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INFLUENCE OF CONSTRUCTIVE PARAMETERS ON THE PERFORMANCE OF TWO INDIRECT EVAPORATIVE COOLER PROTOTYPES

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Abstract

Two equally-sized cross-flow heat exchanger prototypes have been designed with a total heat exchange area of 6 m² and 3 m² respectively, constructed with polycarbonate hollow panels of different cross-section. They are connected into a heat recovery cycle within the whole experimental setup constructed for the tests, which mainly consists of: an Air Handling Unit to simulate the outdoor airstream conditions, a conditioned climate chamber, and a water circuit to provide the water supply required. They have been experimentally characterised in two operating modes in order to determine how evaporative cooling improves heat recovery in each case, focusing on the influence of modifying the constructive characteristics. To perform the evaporative cooling process, water is supplied to the exhaust airstream. Results are studied considering how constructive issues, outdoor air volume flow rate and temperature, as well as operating mode influence on the performance obtained. An Analysis of Variance shows how outdoor airflow has a key role in the performance of the systems; whereas entering outdoor air temperature determines cooling capacities. Improvements introduced by larger heat exchange areas compensate with their corresponding smaller cross sections, which hinder water-air distribution on the exhaust air side of the heat exchanger. Finally, these small devices achieve cooling capacities of up to 800 W, being able to partly support ventilation load and achieving around 50% of energy saving in ventilation cooling.

Keywords

Indirect Evaporative Cooling; Heat Recovery; Plastic Heat Exchanger; Heat Exchange Area; Cooling Capacity; Thermal Conductance.



1 Introduction

Our way of life nowadays requires an availability of energy that is however restricted by serious problems such as dependency on sources, rise in fossil fuel prices or the environmental awareness on the impact involved. Consequently, actions are being driven to reduce energy consumption [1].

Particularly, the building sector appears to entail great potential of energy reduction, as it involves around 40% of the total European energy demand, mainly due to the high requirements of ventilation and thermal comfort expected in conditioned spaces. It is also worth pointing out that space cooling in summer is gaining importance, with the consequent electric energy demand peaks. Taking into consideration that up to a 20% of the energy consumed in buildings could be saved, it seems obvious that the European Union focuses many of the new dispositions on energy saving in this field [2].

One of the main targets for energy saving is the recovering of residual energy of exhaust airstreams coming from conditioned spaces, due to the high ventilation rates established to achieve the expected Indoor Air Quality. Heat-recovery systems allow reducing the energy consumption of HVAC systems installed to meet the thermal comfort required by preconditioning the airstream supplied for ventilation [3, 4]. Air to air systems can be simply sensible heat exchangers, or permit total heat recovery. However, their implantation is restricted by the expensive investment involved and their considerable size [5, 6].

Exhaust airstream heat recovery can be improved by implementing an evaporative cooling process [7, 8, 9]. The great feature of this process is that it is a natural phenomenon which occurs when non-saturated air comes into contact with water; water evaporates into the air, reducing this way its dry-bulb temperature. Thus, it is a process of heat and mass transfer, based on the transformation of sensible heat into latent heat [10].

The ideal adiabatic saturation process determines the operation of most evaporative systems. In the theoretical process, water is recirculated, maintaining its temperature close to the adiabatic saturation temperature of inlet air; then the airstream dry bulb temperature decreases during the evaporative process, reaching the value of its saturation temperature when leaving the system. However, the adiabatic saturation temperature is merely the theoretical limit up to which air could be ideally cooled. Furthermore, the possibilities of evaporative cooling depend inversely on the relative air humidity [10]. Consequently, the application of this phenomenon in air-conditioning is an interesting alternative to reduce energy demand, provided that outdoor air relative humidity falls within certain limits, and so it is particularly interesting in dry climates [11]. In this line, Spain presents a wide range of different climates, among which performance of evaporative cooling devices would vary considerably [12]. However, performance of an indirect evaporative cooler operating with return air into a heat recovery cycle will be independent of the entering outdoor air humidity [13]. Maheshwari et al [14] successfully developed an indirect evaporative cooler operating with airflows of 1180 l/s to achieve energy savings of 12418 and 6320 kWh in interior and coastal areas in Kuwait. Moreover, application in the Mediterranean region of indirect



evaporative cooling operating with return air has been proved to reduce energy consumption maintaining thermal comfort requirements, as shows simulation results given by Jaber& Ajib [15], in whose work 1100 l/s were treated covering over 60% of the cooling demand.

In this study, two heat exchanger prototypes for heat recovery permitting their operation as indirect evaporative coolers are proposed. The evaporative cooling phenomenon is performed through one side of the heat exchanger, thus avoiding humidification of the airstream supplied to the conditioned space [16]. Both systems are made of the same material and have same global size, differing only on their total heat exchange area and air-paths width.

For the manufacturing, polycarbonate is considered regarding its low cost and weight, preventing corrosion additionally. Due to the negligible effect of low thermal conductivity in thin plates [17], plastic materials were firstly used for heat exchangers long ago [18] and are still being considered for their feasibility [19]. Within this research line, Delfani et. al [20] proved a plastic indirect evaporative cooler to successfully pre-cool 0.472 m³/s of supplied air, being able to reduce cooling loads up to 75% in temperate outdoor conditions. They also proved that acceptable operating outdoor conditions could be widened by combining it with a direct evaporative cooling unit [21]. Moreover, polycarbonate panels were already considered by Kragh et al. [22] for a successful heat-recovery during heating season, taking advantage of plastic material to face condensing problems.

In previous studies, a heat exchanger prototype made of narrow polycarbonate panels was constructed and characterised [8]. In present work, the main objective is constructing a new prototype with wide polycarbonate panels, only modifying the original constructive characteristics to maintain the main advantages of simplicity, cheapness and small size.

The aim presented here is thus to experimentally compare the two equally sized plastic heat-exchanger systems, of different total heat exchange area and air-water paths, operating with return air from a conditioned space. This allows determining how these two characteristics influence on their behaviour, throwing light on the performance of the evaporative cooling phenomenon inside the panels. Then it will be possible to determine which design would be of most interest.

2 Experimental Facility and Methodology

2.1 Description of the two prototypes: manufacturing, installation and operation

Two heat exchanger prototypes made of polycarbonate hollow panels arranged vertically and equally spaced are tested to study the possibilities of recovering the cooling potential of an exhaust airstream from a conditioned space, to pre-cool ventilation air in summer conditions. The systems differ from each other in the polycarbonate panels' width, namely 4 mm and 9 mm respectively (figure 1). Then, they will be called from now NP ("narrow" polycarbonate panels prototype) and WP ("wide" panels prototype). Figures 2a and 2b show



a view of both prototypes under construction. The main geometric characteristics of both systems are gathered in table 1, and the most representative are shown in figure 3.

To experimentally characterise each system, they are installed in the laboratory within the whole experimental setup operating in a heat-recovery cycle (figure 4). This experimental facility mainly consists of:

- An Air Handling Unit (AHU) that supplies the outdoor airstream in summer conditions to be treated.
- A climate chamber where pre-cooled air from the prototype is supplied. Comfort conditions are ensured here thanks to an auxiliary heat pump.
- A water circuit consisting of a lower tank, a water pump, an upper water distributor and the required ducts.
- The air ducts connecting these systems into a heat-recovery cycle.
- The measurement equipment in the key points of the installation (at the prototype's inlets and outlets).

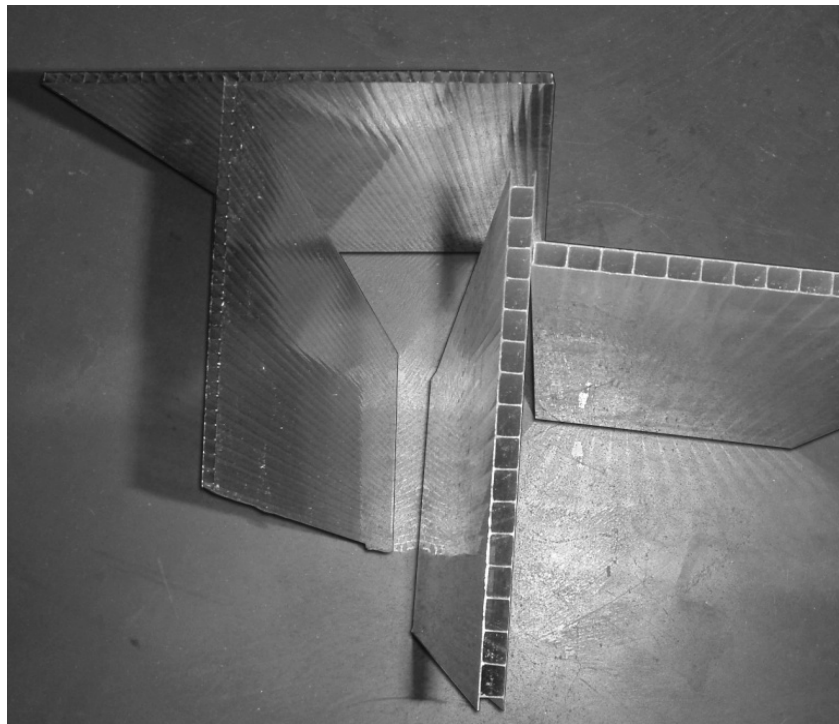


Fig. 1. Detail of the polycarbonate panels cross section

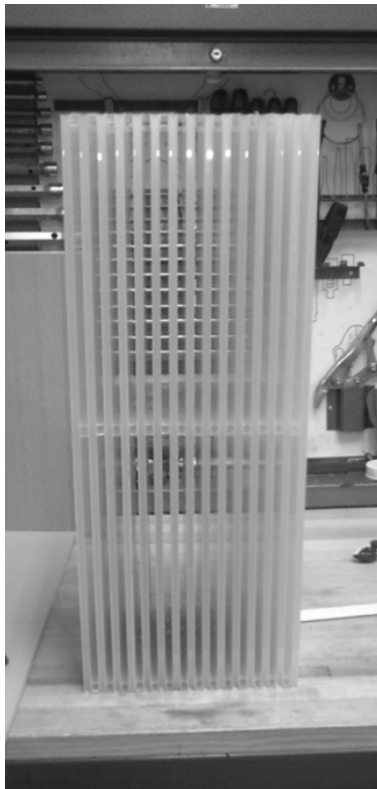


Fig. 2. a. View of prototype WP under construction Fig. 2. b. View of prototype NP under construction, assembled

During the basic operating mode, the system behaves as a mere heat exchanger to pre-condition outdoor air that flows through the space between panels, recovering the residual energy of exhaust air returning from a conditioned space, which flows upwards inside the hollow panels. In this case, heat transfer occurs from the outdoor airstream in summer conditions to the exhaust airstream, which is in comfort conditions, through the system plastic walls. This operating mode is shown in figure 3.

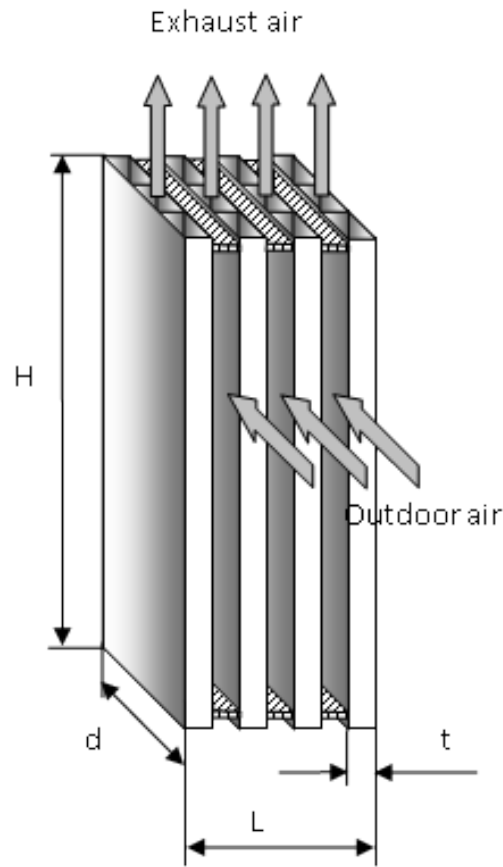


Fig. 3. Scheme of the systems' operation in the first mode.

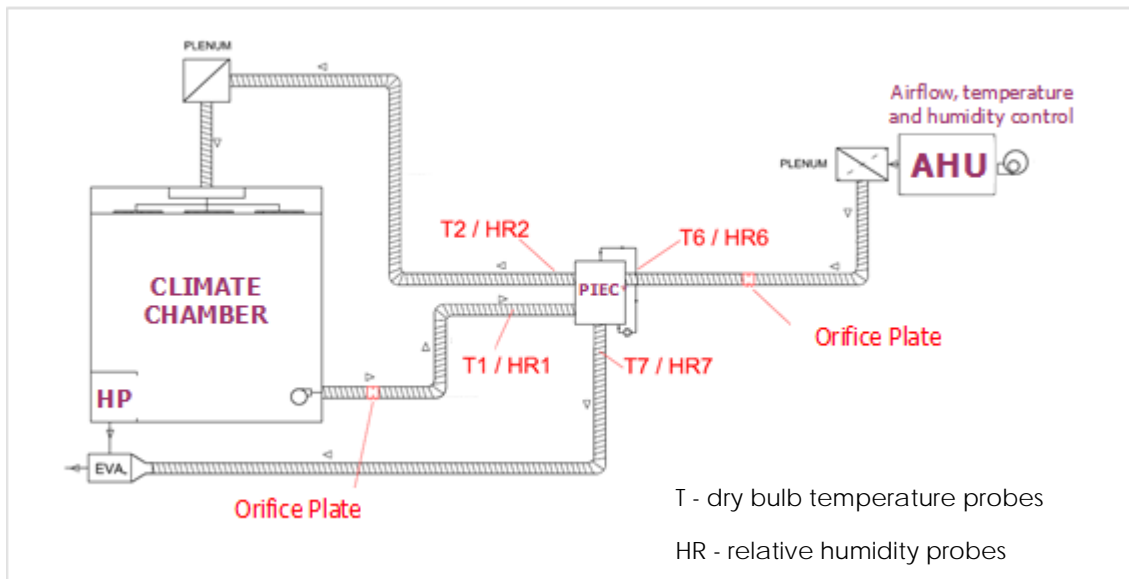


Fig. 4. Scheme of the experimental setup.

A second operating mode is performed by supplying water inside one of the system sides from an upper water distributor. This distributor consists on a shower that provides water that is afterwards uniformly supplied to the upper inlet of the hollow panels with the aid of a wire net. Then, water flows downwards inside the panels in direct contact and counter-flow with



the exhaust airstream. Consequently, the systems operate as indirect evaporative coolers, ideally following an adiabatic process. This mode is developed to study how recovery of the exhaust air energy potential improves in each case by enabling evaporative cooling. Combining heat recovery and evaporative cooling could also be implanted by spraying water to the exhaust airstream before passing through the heat exchanger, as proposed in the Spanish norm [23]. However, the first alternative is preferred in this case because of its compactness.

2.2 Design of experiments

To experimentally determine both prototypes behaviour in both operating modes introduced, outdoor airstream volume flow and dry bulb temperature have been varied and controlled. For each system working in each operating mode, air volume flow is varied from 125 to 400 m³/h whereas its dry bulb temperature is varied from 25 to 40 °C, as shown in table 2. Thus, a total of 64 tests have been performed (32 tests for each system). Entering outdoor air humidity has not been considered a relevant factor for the experimental study because indirect evaporative cooling would be implemented in the return airstream [14].

During the second operation mode (M2), water mass flow supplied has been maintained constant at approximately 0.11 kg/s for all tests. Water temperature varies between 18 and 19.5°C, depending on the particular test, which equals that of wet bulb of exhaust air at the system's outlet, as water supplied is recirculated; though it is influenced by the thermal gains due to the water pump and the heat exchanged with the outdoor airstream.

In every case exhaust air from the conditioned space is supplied to the system at a rate of 260 m³/h and is in comfort conditions. Thus, dry bulb temperature of exhaust air at the system's inlet is 22°C ($\pm 0.5^\circ\text{C}$ due to the sensitivity of the heat pump thermostat), and relative humidity variation is restricted between 50 and 65%; consequently the exhaust air wet bulb temperature varies between 14.5 and 17.5°C, considering psychrometric conditions at 698 m above sea level (tests developed in Valladolid, Spain).

Once operating in steady state, outdoor and exhaust airstreams dry bulb temperature and relative humidity at the system inlet and outlet are registered by the probes and measurement equipment.

To study how the different polycarbonate heat exchangers behaviour is influenced not only by outdoor airstream conditions, but also by evaporative cooling implementation, three parameters have been determined and compared: cooling capacity, thermal effectiveness, and the global heat transfer coefficient. Finally, these parameters are compared for the two different structures designed, to analyse how performance could be improved by acting in the initial system design.

3 Calculation

Parameters considered for the analysis are defined following.



For the basic operating mode (M1), dry bulb thermal effectiveness is studied. When operating with outdoor airflows of 125 and 200 m³/h, heat capacity rate of this cooled air would be determinant and thus the effectiveness should be expressed through equation (1) [24]:

$$\varepsilon_T = \frac{T_{o1} - T_{o2}}{T_{o1} - T_{e1}} \quad (1)$$

For the two remaining cases in which return airflow is lower (260 m³/h), dry bulb thermal effectiveness is then calculated as:

$$\varepsilon_T = \frac{T_{e1} - T_{e2}}{T_{o1} - T_{e1}} \quad (2)$$

Table 1. Geometric characteristics

Characteristic	Type W	Type N
Wall thickness	0.1 mm	0.1 mm
Panel thickness "t"	9 mm	4 mm
Height "H"	0.62 m	0.62 m
Width "d"	0.18 m	0.18 m
Length "L"	0.23 m	0.23 m
Number of plates	15	28
Geometry	Flat plates	Flat plates
Heat exchange area	3 m ²	5.6 m ²

However, for the second operating mode (M2) in which the system works as an indirect evaporative cooler, studying the wet bulb thermal effectiveness is preferable [8], and as the heat capacity rate of return air is always determinant in this case due to the energy involved in the water evaporation, it can be expressed as [24]:

$$\varepsilon_{WBT} = \frac{T_{o1} - T_{o2}}{T_{o1} - T_{WB_{e1}}} \quad (3)$$

To determine the amount of energy involved in the process, and thus quantify the cooling achieved in the outdoor airstream used for ventilation, cooling capacity is defined:

$$E_{CC} = \dot{m} \cdot (h_{o1} - h_{o2}) \quad (4)$$

This parameter can be calculated as follows, because only sensible heat is involved:

$$E_{CC} = \dot{m} \cdot C_{p_a} \cdot (T_{o1} - T_{o2}) \quad (5)$$

The heat exchangers global heat transfer coefficient is determined through:

$$U = \frac{E_{CC}}{A \cdot \Delta T_{LM}} \quad (6)$$

Where:

$$\Delta T_{LM} = \frac{(T_{o1} - T_{e2}) - (T_{o2} - T_{e1})}{\ln\left(\frac{T_{o1} - T_{e2}}{T_{o2} - T_{e1}}\right)} \quad (7)$$

Instead of directly studying this last parameter, thermal conductance will be defined as the product of the global heat transfer coefficient and the total heat exchange area, considered for the particular configuration of the heat-exchanger, to take these constructive characteristics into consideration:



$$U \cdot A \cdot f = \frac{E_{cc}}{\Delta T_{LM}} \quad (8)$$

Table 2. Design of Experiments

Systems studied	Operation modes	Outdoor Air Volume Flow V [m ³ /h]	Outdoor air Dry Bulb Temperature T [°C]
NP	M1- Dry (basic)	V1- 125	T1- 25
		V2- 200	T2- 30
WP	M2- Indirect evaporative cooling	V3- 300	T3- 35
		V4- 400	T4- 40

For low outdoor air temperatures, the scarce difference between this variable and the exhaust air dry bulb temperature, which is in comfort conditions, generates instabilities in the thermal effectiveness expression, eq. (1), and the logarithmic mean temperature difference, eq. (7), because the difference between these two temperatures in the denominator tends to 0. Thus, the parameters: dry bulb thermal effectiveness and thermal conductance are not representative for an outdoor air temperature at the inlet of 25°C, and will not be studied in these conditions.

4 Results and Discussion

4.1 Thermal effectiveness

Dry bulb thermal effectiveness is studied for both systems in the first operating mode, whereas wet bulb thermal effectiveness is observed when operating as indirect evaporative coolers. Table 3 gathers results obtained in these cases.

For both systems, thermal effectiveness does not vary significantly with outdoor air entering temperature. The same happens with the wet bulb thermal effectiveness. This is because the temperature drop clearly increases for higher values of outdoor air dry bulb temperature at the inlet, due to the increased heat transfer obtained as a consequence of the also higher temperature difference between both airstreams; but as the maximum temperature drop achievable also increases with the outdoor air temperature, this last factor does not have effect on these two parameters. On the other hand, an increase in the outdoor air volume flow improves heat transfer, as will be seen in subsection 4.3. Consequently, for cases where outdoor air heat capacity is determinant, higher airflows provide better results. This can be observed comparing thermal effectiveness between 400 and 300 m³/h in M1. For the remaining cases in which exhaust air flow heat capacity is the determinant one, however, higher airflows incur in lower effectiveness, due to the expression of the parameter (see eq. (1)).

Studying both systems as simple heat exchangers, it can be seen that values obtained in M1 range from 0.28 to 0.46 and from 0.33 to 0.42 for the NP and WP, respectively. These results fall below the ones that could be expected in a theoretical unmixed cross-flow heat exchanger, which could reach from 0.4 up to 0.59 and 0.48 for each device.

Results for the NP in terms of dry bulb thermal effectiveness are significantly better than those for the WP, whereas this difference is not so outstanding when comparing both prototypes in



M2, in terms of wet bulb thermal effectiveness. This implies a negligible effect of modifying the heat exchange area maintaining the overall size when operating as indirect evaporative coolers, which could be due to the more appropriate constructive characteristics of the WP (hollow panels' width) in terms of air-water distribution in the return air side of the heat-exchanger.

These trends observed did not seem to apply so clearly to the lowest exhaust air volume flow, because operating conditions in some tests were not stable. It was checked that tests performed in these conditions showed deviations in the values registered for the exhaust air volume flow, which is successfully maintained close to 260 m³/h for the remaining tests. Unfortunately, for ventilation rates of 125 m³/h this variability resulted to be unavoidable in most cases, due to the higher exhaust air volume flow and consequent depression in the climate chamber. This generates uncontrolled conditions which affect operation of the whole installation, as for example during some tests when the relative humidity in the laboratory was afterwards checked to have been particularly high. Consequently, results for the lowest airflow tested would not be especially representative and thus are not provided.

Table 3. Thermal and wet bulb thermal effectiveness.

V [m ³ /h]	T _{o1} [C]	M1		M2	
		NP	WP	NP	WP
		ε _T [%]	ε _T [%]	ε _{WBT} [%]	ε _{WBT} [%]
400	40	0.42	0.36	0.29	0.31
	35	0.46	0.34	0.27	0.32
	30	0.39	0.33	0.23	0.34
	25	-	-	0.23	0.43
300	40	0.40	0.33	0.35	0.28
	35	0.33	0.35	0.30	0.28
	30	0.28	0.42	0.30	0.29
	25	-	-	0.27	0.27
200	40	0.46	0.37	0.40	0.33
	35	0.44	0.37	0.39	0.36
	30	0.43	0.38	0.37	0.34
	25	-	-	0.35	0.31

4.2 Cooling capacity

Contrary to the temperature drop and the thermal effectiveness, the cooling capacity is related to the air mass flow treated by the system. Consequently, it appears as a more interesting parameter in terms of performance, as energy involved is considered.

As can be seen in figures 5a and 5b, for the WP, it shows an increasing trend with the variation of the outdoor air temperature, and improves with higher volume flows, due to the improvement introduced in the heat transfer convective coefficients. The same behaviour was observed from the characterization of the NP [8].

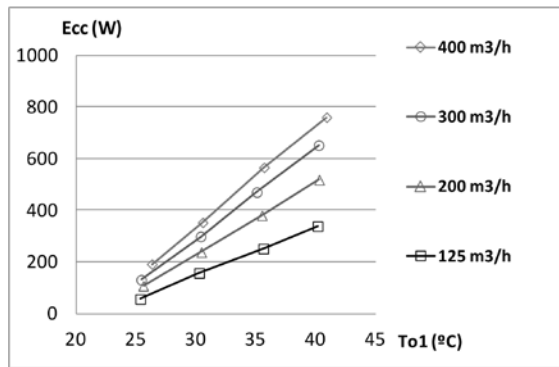


Fig. 5. a. Cooling capacity for WP, M1

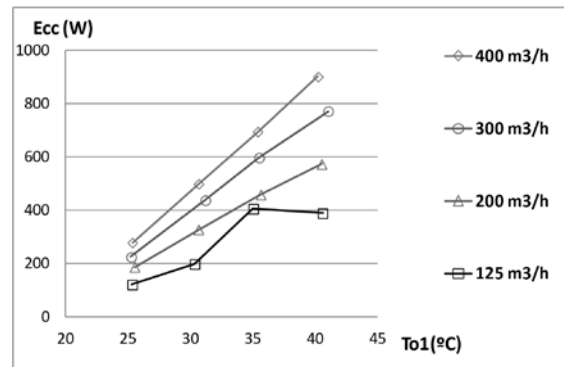


Fig. 5. b. Cooling capacity for WP, M2

Evaporative cooling implementation permits achieving higher cooling capacities in both systems. However, comparing results obtained for both prototypes, performance scarcely improves for NP system, in which the heat exchange area is twofold. Table 4 gathers the linear regressions obtained for the tests in M2. It was also seen in this case that results for the lowest air volume flows were less representative, especially for the WP.

Table 4. Linear regressions for systems operating as indirect evaporative coolers.

V [m ³ /h]	NP – M2		WP – M2	
	Linear regression	R ²	Linear regression	R ²
125	$Ecc = -385.80 + 20.973 \cdot T_{o1}$	0.9978	$Ecc = -370.48 + 19.754 \cdot T_{o1}$	0.8335
200	$Ecc = -584.36 + 30.45 \cdot T_{o1}$	0.999	$Ecc = -467.76 + 25.82 \cdot T_{o1}$	0.9989
300	$Ecc = -789.59 + 39.99 \cdot T_{o1}$	0.9989	$Ecc = -651.88 + 34.87 \cdot T_{o1}$	0.9988
400	$Ecc = -817.36 + 42.00 \cdot T_{o1}$	0.9866	$Ecc = -782.96 + 41.82 \cdot T_{o1}$	1

4.3 Thermal conductance

The parameter defined as thermal conductance for the particular configuration (table 5) remains fairly constant with the outdoor air entering temperature. These results appear predictable, considering that the global heat transfer coefficient is related to the achieved cooling capacity, but neutralises the temperature drop value by introducing the logarithmic mean temperature differences. On the other hand, values for the thermal conductance improve when higher airflows are tested, which could be due to convective coefficients improvement in the heat exchanger outdoor air side as a consequence of being operating with such higher airflows.

Table 5. Thermal conductance

V [m ³ /h]	T _{o1} [C]	M1		M2	
		NP	WP	NP	WP
		UA·f [W/°C]	UA·f [W/°C]	UA·f [W/°C]	UA·f [W/°C]
400	40	263.4	181.4	258.0	194.3
	35	275.5	176.5	246.8	195.4
	30	258.8	190.7	238.7	207.9
300	40	235.6	166.1	224.1	162.3
	35	189.4	158.0	206.6	170.2



	30	164.1	168.4	229.7	182.5
200	40	198.4	136.8	185.3	134.9
	35	190.2	136.8	183.4	141.3
	30	188.8	142.1	192.2	150.2
125	40	74.2	89.2	135.8	92.5
	35	85.3	95.9	137.0	113.4
	30	97.8	94.2	154.2	93.4

The main comparison among systems performance must be raised in terms of heat transfer, where the two constructive differences have key roles. Actually, thermal conductance obtained in M1 is better for the NP, whereas in M2 improvement introduced by doubling the heat exchange area is not as good. Results also show that heat transfer is being improved in M2 for WP in a way that does not have effect on the NP. Considering that the thermal conductance for NP in M2 does not improve in the same proportion as it does for WP, it can be inferred that the narrow geometry of the polycarbonate panels in this case hinders the water distribution. A possible explanation could be that in the NP some paths inside the hollow panels might be filled with water while others would be completely dry. As in that case part of the device would be operating as a water-to-air heat exchanger and other part as it did in M1, its performance would worsen in comparison to the case where the entire device would be operating in an indirect evaporative cooling mode. This is because air evaporatively cooled in the return side would ideally reach the adiabatic saturation temperature, which coincides with that of recirculated water, and thus in the NP temperature differences between both sides all throughout the heat exchanger would not be the maximum achievable in M2.

Consequently, wider panels in the WP could be favouring air-water flow through them and thus the air evaporative cooling, improving the heat transfer process. The merely slightly better results observed in some parameters when doubling the heat exchange area (facing NP to WP) would be due to the effect of this characteristic being compensated by the improvement introduced by wider cross sections.

4.4 Analysis of Variance

Finally, to study the relative influence that each constructive issue has on results obtained, an Analysis of Variance is developed for every parameter calculated. Table 6 gathers the various factors percentage contributions.

Table 6. Percentage contributions of the factors varied in the tests and their interaction, obtained through an Analysis of Variance.

	SSP [%]	SSM [%]	SSV [%]	SST [%]	SSPM [%]	SSPV [%]	SSPT [%]	SSMV [%]	SSMT [%]	SSVT [%]	SSRError [%]
εT	3.21	-	44.01	0.24	-	14.41	10.17	-	-	14.10	13.86
εTBH	1.13	-	38.22	6.36	-	28.49	3.79	-	-	9.91	12.10
ECC	0.03	7.37	13.13	52.63	0.25	1.36	1.40	1.07	1.40	6.82	14.54
U·A	9.91	6.01	46.49	18.93	0.04	0.65	0.33	0.62	5.93	1.12	9.97



Results highlight the important role that factor outdoor air dry bulb temperature has on parameter cooling capacity. It is also proved the effect of air volume flow expected. This last factor has great weight on the thermal conductance, as it clearly determines the convective coefficients in the heat transfer process; as well as on the thermal effectiveness.

Finally, effects of factors “prototype” and “mode” do not appear to be so outstanding. This could prove the explanation proposed concerning the possible compensation of favourable effects introduced by increased heat exchange area and wider flow-paths through the polycarbonate panels. Larger heat exchange implemented in the design of the NP implies narrower flow-paths, and the opposite happens in the WP. Moreover, it has been deduced that larger areas improve systems performance when operating as simple heat exchangers, and that wider paths favour air-water distribution as indirect evaporative coolers. Consequently, their effect on each prototype and mode results would compensate, which is now corroborated in the ANOVA.

5 Ventilation cooling loads covered and energy savings

It can be observed from the experimental study developed on the prototypes behaviour that such simple and small devices can be interesting for pre-cooling ventilation air taking advantage of return air residual energy, especially when operating as indirect evaporative coolers; and thus reduce energy consumed by conventional systems to reach comfort conditions indoors. In this section, experimental results obtained are used to estimate success in covering ventilation cooling loads, and predict energy savings introduced with these devices.

In table 4 linear regressions for the cooling capacity were gathered, which appears to be the most interesting parameter when aiming to study the prototypes actual cooling potential. The second operating mode is the one considered, due to the better results observed in these cases. Only results for the three highest levels of outdoor airflow are considered, because the lowest airflow would not fit to the minimum real ventilation airstreams required in most cases, and furthermore might not be representative due to experimental variability already indicated.

As ventilation cooling demand would vary among different climates, four cases are studied, corresponding to different Spanish cities classified in the four categories distinguished in the Spanish norm according to climate harshness during summer [25], from milder to harsher climate: Bilbao, Valladolid, Madrid and Seville. Ventilation cooling load during summer season (from 1st of May to 30th September) is calculated from meteorological data through equation:

$$Q = \frac{V}{v_e} \cdot c_p \cdot (T_{o1} - 22^{\circ}\text{C}) \quad (9)$$

Where C_p is the air specific heat capacity (1012 J/(kg°C)), and the specific volume v_e would be calculated depending on the city altitude (a). Load is calculated considering an indoor set temperature of 22°C, for being this one the thermal comfort temperature set inside the climate chamber during the tests. Then, for the whole cooling period, the total ventilation



cooling demand will be the sum of the loads during the hours when $T_{o1} > 22^{\circ}\text{C}$. This will be the number of hours during which the devices will be operating (n). Values for a required ventilation airstream of 200, 300 and 400 m^3/h in the four cities considered are given in table 7.

Considering that the prototypes will be capable of supporting ventilation airstream cooling loads corresponding to their cooling capacity, there will be periods of time during which the systems will be able to cover the whole ventilation cooling load and even supply extra cooling capacity; and the remaining period they will only be capable of covering a certain percentage of this load. Percentages of covered ventilation load in each case have been calculated on this base, and are shown in table 7.

Table 7. Percentage of ventilation demand supported and percentage of energy saved expected.

Location	V [m^3/h]	E_{vcd} [kWh]	NP – M2		WP – M2	
			E_{vcc} [%]	E_{s} [%]	E_{vcc} [%]	E_{s} [%]
BILBAO a = 6 m n = 744 hours	200	130.02	80.35%	51.51%	79.48%	50.48%
	300	195.03	67.27%	48.14%	68.88%	49.67%
	400	260.04	57.47%	42.93%	58.43%	44.08%
VALLADOLID a = 698 m n = 1082 hours	200	303.54	75.21%	57.36%	71.90%	54.05%
	300	455.32	61.98%	50.04%	62.41%	50.42%
	400	607.09	50.91%	41.95%	54.71%	45.74%
MADRID a = 655 m n = 1430 hours	200	391.37	74.52%	56.23%	71.63%	53.17%
	300	587.06	61.21%	48.95%	61.84%	49.60%
	400	782.74	50.90%	41.68%	53.94%	44.74%
SEVILLA a = 11 m n = 2172 hours	200	880.47	64.87%	52.53%	60.68%	48.38%
	300	132.70	54.22%	46.04%	52.50%	44.30%
	400	1760.94	44.33%	38.18%	46.40%	40.26%

The percentage of energy savings that could be achieved can be calculated as:

$$E_{\text{s}}[\%] = \frac{E_{\text{s}}}{(E_{\text{vcd}}/\text{COP})} \quad (10)$$

Where the energy saved will be the difference between the required electric energy consumed by a heat pump to support the ventilation cooling demand that could be covered by the alternative systems, and the electric energy consumed by them during their operation period:

$$E_{\text{s}}[\text{kWh}] = \frac{E_{\text{vcc}}}{\text{COP}} - n \cdot P \quad (11)$$

For the analysis, a Coefficient of Performance COP of 2.5 is considered for the conventional heat pump air-conditioning system; whereas an electric power consumption of 20 W would be required for the prototypes operation, corresponding to the water pump needed. Increase in fans power due to additional pressure drop introduced is assumed to be negligible due to the insignificant contraction and expansion of the airflow in the heat exchanger, and the short air paths of the prototypes. Measured pressure drops already validated this hypothesis.



Results gathered in table 7 show how such small and simple devices can be expected to cover from 45% to 80% of the ventilation cooling demand during summer period, depending on the ventilation airflow requirements and climate harshness. Furthermore, this leads to energy savings of around 50%.

Additional fan power required might nevertheless reduce these energy savings. The extra pressure drop depends on the air path length and the equivalent diameter, and on the friction factor, which is linearly dependant to the square root of Reynolds number for laminar flows. Consequently, small pressure drops measured and calculated for a plate heat exchanger according to Kays&London [26] validate these results in the case of the WP, where it never exceeds 1.2 Pa in the outdoor air side and is below 17 Pa in the exhaust air side, and thus no significant extra power consumption would be needed. On the other hand, pressure drop in the outdoor air side of the NP scarcely reaches 4 Pa, though in the exhaust air side increases up to 60 Pa due to the longer and narrower air paths. Nonetheless, pressure drop of 60 Pa when operating with an exhaust air flow of 260 m³/h would still involve low extra fan power, in comparison to the already existing pressure drops in the whole air ventilation system.

6 Conclusions

In this work, the recovering of an exhaust airstream from a conditioned space residual energy has been performed through small and simple polycarbonate heat exchangers of same size but different constructive characteristics: total heat exchange area and airstreams paths. They have been tested as indirect evaporative coolers to study how this effect could improve heat-recovery. It has been seen that implementing evaporative cooling in the heat recovery process leads to an increase in the cooling capacities of about a 13%.

These simple devices could achieve energy savings of around 50% by covering from 45% to 80% of the ventilation cooling demand during summer period, depending on the climate, for required outdoor airflows corresponding to the ones tested: 200, 300 and 400 m³/h.

Both prototypes operate better for higher outdoor air temperatures in whatever operating mode, which improves their behaviour in harsher climates, despite the greater difficulty of supporting expected ventilation cooling loads.

cooling capacity is always better when operating with higher air volume flows. The analysis of this parameter appears to be the most interesting one in terms of performance, because thermal effectiveness merely provides a reference to the maximum heat exchange affordable, which actually depends on global heat exchange coefficient and mass flow.

In an indirect evaporative cooling mode, thermal conductance does not vary significantly when modifying the system total heat exchange area maintaining the overall size, due to compensation with the better air-water distribution inside wider hollow panels. Consequently, it can be deduced that polycarbonate panels' width must be a determining characteristic in the systems performance when combining heat recovery and evaporative cooling together in a unique device.



Taking all this into account, the new prototype constructed with wider panels would provide results as interesting as the ones obtained from the original NP, of larger heat exchange area, with less material necessity for its construction. A twofold-area device constructed with wide panels would be much more sizeable and thus disregarded, as its compactness and cheapness particularly make these systems still more interesting than the separated-systems alternative proposed by the Spanish norm.

Nomenclature

t: Panel thickness (mm)

H: Prototype height (m)

d: Prototype width (m)

L: Prototype length (m)

V: Air volume flow (m³/h)

T: Temperature (°C)

ΔT : Temperature drop (°C)

ϵ_T : Dry bulb thermal effectiveness

ϵ_{WBT} : Wet bulb thermal effectiveness

E_{CC} : Cooling capacity (W)

h: Specific enthalpy (kJ/kgda)

U: Global Heat Transfer Coefficient (W/K·m²)

A: Area (m²)

UA: Thermal conductance (W/K)

m: Dry Air mass flow (kgda/s)

ΔT_{LM} : Log mean temperature difference (K)

f: Correction factor on the ΔT_{LM} for non-countercurrent heat-exchanger

SS_i : Percentage contribution of factor i (%)

SS_{ij} : Percentage contribution of interaction between factors i and j (%)

SS_{ERROR} : Percentage contribution of remaining uncontrolled factors and interactions (%)

Q: Ventilation cooling load (kW)

v_e : Specific volume (m³/kg)

C_p : Specific heat capacity of air (1012 J/(kg°C))

a: Altitude above sea level (m)

E_{VCD} : Total ventilation cooling demand (kWh)



E_{vcc} : Ventilation cooling demand covered [%]

E_s : Energy savings [kWh]

N : Number of operating hours (h)

P : Electric power consumed by the prototypes (W)

Abbreviations

NP: Narrow-panels prototype

WP: Wide-panels prototype

M1: Basic operating mode

M2: Indirect evaporative cooling operating mode

PIEC: Polycarbonate Indirect Evaporative Cooler

HP: Heat Pump

AHU: Air Handling Unit

Subindexes

o1: Outdoor air at system's inlet

o2: Outdoor air at system's outlet

e1: Exhaust air at system's inlet

e2: Exhaust air at system's outlet

wb: Wet Bulb temperature

t: Dry bulb temperature

da: Dry air

NP: Narrow panels prototype

WP: Wide panels prototype

P: Factor "prototype"

M: Factor "operating mode"

T: Factor "outdoor air dry bulb temperature"

V: Factor "outdoor air volume flow"

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