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Design of highly compact indirect evaporative coolers

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Abstract

Evaporative cooling units are an effective alternative to conventional air conditioning technologies, due to their high efficiency and reduced primary energy consumption. There are two main types of evaporative cooling systems: the direct evaporative cooling (DEC) system, and the indirect evaporative cooling (IEC) system. DEC is based on direct contact between air and water, while IEC is based on heat and mass transfer between two flows of air, separated by a heat transfer surface with a dry side, where only air is cooled, and a wet side, where water is evaporated into air. The main objective of the present work was to design and manufacture a highly compact indirect evaporative cooler. Firstly, a mathematical model based on ε-NTU numerical method to determine the optimal geometrical and operating parameters of an IEC system was developed. The mathematical model allowed to obtain the temperature, enthalpy and humidity distributions of the air inside the exchanger. Then, the air-cooling system was manufactured. The device consisted of a compact heat and mass exchanger, a water distributing system and an outer casing. Finally, the IEC system was studied experimentally. An experimental facility was designed to study these air-cooling systems. The cooling unit performance indicators were the cooling capacity per unit volume and per unit airflow rate. The experimental results showed that the cooling capacity per unit volume of the device was 177 kW/m^3, and the cooling capacity per unit airflow rate was 10.9 kW/(m^3/s). These results suggested that highly compact indirect evaporative coolers can achieve air-cooling processes with a low energy consumption and a low environmental impact.

1. Introduction

European Union directives reinforced the objective of reducing primary energy consumption and the integration of renewable energies in buildings, instead of using fossil fuels [1]. A large percentage of current energy consumption and CO_2 emissions are due to heating, ventilating and air conditioning, HVAC, systems.

A traditional method widely used in air cooling is that of conventional HVAC systems based on direct expansion units [2]. However, direct expansion systems typically use refrigerant gases, which could emit polluting gases into the atmosphere, and in addition, they depend mainly on electrical energy.

Another air-cooling technology is evaporative cooling. The evaporative cooling units are based on heat and mass transfer between air and cool water [3]. There are two main types of evaporative cooling systems: the direct evaporative cooling, DEC, system and the indirect evaporative cooling, IEC, system. DEC is based on direct contact between air and water, while IEC is based on heat and mass transfer between two flows of air, separated by a heat transfer surface with a dry side, where only air is cooling, and a wet side, where water is evaporated into air [4]. In addition, there are different types of IEC [4]: conventional IEC, which supply air between the dry bulb temperature and the wet bulb temperature; dew-point evaporative cooler (DIEC), including single-stage counter-flow, and finally, Maisotsenko-cycle (MIEC), including multi-stage cross-flow. The last two supply air between the dry bulb temperature and the dew point temperature.

One of the most effective indirect evaporative cooling solutions are the dew-point counter-flow cycles, DIEC, [5]. The DIEC systems have been studied for many applications [6]. A DIEC with a direct expansion system applied to a residential building in China showed high potential for energy savings [7], up to 39% compared to a direct expansion system. DIEC systems were combined with desiccant systems [8–10], managing to control temperature and humidity independently.

IEC systems have been widely analysed by many authors in the available literature [11,12], in particular focusing on the analysis of influential parameters on outlet air conditions, such as inlet air temperature, inlet air humidity. The coefficient of performance of IEC systems was analysed for different operating conditions, evaluating the effect of variable inlet air velocities [13]. The amount of water flow rate was also analysed [14], which has a significant effect on system performance. The influence of the exchanger material was studied in other works [15].

The design of IEC systems was also analysed in recent works to improve their thermal behaviour. The cooling power of a DIEC system with different number of perforations between the dry and wet channels was analysed [16]. The results showed that the configuration with a single air passage increased the cooling power compared to configurations with four air passages. In another work, a mathematical model of DIEC was developed with different numbers of air passages. [17]. Two DIEC systems with cross flow configuration, one with fins and one without fins, were developed and analysed under different inlet air conditions [18]. The DIEC system with fins improved thermal performance up to 40% compared to the same system without fins. An optimization study of a DIEC unit with counter-flow configuration was developed [19]. The goal of the study was to increase the performance of the system by varying the air speed, the flow rate and the length and height of the channels. A DIEC with counter-flow configuration was manufactured from the results obtained of an optimal design with a mathematical model [20]. The energy performance of this system increased up to 19%.

Therefore, managing air with an efficient air-cooling unit, such as IEC systems, which does not depend mainly on electrical energy and does not use any refrigerant, could be an interesting alternative to conventional air-cooling units based on vapor- compression cycles. The main advantages of evaporative coolers are their constructive simplicity and high efficiency.

The main objective of the present work was to design and manufacture a highly compact indirect evaporative cooler. The cooling unit performance indicators were the cooling capacity per unit volume and per unit airflow rate. Firstly, a mathematical model based on modified ε-NTU numerical method to determine the operating parameters and optimal geometrical of an IEC was developed. Then, the air- cooling system was manufactured.

2. Materials and methods

The steps followed to develop the IEC system were as follows: (i) an IEC design was carried out and the ε- NTU mathematical model was applied to the IEC design; (ii) the numerical results were analysed; and (iii) the IEC system was manufactured.

2.1 Description of the indirect evaporative cooler

In the present work was designed an IEC system to handle air in small spaces. The air-cooling device consisted of a compact heat and mass exchanger, a water distributing system and an outer casing.

The heat and mass exchanger was designed with a counter-flow configuration. The air handling process consisted of an inlet air flow, which circulates through dry channels, exchanging heat with the attached channels, see **Figure 1**. The inlet air stream was divided into two air streams at the end of the device, one air stream was recirculated in the inverse direction through wet channels, exchanging heat and mass between air and water, and another stream was supplied, see **Figure 1**. Water only enters wet channels, so the supply air stream was cooled without varying its moisture content.

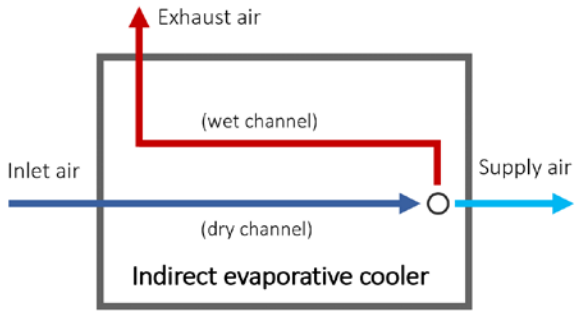

Figure 1. Diagram of the air flows of an IEC system.

2.2 Mathematical model of the indirect evaporative cooler

A mathematical model of IEC systems was developed to study the energy performance of an IEC system. This model was based on ε-NTU numerical method. The main equations of the model are expressed by (1)-(3).

$$NTU = U \cdot dA / C_{min,i} \quad (1)$$

$$U \cdot dA = \left(a \left[\frac{1}{\alpha_{d,i}} + \frac{\delta}{K} + \frac{\delta_w}{K_w} \right] + \frac{1}{\beta_i} \right)^{-1} N_{ch} W \, dx \quad (2)$$

$$\varepsilon = f(NTU, C_{r,i}) \quad (3)$$

Where NTU is number of transfer units; U is overall heat transfer coefficient [W·m^{-2}·K^{-1}]; A is area [m^2]; C is heat capacity rate [kg·s^{-1}]; a is slope of the temperature-enthalpy saturation line [kJ·kg^{-1}·K^{-1}]; δ is thickness of plates [m]; δ_w is thickness of water film [m]; $\alpha_{d,i}$ is; K is thermal conductivity of wall [W·m^{-1}·K^{-1}]; K_w is thermal conductivity of water film [W·m^{-1}·K^{-1}]; β_i is mass transfer coefficient for water vapor [kg·s^{-1}·m^{-2}]; N_{ch} is number of channels; W is width of the exchanger [m]; x is distance of sub-heat exchanger [m]; ε is effectiveness.

The IEC mathematical model developed allows to determine the optimal geometrical and operating parameters. The IEC mathematical model was implemented in *Engineering Equation Solver* (EES) software.

2.3 Manufacturing of the indirect evaporative cooler

The heat and mass exchanger of the IEC system was manufactured by using 3D printing techniques. 3D printing can be used to manufacture complex design prototypes at a low economical cost. However, the main limitation of this manufacturing technique could be the size of the prototypes.

A design of the manufactured IEC system is shown in **Figure 2**. It can be observed the heat and mass exchanger, a reservoir for water and adapter parts for testing the IEC system. The water distribution was carried out through perforations in the upper part of the device. The driven water came from the network. The exchanger was made up of 12 dry channels and 11 wet channels. The dimensions of the exchanger were 130 mm high, 140 mm long and 116 mm wide, and the material used to make the exchanger was resin [21].

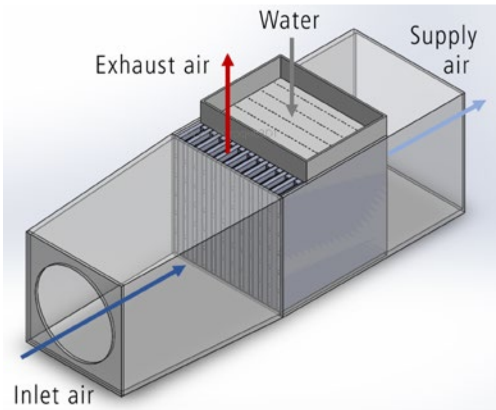

Figure 2. Design of the IEC system.

The design of a dry channel of the IEC system is shown in **Figure 3**.

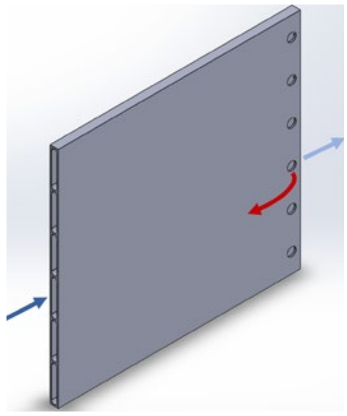

Figure 3. Design of the dry channels.

The thickness of the walls was 0.9 mm. These channels were composed of 6 perforations with a diameter of 5 mm. In addition, the wet channels were covered with a thin layer of cotton in order to retain the maximum amount of water.

2.4 Evaluation indexes of the indirect evaporative cooler

The performance of the IEC system designed was evaluated under the following ratios:

- Cooling capacity per unit volume of the IEC system, see Eq. (4).

$$C_V = Q_{cooling}/Volume \tag{4}$$

- Cooling capacity per unit airflow rate of the IEC system, see Eq. (5).

$$C_R = Q_{cooling}/V_{supply} \tag{5}$$

Where $Q_{cooling}$ was computed with energy balance on the dry-side fluid, see Eq. (6).

$$Q_{cooling} = \dot{m} \cdot (h_{out} - h_{in}) \tag{6}$$

Some commercial IEC systems obtained values of C_V and C_R of 5.1 kW/m³ and 23.5 kW/(m³/s), respectively [22].

A test facility was used to analyse the experimental performance of the IEC system under different inlet air temperatures, from 27°C to 46°C, and different inlet air flow rate, from 100 m³/h to 190 m³/h. The supply air flow rates were 75 m³/h for the inlet air flow rate of 100 m³/h and 115 m³/h for the inlet air flow rate of 190 m³/h. The value of inlet humidity ratio remained constant, 10 g/kg.

All the experimental tests were carried out under steady-state conditions. The sampling time was 3 seconds, and the values are averaged every 30 min. The accuracy of the measuring devices used was ±0.12°C for the temperature sensors, ±0.15°C for the dew-point temperature sensors and ±0.5% for the differential pressure transmitters.

The energy consumption of a fan to overcome the pressure losses of the heat exchanger was calculated with the values of friction losses and minor losses.

3. Results

The numerical and experimental results of the manufactured IEC system are shown in this section.

3.1 Results of numerical modelling

The mathematical model can be individually applied to n sub-heat exchangers of an IEC system. For the present work, 60 sub-heat exchangers were considered. As an example, the air conditions of the dry and wet air streams for each computational element of the exchanger are shown in **Figure 4**. This model

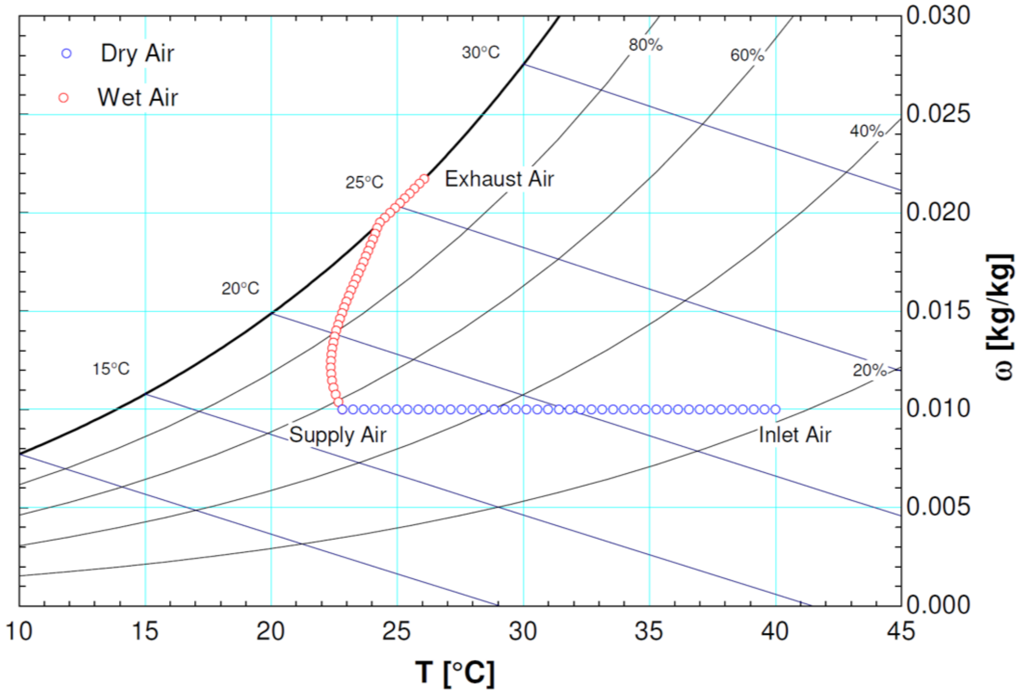

Figure 4. Psychrometric chart with the dry and wet air streams for each computational element.

allowed to obtain the temperature, enthalpy, and humidity distributions of the air inside the exchanger. Moreover, this mathematical model could be used to obtain the length of the heat and mass exchanger from pre-established inlet and outlet air conditions.

3.2 Experimental results

The result of the manufacture of the heat and mass exchanger of the IEC system is shown in **Figure 5**. It can be observed that the exchanger was stacked using dry and wet channels. The wet channels were covered with a thin layer of cotton in order to retain water, which was supplied from the top.

The experimental results of C_V and C_R for each case study are shown in **Figure 6** and **Figure 7**.

Figure 5. Heat and mass exchanger of the IEC system.

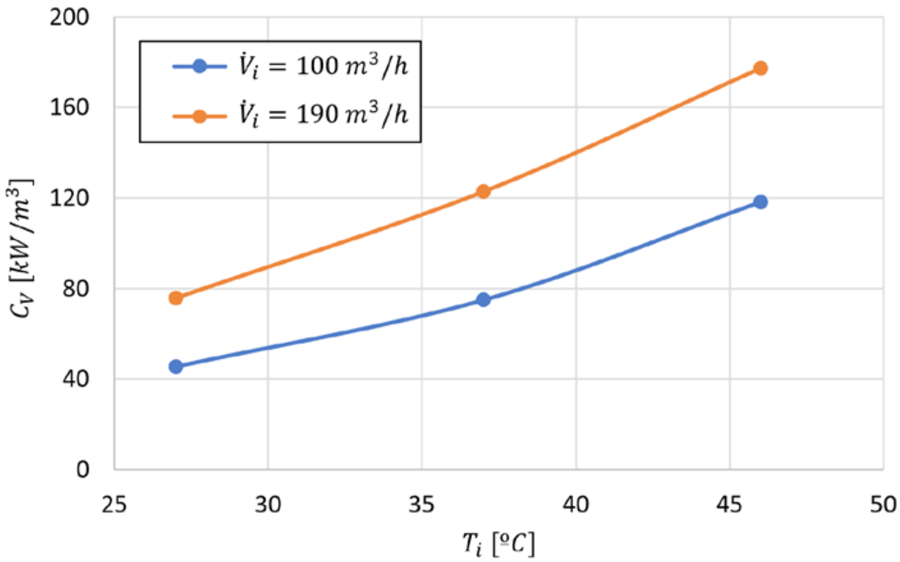

Figure 6. Results of cooling capacity per unit volume of the IEC system.

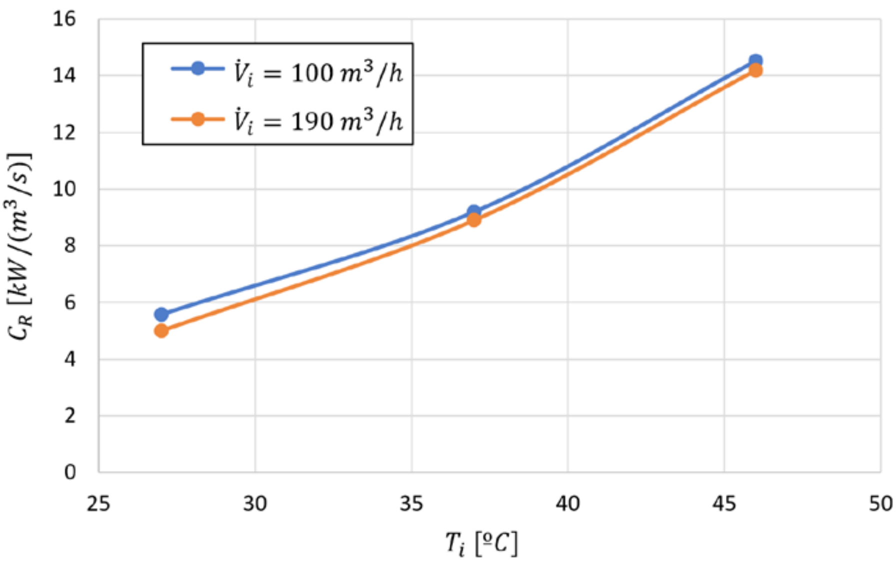

Figure 7. Results of cooling capacity per unit airflow rate of the IEC system.

It can be observed than the C_V values increased when the inlet air temperature was raised, see **Figure 6**. The IEC system also improved the C_V value when the inlet air flow rate increased from 100 m³/h to 190 m³/h, as shown in **Figure 6**. The maximum C_V value was 177 kW/m³ for an inlet air temperature of 46°C and an inlet air flow rate of 190 m³/h. However, the minimum C_V value was 45 kW/m³ for an inlet air temperature of 27°C and an inlet air flow rate of 100 m³/h. Therefore, the air-cooling device increased its cooling capacity for hot inlet air conditions and higher air flow rate.

The C_V values obtained in the present work were significantly higher than those obtained by commercial IEC systems, shown in section 2.4. Therefore, the device was designed and manufactured with high compactness.

Regarding C_R, similar trends to those obtained for C_V were found when the inlet air temperature increased. That is, the higher the air temperature, the higher the C_R value, as shown in **Figure 7**. However, C_R decreased when the inlet air flow rate increased. The maximum C_R value was 14.51 kW/(m³/s) for an inlet air temperature of 46°C and an inlet air flow rate of 100 m³/h, and the minimum C_R value was 5 kW/(m³/s) for an inlet air temperature of 27°C and an inlet air flow rate of 190 m³/h. For this ratio, the C_R values obtained in the present work were lower than those obtained by commercial IEC systems, shown in section 2.4. Therefore, the device needed to supply less flow to cool air.

4. Conclusions

In the present work, the geometric design and manufacture of an indirect evaporative cooling (IEC) system was carried out.

The design of the IEC system was achieved from numerical results obtained with a mathematical model based on ε-NTU numerical method. Moreover, this model allowed to obtain the temperature, enthalpy, and humidity distributions of the air inside the IEC system.

The experimental performance of the air-cooling device was evaluated under two ratios: cooling capacity per unit volume (C_V) and cooling capacity per unit airflow rate (C_R). The results showed high C_V values, up to 177 kW/m³, mainly for high inlet air temperatures and inlet air flow rates. The C_R results were higher for high inlet air temperatures and low inlet air flow rates, with a maximum value of 10.9 kW/(m³/s).

These results suggested that highly compact indirect evaporative coolers can achieve air-cooling processes with a low energy consumption and a low environmental impact.

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Data Statement

The datasets generated during and/or analysed during the current study are not available because the authors continue to investigate the air-cooling system, but the authors will make every reasonable effort to publish them in near future.

References

Please find the complete list of references in the htm-version at https://www.rehva.eu/rehva-journal ■