# EXPERIMENTAL CHARACTERIZATION OF AN INDIRECT EVAPORATIVE COOLING PROTOTYPE IN TWO OPERATING MODES

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## Abstract

The present paper aims to describe the experimental study developed to characterize an indirect evaporative cooling system made of polycarbonate, designed and manufactured by the Thermal Engineering Group of the University of Valladolid; as well as to introduce the main results obtained.

The prototype is characterized by a total heat exchange area of 6 $m^2$ and is installed in a heat recovery cycle in the experimental setup constructed in the laboratory. This setup mainly consists of: an AHU that enables the reproduction of the different climatic conditions to be tested; a climatic chamber where comfort conditions are to be achieved; a circuit to supply water during one of the operating modes; and the due ducts and measurement probes to properly connect the whole system and register the evolution of the interesting parameters.

Two operating modes are performed. In the first one, exhaust air from the climate chamber, in comfort conditions, goes through one side of the heat exchanger, producing heat transfer from the outdoor air stream through the plastic walls of the system. In the second case, an evaporative cooling mode is implemented by supplying water to the exhaust airstream.

Results obtained show that heat transfer through the heat exchanger polycarbonate wall improves in the evaporative cooling mode. Furthermore, both cooling capacity and thermal effectiveness of the system also increase in the second case. Moreover, global heat transfer coefficient and cooling capacity are improved by higher outdoor air volume flow rates. Finally, higher outdoor air temperatures imply better cooling capacities and thermal effectiveness.

## Keywords

Indirect evaporative cooling; heat-recovery cycle; global transfer coefficient; cooling capacity; thermal effectiveness.

# 1 Introduction

Nowadays, energy availability is essential for everyday life and welfare all over the world. Therefore, population and economic growth is expected to involve a faster increase in energy consumption, despite the rise in fossil fuel prices. Taking this into account, many problems such as dependency on sources, increased cost or the environmental impact of energy use and transformation are to be faced. Thus, new legislation to ensure sustainable energy provision at an affordable price is needed [1].

Particularly, around 40% of the total European energy demand is expected to be generated in the building sector, due to the high energy demand required to meet the Indoor Air Quality and thermal comfort expected in conditioned spaces. Furthermore, in some climates electric energy consumption for cooling the air in summer is acquiring more weight each year, incurring in important energy demand peaks. Considering that up to 20% of the energy involved in this field could be saved, it seems obvious that most of the new dispositions introduced by the European Union focus on energy saving in the building sector [2]. Consequently, a number of alternatives are being proposed for cooling buildings with low energy consumption [3, 4].

Recovering of residual energy associated to the exhaust airstreams from conditioned spaces becomes then the first objective for energy saving. The importance of this target is even greater considering the high ventilation rates required to ensure the current acceptable levels of Indoor Air Quality. Heat recovery technologies for building applications existing in the market are various and can be integrated into more complex energy-efficient air-conditioning systems. Heat recovery systems allow preconditioning of the supplied outdoor airstream to comply with the expected ventilation rates, thus reducing the energy demand to be supported by the HVAC systems installed to meet the required thermal comfort conditions [5].

Air to air heat recovery systems can operate either as simple sensible heat exchangers or permit total heat recovery. Despite the role they play in energy saving, the main disadvantages are their large dimensions, which complicates their installation; and high price, which increases the needed initial investment, though the power of the HVAC systems needed to operate in combination with them would be reduced [6, 7].

The application of the evaporative cooling phenomenon as a way to reduce energy demand and consumption in conditioned spaces is widening nowadays due to its cooling potential [8], which results of particular interest in some climates [9]. Moreover, it can be applied to improve the heat recovery process of the exhaust airstreams required in ventilation [10]. The key advantage of evaporative cooling lies in the fact that it is a natural phenomenon which occurs every time non-saturated air and water come into contact; water then simply evaporates into the air, reducing its temperature. Thus, it is a heat and mass transfer process, based on the transformation of sensible heat into latent heat.

Most evaporative systems work depends on the ideal process of adiabatic saturation. Theoretically, in this ideal process water is recirculated, maintaining its temperature close to the inlet air adiabatic saturation temperature. If this hypothesis is accepted, it can be assured that the whole transformation of sensible heat into latent heat is used to cool down the air, not the water. However, water can be expected to gain some heat loads while passing through certain devices such as pumps or pipes. For that reason, the adiabatic saturation temperature is merely the theoretical limit up to which water could be ideally cooled, though it is possible to achieve temperatures below this limit with certain configurations [11]. Furthermore, the possibilities of evaporative cooling are inversely proportional on the relative air humidity.

Particularly, an indirect evaporative cooling process will be proposed in this study. In this case, evaporative cooling is performed through one side of a heat exchanger, thus avoiding humidification of the airstream supplied to the conditioned space [12, 13, 14].

The objective of the study presented here is to characterize the behaviour of a heat exchanger prototype, made of an unusual material in such systems: polycarbonate, which is considered for its low cost and weight, and to prevent corrosion. Despite being unusual, plastic material for indirect evaporative coolers was firstly considered time ago by Pescod [15], disregarding the apparently inconvenient low thermal conductivity thanks to the small plate thickness. Further studies on these materials for heat exchangers have proved their feasibility [16]. Recent works have considered this sort of devices in combination with other systems or to pre-cool make-up air [17, 18].

The characterization is developed through an experimental analysis, in which the system undergoes different conditions of the airstream to be preconditioned, and operates through two different modes, seeking the determination of how implementing evaporative cooling would improve the energy recovery performance. This will permit to complement previous work on the field by combining the idea of implementing the evaporative cooling phenomenon to improve heat recovery in air-to-air systems, with the use of plastic heat exchangers, interesting for their lightness, cheapness and corrosion resistance.

# 2 Experimental setup

## 2.1 System description and construction

A polycarbonate heat exchanger was tested in summer conditions, to study the possibilities of recovering residual energy potential from exhaust air leaving from a conditioned space. The heat exchanger has 28 polycarbonate hollow panels of 4 mm have been arranged vertically and equally spaced. The polycarbonate thickness is 0.1 mm. Figures 1a and 1b show two views of the system under construction at different stages, while figure 2 shows a detail of a hollow polycarbonate panel. The main geometric characteristics are compiled in table 1.

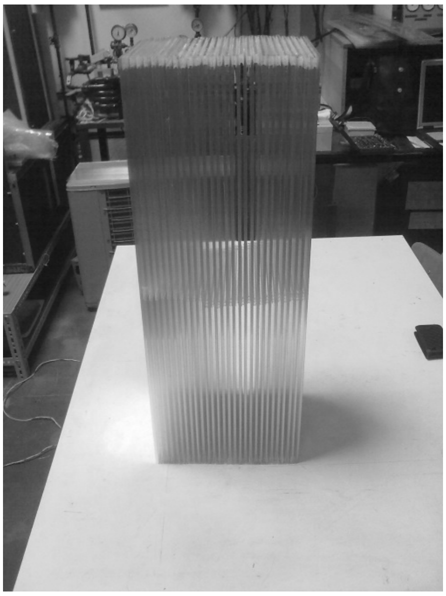

Figure 1a: View of the prototype under construction

Figure 1b: Assembly of the system

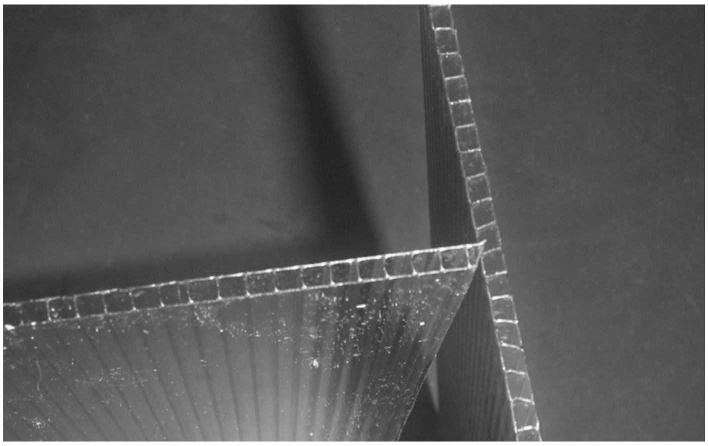

Figure 2: Detail of a hollow polycarbonate panel

Table 1: Geometric characteristics

| | |
|---|---|
| **Wall thickness** | 0.1 mm |
| **Panel thickness "t"** | 0.4 mm |
| **Height "H"** | 0.62 m |
| **Width "W"** | 0.18 m |
| **Length "L"** | 0.23 m |
| **Number of plates** | 28 |
| **Geometry** | Flat plates |
| **Heat exchange area** | 6 $m^2$ |

Like previous prototypes characterised by the authors [19, 20, 21, 22], this system has been designed and manufactured by the Thermal Engineering Group of the University of Valladolid.

The system is installed in a heat recovery cycle into the whole experimental setup constructed in the laboratory. During the normal operation of the system, outdoor air that is aimed to be pre-conditioned flows through the cross section of the panels, while exhaust air from a conditioned space flows upwards inside the hollow panels, as shown in figure 3. In this case, heat transfer is produced from the outdoor airstream in summer conditions to the exhaust airstream, which is in comfort conditions, through the plastic walls of the system.

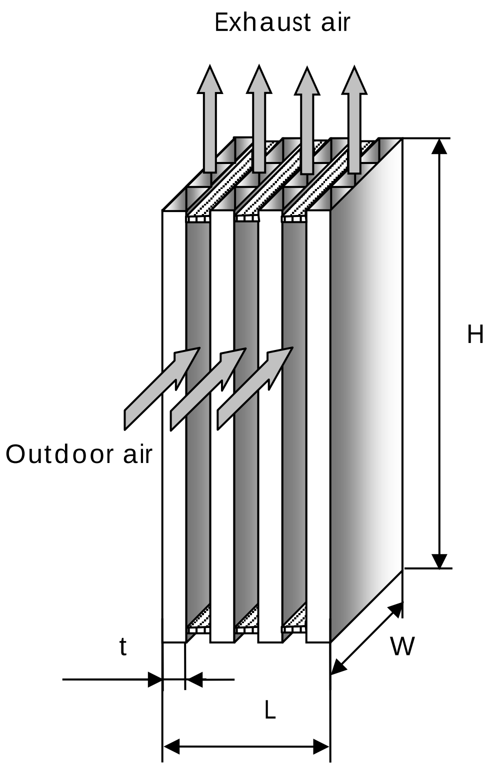


**Figure 3: Scheme of the system operation in the first mode**

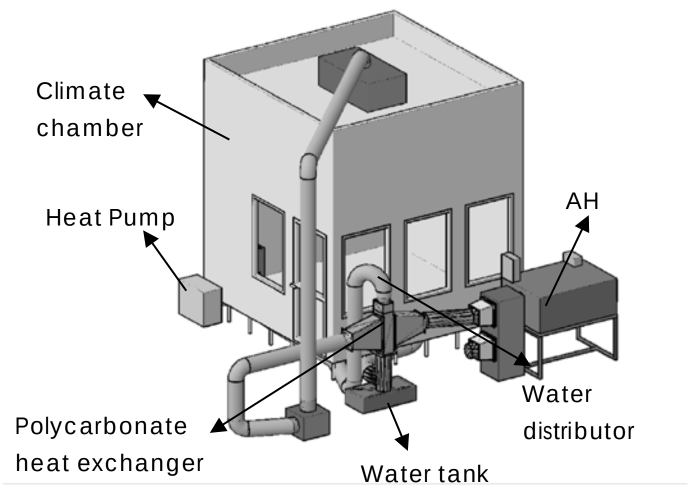


**Figure 4: Experimental setup**

A second operation mode is tested to study how the exploitation of the exhaust air energy potential is improved enabling evaporative cooling. For this purpose, two alternatives were considered initially. One possibility was to perform an adiabatic cooling of the exhaust airstream before passing through the system, by spraying water. However, this option, which is the one proposed in the Spanish norm [23], was rejected in favour of a second one, which consisted of performing the evaporative cooling inside one of the sides of the system. This second possibility was preferred because it could be expected better heat transfer through the polycarbonate walls due to the presence of water on the surface of one side of the heat exchanger.

To implement this last option, water is supplied from an upper water distributor, flowing downwards and counterflow in direct contact with the exhaust airstream. Thus, the system operates as an indirect evaporative cooler.

The whole experimental setup, as shown in figure 4, consists of:

- An Air Handling Unit (AHU) that provides the outdoor airstream volume flow needed for ventilation, in the studied summer conditions.
- A climate chamber, whose dimensions are 4x4x3m3, where the environmental comfort conditions have been created with the aid of a heat pump.

Table 2: Design of Experiments

| Operation modes | Outdoor Air Volume Flow V [m³/h] | Outdoor air-Dry Bulb Temperature T [ºC] |
|---|---|---|
| M1- Dry (basic) | V1- 125 | T1- 25 |
| | V2- 200 | T2- 30 |
| M2- Indirect evaporative cooling | V3- 300 | T3- 35 |
| | V4- 400 | T4- 40 |

– The air pipes connect the different elements of the air circuit mentioned above with the system. They allow directing the outdoor airstream from the AHU and the exhaust air from the climate chamber to the system, as well as supplying the pre-conditioned air from the system to the conditioned space. They also allow the exhaust air to be discharged to the environment at the outlet of the system.
– A water circuit including a lower water tank, a water distributor and a pump, that supplies water required in the second operation mode tested.
– Measurement equipment: to measure dry bulb temperature and relative humidity, Pt100 temperature probes and capacitive hygrometers are arranged in the outdoor and exhaust airstreams, at the inlet and outlet of the system. Besides, hardware and software are required to register these parameters. A previously calibrated orifice plate is used to measure pressure drop and air volume flow.

## 2.2 Design of Experiments

To perform the experimental characterisation of the prototype made of polycarbonate in the two operation modes introduced, two parameters have been varied and controlled, both related to the outdoor airstream used for ventilation: air volume flow and dry bulb temperature. Four levels of air volume flow and 4 further levels of dry bulb temperature are considered for both operation modes, as shown in table 2. Thus, a total of 32 tests have been performed. For the second mode of operation, water mass flow supplied by the distributor has been maintained constant at approximately 0.11 kg/s during all tests.

Dry bulb temperature and relative humidity have been measured for both outdoor and exhaust airstreams at the inlet and outlet of the system, once stationary operating conditions have been reached. From the registered measures, three parameters have been defined to study the operation performance of the system when both operating modes and the different outdoor airstream conditions are tested: cooling capacity, thermal effectiveness, and the global heat transfer coefficient.

# 3 Results and discussion

The following parameters have been defined:

– Thermal effectiveness [24]:
The dry bulb thermal effectiveness can be defined as follows:

$$\varepsilon_T = \frac{T_{O1}-T_{O2}}{T_{O_1}-T_{e1}} \quad (1)$$

However, for the second operation mode, in which the system works as an indirect evaporative cooler, it appears more interesting to study the wet bulb thermal effectiveness, because the minimum temperature up to which the air could be ideally cooled is the wet bulb temperature of exhaust air. Thus, it can be expressed as [24]:

$$\varepsilon_{WBT} = \frac{T_{o1} - T_{o2}}{T_{o1} - T_{WBe1}} \tag{2}$$

– The Cooling Capacity will permit determining the amount of energy involved in the process, and thus quantifying the cooling achieved in the outdoor airstream used for ventilation and that is aimed to be pre-conditioned.

$$E_{CC} = \dot{m} \cdot (h_{o1} - h_{o2}) \tag{3}$$

However, as the outdoor airstream does not have its humidity rate modified, this parameter can be calculated as follows:

$$E_{CC} = \dot{m} \cdot Cp_a \cdot (T_{o1} - T_{o2}) \tag{4}$$

– The Global Heat Transfer Coefficient determines the heat transfer performance of the system as a heat exchanger, and is defined as:

$$U = \frac{E_{CC}}{A \cdot \Delta T_{LM}} \tag{5}$$

Where:

$$\Delta T_{LM} = \frac{(T_{o1} - T_{e2}) - (T_{o2} - T_{e1})}{ln\left(\frac{(T_{o1} - T_{e2})}{(T_{o2} - T_{e1})}\right)} \tag{6}$$

The product of the global heat transfer coefficient and the exchange area will be called thermal conductance.

## 3.1 Thermal effectiveness

As can be seen from results represented in figures 5a and 5b, outdoor air temperature drop achieved in the system improves for higher values of this temperature at the inlet. This is caused by the higher temperature difference in relation to the exhaust airstream maintained at the comfort conditions established inside the climate chamber, thus increasing the heat transfer.

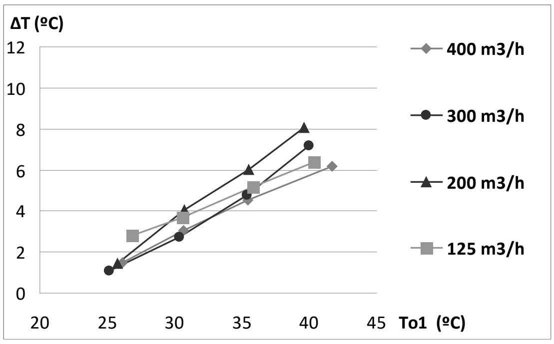


Figure 5a: Temperature drop of outdoor air, dry operation mode.

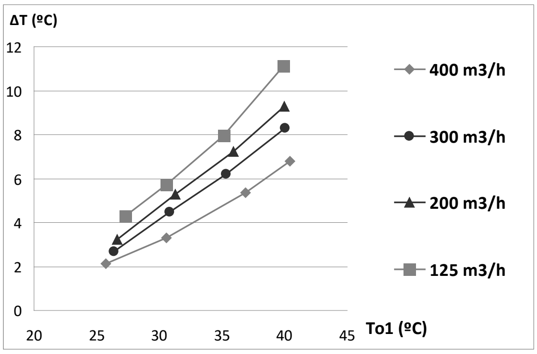


Figure 5b: Temperature drop of outdoor air, indirect evaporative cooling operation mode.

Moreover, the temperature drop is higher with lower air volume flows, because the residence time is longer and so it is the time while air is treated. It is also noticeable that the air volume flow has little relevance for low outdoor air temperatures at the inlet. Consequently, the effect of the temperature difference between the airstreams involved in the process is the most determinant factor in this case.

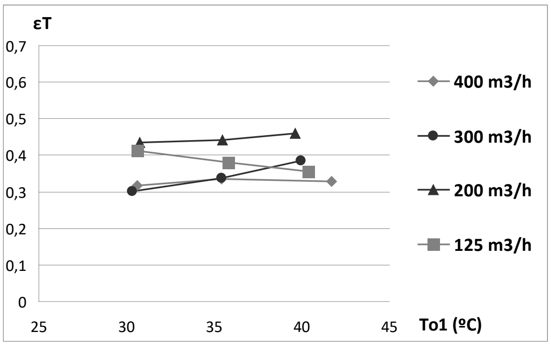


Figure 6a: Thermal effectiveness, dry operation mode.

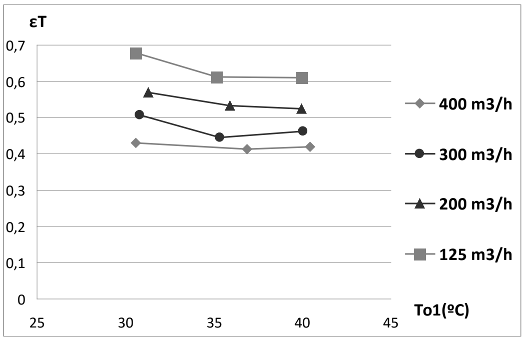


Figure 6b: Thermal effectiveness, indirect evaporative cooling operation mode.

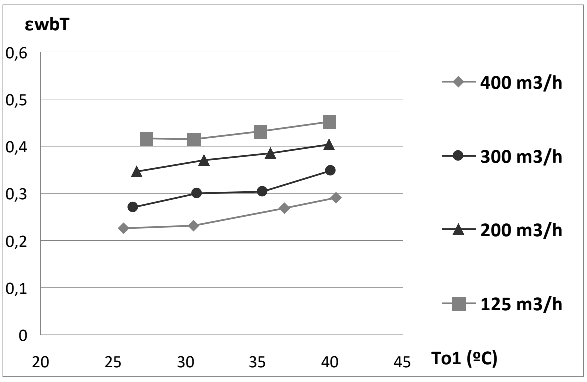


Figure 7: Wet Bulb Thermal Effectiveness, indirect evaporative cooling operation mode.

Another interesting result is the improvement in the temperature drops obtained in the second operation mode. As can be seen, the presence of water in the exhaust air side of the prototype allows obtaining higher temperature drops. This can be explained considering, firstly, the cold temperature of the water supplied, existing an important heat exchange between this liquid and the outdoor airstream; and secondly because the exhaust airstream temperature is also reduced by the effect of the evaporative cooling, and thus the temperature difference between both airstreams is increased, achieving a higher heat exchange between them.

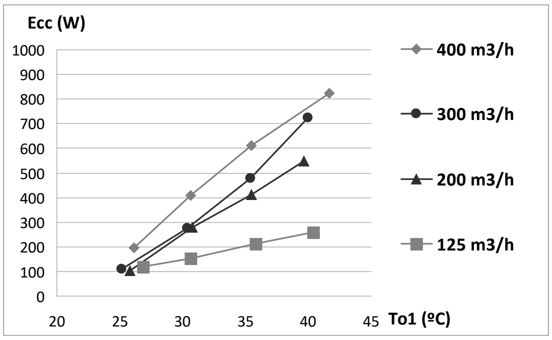


Figure 8a: Cooling capacity, dry operation mode.

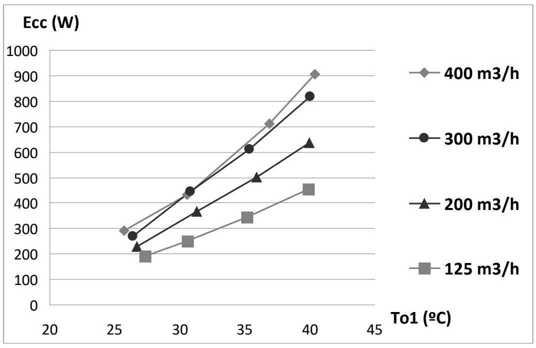


Figure 8b: Cooling capacity, indirect evaporative cooling operation mode.

The increase in film coefficients due to the presence of water would be a less remarkable cause, as it has not been observed any improvement in the global heat transfer coefficient (see figures 9a and 9b for thermal conductance).

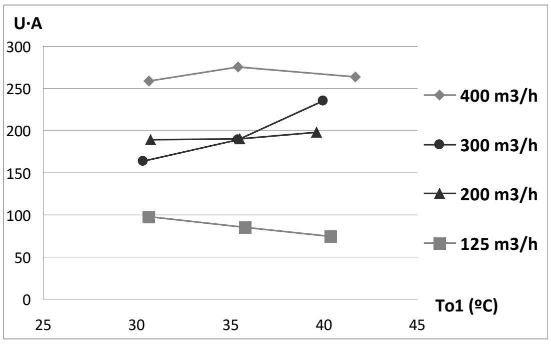


**Figure 9a: Thermal conductance, dry operation mode**

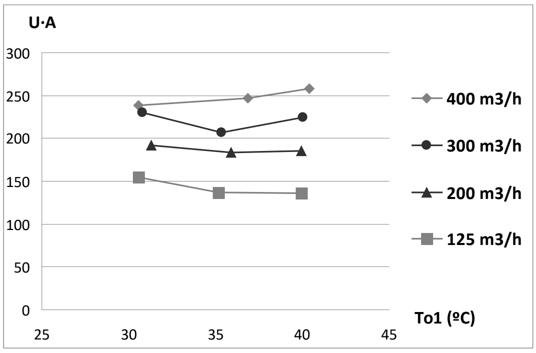


**Figure 9b: Thermal conductance, indirect evaporative cooling operation mode.**

All these trends are more clearly shown by the results regarding the second operation mode. However, those tests performed in the dry operation mode that adjust less clearly to the tendency expected showed deviations in the values registered for the exhaust air volume flow, which is successfully maintained around 260 $m^3/h$ for the remaining tests. Actually, exhaust air volume flows registered for tests M1-V1 at temperature levels T2, T3 and T4; and tests M1-V2-T1, M1-V3-T2, scarcely reach 180 m3/h. In the case of the lowest tested supply air volume flow, this is unavoidable due to the higher exhaust air volume flow and consequent depression in the climate chamber. Nonetheless this is a rare case in air conditioning, as usually conditioned spaces are required to be over pressurised. Deviation in the other three cases would be due to variability in the experimental work.

Nonetheless, the thermal effectiveness, defined in relation to exhaust air dry bulb temperature at the inlet, does not vary significantly with the entering outdoor air-dry bulb temperature, but it really does with different air volume flows (figures 6a and 6b). Nevertheless, for the first operation mode in the same particular cases indicated before, deviations when the exhaust air volume flow varied from the level established can also be appreciated, though less clearly. Thermal effectiveness appears to be improved by lower air volume flows, which can be explained as it was done for the temperature drop.

Notice that, as happens in the case of the thermal conductance (figures 9a and 9b), results for an outdoor air temperature at the inlet of 25 ºC have not been considered in the figures below. This is because results for low temperatures are not representative, due to the scarce difference between this value and that of exhaust airstream, which incurs in instabilities in the expression of the logarithmic mean temperature difference.

However, for the second operation mode the wet bulb thermal effectiveness was also defined and considered a more interesting parameter than the expression related to the exhaust air dry bulb temperature in this case, for the reasons already introduced. Actually, results at low outdoor air temperatures can be considered now, as their difference from the value of exhaust air wet bulb temperature does not incur in instabilities of the parameter expression in any case.

It can be inferred from figure 7 that wet bulb thermal effectiveness shows a similar trend for lower air volume flows. A slightly increasing tendency for higher outdoor air temperatures can, however, be appreciated in this case, though in any case turns to be representative (see slopes given in figure 7). This could be due to a possible increase in the exhaust air dry bulb temperature, and consequently in its wet bulb temperature.

## 3.2 Cooling Capacity

The most interesting parameter in terms of performance is, however, the system cooling capacity (figures 8a and 8b). The cooling capacity presents the same increasing trend as the thermal effectiveness with the variation of the outdoor air temperature. However, in this case the parameter value improves for higher volume flows. This is because the cooling capacity considers the air flow rate (though expressed in terms of air mass flow instead of volume flow) treated by the system, which is not represented by the thermal effectiveness. Consequently, although lower air volume flows allow higher temperature drops due to the longer residence time, the amount of treated air is also lower, and thus the amount of involved energy. The smaller slope obtained at lower outdoor air volume flows in the first operation mode could be due to the already proposed cause that also lower exhaust air volume flows were obtained.

Comparing the cooling capacity between both operation modes introduced, it can be seen that this parameter also improves if the evaporative cooling phenomenon is involved in the process. In this case, better performance results can be explained abiding by the same reasons exposed before concerning the thermal effectiveness.

## 3.3 Global Heat Transfer Coefficient and Thermal Conductance

The system is basically a heat exchanger, which can operate as an indirect evaporative cooler if water is supplied through one of its sides, being in direct contact with the exhaust airstream.

Notice that the expression defined for 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. Thus, the results obtained in figures 9a and 9b, in which the thermal conductance maintains an approximately constant value, appear predictable.

During the design of the prototype, the main advantage that justified performing the evaporative cooling inside the system instead of upstream, was the fact that the presence of liquid improves the film convection coefficient in the heat exchange.

However, results do not clearly prove this, as can be seen in figures 9a and 9b, where only for the lowest tested air volume flow the thermal conductance presents a significant increase for the second operation mode. This could be explained considering that, due to the narrow section of the polycarbonate panels used for the manufacturing, for high air volume flows, which could be obtained in the exhaust airstream due to an overpressure in the climate chamber generated by high supply airflows, the section would not be wide enough to allow water flow through, and thus some paths of the section would be filled by downstream water, operating the remaining paths like they did in the first mode.

It is remarkable, however, the fact that overall values improve for the studied higher air volume flows. This could be expected because higher flows cause better convection film coefficients, and thus heat transfer is improved.

Notice that, as it was indicated in the case of the thermal effectiveness, results for the lowest outdoor air temperature at the inlet are not representative. This is due to the same reason already presented, as the defined expression of the global heat transfer coefficient becomes instable if this temperature approaches that of exhaust air at the inlet.

Moreover, for the first operation mode, those tests that show higher deviations from the expected constant tendency are the ones for which also deviations were registered in the exhaust air volume flow.

## 3.4 Comparison with existing alternatives

It has been argued that implementing the evaporative cooling phenomenon in one side of the heat exchanger was preferred to perform both processes separately, being this last option the one proposed in the Spanish Norm [23]. The proposed system usually consists of a Rigid Media Pad Evaporative Cooler installed in the exhaust airstream, then passing this airstream through a Rotary Heat-Recovery system to precondition the supply airstream.

The alternative selected thus results to be interesting for its greater compactness. Moreover, the use of plastic materials makes the heat exchanger lighter and cheaper and enables the implementation of the evaporative cooling phenomenon inside, avoiding corrosion. This also avoids possible mixing between exhaust and supply airstreams.

During the design, however, better convective film coefficients were expected in this alternative thanks to the existence of a water film on the heat exchange walls, which have not been perceived in the results for the operating conditions tested. Consequently, no noticeable improvement in the performance could be highlighted, in comparison to the separated-systems alternative. For this reason, an improvement in the design is proposed by increasing the polycarbonate panel thickness in order to permit higher water flows and better water distribution inside the paths, and thus better global heat transfer coefficients.

# 4 Conclusions

The prototype made of polycarbonate panels works as a heat exchanger used for exploiting the cooling potential of exhaust air from a conditioned space, and thus recovering residual energy. It has been designed considering the possibility of an alternative operation mode like an indirect evaporative cooler, and results have been studied to determine the improvement introduced in the energy recovery performance.

The global heat transfer coefficient has been defined to characterize the behaviour of the system as a heat exchanger. This parameter remains approximately constant with change of the outdoor air temperature, though increases for higher air volume flow rates. However, it is not a representative parameter for low temperatures. The introduction of an alternative parameter that could better represent the operation mode of the heat recovery systems by considering the relevant temperature differences involved in the heat exchange, as for example between airstreams at the inlet, appears necessary as future work to be developed.

The presence of water in one side of the heat exchanger does not provide noticeable better results, except for the case of the lowest air volume flow rate, apparently due to the narrow section of the polycarbonate panels. Thus, further interesting work is the redesign of the prototype using wider polycarbonate panels to avoid the dryness of some paths of the section in the second operation mode and consequently improve the film convection coefficients by the presence of water.

Both the achieved temperature drop, and the thermal effectiveness are improved by higher outdoor air temperatures at the inlet as well as for lower air volume flows, as expected. Better results are also obtained when the system works as an indirect evaporative cooler. However, the thermal effectiveness cannot either be studied for low temperatures on account of the inconsistencies introduced in the expression used for its calculation. It is more interesting for the second operation mode to consider the wet bulb thermal effectiveness.

The cooling capacity improves with higher air flow rates, as it considers the amount of air treated by the system. It also increases with outdoor air temperatures and improves significantly if evaporative cooling is performed. This parameter appears to be the one that best represents the system performance, since it considers the amount of involved energy.

Globally, higher outdoor air temperatures at the inlet incur in better performance. Although higher air volume flows undergo lower temperature drops, the amount of recovered energy is higher and thus the performance of the system is better.

The supply of water induces better performance results for the second operation mode, probably due to the increase in the heat exchange between outdoor air and exhaust air, which is evaporative cooled; and to the existence of heat exchange between outdoor air and cold water. The improvement in film coefficients is not remarkable, and wider polycarbonate panels should be used to permit this effect, avoiding preferable paths for water and air and thus ensuring their contact.

One advantage of the use of polycarbonate panels in a heat exchanger is that water can be supplied through one side without special treatments to avoid corrosion of the material. Moreover, although plastic materials have worse conduction coefficients than metals or other materials more commonly used in heat exchangers, the wall is thinner and thus the conduction heat transfer would not be necessarily decreased, being possibly increased. Consequently, heat recovery systems made of polycarbonate are expected to be smaller than those currently in the market, with the further advantage of being lighter. Moreover, this material presents another interesting advantage: its low price.

Recovering energy from exhaust air of conditioned spaces in summer is a wide field of study as well as one of great interest nowadays. Consequently, research on improving the heat exchange studying new materials and exploiting alternatives such as evaporative cooling appears not only as an interesting work but also necessary.

## Nomenclature

$\varepsilon_T$ : Thermal effectiveness.

$\varepsilon_{WBT}$ : Wet bulb thermal effectiveness.

$T_{o1}$: Outdoor air-dry bulb temperature at the inlet [ºC]

$T_{o2}$: Outdoor air-dry bulb temperature at the outlet [ºC]

$T_{e1}$: Exhaust air dry bulb temperature at the inlet [ºC]

$T_{e2}$: Exhaust air dry bulb temperature at the outlet [ºC]

$T_{WBe1}$: Exhaust air wet bulb temperature at the inlet [ºC]

$E_{CC}$: Cooling capacity [W]

$\dot{m}$: Dry Air mass flow [$kg_{da}$/s]

$h_{o1}$: Outdoor air specific enthalpy at the inlet [$kJ/kg_{da}$]

$h_{o2}$: Outdoor air specific enthalpy at the outlet [$kJ/kg_{da}$]

$Cp_a$: Air specific heat capacity [J/kgºC]

U: Global heat transfer coefficient [$W/m^2K$]

A: Heat exchange Area [$m^2$]

$\Delta T_{LM}$: Log mean temperature difference [K]

$\Delta T$: Outdoor air temperature drop [K]

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