<4,5 Pa
195,0 cm	195,0 cm	10 cm	38025 cm^2	<2 Pa

Air flow rate: 2000 m3/h				
Crossed Flow				
Width	Height	Length	Area	ΔP
60,0 cm	60,0 cm	20 cm	3600 cm^2	15,4 Pa
113,0 cm	113,0 cm	15 cm	12769 cm^2	<4,5 Pa
275,5 cm	275,5 cm	10 cm	175900 cm^2	<2 Pa

Air flow rate: 3000 m3/h				
Crossed Flow				
Width	Height	Length	Area	ΔP
73,5 cm	73,5 cm	20 cm	5402 cm^2	15,4 Pa
138,4 cm	138,4 cm	15 cm	19154 cm^2	<4,5 Pa
337,5 cm	337,5 cm	10 cm	113906 cm^2	<2 Pa

The values of head loss in the operational configuration represented in Fig. 2B remain the same, because the head loss is a function of the evaporative panel length and of the velocity (as easily observed in the data given by the manufacturer).

5. CONCLUSIONS

This study was dedicated to evaluate the thermal performance of evaporative coolers, with the intention of build them, from the geometrical point of view, with a quite different shape from the conventionally commercialized models. In other words, devices that could present a slim shape due its reduced thicknesses. To do that, a mathematical model for quantification of the convective heat transfers in the panel was computationally plotted, having in view obtaining results that could show a good thermal performance for evaporative coolers with the characteristics or geometrical features desired, as it is, comprised by thin panels.

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7. RESPONSIBILITY NOTICE