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Potential of indirect evaporative cooling to reduce the energy consumption in fresh air conditioning applications

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Abstract

The study presents a complex evaluation of the indirect evaporation cooling technology potential to reduce the energy consumption of the fresh air conditioning system, of an office building in the climatic conditions of Cluj-Napoca, Romania. The study was realized for the cooling season of one year based multiannual values of ambient temperature and relative humidity. It was considered two types of fresh air conditioning systems. The classic system was considered composed of a cooling battery powered by an electric chiller and the evaporative system was considered composed by a recuperative heat exchanger, indirect evaporative cooling equipment and the same classic cooling battery. It was found that the fresh air cooling period starts in May and ends in September. The performances of the two air conditioning technologies were compared and it was concluded that the evaporative system allowed the reduction of energy consumption for the fresh air cooling with almost 80%.

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Keywords: Indirect evaporative cooling; air conditioning; fresh air; wet bulb; energy efficiency; chiller

1. Introduction

The indirect evaporative cooling (IEC) represents old water based cooling technology which is rediscovered in our days, being still relatively unknown [1]. In this study IEC is evaluated from the energy efficiency point of view in fresh air cooling systems.

The most important particularity of IEC is represented by the use of water in the cooling process. Different reviews are presenting technical, technological, economical and scientific aspects about IEC [2-4].

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In the IEC systems, water is extracting heat by evaporation. IEC is based on heat and mass transfer between two streams of air, separated by a heat transfer surface with a dry side where only air is cooling and a wet side where both air and water are cooling.

Fundamental theoretical and physical aspects concerning the heat and mass transfer including those concerning the mixture between air and moisture, are presented in several textbooks [5-7].

Due to its very low energy consumption, IEC is considered suitable for many applications like heating ventilation and air conditioning (HVAC) for office buildings, supermarkets, cinemas, sport centres, data centres, etc.

In this context, the study and the evaluation of IEC potential to reduce the energy consumption in air conditioning applications is of high importance and interest. This study refers to an application located in Cluj-Napoca, Romania.

This evaluation is continuing the previous studies in the field of IEC and of heat and mass transfer, at the Technical University of Cluj-Napoca [8, 9].

The goal of the study is to elaborate and apply a methodology for evaluation the potential of IEC for energy reduction in the fresh air conditioning system of an office building, based on multiannual climatic data.

2. Material and method

The study is focused on the comparative analysis of two different fresh air conditioning systems. The classic system is based on a single heat transfer battery powered in cooling mode by a chiller and in heating mode by a boiler. The evaporative system is designed with a recuperative heat exchanger also called recovery unit, **IEC** equipment and the same classic heat transfer battery. The first system is referred as *classic* and the second system is referred as *evaporative*.

Both technologies are used for the fresh air conditioning of a virtual office building with ground floor and two floors, located in Cluj-Napoca, Romania, at 46.8° N latitude and 23.571° E longitude, with the dimensions of (80 x 18.75 x 12) m. The *classic* and the *evaporative* fresh air conditioning systems are considered to be placed on the roof terrace. The image of the building alone is presented in figure 1.

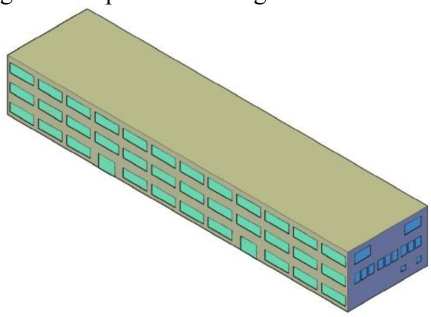

Fig. 1. The image of the virtual office building

The principle scheme of the *classic* fresh air cooling system is presented in figure 2.

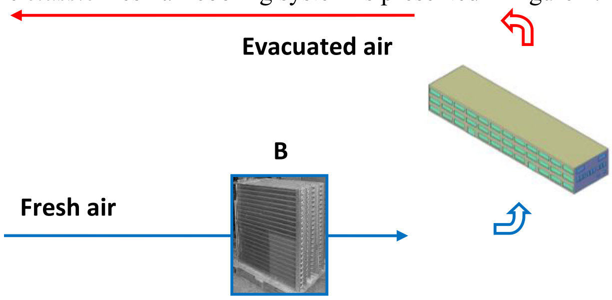

Fig. 2. Principle scheme of the classic fresh air cooling system. B – Heat transfer battery

The principle scheme of the *evaporative* fresh air cooling system is presented in figure 3.

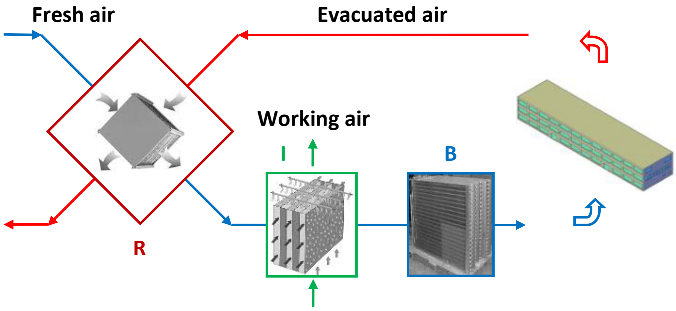

Fig. 3. Principle scheme of the evaporative fresh air cooling system
R – Recovery unit, I – Indirect evaporative cooler (IEC), B – Heat transfer battery

As input data of the study, were considered multiannual average climatic data corresponding to Cluj-Napoca, Romania, for the ambient temperature (t_e [°C]) and relative humidity (φ_e [%]). Both parameters are available as hourly variation, for the whole 8760 hours in one year.

The wet bulb temperature (t_{WB} [°C]) was calculated as function of temperature and relative humidity [10]:

$$t_{wb} = t_e \cdot atan[0.151977 \cdot (\varphi_e + 8.313659)^{1/2}] + atan(t_e + \varphi_e) - \\ - atan(\varphi_e - 1.676331) + 0.00391838 \cdot \varphi_e^{3/2} \cdot atan(0.023101 \cdot \varphi_e) - 4.686035 \tag{1}$$

The wet bulb temperature is important because it is also equal with the water temperature in the **IEC** equipment.

The yearly average variations of ambient temperature and of wet bulb temperature are presented in figure 4 and the yearly average variation of relative humidity is presented in figure 5.

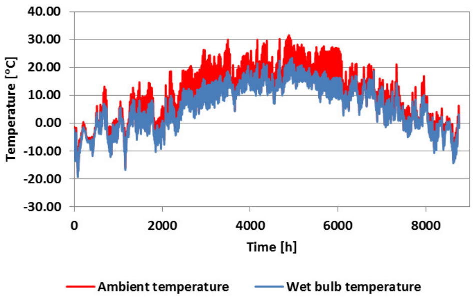

Fig. 4. Multiannual average variation of ambient temperature and of wet bulb temperature, for Cluj-Napoca, RO (one year).

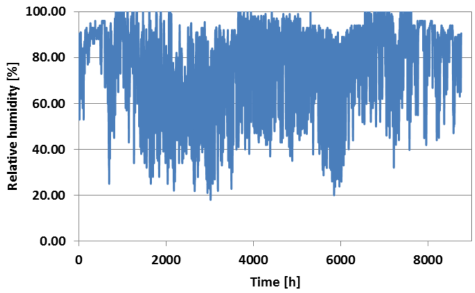

Fig. 5. Multiannual average variation of relative humidity for Cluj-Napoca, RO (one year)

The hourly cooling load of the office building ($\dot{Q}_{0b}$ [kW]) was calculated according to the national regulations, similar with the European regulations:

$$\dot{Q}_{0b} = \dot{Q}_{0t} + \dot{Q}_{0a} + \dot{Q}_{0g} \tag{2}$$

where:

$\dot{Q}_{0t}$ [kW] – Cooling power due to the heat transmitted through the building envelope

$\dot{Q}_{0a}$ [kW] – Cooling power requested by the fresh air

$\dot{Q}_{0g}$ [kW] – Cooling power due to building gains (illumination, computers, peoples, etc.)

The study was focused only on the fresh air cooling, because the **IEC** can contribute only to the cooling of fresh air. The cooling power requested to cover the heat loses through the building envelope and the building gains must be covered by other cooling technologies, most likely based on classic chillers.

The cooling power requested by the fresh air ($\dot{Q}_{0a}$ [kW]), was calculated as follows:

$$\dot{Q}_{0a} = \rho_{fa} \cdot (n \cdot q_n + S \cdot q_S) \cdot c_{fa} \cdot (t_e - t_b) \tag{3}$$

where:

ρ_{fa} = 1.25 kg/m^3 – Density of the fresh air, considered constant
n – The number of people in the building. It was considered that in the building are working 480 peoples between (8:00 – 16:00)
q_n = 25 m^3/h/pers – The fresh air flow rate requested by each person
S = 4500 m^2 – The total surface covered by the fresh air conditioning system
q_S = 1.26 m^3/h/m^2 – The fresh air flow rate requested for the unit of surface, considered only when the building is occupied
c_{fa} = 1 kJ/kgK – The specific heat of the fresh air, considered constant
t_b = 23°C – The blowing temperature of the air conditioned, considered constant

It can be observed that cooling of the fresh air is requested only when the building is occupied (8h/day).

It was considered that the fresh air cooling system must drop the fresh air temperature from the ambient temperature (t_e) to the blowing temperature (t_b), meaning that the requested cooling power is influenced by the ambient temperature.

It was assumed that the fresh air conditioning system must maintain inside the building a constant air temperature of 25°C. Taking into account the temperature variation on vertical direction, due to thermal stratification, at this temperature corresponds a blowing temperature of the air (t_b=23°C) and a temperature of the evacuated air (t_i=28°C). This thermal regime inside the building was considered constant.

The components of the *evaporative* system (**R**ecovery unit, **IEC** equipment and **B**attery) are functioning or not, depending by report between the air temperature at the inlet of each device and the different set point temperatures and temperature differences, as presented in figure 6.

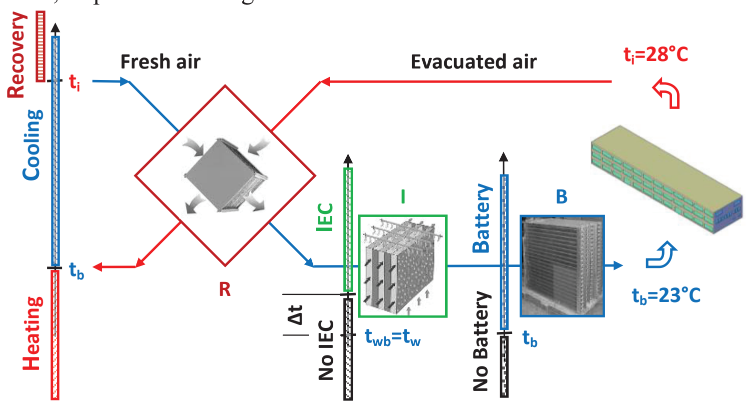

Fig. 6. Operating ranges and modes for the evaporative system components

The operating conditions and operating regimes of the evaporative system components are presented in table 1 function of the air temperature (t) at the inlet of each device.

Table 1. Operating conditions and operating regimes of the evaporative system components

Condition	Operating regime	Components functioning	Obs.
At inlet of R			
$t \le t_b$	Heating mode	R: No; IEC: No; B - Heating	B: Boiler is on
$t > t_b$	Cooling mode	B - Cooling	B: Chiller is on
$t > t_i$	Recovery mode	R: Yes	R: Normal operation
$t_i \le t < t_b$	Non recovery mode	R: No	R: By-pass is on
At inlet of I			
$t > t_{wb} + (\Delta t = 7°C)$	IEC mode	I: Yes	I: Water is on
$t \le t_{wb} + (\Delta t = 7°C)$	Non IEC mode	I: No	I: Water is off
At inlet of B			
$t > t_b$	Cooling mode	B: Yes	B: Chiller is on
$t \le t_b$	Non cooling mode	B: No	B: Chiller is off

The electric power of the chiller (P_c [kW]) was calculated with the relation:

$$P_c = \frac{\dot{Q}_{0a}}{COP} \tag{4}$$

where COP [-] – The coefficient of performance of the chiller.

COP is depending by the operating conditions, represented by the temperature of the cooled water and the ambient temperature of the cooling air. The thermal regime of the cooled water was considered constant of (5…12)°C and the evaporating temperature of the refrigerant in the chiller was considered equally constant of 3°C, corresponding to the thermal regime of the cooled water. The condensing temperature of the refrigerant in the chiller (t_k [°C]), was calculated as:

$$t_k = t_e + 15°C \tag{5}$$

The refrigerant was considered to be R410A which is very common in the electric chillers. The thermal calculation of the refrigerating cycle was realised with the Coolpack software. The values calculated for different ambient temperatures, were interpolated and it was obtained the following original equation of interpolation, for calculating the COP [%] as function only of ambient temperature t_e [°C]:

$$COP = 0.0035 \cdot t_e^2 - 0.3372 \cdot t_e + 10.287 \tag{6}$$

The equation is valid for ambient temperatures between (0…45)°C, for electric chillers with R410A, providing air conditioning.

The yearly electric energy consumption of the chiller (E_c [kWh]) was calculated as the sum of the instant electric power of the chiller for the whole number of hours in a year.

The *evaporative* technology configuration requires energy consumptions for the water pump operation, for the working (secondary) air fan operation, for covering the pressure drop of the fresh air in the **IEC** equipment and for covering the pressure drop of the heat recovery unit.

The electric power of the water pump used for the water recirculation in the **IEC** equipment (P_p [kW]) was calculated with the relation:

$$P_p = \left(\dot{V}_w \cdot \Delta p_w\right)/1000 \tag{7}$$

where:

$\dot{V}_w$ [m^3/s] – The volume flow rate of the water in the IEC equipment

Δp_w [Pa] – The pressure drop of the water in the IEC equipment

In similar application, the volume flow rate of the water in the IEC equipment is recommended at the value $\dot{V}_w$ = 0.13 l/s per 1000 cfm = 0.0765 l/s per 1000 m^3/h of air [11] and the pressure drop in IEC equipment is recommended as Δp_w = 9.1 m_{H2O} = 9.1 · 9800 Pa = 89180 Pa ≈ 0.89 bar [11].

Unlike the electric power of the chiller, the electric power of the recirculating water pump is constant in all the periods of pump operation ($P_p \approx 0.12$ kW).

The yearly electric energy consumption of the recirculating water pump (E_p [kWh]) was calculated as the sum of the instant electric power of the recirculating water pump for the whole number of hours in a year.

The electric power of the working air fan used in the IEC equipment (P_f [kW]) was considered similar with the electric power required to cover the fresh air pressure drop in the IEC equipment and was calculated with the relation:

$$P_f = \left(\dot{V}_a \cdot \Delta p_a\right)/1000 \tag{8}$$

where:

$\dot{V}_a$ [m^3/s] – The volume flow rate of the fresh and working air in the IEC equipment

Δp_a [Pa] – The pressure drop of the fresh and working air in the IEC equipment

The volume flow rate of the working air in the IEC equipment was considered similar with the volume flow rate of the fresh air and the pressure drop of both fresh and working air in the IEC equipment was considered equal and as recommended in a similar application [11]:

$$\Delta p_a = 22.86\ mm_{H2O} = 224.256\ Pa \approx 224\ Pa.$$

The electric power of the working air fans and required to cover the fresh air pressure drop in the IEC equipment were considered constant in all the operating periods ($P_f \approx 2 \cdot 1.099$ kW).

The yearly electric energy consumption of the working air fans (E_f [kWh]) was calculated as the sum of the instant electric power of the secondary air fan for the whole number of hours in a year, and was considered equal with the yearly electric power to cover the fresh air drop in the IEC equipment.

The pressure drop in the heat recovery unit was considered as 10% of the fresh air pressure drop in the IEC equipment. Similar percentage was considered for the energy consumption of the heat recovery unit.

By calculating the whole algorithm, the potential to reduce the energy consumption using the IEC in the fresh air conditioning, was completely evaluated by comparing the results obtained for the *classic* and for the *evaporative* technology.

3. Results and discussions

The mathematical model presented above, was implemented in Excel for the whole 8760 numbers of hours in the year.

The variation of the total cooling load of the building, corresponding to the climatic conditions of Cluj-Napoca, is presented in figure 7 and the variation of the cooling power requested by the fresh air, corresponding to the climatic conditions of Cluj-Napoca, is presented in figure 8.

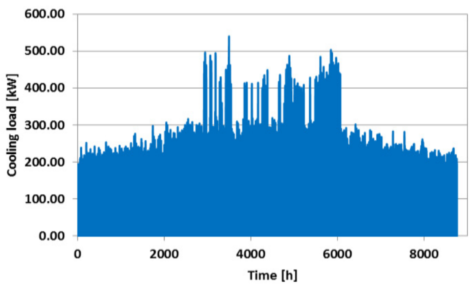

Fig. 7. The variation of the total cooling load of the building.

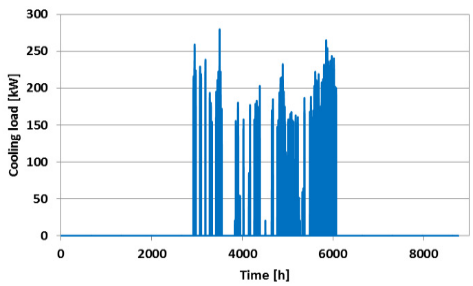

Fig. 8. The variation of the total cooling power requested by the fresh air.

The existence of cooling load from even in the cold season is determined by the high value of the heat gain from the building due to the 480 peoples (60 kW), to the 480 computers (96 kW) and to the illuminating system (31-63) kW. All these consumptions and loads appear only during the daily working time (7:00 - 20:00) and only in the working days of the week (Monday - Friday). The values of the instant solar radiation were considered when the instant power of the illuminating system was calculated.

It was observed that the period requesting cooling of the fresh air was (2 May – 8 September). The IEC equipment operating time in each month situated in the mentioned interval is presented in figure 9.

The total number of operating hours is 346 with an average of 69.2 hours/month. The operating period is in good agreement with [12] reporting (124…1407) hours/year in different regions of Italy, with [13] reporting (4.4 … 19.3) hours/month in Athens and [14] reporting (1300…1700) in Kuwait. Differences between the numbers of operating hours are normal for different climatic regions, because both temperature and humidity are influencing the conditions required for IEC operation.

Each component of the evaporative system (recovery unit, IEC equipment and battery) is functioning if the operating conditions are fulfilled. An example of operating of the three components is presented in figure 10, corresponding to the period: (13-14) July.

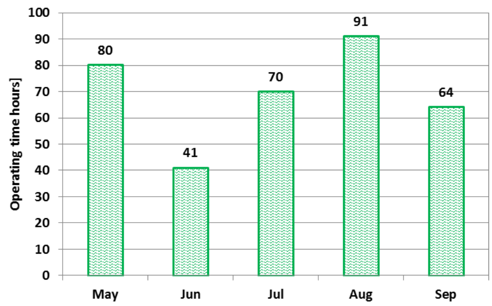

Fig. 9. The IEC operating time in each month.

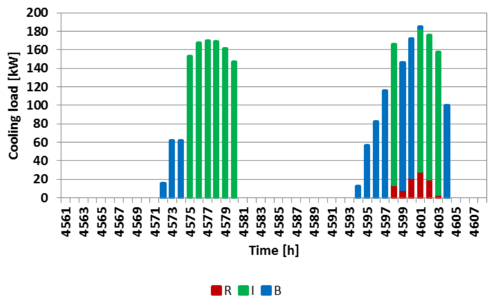

Fig. 10. Example of the three R, I and B operating shares (13-14) July.

The recovery unit (R) is operating only 6 hours in the selected period, sharing the cooling power with IEC (I) or with the Battery (B). In a single hour of the selected period all the three components of the evaporative system are operating in the same time. During the first day of the selected period, the operating conditions of the recovery unit are not at all fulfilled and the battery (B) must operate alone in the early hours while the IEC equipment (I) operate alone in the last hours.

It was found that the IEC equipment allowed the reduction of maximum requested cooling power of the battery from 277.6 kW in the classic system, at 161.8 kW in the evaporative system, representing an equivalent reduction of the cooling power of the chiller of 41.7%, corresponding to a significant reduction of the initial investment in the chiller.

The yearly cooling load is divided between the recovery unit (R) IEC equipment (I) and battery (B) as presented on figure 11.

The cooling load of the battery drops in the *evaporative* design from 100% to 16.3%. The IEC equipment takes over the largest part of the cooling load, representing 83.7% of the total cooling load and the recovery unit takes over only 1.5% of the total cooling load, proving to be inefficient in cooling mode.

The effect of the cooling load redistribution, mainly between the IEC equipment and the battery is the consistent reduction of the electric energy consumption. The yearly energy consumption in the *classic* design is of 19550 kWh/year and the yearly energy consumption in the *evaporativ*e design is of only 3127 kWh/year, of which 3085 kWh corresponds to the chiller and 42 kWh correspond to the water pump of the IEC equipment.

The shares of the yearly electric energy consumptions corresponding to the considered designs are presented in figure 12.

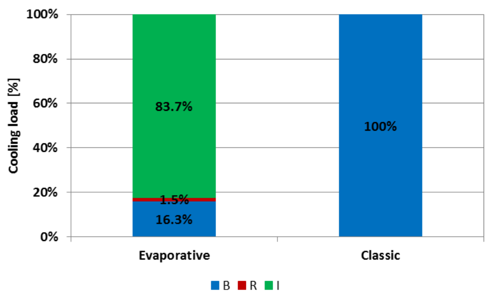

Fig. 11. The yearly shares of cooling load divided between R, I and B.

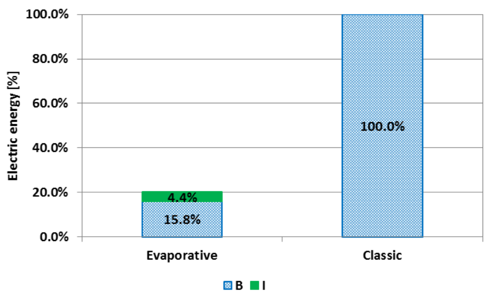

Fig. 12. The shares of the yearly electric energy consumptions between classic (B) and evaporative (I).

The total electric energy consumption due to the evaporative design is of (79.9 ≈ 80)% from the energy consumption of the chiller in the classic system.

A comparison between reported energy reductions in buildings air conditioning systems due to **IEC** is presented in table 2.

Table 2. Reported energy reduction is buildings air conditioning systems due to **IEC**

Reference	Year	Reported energy reduction
[12]	1999	(17…76)%
[14]	2001	(30…60)%
[15]	2009	(64…68)%
[16]	2011	90%
[2]	2012	80%, 90%
[17]	2013	(50…89)%
This	*2015*	*80%*

The highest degrees of reductions of energy consumptions were reported for data centre cooling and for warm and dry regions.

4. Conclusions

The study proved that IEC presents tremendous potential for energy saving in fresh air conditioning systems in large office buildings or similar applications. As larger is the air based air conditioning system the larger is the potential of IEC to save electric energy for cooling.

It was compared the performances of two air cooling technologies: a *classic* one based on a chiller and an *evaporative* one based on a combination between IEC equipment and a chiller. The considered *evaporative* design included also a heat recovery unit that proved to be inefficient in cooling mode because it reduced the cooling load with only 1.5%. By contrary it is estimated that during the heating season, the heat recovery unit became very efficient.

In the climatic conditions of Romania, particularly of Cluj-Napoca, the operating period of fresh air cooling was found to be from May to September, with 346 hours/year and an average of almost 70 hours/month. These data are in agreement with similar reported information.

It was found that IEC equipment was capable to reduce the requested maximum cooling power of the chiller with 41.7%. This conclusion lead to the possibility of using a lower size chiller with lower investment cost.

The IEC equipment was capable to take over 83.7% from the total cooling load, reducing the yearly electric energy consumption by 80%.

The obtained results, strongly recommend the use of IEC technology in air conditioning of buildings, but also in any type of cooling systems based on fresh air.

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