A SOLAR AIR-COOLED HIGH EFFICIENCY ABSORPTION SYSTEM IN DRY HOT CLIMATES
Reduction of Water Consumption and Environmental Impact

by

Jose D. MARCOS [a*], Raquel LIZARTE [b], Fernando VARELA [a], Maria E. PALACIOS-LORENZO [c], and Ana M. BLANCO-MARIGORTA [d]

[a] Department of Energy Engineering, National University of Distance Education (UNED), Madrid, Spain
[b] Department of Thermal and Fluids Engineering, Charles III University of Madrid, Madrid, Spain
[c] Department of Mechanical, Chemical and Industrial Design Engineering, Technical University of Madrid, Madrid, Spain
[d] Department of Processes Engineering, University of Las Palmas de Gran Canaria, Las Palmas de Gran Canaria, Spain

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A solar cooling system with an optimized air-cooled double-effect water/LiBr absorption machine is proposed as a sustainable alternative to meet cooling demands in dry hot climates. This system allows eliminating the cooling towers in those regions of the planet where water is scarce. This work analyses the environmental benefits of this air-cooled system, as well as its environmental footprints, compared to a solar water-cooled single effect. In this regard, a methodology has been applied to calculate the annual saving in water consumption produced in a case study: a hospital located in Almería, in South of Spain. Furthermore, the reduction in energy consumption and CO_2 emissions is also quantified since this machine can be driven by solar energy and with higher efficiency than those of single effect.

Key words: *solar cooling, air-cooled, water-energy nexus, water consumption, CO_2 emissions reduction, absorption machine, double effect, environmental footprints*

Introduction

Water has become a highly prized good in those regions of the planet where it is scarce. Several publications have warned that water scarcity will become one of the most serious problems of the next decade. Rosegrant *et al.* [1] warned of the arrival of an impending crisis and Martin [2] stated that energy and water consumption are major concerns nowadays. The growth of the world population and the effects of climate change reduce year by year the availability of this essential resource for the life of living beings on the planet. The integration of energy production systems with low water consumption is a concept of great relevance for the development of these regions. It also coincides that these areas are the ones that present the highest cooling demands, as it is the case of the south of Europe and the Mediterranean countries. Among commercial air conditioning equipment there are air-cooled mechanical compression chillers. In particular, the small power units for the residential sector

* Corresponding author, e-mail: jdmarcos@ind.uned.es

are the most used for their ease of installation and relative low price. However, the major disadvantage of these systems is the high global warming potential (GWP) of the refrigerants they use. For example, one of the most used, the R410A, has a GWP 3400 times higher than the estimated for CO_2 over a time horizon of 20 years and 1300 times greater for a time horizon of 100 years [3]. In recent years, both research and regulation are committed to the development of refrigerants with low GWP to replace current ones. This is the case of natural refrigerants and new generation refrigerants [4, 5]. In any case, it is necessary to develop refrigeration equipment combining two characteristics: on the one hand, the possibility of operating using low GWP renewable energies, and on the other hand, to avoid water consumption due to heat rejection using air-cooled systems.

A sustainable alternative that meets both objectives are absorption solar cooling systems. However, the most widely used absorption chillers on the market, water/lithium bromide (LiBr), are all water-cooled [6]. All these models, whether single, double or triple effect, are water-cooled. This implies that they require the use of cooling towers to remove the absorption and condensation heat, which supposes great water consumption [7]. This fact makes the process of heat rejection in the absorption machine expensive and unsustainable as it consumes a scarce resource in regions with high cooling demands. Added to the cost of installing cooling towers is the space required for their placement in buildings because of their large size. In addition, it is necessary to add the maintenance cost in this type of towers to prevent the proliferation of legionella. All these factors support the importance of eliminating cooling towers in HVAC equipment in this type of regions.

Therefore, the best sustainable option to meet the cooling demand in buildings located in areas with water scarcity is through air-cooled solar absorption machines. This type of machines avoid the problems derived from the use of cooling towers and they suppose saving in water consumption. In the recent literature there are works on the development of air-cooled absorption machines from a theoretical point of view: in extremely hot weathers [8], driven by low grade energy [9], comparing [10], and optimizing [11] air-cooled and water-cooled systems. From the experimental point of view, there are also a number of articles that contribute to the development of new air-cooled absorption prototypes: double effect [12, 13], single effect [14, 15]. As far as the authors are aware, only one air-cooled water/LiBr absorption machine was commercialized and was available between 2005 and 2008 [16]. Currently the only air-cooled absorption machine on the market is of ammonia/water [17]. But this machine has lower performances than double effect water/LiBr absorption chillers.

In recent years, a great effort has been made in the development of solar-powered absorption machines in order to protect the environment: with various heating source temperatures [18], collecting recent improvements [19], applied to a hospital [20], with a warming impact assessment [21] and in Mediterranean climates [22]. The main advantage of solar cooling facilities is that the period of highest cooling demand coincides with that of highest solar radiation (summer). The main disadvantages include their dependence on environmental parameters (ambient temperature, solar radiation, wind speed) as well as their high initial investment cost, which make solar cooling systems not yet economically competitive with conventional air conditioning systems [8, 23]. However, when the installation is used to provide cooling in summer and heating and domestic hot water (DHW) in winter, the difference in economic costs is markedly reduced [24].

Tsoutsos *et al.* [20] designed a solar cooling system cooled by water to meet the cooling demand in a hospital in Crete (Greece) optimizing the design from a financial and environmental point of view. The solar collectors were used in winter to provide heating and

DHW in order to get more advantage of the solar facility reducing the payback period. The authors stated that this type of systems present advantages with respect to the conventional systems and for that reason, they should dominate the future market.

The present work has carried out a modelling and annual simulation of an air-cooled high efficiency water/LiBr solar cooling system to meet the cooling demand in hot regions with water scarcity as an alternative to the water-cooled solar cooling systems.

The main objective and the novelty of the work is to determine the advantages of air-cooled absorption solar cooling systems compared to the solar water-cooled single effect systems and conventional mechanical compression refrigeration systems in terms of efficiency, savings in CO_2 emissions and water consumption, comparing the annual performance of these three systems.

As was done by Tsoutsos *et al.* [20], a hospital located in a city in the south of Spain with high solar radiation and little rainfall [25] was chosen as a case study. This study therefore aims to integrate air-cooled high efficiency solar absorption systems in those regions where heat and the lack of water are the predominant characteristics.

Models and methodology

This section includes the description of the case study, the applied methodology and the models and software tools used for the simulations.

Description of the case study

The building used as case study in the simulation is a generalized hospital (whose envelope and schedules have been obtained from an examples database) placed in the city of Almeria, Spain. This city has been chosen because it presents a hot climate with abundant solar radiation and at the same time water scarcity.

Description of the building

The building chosen as a case study is a hospital taken from the database of buildings included in EnergyPlus, fig. 1. It has a constructed area of 48,975 m^2 which corresponds to a hospital of medium size. The surface of walls in contact with the exterior is 11,248.7 m^2 while the surface of windows is 4,499.45 m^2. On the other hand the surface corresponding to the roof is 13,249.47 m^2.

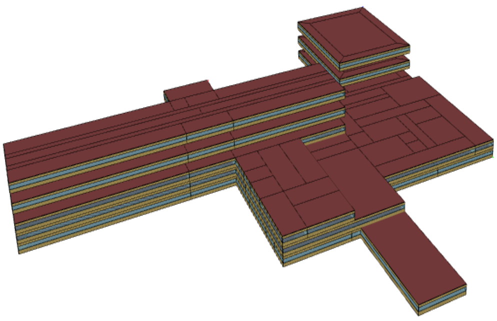

Figure 1. Openstudio 3-D view of the hospital

Applied methodology

The methodology carried out in this work involves the following steps:

– obtaining meteorological data from the locality where the energy simulation is to be performed (Almería); these data have been provided by the Spanish weather for energy calculations (SWEC) database,
– calculation of the annual cooling thermal loads; the hospital has cooling thermal loads throughout the year,
– modelling and simulation of a solar water-cooled single effect (WC-SE) water/LiBr system; this system is simulated to meet the cooling demand of the case study, obtaining the heat of absorption and condensation for the entire year,

– calculation of saving in water consumption; using the methodology described by the Institute for Energy Diversification and Saving (IDEA) [26], the annual water consumed by the WC-SE system is calculated,
– modelling and simulation of a solar air-cooled double effect (AC-DE) water/LiBr system; Marcos *et al.* [11] carried out a COP optimization for AC-DE cycles; using their conclusions, the annual performance of the air-cooled system has been simulated and optimized, and
– calculation of energy saving and emissions reductions with the proposed AC-DE system versus a WC-SE; solar fractions (defined as the ratio between cooling demand covered only using solar energy as energy source, and the total cooling demand) need to be higher than about 50% to start saving primary energy [27]; on the other hand, Tsoutsos *et al.* [20] proposed an optimized scenario with a solar fraction of almost 75% for a WC-SE system; this criterion has been adopted as an input in order to make an appropriate comparison between both absorption systems.

Meteorological data

In this case, the meteorological data are taken from the SWEC MET database and they are synthetic typical meteorological year (TMY) generated with the software Climed 1.3 from climatic records of the Spanish Meteorological State Agency. These data include hourly values of dry bulb temperature, sky temperature, direct normal irradiation, horizontal diffuse radiation, specific humidity, relative humidity, wind speed, and wind direction for a typical year. Figure 2 shows the variation of the mean ambient temperature and relative humidity for the typical meteorological year corresponding to the city of Almeria, Spain.

Cooling load calculation

The software EnergyPlus, through the interface Openstudio is used to reproduce the thermal behaviour of the building, *i. e*. the calculation of the thermal loads. Openstudio is merely an interface for the calculation engine, EnergyPlus, which is an internationally recognized building energy simulation program, developed by the Department of Energy of the United States.

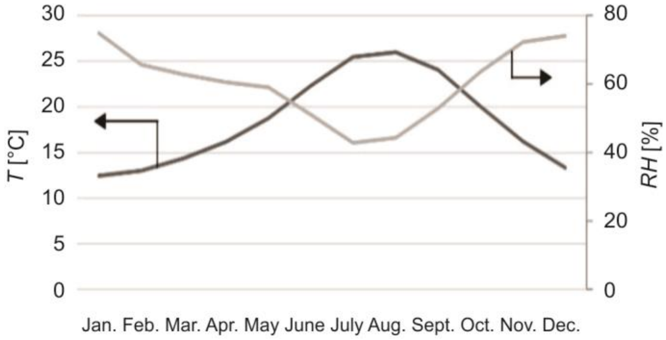

Figure 2. Mean monthly ambient temperature and relative humidity (Almería, Spain)

Cooling systems modelling

Engineering equation solver (EES) has been used for the detailed modelling and simulation of the cooling systems, since it includes, among other useful characteristics, the correlations to calculate the thermodynamic properties of water/LiBr mixtures. The equations are described in the next paragraphs.

Modelling the WC-SE and AC-DE systems

As stated in the introduction, the most commonly used solar cooling system on the market is the WC-SE water/LiBr, fig. 3(a). The objective is to model a system of this type that is capable of meeting the annual cooling demand of the proposed case study. The main goal is to simulate the system in order to get the heat of absorption and condensation to be rejected with a cooling tower for the entire cooling season.

The alternative configuration proposed in this work corresponds to an air-cooled high efficiency absorption solar cooling, fig. 3(b). The acronyms for states 1H, 1L and 1CH means 1 high, 1 low and 1 condensed-high, respectively.

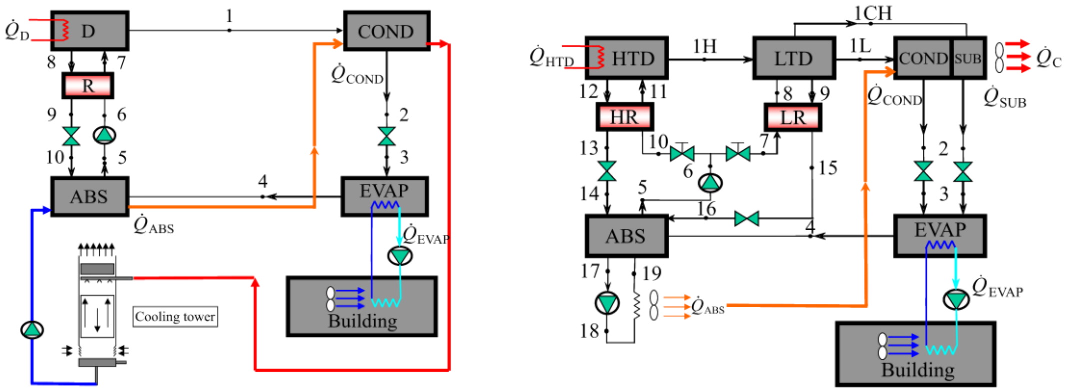

Figure 3. Schemes of a WC-SE (a) and AC-DE (b) water/LiBr machines

Concerning condensation temperatures, for this AC-DE system has been considered that it is 15 °C higher than the ambient dry bulb temperature [28], while for the WC-SE is maintained fixed at 18 °C + T_{wbextmax} = 39.1 °C.

In order to achieve the highest efficiency possible, equal increments of the solution concentration as it passes through the two desorbers have been considered [11]. Taking into account this criterion, our program has simulated the performance of the AC-DE system to meet the annual cooling demand for the hospital.

The equations primarily describing the absorption chillers operation are based on the principle of mass, energy, and species conservation. All these equations for both systems have been published by Marcos *et al.* [11].

Water consumed by the cooling tower of the WC-SE system

The total mass-flow of water consumed in a cooling tower, $\dot{m}_{\text{wc}}$, is the sum of three components: evaporated water, $\dot{m}_{\text{ev}}$, drift water, $\dot{m}_{\text{dr}}$, and blowdown water, $\dot{m}_{\text{bd}}$:

$$\dot{m}_{\text{wc}} = \dot{m}_{\text{ev}} + \dot{m}_{\text{dr}} + \dot{m}_{\text{bd}} \tag{1}$$

The calculation of each of these three components is described below to finally be able to determine the mass of total water consumed for the entire year.

Total water mass-flow rate entering the cooling tower:

$$\dot{Q}_{\text{COND}} + \dot{Q}_{\text{ABS}} = \dot{m}_{\text{w}} c_{\text{p}} \Delta T \tag{2}$$

$$\dot{m}_{\text{w}} = \frac{\dot{Q}_{\text{COND}} + \dot{Q}_{\text{ABS}}}{c_{\text{p}} \Delta T} \tag{3}$$

Being ΔT the difference between the water inlet temperature to the cooling tower and the outlet temperature, obtained from the expression:

$$T_{\mathrm{COND}} - T_{\mathrm{wbext}} = \Delta T_{\mathrm{app}} + \Delta T + (T_{\mathrm{COND}} - T_{\mathrm{wo}}) \tag{4}$$

where $\Delta T_{\mathrm{app}} = T_{\mathrm{wo}} - T_{\mathrm{bh}} = 7$ °C is the tower approach, and the minimum condenser temperature difference is controlled at $T_{\mathrm{COND}} - T_{\mathrm{wi}} = 5$ °C.

Evaporated water mass-flow rate. The mass-flow rate of water evaporated in a cooling tower corresponds to the evaporation produced by the latent heat of vaporization.

In a cooling tower of a WC-SE system the total heat to be evacuated is the sum of absorption plus condensation heat. This total heat has a sensible part (estimated at 15% on average) and a latent heat part (85%) [29]:

$$\dot{Q}_{\mathrm{tower}} = \dot{Q}_{\mathrm{COND}} + \dot{Q}_{\mathrm{ABS}} \tag{5}$$

$$\dot{Q}_{\mathrm{lat}} = 0.85(\dot{Q}_{\mathrm{COND}} + \dot{Q}_{\mathrm{ABS}}) \tag{6}$$

$$\dot{Q}_{\mathrm{lat}} = \dot{m}_{\mathrm{ev}} h_{\mathrm{fg}} \tag{7}$$

From eq. (7) evaporated water mass flow rate is obtained:

$$\dot{m}_{\mathrm{ev}} = \frac{\dot{Q}_{\mathrm{lat}}}{h_{\mathrm{fg}}} \tag{8}$$

Drift water mass-flow rate. The drift water is estimated as a function of the efficiency of the droplet separator. If it is of average quality, it means that 0.01% of the total recirculating water is consumed [26]:

$$\dot{m}_{\mathrm{dr}} = 0.0001 \dot{m}_{\mathrm{w}} \tag{9}$$

Blowdown water mass-flow rate. Blowdown water is the portion of the circulating water that is removed from the system to avoid the excess of dissolved solids/impurities contained in the water. In order to calculate the blowdown water mass flow first it is necessary to calculate the so-called *cycles of concentration* (*CC*):

$$CC = \frac{\text{Level of total dissolved solids of the recirculating water}}{\text{Level of total dissolved solids of the make-up water}} \tag{10}$$

Water consumption in cooling towers is reduced by increasing the concentration cycles. Determining the optimum number of concentration cycles is a balancing act between the reduced chemical, water and sewage costs at higher *CC vs*. the increased risk of scale formation, corrosion and biofilm growth. Therefore, with higher *CC*, raw water quality, pretreatment and using inhibitors in the cooling tower become more important [30].

A typical galvanized enclosure with an epoxy coating is taken. Selecting the parameters of quality of the re-circulation water in cooling towers recommended by Spanish regulations (R.D. 865/2003) the determinant ratio in the concentration cycles is equal to:

$$CC = \frac{1{,}200 \text{ ppm}}{500 \text{ ppm}} = 2.4 \tag{11}$$

The blowdown water can be expressed as [26]:

$$\dot{m}_{\mathrm{bd}} = \frac{\dot{m}_{\mathrm{ev}} + \dot{m}_{\mathrm{dr}}}{CC - 1} \tag{12}$$

After calculating the three water flows that are subtracted from the total re-circulating water of the cooling tower, the total water consumption can be calculated from eq. (1).

Next, the simulation of all these water mass-flow rates is presented in detail for the whole year. By means of integration, the total water consumption is calculated.

Modelling the mechanical vapour compression refrigeration system

The system modelled is a 410A simple cycle vapour compression system, defined by its evaporation and condensation pressures:

$$P_{\mathrm{EVAP}} = P_{\mathrm{sat}}\ (5\ ^{\circ}\mathrm{C}) \tag{13}$$

$$P_{\mathrm{COND}} = P_{\mathrm{sat}}\ (\mathrm{T}_{\mathrm{ext}} + 5\ ^{\circ}\mathrm{C}) \tag{14}$$

where T_{ext} is the exterior dry bulb temperature, its isentropic efficiency $\eta_{\mathrm{s}} = 0.85$ and its four characteristic cycle points are:

– evaporator outlet: saturated vapour at evaporation pressure P_{EVAP}:

$$h_1 = h_{\mathrm{g}}(P_{\mathrm{EVAP}}) \tag{15}$$

$$P_1 = P_{\mathrm{EVAP}} \tag{16}$$

– isoentropic compression outlet:

$$s_2 = s_1 \tag{17}$$

$$P_2 = P_{\mathrm{COND}} \tag{18}$$

– real compressor outlet:

$$h_3 = h_1 + \frac{h_2 - h_1}{\eta_{\mathrm{s}}} \tag{19}$$

$$P_3 = P_{\mathrm{COND}} \tag{20}$$

– condenser outlet: saturated liquid at condensing pressure P_{COND}:

$$h_4 = h_{\mathrm{f}}(P_{\mathrm{COND}}) \tag{21}$$

$$P_4 = P_{\mathrm{COND}} \tag{22}$$

– expansion valve outlet:

$$h_5 = h_4 \tag{23}$$

$$P_5 = P_{\mathrm{EVAP}} \tag{24}$$

Model validation of the single and double effect absorption chillers

The model for the simple effect water-cooled absorption chiller is taken from the literature and is widely validated. The new model for the double effect air-cooled absorption chiller is validated in this previous publication [11], where it is shown that the deviation between the model and experimental data do not exceed 3% in both COP and HTD heat transfer rate, as can be seen in tab. 1.

Table 1. Model validation for a double effect chiller

Parameters: T_{amb} = 32 °C; T_{COND} = 44 °C; T_{EVAP} = 5 °C; T_{HTD} = 164 °C; $\dot{Q}_{EVAP}$ = 4 kW			
	Experimental	Simulated	Difference
$\dot{Q}_{HTD}$ [kW]	4	3.9	2.5%
COP [a]	0.85	0.87	2.4%
Parameters: T_{amb} = 29 °C; T_{COND} = 41 °C; T_{EVAP} = 5 °C; T_{HTD} = 157 °C; $\dot{Q}_{EVAP}$ = 4.3 kW			
	Experimental	Simulated	Difference
$\dot{Q}_{HTD}$ [kW]	4.5	4.47	0.6%
COP [a]	1.15	1.12	2.6%

[a] COP calculated excluding the power demand of the auxiliary equipment

Results and discussion

This section shows the different data extracted from the different simulations, including thermal loads, heat rates, water consumption, performance of the systems and energy/CO_2 emission saving figures.

Thermal loads

The calculation of cooling thermal loads for the entire year was carried out with EnergyPlus. The maximum cooling power required is 2,549 kW, fig. 4. On the other hand, the energy demand for air conditioning has been calculated reaching its maximum in the month of August with a value of 947 MWh, fig. 5.

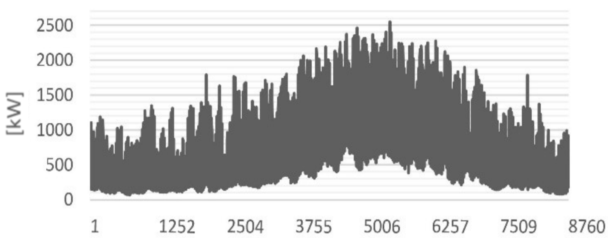

Figure 4. Cooling loads on hourly base

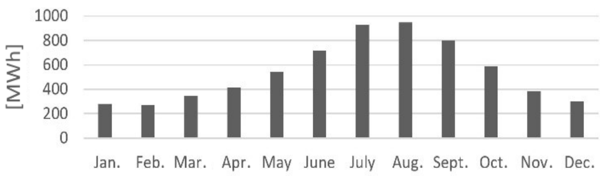

Figure 5. Cooling energy on monthly base

Absorption and condensation heat transfer rates for the WC-SE system

The results provided by EnergyPlus provide the annual cooling thermal loads for the analyzed hospital, fig. 1. These thermal loads together with the ambient temperature values corresponding to the city of Almeria are inputs for the simulation program of the WC-SE system. This program has been carried out with the EES.

Figure 6 shows the annual variation of the sum of absorption and condensation heat transfer rates. As can be seen, the maximum power to be evacuated by the cooling tower is 5,695 kW.

Water consumption by the WC-SE system

Figure 7 represents mass-flows rates involved in a cooling tower modelled in the previous section. From this figure it can be concluded that the water consumption in the cooling tower comes mainly from the evaporation and blowdown water. Drift water plays a role much less relevant.

The annual water consumed by the cooling tower was 35376.5 m^3. This is the total water saved by the system proposed.

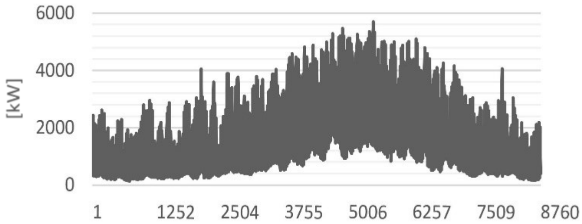

Figure 6. Sum of absorption and condensation heat transfer rates for the WC-SE system

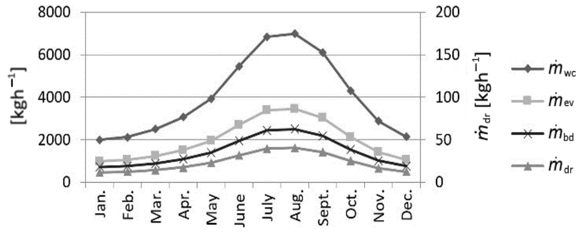

Figure 7. Total water consumed, $\dot{m}_{wc}$, evaporated, $\dot{m}_{ev}$, blowdown, $\dot{m}_{bd}$, and drift water, $\dot{m}_{dr}$

The COP optimization for the AC-DE system

Figure 8 shows the results of the simulation comparing the COP provided by the WC-SE system *vs*. the AC-DE. It is appreciated that, although an air-cooled system decreases the efficiency with respect to a water-cooled one, the use of a double effect machine provides a higher COP. It is also noted that the decrease of the COP for WC-SE system in summer is much smaller than for the AC-DE. This is due to the fact that summer humidity drops significantly and causes the wet bulb temperature in the machine to be much lower. This effect leads to a lower condensation temperature and therefore favours the COP of the WC-SE system.

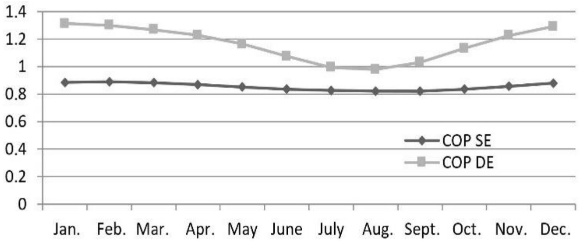

Figure 8. Mean monthly COP of AC-DE *vs.* WC-SE

Saving energy and CO_2 emissions reduction

This section analyses the environmental benefits in energy savings and reduction of emissions that entails the use of the AC-DE system. As previously specified, a solar fraction of 75% is used as the reference criterion [20]. A comparison is first made with respect to a conventional mechanical compression system and, secondly, respect to a WC-SE. The refrigerant in the conventional mechanical compression system is R410A.

Table 2 presents an annual energy balance for all three systems. The AC-DE system requires 35% less energy to meet the cooling demand than the WC-SE. The saving in electricity consumption of the absorption systems compared to the mechanical compression is in the order of 1,250 MWh per year.

Table 2. Annual energy balance for a conventional, WC-SE and AC-DE system

Annual energy balance	Conventional	WC-SE	AC-DE
Annual electricity consumption [MWh]	1,253.95	0.12	1.13
Annual required heat for cooling [MWh]	–	7,740.4	5,037.8
Annual heat from 2nd heat source (fossil fuel) [MWh]	–	1,935.1	1,259.5

The environmental impact of the refrigerants is evaluated by means of the parameter total equivalent warming impact (*TEWI*), eq. (25). This indicator consists of two terms: the direct emissions, that takes into account the refrigerant that leaks directly into the air, and the indirect emissions, that refers to the emissions due to the generation of electricity:

$$TEWI = \mathrm{GWP} \times f \times M_\mathrm{r} \times N + E \times N \times \alpha \tag{25}$$

where M_r is the mass of refrigerant, and it is determined according to the dimensions of the condenser, evaporator, compressors, and tubing. In this work it was estimated 750 kg of refrigerant taking into account the cooling capacity of the system (2,550 kW) [31], the f is the annual leakage amount, and is described as percentage for total mass of refrigerant, f was assumed as 7% [32], N refers to the operating life of the system. It can be evaluated as 20 years horizon. The GWP for R410A is 3,400 [3], E is the electric consumption in kWh per year, and α is the indirect emission factor (kgCO_2/kWh). In the present work it has been considered 0.4 kgCO_2/kWh [33].

Table 3 shows the environmental benefits provided by the use of the AC-DE system. It also includes savings in water consumption.

Finally, a quantitative analysis of the environmental footprints generated by the three systems was carried out. Among the different categories inserted within the environmental footprints the two most influential in the systems studied were quantified: carbon and water footprints, tab. 4.

Table 3. Annual environmental benefits produced by the AC-DE system

Environmental benefits respect to compression	AC-DE
Saved electricity [MWh]	1,252.8
CO_2 savings due to electricity savings (*TEWI*) [tons]	13,592.6
CO_2 emissions due to 2nd heat source (fossil fuel) [tons]	2,585
Total CO_2 savings [tons]	11,007.6
Environmental benefits respect to WC-SE	
Total water consumption savings [m^3]	35,376.5

Table 4. Environmental footprints for a conventional, WC-SE and AC-DE system

Environmental footprints	Conventional	WC-SE	AC-DE
Carbon footprint [tCO_2 per year]	13,592.6	0.952	9.04
Water footprint [m^3 per year]	–	35,376.5	–

Conclusions

The present work analyses the environmental and efficient benefits of an optimized AC-DE system *vs*. a WC-SE in hot and water scarce regions. For this, both systems have been simulated for an entire year by choosing as a case study for comparison a hospital located in Almeria (southern Spain).

The main conclusion of this work is the clear advantage of the absorption system proposed (air-cooled double effect) over the traditional absorption system (water cooled-single effect) in the case proposed, endorsed by the obtained results.

Concerning the system performance, the COP of the system proposed is between 19% and 48% higher than WC-SE system and it consumes a 35% less energy to meet the cooling demand. On the other hand, comparing to a conventional mechanical compression system the saving in electricity consumption was 1,253 MWh.

With regard to environmental issues, this system saves annually around 35,377 m^3 of water and 11,007.6 tons of CO_2 compared to the water-cooled one, being its environmental footprint the lowest among the analysed systems.

Nomenclature

c_p – specific heat at constant pressure, [$kJkg^{-1}$]
D – desorber or generator
E – electric consumption, [kWh $year^{-1}$]
f – annual leakage amount
h – specific enthalpy, [$kJkg^{-1}$]
h_{fg} – latent heat of vaporization, [$kJkg^{-1}$]
M_r – refrigerant mass, [kg]
$\dot{m}$ – mass-flow, [kgs^{-1}]
N – operating life, [years]
P – pressure, [bar]
$\dot{Q}$ – heat transfer rate, [W]
R – regenerator
T – temperature, [°C]

Greek symbols

α – indirect emission factor, [$kgCO_2kWh^{-1}$]
η_s – isoentropic efficiency, [–]

Subscripts

bd – blowdown water
dr – drift water
ev – evaporated water
ext – exterior
f – saturated liquid
g – saturated vapour
lat – latent
sat – saturation
wb – wet bulb
wbext – exterior wet bulb
wbextmax – maximum exterior wet bulb
wi – inlet water cooling tower
wo – inlet water cooling tower
wc – total water consumption

Acronyms

ABS – absorber
AC-DE – air-cooled double effect
CC – cycles of concentration
COND – condenser
COP – coefficient of performance
DHW – domestic hot water
EES – engineering equation solver
EVAP – evaporator
GWP – global warming potential
HR – high temperature regenerator
HTD – high temperature desorber
LR – low temperature regenerator
LTD – low temperature desorber
MET – weather format for SWEC climatic database
RH – relative humidity
SUB – sub-cooler
SWEC – spanish weather for energy calculations
TEWI – total equivalent warming impact
TMY – tipical meteorological year
WC-SE – water-cooled single effect

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