AiCARR Journal / Vol 67, n. 2, 44 - 48, 2021

Marco Noro[1]*

Sensitivity analysis of base processes and equipment in HVAC plants using low and medium temperature heat sources Part 2

Analisi di sensitività dei processi di base e apparecchiature per la climatizzazione che utilizzano calore a media e bassa temperatura Seconda parte

[1] Department of Management and Engineering, University of Padova, Vicenza, Italy

***Corresponding author:**
Marco Noro
Department of Management and Engineering
University of Padova
Stradella S. Nicola 3
36100 Vicenza, Italy
marco.noro@unipd.it
tel +39 0444 998704

DOI: 10.36164/AiCARRJ.67.02.02

Abstract

Useful utilization of low and medium temperature heat can be a valuable issue when designing nearly zero-energy buildings (2010/31/UE Directive) and considering 2018/2001/UE Directive on the promotion of the use of energy from renewable sources. From the thermodynamic point of view, heat recovery covers a lot of technologies: vapor compression chillers and heat pumps, sorption machines, power cycles, cogeneration. In the first part of this study, the impact of heat source, heat sink and environment temperatures is analyzed for such technologies, both from the energy and exergy point of view. In the second part of this study, such analysis is extended to humid air in the main psychrometric processes. On this basis, as in Mediterranean climates new and energy retrofitted buildings are going to have increasing cooling demand with respect to heating, direct and indirect evaporative cooling air conditioning plant schemes are studied from the exergy point of view.

Keywords:
- Heat source
- Cogeneration
- Trigeneration
- Exergy analysis
- Absorption chiller
- Evaporative cooling

Sommario

L'opportuna utilizzazione di calore a bassa e media temperatura può essere un elemento importante nella progettazione di un edificio ad energia quasi zero (Direttiva 2010/31/UE), anche in considerazione della Direttiva 2018/2001/UE sulla promozione dell'utilizzo delle fonti rinnovabili di energia. Dal punto di vista termodinamico, il recupero termico comprende molte tecnologie: macchine frigorifere e pompe di calore a compressione meccanica, macchine ad assorbimento, impianti di produzione di potenza, cogenerazione. Nella prima parte di questo lavoro viene analizzato l'impatto della variazione delle temperature delle sorgenti e dei pozzi termici, nonché della temperatura ambiente, sull'efficienza energetica ed exergetica di tali tecnologie. Nella seconda parte tale analisi viene estesa all'aria umida nei principali processi psicrometrici. Su tale base, poiché nei climi mediterranei il peso dei fabbisogni per il raffrescamento sta diventando sempre maggiore rispetto a quello per il riscaldamento, soprattutto nei nuovi edifici e in quelli ristrutturati, vengono analizzate le efficienze exergetiche degli schemi di climatizzazione tramite raffreddamento evaporativo diretto e indiretto.

Parole chiave:
- Sorgente termica
- Cogenerazione
- Trigenerazione
- Analisi exergetica
- Chiller ad assorbimento
- Raffreddamento evaporativo

NOMENCLATURE

Symbols

AHU	Air Handling Unit
CHP	Combined Heat and Power plant
COP	Coefficient Of Performance
DEC	Direct Evaporative Cooling
EER	Energy Efficiency Ratio
$\dot{E}X$	exergy (W)
h	specific enthalpy (J kg^{-1})
HP	Heat Pump
HVAC	Heating, Ventilation and Air Conditioning
IEC	Indirect Evaporative Cooling
$\dot{m}$	mass flow rate (kg s^{-1})
ORC	Organic Rankine Cycle
Q	heat or cold (thermal energy) (J)
$\dot{Q}$	thermal power (W)
s	specific entropy (J kg^{-1} K^{-1})
$\dot{S}$	stream flow entropy (W K^{-1})
T	temperature (°C)
TER	Total Energy Ratio
VC	vapor compression
W	mechanical work (J)
x	humidity ratio (kg$_{vap}$ kg$_a^{-1}$)
X	can be η, *EER*, *COP*, *TER*, η_I

Greek symbols

β	ratio between recirculated and inlet air flows in IEC
ε	adiabatic saturator efficiency
Φ	relative humidity
η	efficiency
σ	specific cost ratio
ξ	heat exchanger (recuperator) effectiveness
Ψ	stream flow exergy (J kg^{-1})

Subscripts

0	environment, reference state
1	refers to the inlet thermodynamic state in DEC and IEC
1'	refers to the second air flow in adiabatic mixing process
2	refers to the thermodynamic state of inlet air flow after the (direct) adiabatic saturator in DEC and of outdoor air flow after the heat exchanger (recuperator) in IEC
I	first law of thermodynamics
a	dry air
A	refers to the thermodynamic state of indoor ambient air flow
c	cold
C	refers to the thermodynamic state of outlet air flow after the (indirect) adiabatic saturator in IEC
d	drive
dest	destroyed
el	electric
ex	exergy
gen	generated
i	ideal
in	inlet
inf	infinite
max	maximum
min	minimum
O	refers to the thermodynamic state of outlet air flow after the heat exchanger (recuperator) in IEC
out	outlet
r	real
s	source
u	upgrade
useful	useful thermal energy or mechanical work
v	vapor
w	water

Introduction

In the first part of this study, different options for revalorization of low and medium temperature heat in buildings application were analyzed. The impact of the variation in heat source, sink and environment temperatures from both energy and exergy point of view were determined, also accounting for the cost of heat exchange. The main direct power and reverse cycles were considered. As the main conclusions of the first part of the study:

- the impact of sink temperature is predominant in direct cycles and cogeneration plants;
- heat pumps are more sensitive to environment (heat source) temperature when considering vapor compression technology, and to useful heat (heat sink) temperature when thermal compression is considered;
- refrigerators and multiuse units feature a similar behavior: they are more sensitive to the temperature of the "cold produced" (heat source) when considering vapor compression, and to the temperature of the environment (heat sink) with absorption chillers.

In this second part of this study, another aspect related to the evaluation of exergy analysis in buildings is considered. As a matter of fact, new and energy retrofitted buildings are going to have more and more cooling needs with respect to heating ones, due to the reduction of thermal losses and thus to the increase of the internal heat gains weight. Air conditioning is therefore increasing its relevance, especially in milder climates as the Italian one; in this context, air-side free cooling techniques can be a very interesting way to face cooling loads when outside air temperature is lower than desired indoor air temperature (ASHRAE, 2015) (Lazzarin and Noro, 2009). In particular, wet bulb economizer cycles (worth of a special attention because they allow to obtain greater energy savings) such as direct evaporative cooling (DEC) and indirect evaporative cooling (IEC) are analyzed in this study by the energy and exergy point of view. Previously, the key psychrometric processes (sensible heat and cooling, cooling with dehumidification, heating and humidification, evaporative cooling, adiabatic mixing) are considered, studying how the exergy performance of the processes vary in function of the inlet air and of the outdoor environmental temperatures.

METHODS

Psychrometric Processes and Air Conditioning

Low temperature heat is widely available in HVAC plants when considering the air flows treated by air handling units. Previous few studies were developed on exergy analysis of key psychrometric processes. Qureshi and Zubair (2003) analyzed various psychrometric processes and performed some parametric studies to see the effect of mass ratio and relative humidity on the performance of the system. Energy and exergy analyses of an experimental open-cycle desiccant cooling system were conducted by Kanoglu et al. (2004) and obtained an exergy efficiency of 11% for the system. Kanoglu et al. (2007) studied each of the five basic psychrometric processes from exergy point of view and conducted a parametric study of heating with humidification process. In another study, Kilkis (2004) analyzed the environmental problems associated with the amount of exergy destructed elaborating an analytical optimization and control algorithm. Ratlamwala and Dincer (2013) studied five basic psychrometric processes both energetically and exergetically based on three different energy and exergy efficiency definitions: the parametric studies conducted to investigate the effect of variations in ambient temperature, ambient relative humidity and inlet temperature showed how different definitions of efficiencies can give different results which are far away from each other. Marletta (2010) used exergy analysis to

AiCARR Journal / Vol 67, n. 2, 44 - 48, 2021

Table 1 – Mass, energy, entropy and exergy balances for the psychrometric processes considered. The exergy efficiency and the main hypotheses are reported on the last rows of the table

Tabella 1 – Bilanci di massa, energetico, entropico ed exergetico dei processi dell'aria umida considerati. L'efficienza exergetica e le principali ipotesi sono riportate nelle ultime righe della tabella

	Sensible heating or cooling	Cooling and dehumidification	Heating and humidification	DEC	IEC $\left(\beta = \frac{\dot{m}_{a,recirculated}}{\dot{m}_{a1}}\right)$	Adiabatic mixing
Mass balance	$\dot{m}_{a1} = \dot{m}_{a2}$ $\dot{m}_{a1}x_1 = \dot{m}_{a2}x_2$	$\dot{m}_{a1} = \dot{m}_{a2}$ $\dot{m}_{a1}x_1 = \dot{m}_{a2}x_2 + \dot{m}_w$	$\dot{m}_{a1} = \dot{m}_{a2}$ $\dot{m}_{a1}x_1 + \dot{m}_{vap} = \dot{m}_{a2}x_2$	$\dot{m}_{a1} = \dot{m}_{a2}$ $\dot{m}_{a1}x_1 + \dot{m}_w = \dot{m}_{a2}x_2$	$\dot{m}_{a1} = \dot{m}_{a2}$ $\dot{m}_{a1}x_1 = \dot{m}_{a2}x_2$ $(1-\beta)\cdot\dot{m}_{a1} = \dot{m}_{aO}$ $(1-\beta)\cdot\dot{m}_{a1}x_A + \dot{m}_w = \dot{m}_{aO}x_O$	$\dot{m}_{a1} + \dot{m}_{a1'} = \dot{m}_{a2}$ $\dot{m}_{a1}x_1 + \dot{m}_{a1'}x_{1'} = \dot{m}_{a2}x_2$
Energy balance	$\dot{Q} + \dot{m}_{a1}h_1 = \dot{m}_{a2}h_2$	$\dot{m}_{a1}h_1 = \dot{m}_{a2}h_2 + \dot{m}_w h_w - \dot{Q}$	$\dot{Q} + \dot{m}_{a1}h_1 + \dot{m}_{vap}h_{vap} = \dot{m}_{a2}h_2$	$\dot{m}_{a1}h_1 + \dot{m}_w h_w = \dot{m}_{a2}h_2$	$\dot{m}_{a1}h_1 + (1-\beta)\cdot\dot{m}_{a1}h_A + \dot{m}_w h_w =$ $= \dot{m}_{aO}h_O + \dot{m}_{a2}h_2$	$\dot{m}_{a1}h_1 + \dot{m}_{a1'}h_{a1'} = \dot{m}_{a2}h_2$
Entropy balance	$\frac{\dot{Q}}{T} + \dot{m}_{a1}s_1 + \dot{S}_{gen} = \dot{m}_{a2}s_2$	$\dot{m}_{a1}s_1 + \dot{S}_{gen} = \dot{m}_{a2}s_2 + \dot{m}_w s_w - \frac{\dot{Q}}{T}$	$\frac{\dot{Q}}{T} + \dot{m}_{a1}s_1 + \dot{m}_{vap}s_{vap} + \dot{S}_{gen} = \dot{m}_{a2}s_2$	$\dot{m}_{a1}s_1 + \dot{m}_w s_w + \dot{S}_{gen} = \dot{m}_{a2}s_2$	$\dot{m}_{a1}s_1 + (1-\beta)\cdot\dot{m}_{a1}s_A + \dot{m}_w s_w +$ $+\dot{S}_{gen} = \dot{m}_{aO}s_O + \dot{m}_{a2}s_2$	$\dot{m}_{a1}s_1 + \dot{m}_{a1'}s_{1'} + \dot{S}_{gen} = \dot{m}_{a2}s_2$
Exergy balance	$\dot{Q}\cdot\left(1-\frac{T_0}{T}\right) + \dot{m}_{a1}\Psi_1 - \dot{E}X_{dest} = \dot{m}_{a2}\Psi_2$	$\dot{m}_{a1}\Psi_1 - \dot{E}X_{dest} =$ $= \dot{m}_{a2}\Psi_2 + \dot{m}_w\Psi_w - \dot{Q}\cdot\left(1-\frac{T_0}{T}\right)$	$\dot{Q}\cdot\left(1-\frac{T_0}{T}\right) + \dot{m}_{a1}\Psi_1 +$ $+\dot{m}_{vap}\Psi_{vap} - \dot{E}X_{dest} = \dot{m}_{a2}\Psi_2$	$\dot{m}_{a1}\Psi_1 + \dot{m}_w\Psi_w -$ $-\dot{E}X_{dest} = \dot{m}_{a2}\Psi_2$	$\dot{m}_{a1}\Psi_1 + (1-\beta)\cdot\dot{m}_{a1}\Psi_A + \dot{m}_w\Psi_w -$ $-\dot{E}X_{dest} = \dot{m}_{aO}\psi_O + \dot{m}_{a2}\psi_2$	$\dot{m}_{a1}\Psi_1 + \dot{m}_{a1'}\Psi_{1'} -$ $-\dot{E}X_{dest} = \dot{m}_{a2}\Psi_2$
Exergy efficiency	$\eta_{ex} = \frac{\dot{m}_{a2}\Psi_2}{\dot{Q}\cdot\left(1-\frac{T_0}{T}\right) + \dot{m}_{a1}\Psi_1}$ heating $\eta_{ex} = \frac{\dot{m}_{a2}\Psi_2 - \dot{Q}\cdot\left(1-\frac{T_0}{T}\right)}{\dot{m}_{a1}\Psi_1}$ cooling	$\eta_{ex} = \frac{\dot{m}_{a2}\Psi_2 + \dot{m}_w\Psi_w - \dot{Q}\cdot\left(1-\frac{T_0}{T}\right)}{\dot{m}_{a1}\Psi_1}$	$\eta_{ex} = \frac{\dot{m}_{a2}\Psi_2}{\dot{m}_{a1}\Psi_1 + \dot{m}_{vap}\Psi_{vap} + \dot{Q}\cdot\left(1-\frac{T_0}{T}\right)}$	$\eta_{ex} = \frac{\dot{m}_{a2}\Psi_2}{\dot{m}_{a1}\Psi_1 + \dot{m}_w\Psi_w}$	$\eta_{ex} = \frac{\dot{m}_{a2}\psi_2}{\dot{m}_{a1}\Psi_1 + (1-\beta)\cdot\dot{m}_{a1}\Psi_A + \dot{m}_w\Psi_w}$	$\eta_{ex} = \frac{\dot{m}_{a2}\Psi_2}{\dot{m}_{a1}\Psi_1 + \dot{m}_{a1'}\Psi_{1'}}$
Hypotheses on coils heat exchangers and saturators	Heating $\begin{cases}\dot{Q} = 40\,\text{kW}\\ T_{in} = 60\,°\text{C}\quad T_{out} = 50\,°\text{C}\end{cases}$ Cooling $\begin{cases}\dot{Q} = -8\,\text{kW}\\ T_{in} = 7\,°\text{C}\quad T_{out} = 12\,°\text{C}\end{cases}$	Cooling coil: By Pass Factor = 0.2 $T_{in} = 7\,°\text{C}\quad T_{out} = 12\,°\text{C}$	Vapor humidifier: $\dot{Q} = 10\,\text{kW}$ $T_{vap} = 100\,°\text{C}\quad \dot{m}_{vap} = 0.05\,\text{kg s}^{-1}$	Adiabatic saturator: $\varepsilon_{sat} = 95\%$ $T_w = 15\,°\text{C}$	Adiabatic saturator: $\varepsilon_{sat} = 95\%\quad T_w = 15\,°\text{C}$ Recirculated air: $T_A = 25\,°\text{C}\quad \Phi_A = 50\%$ Heat exchanger eff.: $\xi = 50\%$	Second air flow: $T_{1'} = 15\,°\text{C}$ $\Phi_{1'} = 50\%$

assess the performance of the three most common air conditioning plant schemes: all-air, dual-duct and fan-coil systems.

In this study, six key psychrometric processes (sensible heating or cooling, cooling and dehumidification, heating and humidification, evaporative cooling — direct and indirect —, adiabatic mixing) are evaluated from energy and exergy points of view. Each of the six systems is modeled by mass, energy and exergy balance in order to investigate the effect of inlet air temperature T_1 and environment temperature T_0 (reference state) on the energy and exergy efficiencies (Table 1). There is some discussion in literature about the definition of exergy efficiency of psychrometric processes (Ratlamwala and Dincer, 2013) (Marletta, 2010) (Wepfer et al., 1979) (Bejan, 2006) (Wark, 1995) (Ren and Li, 2002). As for the first part of this study, here we consider the "rational" exergy efficiency as described in (Kanoglu et al., 2007) and (Ratlamwala and Dincer, 2013).

In Table 1 the following definitions (Equations 1-3) have to be considered respectively for the thermal power exchanged by coils $\dot{Q}$, the stream flow exergy Ψ, and the destroyed exergy $\dot{E}X_{dest}$:

$$\dot{Q} > 0 \quad \text{for heating} \qquad \dot{Q} < 0 \quad \text{for cooling} \tag{1}$$

$$\Psi = h - h_0 - T_0\,(s - s_0) \qquad T_0 = 10\,°\text{C} \qquad \Phi_0 = 50\% \tag{2}$$

$$\dot{E}X_{dest} = T_0 \cdot \dot{S}_{gen} \tag{3}$$

For the evaporative cooling techniques, we refer to Figure 1 and Figure 2 for the configuration of the Air Handling Units (AHU), and to Equations 4–7 for the calculations of air thermodynamic variables in Figure 3 (Lazzarin and Noro, 2009) (Lazzarin, 2007).

In particular, in the case of Indirect Evaporative Cooling (IEC) (Figure 2), inside air flow G_O is humidified before being exhausted: its temperature decreases so it can be possible to cool outside air flow G_I by means of a sensible heat exchanger. In this case, it is possible a free cooling effect also when outside air enthalpy h_I is greater than inside air h_A: it is sufficient that enthalpy after indirect heat exchange (h_2) is lower than h_A. The effect is to extend the free cooling zone in the psychrometric diagram, using the lower enthalpy of inlet air (outside air flow G_I can be increased till the whole inlet flow to maximize free cooling effect).

It is possible to calculate the temperature t_C of the exhausted air flow G_O just after the adiabatic saturator ε_{sat} (Equation 4) (usually $G_O = (1-\beta)\cdot G_E$ with $\beta \leq 1$ to have a slight extra-pressure in the inside ambient):

$$\varepsilon_{sat} = \frac{x_C - x_A}{x_{C,\max} - x_A} \quad \Rightarrow \quad x_C = x_O = x_A + \varepsilon_{sat}\cdot(x_{C,\max} - x_A) \tag{4}$$

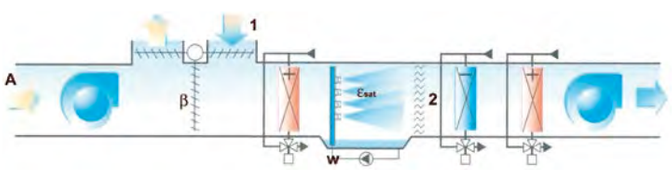

Figure 1 – AHU scheme with direct evaporative cooling (courtesy Carel Spa) (Lazzarin and Noro, 2009)

Figura 1 – Schema dell'unità trattamento aria con raffreddamento evaporativo diretto (per cortesia di Carel Spa) (Lazzarin e Noro, 2009)

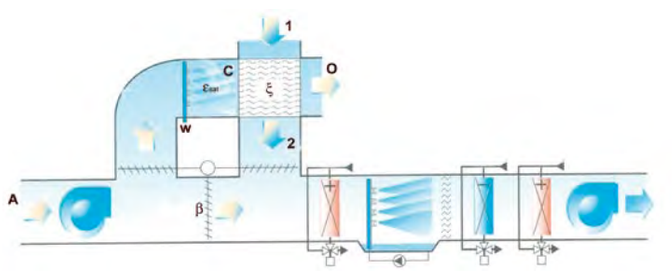

Figure 2 – AHU scheme with indirect evaporative cooling (courtesy Carel Spa) (Lazzarin and Noro, 2009)

Figura 2 – Schema dell'unità trattamento aria con raffreddamento evaporativo indiretto (per cortesia di Carel Spa) (Lazzarin e Noro, 2009)

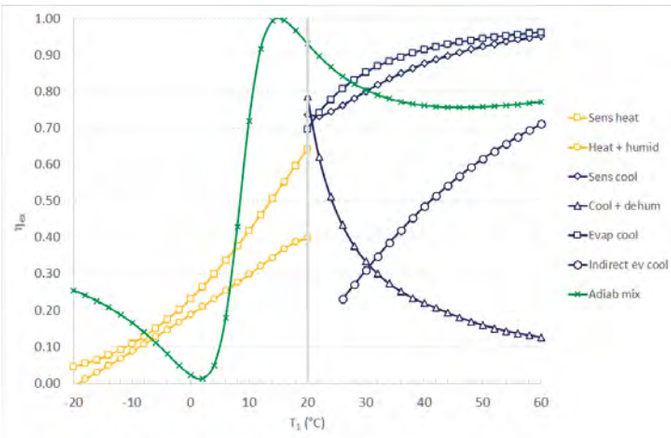

Figure 3 – Effect of variation in inlet temperature T_1 on exergy efficiency of the six processes described in Table 1. Values refer to typical ranges of temperature for each process, considering $T_1 = 20$ °C as a threshold between heating and cooling processes

Figura 3 – Effetto della variazione della temperatura di ingresso T_1 sull'efficienza exergetica dei sei processi descritti in Tabella 1. I valori si riferiscono agli intervalli tipici di temperatura per ogni processo, considerando T_1=20 °C come un valore soglia tra il processo di riscaldamento e quello di raffrescamento

In the best case, G_1 can reach t_C temperature (and x_1 humidity ratio): these two conditions determinate the lowest enthalpy value of outside air flow ($h_{2,min}$) (Equation 5).

$$h_{2,\min} = h_1 - \xi \cdot (1-\beta) \cdot (h_1 - h(T_A - \varepsilon_{sat} \cdot (T_A - T_{sat}(T_A; x(T_A; \Phi_A))); x_1) \qquad (5)$$

So h_2 can be calculated by the knowledge of the sensible heat exchanger effectiveness ξ (Equation 6):

$$\xi = \frac{h_1 - h_2}{(1-\beta) \cdot (h_1 - h_{2,\min})} \quad \Rightarrow \quad h_2 = h_1 - \xi \cdot (1-\beta) \cdot (h_1 - h_{2,\min}) \qquad (6)$$

Now it is possible to determinate humidity ratio x_C and temperature t_2 (Equation 7):

$$x_C = x(t_C; h_A) \quad \Rightarrow \quad t_2 = t(h_2; x_1) \qquad (7)$$

It must be $h_2 < h_A$ to get energy saving by increasing outside air flow besides the minimum value (i.e. ventilation air flow).

RESULTS AND DISCUSSION

Inlet temperature (T_1) and ambient temperature (T_0) are varied to see their effect on exergy efficiency of each of the system in Table 1 (Figure 3 and Figure 4). The results are here summarized for each process:

- *sensible heating*: exergy destruction in the system increases till certain ambient temperature and then start decreasing, that is why the exergy efficiencies decrease until 25 °C (from 0.44 to 0.12) and then start increasing (from 0.12 to 0.17). Exergy efficiency shows a significant increase with the increase in inlet temperature, from 0.05 at T_1=-20 °C till 0.64 at 20 °C (less exergy destruction with increase in inlet temperature);
- *sensible cooling*: exergy efficiency is very high during winter (low ambient temperature T_0), when nevertheless this process is useful for refrigeration only (not for air conditioning unless particular cases). Exergy efficiency strongly decreases when ambient temperature increases. The efficiency increases varying the inlet temperature T_1, going from 0.73 to 0.95 in the temperature range 20-60 °C;
- *heating with humidification*: similar considerations as the previous ones for sensible heating can be done, but with lower values of exergy efficiency as a larger amount of exergy is destroyed with the humidification process;
- *cooling with dehumidification*: Figure 3 explains how variation in inlet temperature affects the performance of the system. Exergy efficiency decreases from 0.79 to 0.12 with the increase in inlet temperature from 20 °C to 60 °C, while it decreases from 0.6 to zero with the increase in ambient temperature (Figure 4);
- *adiabatic mixing*: the exergy destruction during the mixing process increases with increase in ambient temperature, so a lesser exergy efficiency of the process is obtained (Figure 4). Given the hypotheses in Table 1, Figure 3 depicts the performance varying inlet temperature: when T_1 is much lower than the other mixing air flow, the exergy destruction is high and increases with T_1. On the contrary, when T_1 assumes values around or higher than $T_{1'}$, the exergy destruction is much lower (exergy efficiency reaches 100% and then decreases);
- *direct and indirect evaporative cooling*: the exergy efficiency is observed to be increasing from 0.23 to 0.71 with the increase in inlet temperature from 20 °C to 60 °C for IEC; DEC features a much better performance, above all at medium temperatures (Figure 3). Increasing

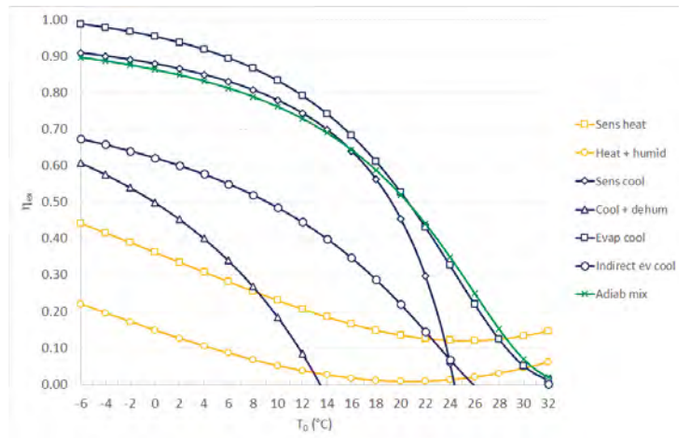

Figure 4 – Effect of variation in ambient temperature T_0 on exergy efficiency of the six processes described in Table 1

Figura 4 – Effetto della variazione della temperatura ambiente T_0 sull'efficienza exergetica dei sei processi descritti in Tabella 1

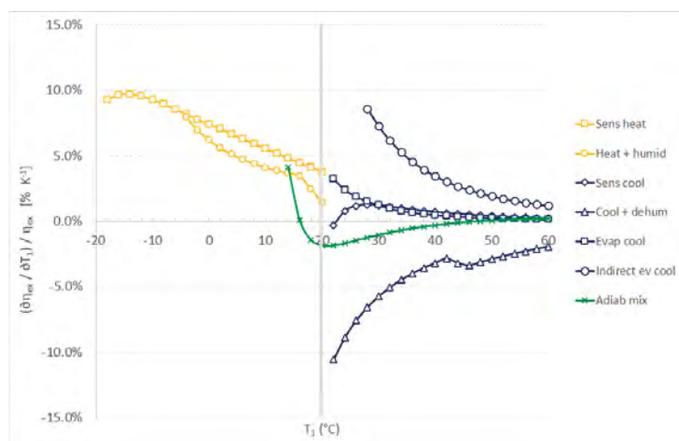

Figure 5 – Relative change of the exergy efficiency with inlet temperature. Values refer to typical ranges of temperature for each process, considering $T_1 = 20$ °C as a threshold between heating and cooling processes

Figura 5 – Variazione relativa dell'efficienza exergetica con la temperatura di ingresso. I valori si riferiscono agli intervalli tipici di temperatura per ogni processo, considerando T_1=20 °C come un valore soglia tra il processo di riscaldamento e quello di raffrescamento

the ambient temperature, exergy performances decrease (Figure 4), from 0.98 to zero for direct evaporative cooling, and from 0.67 to zero for indirect process when T_0 varies, respectively, in the ranges -6–32 °C and -6–26 °C.

Finally, a similar evaluation to the one depicted in the first part of this study for low and medium temperature heat valorization is conducted: the sensitivity of exergy efficiency on the inlet temperature T_1 is investigated by using the derivatives, normalized with the efficiency and plotted against the inlet temperature (Figure 5). Substantially, all the processes feature a decrease in the normalized sensitivity with the increase of T_1, that means the hotter the inlet temperature, the less sensible the variation of exergy efficiency.

CONCLUSIONS

Studying psychrometric processes from the exergy point of view, the main results are that sensible cooling and evaporative cooling techniques are the most efficient technologies, while cooling with dehumidification is much penalized above all at high inlet air and environment temperature. The normalized sensitivity of exergy efficiency of these best technologies is decreasing when inlet air temperature is increasing.

As a final consideration of this study, it is worth to highlight that every real practical system has a large number of small heat sinks and other losses. For example, electrically driven heat pumps supply useful heat at the condenser also by electric motor dissipation heat, while the same heat is a further cooling load for the same machine operating as (electrically driven) chiller. Moreover, cogeneration plants can produce useful heat at two different temperature/enthalpy levels, or they can produce useful cooling energy (trigeneration). Furthermore, no real thermal energy losses have been considered in this work (radiation to the environment, friction, etc.). From this point of view, it is worth to stress the idealized character of this study, giving however a theoretical limit to the maximum energy and exergy efficiency of the most diffused HVAC equipment.

CONFLICT OF INTEREST

The author declare the absence of any conflict of interest.

REFERENCES

ARERA (Italian Authority for energy, grids and environment), (2008), Delibera EEN 3/08 – Aggiornamento del fattore di conversione dei kWh in tonnellate equivalenti di petrolio connesso al meccanismo dei titoli di efficienza energetica (Act EEN 3/08 – Updating of the energy conversion factor from kWh to equivalent oil tonnes linked to the energy efficiency credits, in Italian), www.arera.it (accessed 10th January 2021).

AiCARR, (2011), EUROPEAN DIRECTIVE 2009/28/EU: AiCARR's Position about Italian Decree 28/11 for aspects related to heat renewable sources. www.aicarr.org (accessed 10th January 2021).

ASHRAE, (2015). Heating, Ventilating and Air-Conditioning Applications, ASHRAE Handbook 45.5.

Balta, M. T. (2012), "Exergy cost analysis and sustainability assessment of various low exergy heating systems", Energy and Buildings, 55, 721-727.

Balta, M. T., Dincer, I., Hepbasli, A. (2011), "Development of sustainable energy options for buildings in a sustainable society", Sustainable Cities and Society, 1(2), 72–80.

Bejan, A. (2006), Advanced Engineering Thermodynamics, Wiley, New York, ISBN 978-0-471-67763-5.

Boroumand Jazi, G., Rismanchi, B., Saidur, R. (2013), "A review on exergy analysis of industrial sector", Renewable and Sustainable Energy Reviews, 27, 198-203.

Buchin, O., Ziegler, F. (2013), "Valorisation of low-temperature heat: Impact of the heat sink on performance and economics", Applied Thermal Engineering, 50, 1543-1548.

CEN (European Committee for Standardization), EN ISO 52000-1:2018 "Energy performance of buildings – Overarching EPB assessment – Part 1: general framework and procedures".

Chua, K. J., Chou, S. K., Yang, W. M. (2010), "Advances in heat pump systems: a review", Applied Energy, 87(12), 3611–3624.

UNI/TS 11300-5:2016 "Prestazioni energetiche degli edifici – Parte 5: Calcolo dell'energia primaria e della quota di energia da fonti rinnovabili" (Energy performances of buildings – Determination of primary energy consumption and renewable energy quota, in Italian).

Dincer, I., Rosen, M. A. (2007), Exergy. Energy, Environment and Sustainable Development, Elsevier Science Ltd, ISBN 978-0-08-097089-9.

Hepbasli, A. (2011), "A comparative investigation of various greenhouse heating options using exergy analysis method", Applied Energy, 88, 4411-4423.

Hepbasli, A. (2012), "Low exergy heating and cooling systems for sustainable buildings and societies", Renewable and Sustainable Energy Reviews, 16, 73–104.

Kanoglu, M., Carpinlioglu, M. O., Yildirim, M. (2004), "Energy and exergy analyses of an experimental open-cycle desiccant cooling system", Appl. Therm. Eng., 24, 919-932.

Kanoglu, M., Dincer, I., Rosen, M. A. (2007), "Exergy analysis of psychrometric processes for HVAC&R applications". ASHRAE Trans., 113, 172-180.

Kaushik, S.C., Siva Reddy, V., Tyagi, S. K. (2011), "Energy and exergy analyses of thermal power plants: A review", Renewable and Sustainable Energy Reviews, 15, 1857-1872.

Kilkis, B. (2004), "An exergy aware optimization and control algorithm for sustainable buildings", Int. J. Green Energy, 1, 65-77.

Kim, Y. M., Kim, C. G., Favrat, D. (2012), "Transcritical or supercritical CO_2 cycles using both low – and high-temperature heat sources", Energy, 43(1), 402-415.

Köhler, S., Ziegler, F. (2006), "Sorption cooling: best alternative for the use of low temperature heat sources?", Proc. of Heat Powered Cycles 2006, Newcastle upon Tyne, UK, 12th to 14th September (CD-ROM).

Lazzarin, R. M. (2007), "Introduction of a simple diagram-based method for analyzing evaporative cooling", Applied Thermal Engineering, 27, 2011–2025.

Lazzarin, R. M., Macor, A. (1989), Introduzione all'analisi exergya (Introduction to the exergy analysis, in Italian), CLEUP, Padua, ISBN 8871783085.

Lazzarin, R. M., Noro, M. (2009), "Energy and economic savings of free cooling in different European climates", International Journal of Low Carbon Technologies, 4, 213-223.

Liu, J., Lin, Z. (2021), "Energy and exergy analyze of different air distributions in a residential building", Energy and Buildings, 233, 110694.

Marletta, L., (2010), "Air Conditioning Systems from a 2nd Law Perspective", Entropy, 12, 859-877.

Ozyurt, O., Ekinci, D. A. (2011), "Experimental study of vertical ground-source heat pump performance evaluation for cold climate in Turkey", Applied Energy, 88, 1257–1265.

Qureshi, B. A., Zubair, S. M. (2003), "Application of exergy analysis to various psychrometric processes", Int. J. Energy Res., 27, 1079-1094.

Ratlamwala, T. A. H., Dincer, I. (2013), "Efficiency assessment of key psychrometric processes", International Journal of Refrigeration, 36, 1142-1153.

Ren, C. Q., Li, N. P. (2002), "Principles of exergy analysis in HVAC and evaluation of evaporative cooling schemes", Build. Environ., 37, 1045–1055.

Sala Lizarraga, J. M. P., Picallo-Perez A. (2020), Exergy Analysis and Thermoeconomics of Buildings. Design and Analysis for Sustainable Energy Systems, Butterworth-Heinemann, ISBN 978-0-12-817611-5.

Sayyaadi, H., Sabzaligol, T. (2009), "Various approaches in optimization of a typical pressurized water reactor power plant", Applied Energy, 86(7–8), 1301-1310.

Torio, H., Angelotti, A., Schmidt, D. (2009), "Exergy analysis of renewable energy-based climatisation systems for buildings: A critical view", Energy and Buildings, 41, 248-271.

Wark, K. (1995), Advanced Engineering Thermodynamics for Engineers, McGraw Hill, New York, ISBN 978-0070682924.

Wepfer, W. J., Gaggioli, R. A., Obert, E. F. (1979), "Proper evaluation of available energy for HVAC", ASHRAE Trans., 85, 667–677.

Yucer, C. M., Hepbasli, A. (2011), "Thermodynamic analysis of a building using exergy analysis method", Energy and Buildings, 43, 536-542.