Energetic and economic savings of free cooling in different European climates

Renato M. Lazzarin and Marco Noro*

Department of Management & Engineering, University of Padova, Stradella S. Nicola, 3, 36100 Vicenza, Italy

Abstract

Free cooling is sometimes useful to face up not only ventilation load but also cooling load as well, taking advantage, directly or indirectly, of the enthalpy difference between inside and outside air in summer. The paper reports on the analysis of the energetic advantage of direct and indirect evaporative cooling (IEC) techniques conducted in summer in different European climates, in which external air flow is established by outdoor, indoor and inlet air enthalpies comparison. Psychrometric diagram has been divided into seven zones as a function of external air conditions; for each zone outside air flow control logic and obtainable free cooling effect have been developed and evaluated. As a function of indoor conditions, outdoor conditions (expressed hourly by Typical Meteorological Year), inlet temperature, external ventilation air flow and building cooling load, energy savings with free cooling techniques have been calculated, considering water consumptions in the adiabatic saturators. Interaction of free cooling, sensible heat recovery and IEC allows very interesting seasonal cooling load reductions, with the additional advantage of decreasing the chiller cooling capacity. All the energetic advantages have been evaluated by an economic point of view, in terms of net present worth and discounted payback period of the investment.

Keywords: free cooling; DEC; IEC; energy saving; economic saving

*Corresponding author: marco.noro@unipd.it

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1 INTRODUCTION

Free cooling techniques are dedicated to:

- reduce cooling load of buildings using their thermal inertia;
- reduce chiller cooling power for a given cooling load of the building.

The first is related to a deferred cooling of the buildings with respect to occupation time (i.e. during night time) in order to have better conditions for heat exchange with cold sinks and to reduce cooling load during occupation time [1,2]. The latter is related to the use of mechanical ventilation and/or heat exchangers, adiabatic saturators and evaporative cooling towers. In this case we usually differentiate between air-side free cooling and water-side free cooling techniques, as a function of the thermal fluid.

This paper reports on air-side free cooling techniques. They are usefully used in buildings where internal heat gain is high enough to need cooling also when outside air temperature is lower than desired inside air temperature. They are all founded on the possibility of varying outside air flow treated by Air Handling Unit (AHU), from the minimum (ventilation air flow G_{vent}) to the whole inlet air flow G_I. This can surely be done in variable air volume Heating, Ventilating and Air Conditioning (HVAC) plants (where cooling load is faced by fixing inlet air temperature and varying air flow), but in constant air volume plants as well (where cooling load is faced by fixing air flow and varying inlet air temperature).

These characteristics allow to realize the economizer cycles (they control outside air flow to reduce cooling load of the building) [3]. They can use evaporative cooling potentials, so they are called wet bulb economizer cycles (outside air flow is determinated by enthalpies comparison between outside, inside and inlet air flows), or they cannot, so they are called dry bulb economizer cycles (determination of outside air flow is done by respective temperature comparison).

Wet bulb economizer cycles (worth of a special attention because they allow to obtain greater energetic savings against a more critical control logic) can be classified into direct evaporative cooling (DEC) techniques and indirect evaporative cooling (IEC) techniques.

A simulation worksheet has been developed to evaluate, by means of hourly Typical Meteorological Year of four typical

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European climates [4], the possible energetic advantages using such techniques in an assigned room to be conditioned; the analysis has been extended to the economical aspects, pointing out convenience limits of different techniques. The software used for the present analysis has been tested and verified under experimental conditions [5].

2 CALCULUS PROCEDURE

2.1 Direct evaporative cooling

Cooling effect deriving from an adiabatic saturation process of an air flow (evaporative cooling) is well known: injecting liquid water at (quite) constant enthalpy causes humidity to increase and temperature to decrease. When the air flow is outside (external) air, we have direct evaporative cooling. Such a technique is useful when outside air enthalpy (h_E) is lower than inside air (h_A) and, at the same time, humidity ratio (x_E) is enough lower than inside air (x_A) to be able to face internal latent load (Figure 1).

Starting point of the analysis [6] is the knowledge of inside air conditions (t_A, φ_A), outside air hourly conditions, inlet air temperature t_I and sensible and latent cooling loads (Equation 1):

$$P_{sens} = G_I \cdot c_{p,a} \cdot (t_A - t_I), \quad P_{lat} = G_I \cdot r_s \cdot (x_A - x_I) \tag{1}$$

From the two possible expression of latent load P_{lat}, inlet air humidity ratio x_I and enthalpy h_I can be deduced (Equation 2):

$$\left.\begin{aligned} P_{lat} = G_v \cdot r_s = G_{vent} \cdot \Delta x_{sp} \cdot r_s \\ P_{lat} = G_I \cdot r_s \cdot (x_A - x_I) \end{aligned}\right\} \Rightarrow x_I = x_A - \frac{G_{vent}}{G_I} \cdot \Delta x_{sp}$$
$$h_I = c_{p,a} \cdot t_I + x_I \cdot (r_0 + c_{p,v} \cdot t_I) \tag{2}$$

There is a minimum value of the outside air enthalpy ($h_{E,min}$) for which total cooling load is completely faced by ventilation air flow G_{vent} (Equation 3).

$$\begin{aligned} P_{tot} = P_{sens} + P_{lat} = G_I \cdot (h_A - h_I) = G_{vent} \cdot (h_A - h_{E,min}) \\ \Rightarrow h_{E,min} = h_A - \frac{G_I}{G_{vent}}(h_A - h_I) \end{aligned} \tag{3}$$

h_A, h_I, $h_{E,min}$ and x_I allow to divide the psychrometric chart into seven parts, as a function of outside air enthalpy h_E (Figure 2).

Table 1 reports the zones division with respective description, outside air flow G_E that maximize free cooling effect and the two components (sensible and latent) of the latter.

2.2 Indirect evaporative cooling

In this case (Figure 3), inside air flow G_O is humidified before being exhausted: its temperature decreases so that it can be possible to cool outside air flow by means of a sensible heat exchanger.

In this case it is possible to get a free cooling effect also when outside air enthalpy h_E is greater than inside air h_A. It is sufficient that enthalpy after indirect heat exchange (h_X) is lower than h_A. The effect is to extend D2 zone, using the lower enthalpy of inlet air (outside air flow G_E can be increased till the whole inlet flow G_I to maximize free cooling effect).

To establish to new separation line between D1 and D2 zones, it is necessary to calculate, as in the previous case, the $h_{E,min}$ value. So it is possible to calculate the temperature of the exhausted air flow G_O just after the adiabatic saturator AC, t_C (Equation 4) (usually $G_O = \alpha \cdot G_E$ with $\alpha \leq 1$ to have a slight extra-pressure in the inside ambient):

$$\begin{aligned} \eta_{AC} = \frac{t_A - t_C}{t_A - t_{satadiab}(t_A; x_A)} \\ \Rightarrow t_C = t_A - \eta_{AC} \cdot (t_A - t_{satadiab}(t_A; x_A)) \end{aligned} \tag{4}$$

In the best case, G_E can reach t_C temperature (and x_E humidity ratio): these two conditions determinate the lowest enthalpy value of outside air flow ($h_{X,min}$) (Equation 5).

$$h_{X,min} = h(t_C; x_E) \tag{5}$$

So h_X can be calculated by the knowledge of the sensible heat exchanger efficiency ξ (heat exchanger efficiency is not constant at nominal value, but it is slightly increased in the presence of fog caused by extra-saturation of air immediately after saturation of exhausted air; we take into account of this factor by mathematical relations based on experimental data [5]) (Equation 6):

$$\begin{aligned} \xi = \frac{q}{q_{max}} = \frac{h_E - h_X}{\alpha \cdot (h_E - h_{X,min})} \\ \Rightarrow h_X = h_E - \xi \cdot \alpha \cdot (h_E - h_{X,min}) \end{aligned} \tag{6}$$

Now it is possible to determinate humidity ratio x_C and temperature t_X (Equation 7):

$$x_C = x(t_C; h_A) \Rightarrow t_X = t(h_X; x_E) \tag{7}$$

It must be $h_X < h_A$ to get energy saving increasing outside air flow besides the G_{vent} value. This implies that (Equation 8):

$$\begin{aligned} h_E - h_X \geq h_E - h_A \Rightarrow h_E \\ \leq \frac{1}{1 - \alpha \cdot \xi} \cdot (h_A - \alpha \cdot \xi \cdot h_{X,min}) = h_{E,max} \end{aligned} \tag{8}$$

Such relation represents the division line between D1 zone (where no energy saving is obtainable by AC saturator, while there is an energy saving due to heat recovery on exhausted air—but outside air flow has to be maintained to the minimum G_{vent}–) from D2 zone (where there is free cooling effect). The position of the division line depends on air flows

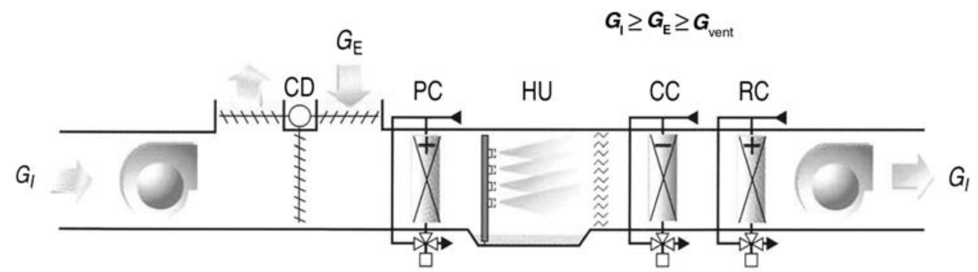

Figure 1. *AHU scheme with direct evaporative cooling (courtesy Carel Spa).*

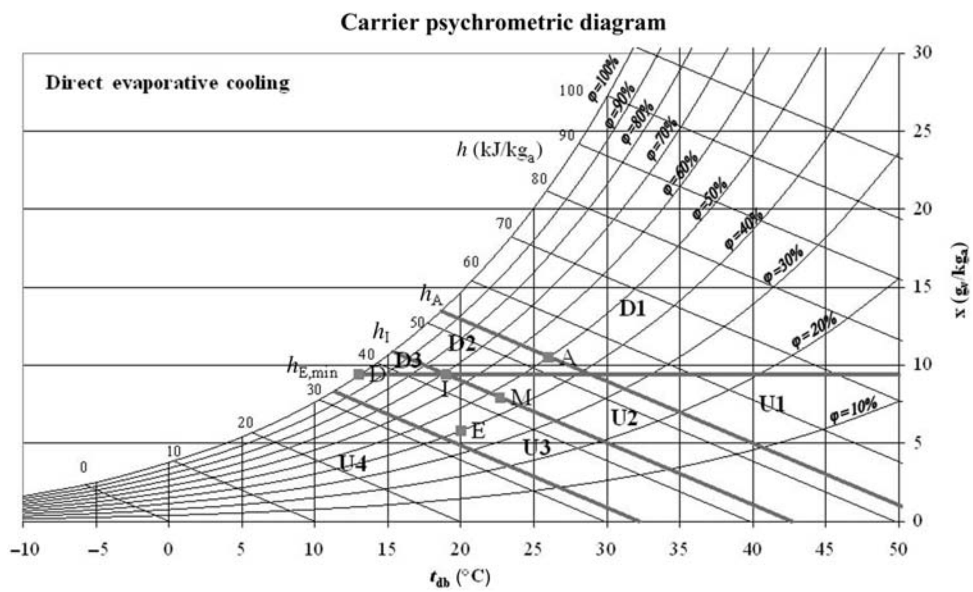

Figure 2. *Psychrometric chart with zones division with DEC and input data of Table 2.*

ratio α, heat exchanger efficiency ξ and on outside air humidity ratio x_E (that affects $h_{X,min}$ (Figure 4)).

2.3 Conclusion of the procedure

From this point the calculus proceeds in the same way for both the techniques, direct and indirect. Thermal power of the different coils (pre-heating, cooling, post-heating) to face sensible, latent and ventilation loads are calculated. So it is possible to calculate free cooling effect as depicted in Table 1 (with some little variations for IEC) and HU saturator efficiency (that is considered adjustable with fine control by means of high pressure water spray humidifier). To determine effective energy saving thermal power to dehumidify the possible outside air flow extra ventilation needs is subtracted. Water consumption of HU and AC saturators are then calculated. Finally, electrical energy saving is calculated as the cooling energy saving just determined divided by the electrical efficiency ratio (EER) of the chiller (Equation 9):

$$P_{\text{saved,input}} = \frac{P_{\text{saved}}}{\text{EER}_{\text{chiller}}} \qquad (9)$$

A more detailed description of the calculus procedure can be found in refs. [7, 8].

3 ENERGETIC ANALYSIS: AN EXAMPLE

It can be of use to consider a numerical example to better understand calculus conditions just described. Consider a room to be air conditioned with indoor conditions $t_A = 26°C$, $\varphi_A = 50\%$, outdoor conditions $t_E = 30°C$, $\varphi_E = 40\%$ and inlet air temperature $t_I = 19°C$. The HVAC plant is designed for 50 people occupancy, office work: latent and sensible loads are both 75 W per person [9], plus another 6 kW sensible load due to heat gain from external and internal loads such as PC, printers, photocopiers, etc. Outdoor ventilation air rate is 11 l/s per person [10], with a 0.9 ratio between exhaust air and ventilation air. Finally, efficiencies have been fixed at 50% for the sensible heat exchanger (recuperator) and 90% for HU and AC adiabatic saturators.

Figure 5 depicts values of the main variables in the AHU in case of DEC: because outdoor air enthalpy is greater than indoor air ($h_E = 57.3$ kJ/kg$_a$ $> h_A = 52.9$ kJ/kg$_a$) we are in D1

Table 1. *Psychrometric chart zones division with respective description, outside air flow G_E that maximize free cooling effect and the two components (sensible and latent) of the latter.*

Zone	Description	G_E	Free cooling	
			Sensible	Latent
D1	No energy saving zone	G_{vent}	0	0
D2	Zone with possibility of energy saving but with dehumidification	G_I	$G_I \cdot c_{pa} \cdot (t_A - t_M)$	$G_I \cdot r_s(t_M) \cdot (x_A - x_M)$
D3	Zone with possibility of energy saving but with dehumidification	$G_I \cdot (h_A - h_I)/(h_A - h_E)$	$G_I \cdot c_{pa} \cdot (t_A - t_M)$	$G_I \cdot r_s(t_M) \cdot (x_A - x_M)$
U1	Zone without energy saving and with dehumidification	G_{vent}	$G_I \cdot r_s(t_M) \cdot (x_I - x_M)$ + if ($t_A > t_M$; $G_I \cdot c_{pa} \cdot (t_A - t_M)$; 0)	$G_I \cdot r_s(t_M) \cdot (x_A - x_I)$
U2	Zone of partial free cooling and humidification	G_I	$G_I \cdot r_s(t_M) \cdot (x_I - x_M)$ + if ($t_A > t_M$; $G_I \cdot c_{pa}{}^* (t_A - t_M)$; 0)	$G_I \cdot r_s(t_M) \cdot (x_A - x_I)$
U3	Zone of total free cooling and humidification	$G_I \cdot (h_A - h_I)/(h_A - h_E)$	$G_I \cdot r_s(t_M) \cdot (x_I - x_M)$ + if ($t_A > t_M$; $G_I \cdot c_{pa} \cdot (t_A - t_M)$; 0)	$G_I \cdot r_s(t_M) \cdot (x_A - x_I)$
U4	Zone of heating and humidification	G_{vent}	$G_I \cdot r_s(t_M) \cdot (x_I - x_M)$ + if ($t_A > t_M$; $G_I \cdot c_{pa} \cdot (t_A - t_M)$; 0)	$G_I \cdot r_s(t_M) \cdot (x_A - x_I)$

zone, with no free cooling effect. This is the reason why outdoor air rate is the minimum (that is ventilation air rate G_{vent}, $0.011 \times 3600 \times 50 = 1980\ m^3/h$). The whole cooling load ($0.075 \times 50 + 6 = 9.8\ kW_c$ sensible load $+ 0.075 \times 50 = 3.8\ kW_c$ latent load) and the ventilation load ($1980 \times (1.225/3600) \times (57.3-52.9) = 2.9\ kW_c$) are satisfied by cooling coil ($25.3\ kW_c$) and post-heating coil ($8.5\ kW_t$). Cooling load determines also inlet air flow in $4000\ m^3/h$. All the variables in the figure are expressed in terms of power (in kW), but they refer to 1 h of operation so they are also energy (in kWh).

Indirect evaporative cooling, extending D2 zone limits, implies point E to be inside that zone. This determines a cooling power saving of $25.3-20.9 = 4.4\ kW_c$ on the cooling coil (Figure 6) due to a free cooling effect calculated by the program in $7.3\ kW_c$. This satisfies not only the whole ventilation load (always $2.9\ kW_c$) but also allows to face a great part of total cooling load (in this example $(7.3-2.9)/(9.8+3.8) = 32\%$) with 12 l/h water consumption of the AC adiabatic saturator and an increase, with respect to DEC, of the outdoor air flow that assumes the whole inlet air flow G_I value (Figure 6).

4 SEASONAL ENERGETIC AND ECONOMIC ANALYSIS

The choice of the resorts for the analysis has been done taking into account the mean seasonal values of the outdoor air temperature and relative humidity during the operation period of the HVAC plant (May–September). Four resorts have been selected representative of the main European climates according to Koppen classification (www.wikipedia.it) (Figure 7). We recall that *Cf* is a temperate and humid in all seasons climate (*Cfa* with a very hot summer—the hottest month has a mean temperature higher than 22°C, *Cfb* with a hot summer—the hottest month has a mean temperature lower than 22°C) and that *Cs* is a temperate with dry summer climate (*Csa* and *Csb* with the meanings just mentioned). Thus two resorts have been selected for the *Csa* climate, the most widespread in European countries (Hyeres, France and Sevilla, Spain), one for the *Cfb* climate (Milan, Italy) and one for the *Csb* climate (Porto, Portugal). Figure 7 depicts the resorts in the $\varphi_E - t_E$ diagram, justifying the reason of this choice: Sevilla is a meanly dry and hot climate, Hyeres a meanly humid and hot one, Porto a meanly humid and cool one and Milan a meanly not so humid and cool one.

Table 2 reports the inputs of the seasonal energetic and economic analysis.

A first result concerns the percentage incidence of the various psychrometric zones for the two evaporative cooling systems and for the different climates (Figure 8). Note the increase of D2 and U2 zones (the most favorable for a free cooling effect) with IEC technology, in connection with a reduction of D1 and U1 zones (the most disadvantageous).

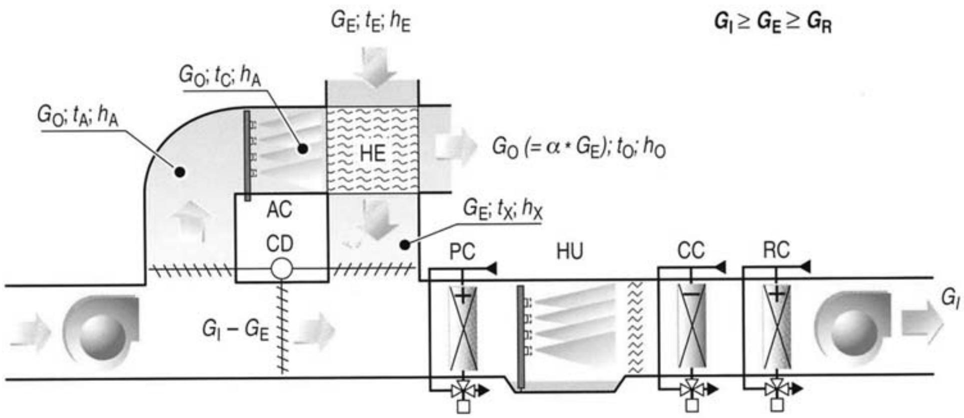

Figure 3. *AHU scheme with indirect evaporative cooling (courtesy Carel Spa).*

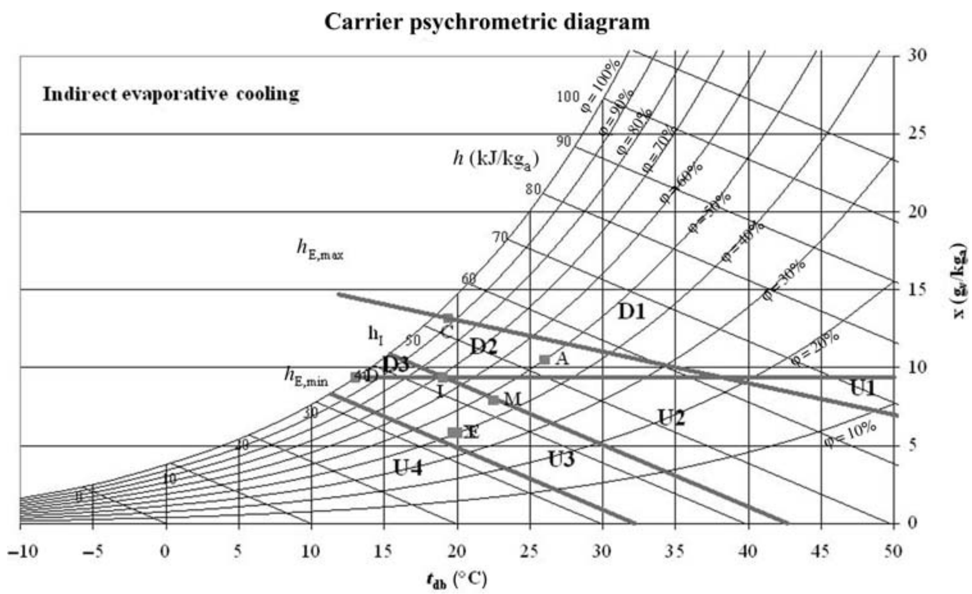

Figure 4. *Psychrometric chart with zones division with IEC and input data of Table 2.*

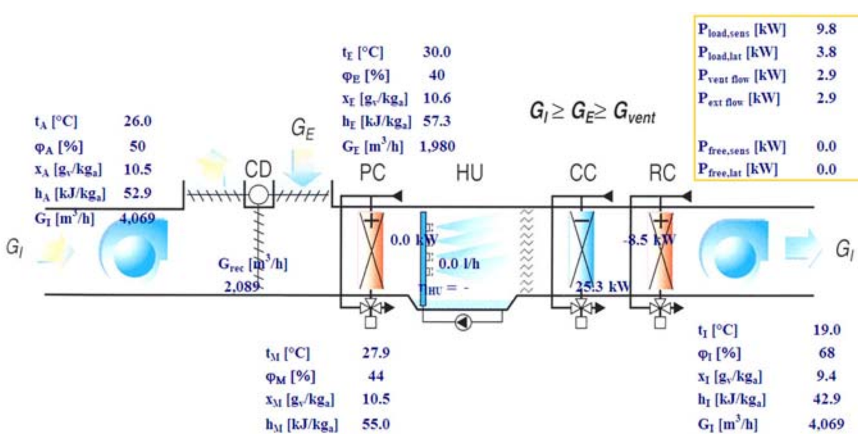

Figure 5. *DEC AHU scheme for the numerical example.*

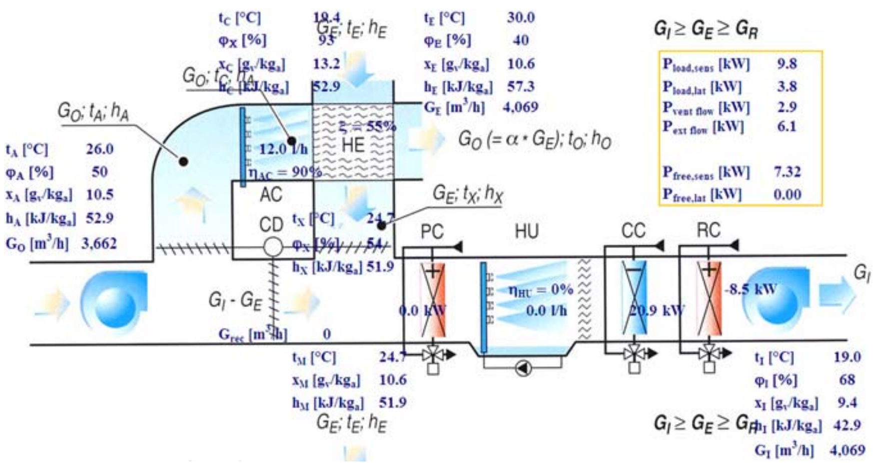

Figure 6. *IEC AHU scheme for the numerical example. Negative values refer to thermal effect gained by air.*

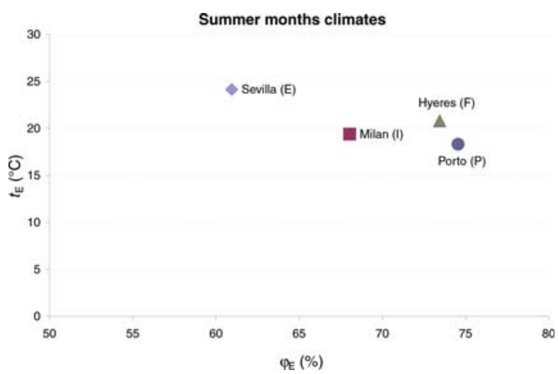

Figure 7. *Position of the four types of European climate in the* $\varphi_E - t_E$ *diagram. Hyeres (France) and Sevilla (Spain) are Csa climates, Milan (Italy) is Cfb climate, Porto (Portugal) is Csb climate.*

Table 2. *Input data for the seasonal analysis.*

t_A (°C)	26
φ_A	50%
t_I (°C)	19
Numbers of persons	50
Latent load per person (W)	75
Sensible load per person (W)	75
Other sensible loads (W)	6000
G_{vent} (11 l/s per person)	11
α	0.9
ξ	50%
η_{AC}	90%
η_{HU}	90%
Mean seasonal EER chiller	2.5
Electrical energy cost (€/kWh)	0.20
Operation period	May–September
Operation time	7–19

Figure 9 depicts the results in terms of obtainable free cooling energy and respective electrical energy saving (with a 2.5 mean seasonal EER for the chiller). The same information basically is provided by Figure 10 that also supplies data concerning the cooling energy savings (in relative terms) with respect to the total cooling load (sensible + latent) and the ventilation air load (Equation 10):

$$\text{Cooling energy saving} = \frac{\text{Free cooling}}{\text{total cooling load} + \text{if (ventilation load} > 0;\ \text{ventilation load}; 0)} \tag{10}$$

Obviously, seasonal ventilation air load is considered only if positive (that is due to an effective cooling load). Note how energy savings with both DEC and IEC are very effective, above all in climates where D1 and U1 zones are relatively less frequent than D2, U2 and U3. Energy savings with DEC vary from 64% (Porto, P) to 23% (Sevilla, E). IEC technique allows greater savings than DEC: this is apparent for resorts with a high frequency of D1 zone (Hyeres and Sevilla, with an advantage with respect to DEC of 22 and 40%, respectively) (Figure 10).

Figure 11 reports water consumption for the two adiabatic saturators.

Figure 12 depicts another worth feature of evaporative cooling techniques, i.e. the reduction of the chiller capacity. The figure reports the cooling power necessary to the user (that is the cooling load) and the cooling power with IEC for the four climates. Such cooling powers have been calculated as the 99th percentile of the hourly power data (to choose the hourly peak power would have been too much penalizing). Distance between horizontal line and histograms on Figure 12 allows to appreciate the positive effect of free cooling in the reduction of chiller capacity that results about 4 kW in most situations.

Economic analysis is based on assumptions reported in Table 3.

Reported data have been derived from literature and from direct experience of the authors. Saturators water absorption

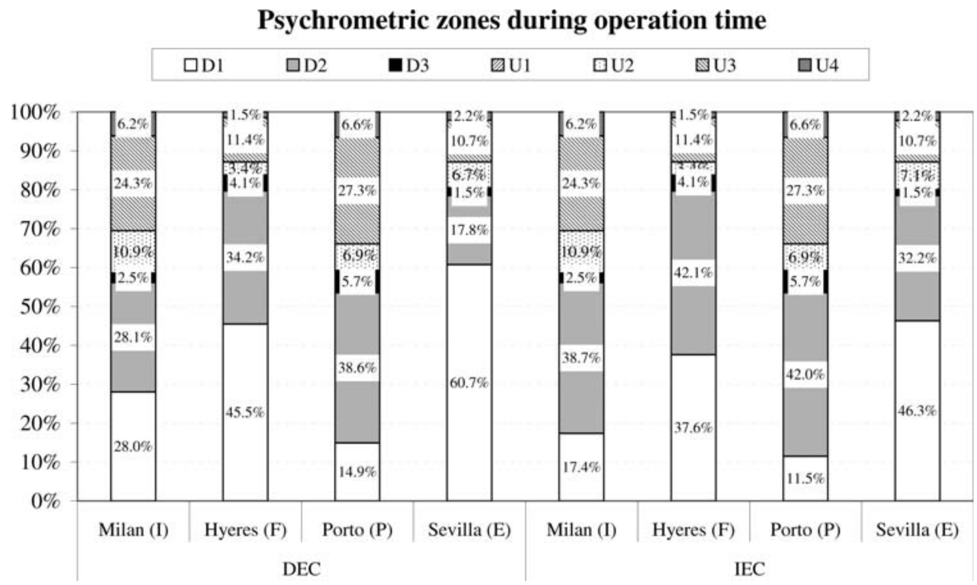

Figure 8. *Percentage incidence of the various psychrometric zones for the two evaporative cooling systems and for the different climates.*

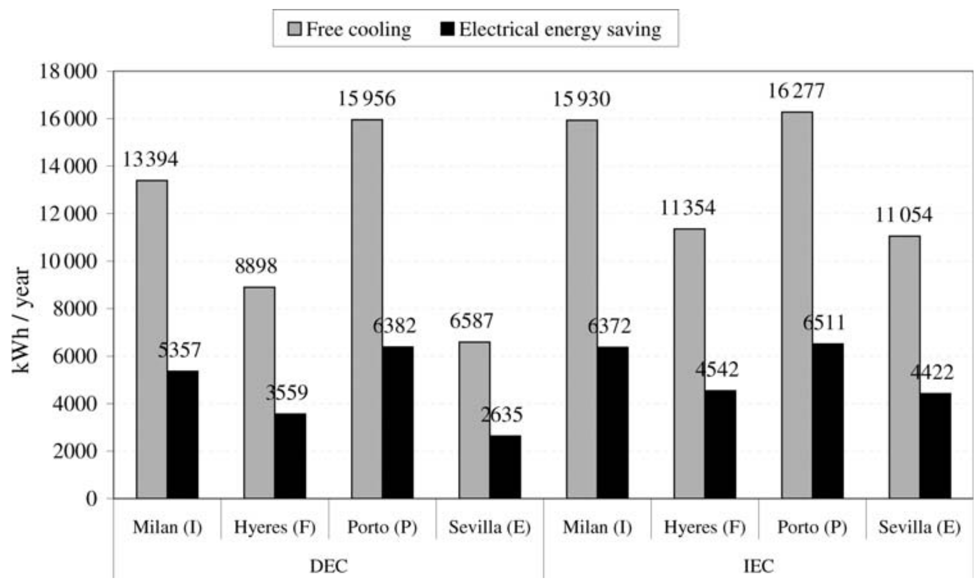

Figure 9. *Free cooling effect and electrical energy saving (in absolute terms) during the considered operation period.*

efficiency (ε) is a factor that takes into consideration the fraction of sprayed water not evaporated in the AHU duct. Energetic advantages of evaporative cooling here calculated are obtained against the extra cost of installing a high pressure water spray humidifier (instead of, for example, a wetted media humidifier) and, in connection with this, a reverse osmosis demineralization plant (instead of, for example, a simple water-softener). Because of the difficulty to find data about cost of these components (as their values are a function of nominal size), the analysis has been developed with a variable extra investment cost, from 1000 to 5000€ for the plant here considered.

Table 4 reports chiller electrical energy and cooling capacity savings for the four climates and the two evaporative cooling techniques. Both are economically evaluated, using data of Table 3. Observe that IEC allows an economic saving in cooling capacity even where DEC does not. Table 4 also reports the cost of electrical energy required by the pumps of the reverse osmosis demineralization plant and of the high-pressure water spray saturators. Costs of fans electricity consumption are considered as well, taking into account the total pressure drop (coils, saturators and heat exchanger) of the AHU and fans efficiency reported in Table 3. The costs of pressure drop are evaluated in relative terms, that is as the

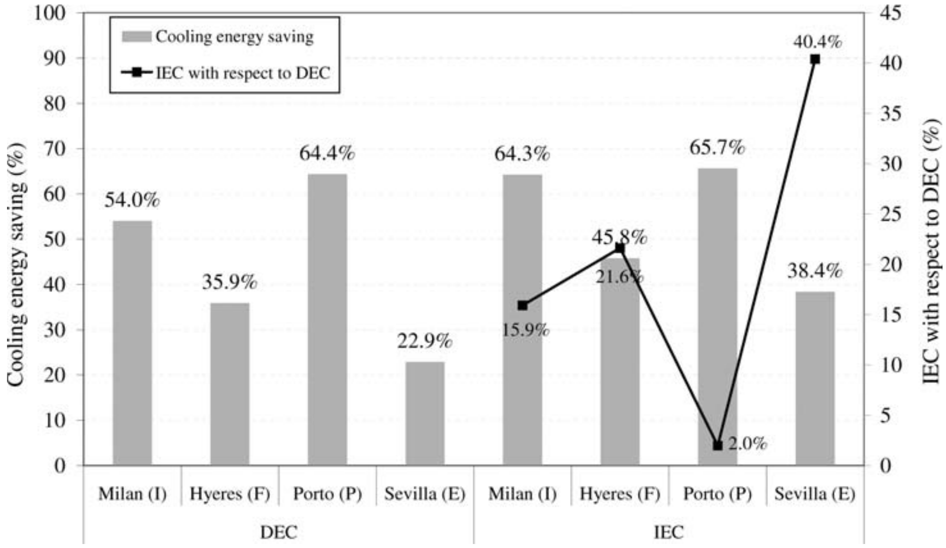

Figure 10. *Free cooling effect in relative terms (that is with respect to seasonal loads due to both total cooling load and (if positive) ventilation load) and increase in cooling energy savings (in relative terms) with IEC technology with respect to DEC.*

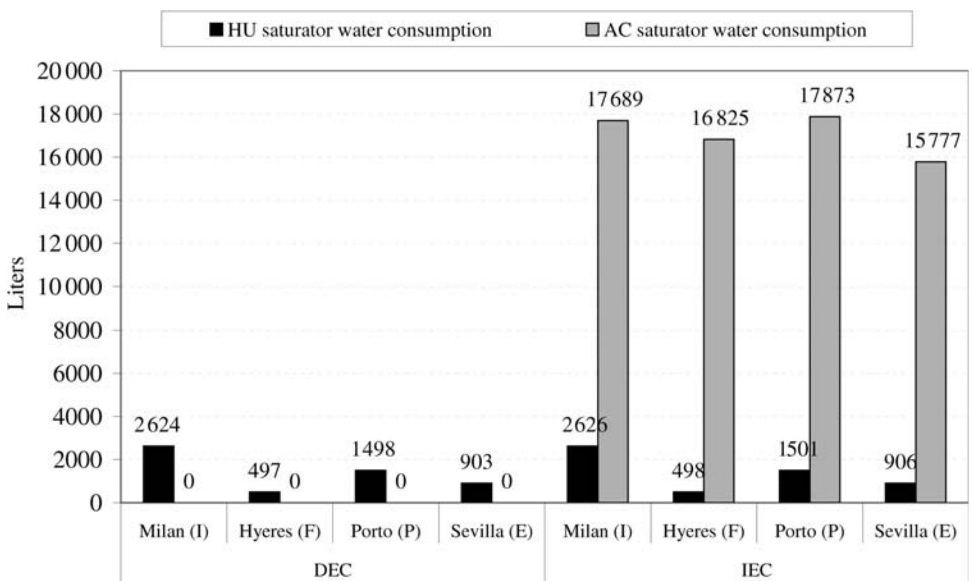

Figure 11. *Seasonal water consumption for HU and AC adiabatic saturators.*

increase in the seasonal energy consumption due to the greater (than ventilation needs) outdoor air flow used in accordance with the control logic described. So, in the net present worth (NPW) and discounted payback period (DPP) calculus, this cost is considered the difference between the two lines in Table 4 ('G_E optimized' and '$G_E = G_{vent}$'). Note that annual costs for water demineralization and humidifiers are lower than fans electricity consumption costs and extremely lower than annual savings. Besides this latter consideration, economic analysis results are very different as a function of the climate and of the extra investment cost.

Figures 13 and 14 depict the results: the investment is very advantageous in colder climates (Milan (I) and Porto (P)), with NPW always positive and DPP quite always lower than 5 years both for DEC and IEC techniques. Note that, while in Milan (I) IEC is more economic and advantageous than DEC, in Porto (P) the lower energy advantage of IEC with respect to DEC is not economically justified. For warmer climates the economic advantage is less safe: Hyeres (F) and Sevilla (E) has a positive NPW only till, respectively, 4000 and 2500€ of extra investment cost for DEC. IEC increases economic advantage considerably, above all for the dryer climate (limits of

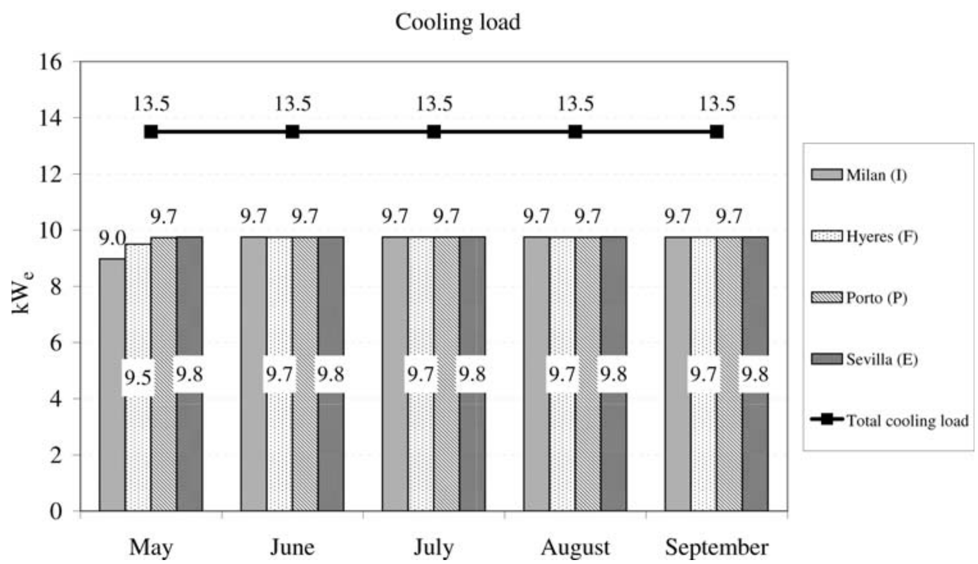

Figure 12. *Total cooling load with and without indirect evaporative cooling for the different climates.*

Table 3. *Assumption for the economic analysis.*

Electrical energy cost (€/kWh$_{el}$)	0.20
Marginal specific cost of cooling capacity (€/kW$_c$)	140
Water cost (€/m^3)	0.8
Extra investment cost (high-pressure water spray humidifier + reverse osmosis demineralization plant) (€)	1000–5000
Saturators water absorption efficiency (ε) [5]	0.8
Water pump electricity consumption (humidifiers) [5] (W h/l)	6
Water pump electricity consumption (demineralization plant) [5] (W h/l)	2
Total pressure drop DEC (Pa)	600
Total pressure drop IEC (Pa)	800
AHU fans efficiency	0.6
Interest rate	5%
Period of the analysis	10

advantage become, respectively, 5000 and 4000€). The payback period looks advantageous for a large interval of extra investment cost only for *Cfb* and *Csb* climates, while for warmer climates payback are acceptable only for low extra investment cost and for IEC technique.

5 CONCLUSIONS

The paper reports on the use of air-side free cooling techniques by means of wet bulb economizer cycles, both DEC and IEC. A simulation worksheet has been developed to evaluate, by means of hourly Typical Meteorological Year of four typical European climates, the possible energetic advantages. Results show very interesting energy savings above all in the colder climates and, for the warmer ones, an important increase of the savings with IEC with respect to DEC. Such a benefit is obtained by means of a greater water consumption (adiabatic saturator in the exhausted air flow) and a higher electricity consumption of the fans (outdoor air flow is increased besides ventilation air flow when advantageous in accordance with control logic described).

Against advantageous energy savings in all the European climates, economic analysis reveals some critical aspects: electrical energy and cooling capacity savings are able to cover the greater costs (due to water consumption and higher electricity consumption of pumps and fans) only for *Cfb* and *Csb* climates (both for DEC and IEC). In warmer climates (*Cfa* and *Csa*), economic profit depends largely on extra investment cost

Table 4. *Economic annual savings and costs.*

		DEC				IEC			
		Milan (I)	Hyeres (F)	Porto (P)	Sevilla (E)	Milan (I)	Hyeres (F)	Porto (P)	Sevilla (E)
Economic savings [€/year]	Electrical energy	1071	712	1276	527	1274	908	1302	884
	Cooling capacity	0	0	0	0	68	68	68	68
Economic costs [€/year]	Water demineralization cost (electricity)	1	0	1	0	10	9	10	8
	Humidifiers cost (electricity + water)	7	1	4	2	51	43	48	42
	Fans (electricity) (G_E optimized)	444	581	611	623	593	781	849	910
	Fans (electricity) ($G_E = G_{vent}$)	404	404	404	404	539	539	539	539

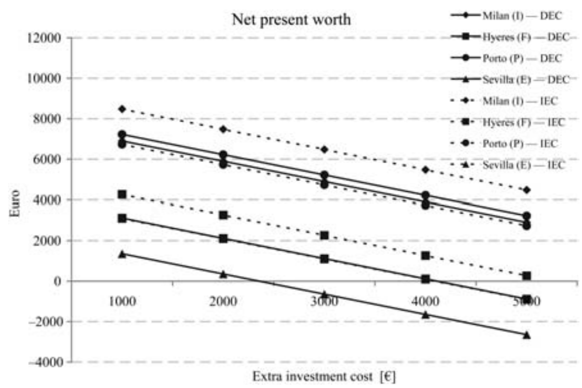

Figure 13. *Net present worth of the DEC and IEC techniques for the four European climates as a function of the extra investment cost (with respect to a wetted media humidifier and water-softener).*

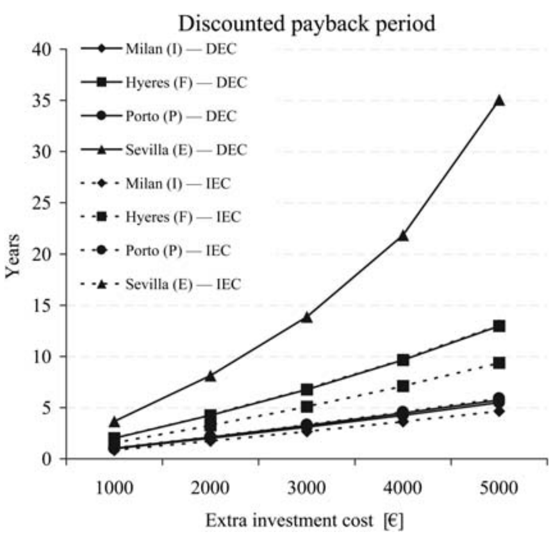

Figure 14. *Discounted payback period of the DEC and IEC techniques for the four European climates in function of the extra investment cost (with respect to a wetted media humidifier and water-softener).*

in evaporative cooling equipment, due to the requirement of high-pressure water spray humidifiers (to have a fine regulation of air humidity) and of a reverse osmosis demineralization plant. In such a case, IEC is definitely more advantageous than DEC, above all in dryer climates.

In conclusion, evaporative cooling techniques are energy saving for a wide range of climates, typically in buildings where internal heat gain is high enough to need cooling also when outside air temperature is lower than the desired inside air temperature. Economic profit should be instead carefully evaluated from time to time. The evaluation must be carried out considering the existence of latent and sensible loads, in the example here represented by 50 people; a further analysis should dealt with in sufficient detail the indoor activities and their impact on the loads and evaporative cooling performances. Moreover, comfort conditions in the indoor space when the free cooling techniques are used are not here considered.

NOMENCLATURE

c	specific heat, J/(kg K)
EER	energy efficiency ratio
G	mass air or steam flow, kg/s
h	humid air enthalpy, J/kg_a
P	thermal or cooling power, W
q	enthalpy difference, J/kg_a
r	water latent heat of vaporization, J/kg_v
t	Temperature, °C
x	humidty ratio, kg_v/kg_a
W	adiabatic saturator water consumption, l
Δx_{sp}	specific latent load, kg_v/kg_a

Greek symbols

α	exhausted and outdoor air flow ratio
φ	relative humidity
ξ	sensible heat exchanger efficiency
η	adiabatic saturator efficiency

Subscripts

a	dry air
A	indoor ambient air
AC	referred to AC abiabatic saturator
B	point at the same humidity ratio of point M with such an enthalpy to become at the same humidity ratio of point I by means of adiabatic saturation with efficiency η_{HU}
c	cool
C	air flow after AC adiabatic saturator
chiller	referred to the chiller
D	point at the same humidity ratio of point I in the saturation curve
dehum	referred to the dehumidification of outdoor air flow $G_E - G_{vent}$
E	outdoor ambient air
free	referred to free cooling effect
HU	referred to HU adiabatic saturator
I	inlet air conditions
input	referred to the input conditions
lat	latent
M	outside (G_E) and recirculated (G_{rec}) air mixture
max	maximum
min	minimum
O	referred to exhaust air flow
p,a	referred to specific heat at constant pressure of dry air
p,v	referred to specific heat at constant pressure of steam
saved	saved

sens sensible
sp specific
t thermal
X air flow after sensible heat exchanger
v steam
vent ventilation

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