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Modelling and control optimization of a solar desiccant and evaporative cooling system using an electrical heat pump

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Abstract

The aim of this work is the control optimization of a new solar assisted air-conditioning concept which combines a desiccant and evaporative cooling (DEC) system with an electrical heat pump. The DEC air handling unit configuration had to prevent return air from mixing with supply air. Therefore, a flat plate sensible heat exchanger is used instead of a rotary. Moreover an additional stream of outdoor air is used for the desiccant wheel regeneration. A reversible water/water heat pump is also included in the system. In summer, the heat pump cools the supply air stream and pre-heats the regeneration air when dehumidification is needed; in winter, the heat pump provides auxiliary heat if a minimum temperature is available in the heat storage, otherwise a backup boiler is used. The depicted system has been modelled and an extensive simulation work has been carried out in order to verify the control capability during the different operation modes. As a result an optimal control strategy has been identified. According to simulations, the system can deliver primary air at the requested temperature and humidity while holding the overall electricity consumption at significant low levels compared to reference system solutions.

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Keywords: Air-conditioning; DEC; desiccant cooling; control; simulation

1. Introduction

Few are the solar assisted desiccant and evaporative cooling (DEC) systems which have been installed worldwide until now. Within the group of experts who worked at the IEA-SHC (e.g., Task 38 – Solar Air-conditioning and refrigeration) [1], [2], data on seventeen solar assisted DECs have been reported. The

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large majority of them have been installed in Central Europe. They often employ dehumidifiers using solid desiccant and are driven by liquid or air flat plate collectors. For the reported systems, among the main drawbacks there is the limited dehumidification capacity achievable in relation to the relatively low regeneration temperature provided by the solar systems. This is a crucial aspect for humid climates, such those encountered in South Europe. In order to overcome this limitation different DEC configurations can be employed. Among them a hybrid concept, in which an electrically driven heat pump (EHP) provides active heat recovery between return and supply air, have been investigated [3]. In a hybrid system, the cooling capacity provided by the EHP can be used to enhance both sensible and latent cooling capacity, according to the specific application needs; whereas the heat rejected by the EHP can become a heat source for preheating the regeneration air.

In the mentioned works dealing with solar DEC systems, it is highlighted that some important technical aspects shall be considered in the design of these plants. In particular the following issues have to be tackled: selection of cold or hot backup, use of direct and/or indirect evaporative cooling, efficiency reduction due to leakages in the heat exchanger section (when a rotary exchanger is in use).

The choice of a backup heat generator, to use when the solar heat is not available, might have severe consequences on the overall system efficiency, in primary energy terms. In particular, being the thermal COP of conventional DEC air handling units, normally in the range of 0.6 to 0.9, the use of a standard backup heater (e.g., a gas boiler) results in a poor primary energy yearly performance in most of the globe [4].

Standard DEC systems employ humidifiers both on the supply and return air channels, allowing t the implementation of both direct and indirect evaporative cooling; and their combination. However in such air handling units (AHU) the humidifiers require frequent maintenance due to hygienic concerns, such as legionella proliferation, and accumulation of solid residues released with water evaporation. Ways to avoid maintenance problems include frequent cleaning of the humidifier basin and water demineralization, which both lead to not negligible additional power consumption. Furthermore, systems running in direct evaporative cooling mode are not always easy to control, depending on the typology of humidifier is employed on the supply stream channel.

In systems using rotary heat exchangers, particularly for low nominal volume flow rates (below 5000 $m^3\ h^{-1}$), there is the risk of leakages through the rotor case. Mixing of return air with supply air decrements both efficiency and air cleanness. Therefore, the positioning of fans and the tightness of the rotors sealing shall be accurately verified. Finally, pressure drops inside the air handling unit shall be kept as low as possible in order to limit fans power.

As mentioned above, the three discussed problems can be faced in DEC systems based on standard configurations. In order to avoid the possible problems linked with the three technical issues a careful design has to be carried out. Another possible solution is to adopt a new solar assisted air-conditioning system based on a particular version of the mentioned hybrid configuration: the desiccant and evaporative cooling concept with an electrical heat pump. An example of this hybrid configuration is given in Fig. 1.

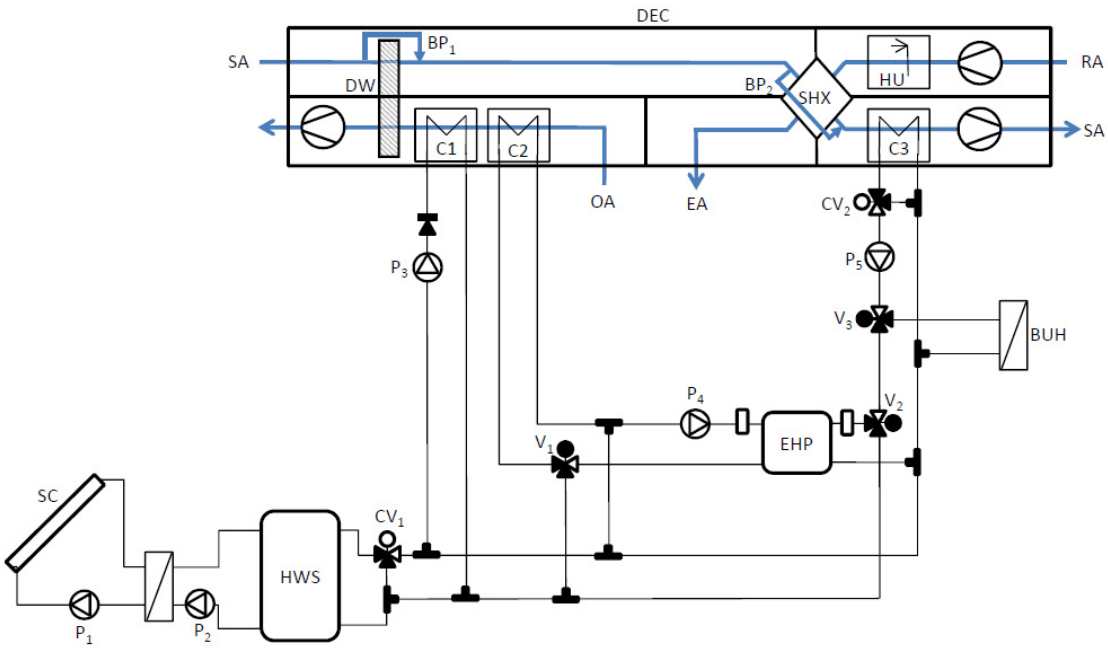

Fig. 1. Scheme of the hybrid solar DEC system

The given scheme has been developed for a real application and will be installed at a university dormitory located in Milan, Italy. The plant is intended to supply primary air (i.e. volume flow needed for hygienic needs) to the building. The rest of the building heating and cooling loads are covered by a distributed terminals system, which is not in the scope of this work.

Due to odour contamination of the exhaust air, the air handling unit configuration had to prevent return air from mixing with supply air. Therefore, a flat plate sensible heat exchanger was selected instead of a rotary one and an additional stream of outdoor air was used for the desiccant wheel regeneration. A reversible water/water heat pump provides both heating in winter and sensible cooling in summer. The peculiarities of the plant lie in the following: a) in summer, the heat pump cools the supply air and pre-heats the regeneration air, when dehumidification is needed; b) in winter, the heat pump upgrades the solar heat to a higher level of temperature, if a minimum temperature is available in the heat storage, otherwise the hot backup is used.

The controller's goal is to maintain the supply air temperature within a given range (e.g., 20 ÷ 24 °C) and to limit supply the air humidity ratio to a certain value (e.g. 10.5 g kg^{-1}). Such conditions can be considered neutral, a well diffuse practice even for conventional air conditioning systems supplying ventilation air [3]. Dehumidification can be provided only by means of the desiccant rotor and the target supply air humidity will be reached only when sufficient solar heat is available. Therefore, the target humidity shall be intended as a desired condition which sometimes could not be met. Moreover, in the considered climate (south European), humidification in winter is not a diffuse practice, thus it is not foreseen. Target temperatures are strict requirements and the heat pump in summer allows reaching the desired supply temperature even if a hot backup is not present on the regeneration stream and the direct evaporative cooling is not employed.

The objective of this work is the definition and verification of the control strategy through modelling and simulation of the whole plant. The energy performance is evaluated in comparison to a conventional air handling unit using a gas boiler and an air-condensed chiller. The experimental evaluation of the plant performance will be carried out in future works.

Nomenclature

AC	Active cooling	AH	Active heating
AHU	Air-handling unit	BP	Bypass
BUH	Backup heater	C	Coil
CC	Climatic control	C-AHU	Conventional AHU
CV	Control valve	DEC	Desiccant evaporative cooling
DH	Dehumidification	DSH	Direct solar heating
DW	Desiccant wheel	PE	Primary energy (kWh_{PE})
EA	Exhaust air	EHP	Electrical heat pump
FC	Free cooling	HU	Humidifier
HR	Heat recovery	HWS	Hot water storage
HX	Heat exchanger	IEC	Indirect evaporative cooling
OA	Outdoor air	P	Pump or Power (kW)
PI	Proportional integral control	Q	Enthalpy increase (kW)
RA	Return air	RH	Relative humidity (%)
nHX	number of HX in series	SA	Supply air
SC	Solar collector	SDEC	Solar DEC system
SHP	Solar and EHP heating	SHX	Sensible heat exchanger
T	Temperature (°C)	V	Valve
h	Specific enthalpy ($J\ kg^{-1}$)		

Greek letters

η_{el}	average electric generation efficiency (-)	ω	Humidity ratio ($g\ kg^{-1}$)

Subscripts

cond	condenser of EHP	evap	evaporator of EHP

PE	non-renewable primary energy

2. Plant modelling

Algebraic and differential models of the plant components have been developed and implemented in MATLAB. The transient state problem was solved for one year using the MATLAB differential algebraic equations solver [6] which uses an internally set variable time step. Thanks to the slow variation of plant states, the controller can be updated at a fixed time step of a few minutes without compromising the quality of the control. Therefore, a simple simulation program was developed with a fixed control time step (180 s) and an internally set variable time step for the time between two adjacent control events. A simple but accurate algebraic model is proposed for the desiccant rotor steady-state outlet conditions assessment. The model was tuned on the basis of manufacturer's data. The heat pump, which can modulate at two capacity steps (50% ÷ 100%), was modelled as algebraic since its reaction time can be assumed negligible with respect to the delay caused by the inertial vessels, one on each side, series connected on each hydraulic loop. The sensible heat exchanger is based on the constant effectiveness model with a modulating effectiveness (between 10% and 100% of the maximum effectiveness) to reproduce the effect of the bypass. The hot water storage is a multinode model based on mass and energy balance at each node, including augmented heat conduction between adjacent nodes [7]. The steady-state solar collector model is based on the Hottler-Whiller-Bliss equation [8]. The effectiveness-NTU model for the counterblow heat exchanger is assumed valid for all coils. Pumps, fans and three-way valves are all modelled as algebraic equations representing simple mass and energy balances. Suitable time constants have been set for the collector field, long distribution pipes, coils, desiccant wheel and heat exchanger in order to reproduce the transient state behaviour. In order to limit the complexity of the model, the building energy balance was not addressed. Instead, a simple temperature profile, ranging from 20 to 26 °C during the year, was assigned to the return air (RA) and a fixed internal latent load was used. In the following, the desiccant rotor and the heat pump models are presented in more detail.

2.1. Desiccant rotor

The steady-state model of the desiccant rotor is an algebraic model of the form:

$$T_{12} = T_{11} + a_1(T_{21} - T_{11}) + a_2 h_{11}(T_{21} - T_{11}) + a_3(T_{21} - T_{11})^2 \tag{1}$$

$$\omega_{12} = \omega_{11} + b_1(T_{21} - T_{11}) + b_2 h_{11}(T_{21} - T_{11}) + b_3(T_{21} - T_{11})^2 \tag{2}$$

where, in the subscripts, the first digit indicates the wheel sector, i.e. processing (1) or regeneration (2), and the second digit indicates inlet (1) or outlet (2).

This model is justified because, in the considered application, the humidity ratios at process stream inlet and regeneration stream inlet are equal, the two flow rates are balanced and the rotational speed is constant. Coefficients a_1, a_2, a_3 and b_1, b_2, b_3 have been identified on the basis of the manufacturer's data. As shown in Fig. 2, percentage deviations of temperature increase and humidity drop of supply air fit manufacturer's data sufficiently well in the range of operating conditions which are of interest for the application ($T_{11} = 22 \div 34$ °C, $\omega_{11} = 8 \div 14$ g kg^{-1}, $T_{21} = 50 \div 80$ °C).

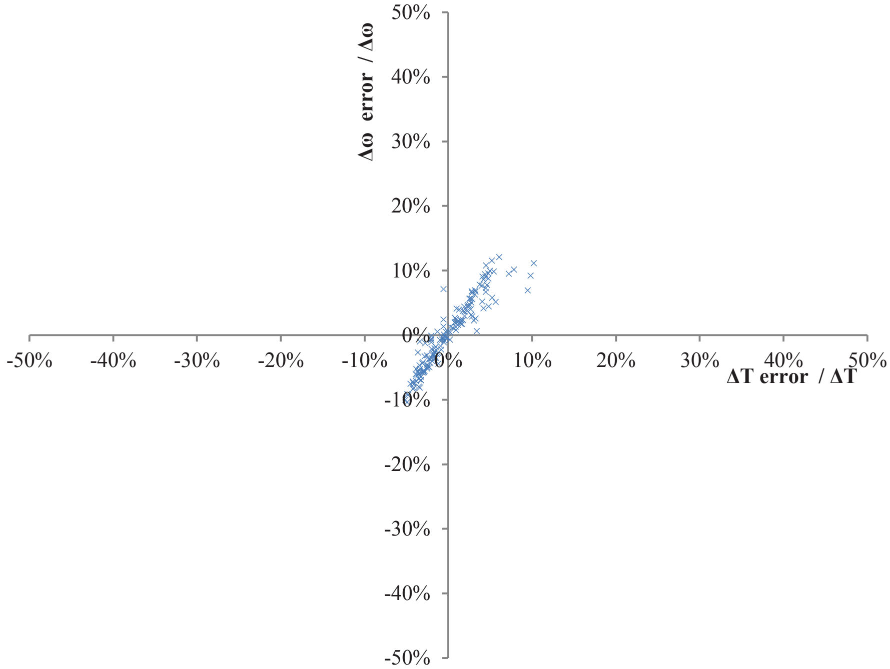

Fig. 2. Desiccant rotor model Vs manufacturer's data

A fictitious time constant is introduced in order to model the transient behaviour. At the typical operating conditions, a suitable time constant is about 500 s. The resulting equations are as follows:

$$\tau_{DW}\frac{dT_{12}}{dx} = T_{11} + a_1(T_{21}-T_{11}) + a_2 h_{11}(T_{21}-T_{11}) + a_3(T_{21}-T_{11})^2 - T_{12} \tag{3}$$

$$\tau_{DW}\frac{d\omega_{12}}{dx} = \omega_{11} + b_1(T_{21}-T_{11}) + b_2 h_{11}(T_{21}-T_{11}) + b_3(T_{21}-T_{11})^2 - \omega_{12} \tag{4}$$

2.2. Electrical heat pump

The heat pump used is water cooled, reversible, and can modulate at two capacity steps. The algebraic model consists in two sets of equations, one for each operational mode (heating or cooling). In the two systems the input variables are the capacity step (50% or 100%) and the return temperatures to condenser and evaporator. The output variables are the condensing and evaporating power. Electrical power is derived from first principle assuming negligible heat losses. The equations are polynomial whose coefficients set *(c, d)* have been identified on the basis of manufacturer's performance data at the design flow rate. The general form, valid for both heating and cooling mode, is as follows:

$$\dot{Q}_{evap} = (c_{00} + c_{10}T_l + c_{01}T_h + c_{20}T_l^2 + c_{11}T_lT_h + c_{02}T_h^2 + c_{30}T_l^3 + c_{21}T_l^2T_h + c_{12}T_lT_h^2 + c_{03}T_h^3)S_{ctrl} \tag{5}$$

$$\dot{Q}_{cond} = (d_{00} + d_{10}T_l + d_{01}T_h + d_{20}T_l^2 + d_{11}T_lT_h + d_{02}T_h^2 + d_{30}T_l^3 + d_{21}T_l^2T_h + d_{12}T_lT_h^2 + d_{03}T_h^3)S_{ctrl} \tag{6}$$

$$\dot{P}_{ele} = \dot{Q}_{cond} - \dot{Q}_{evap} \tag{7}$$

where S_{ctrl} is the capacity step and, in the subscripts, "h" indicates the temperature at the inlet of the condenser and "l" indicates the temperature at the inlet of the evaporator.

3. Control strategy definition

In general, AHU controllers use sequencing logic in order to maintain the desired comfort conditions in the most economic way. One drawback of this approach is the risk of instability when several components are controlled at the same time. This problem can be partly addressed in the tuning phase of the control parameters, an approach which tends to produce a sluggish controller [9]. Sequencing logic applied to DEC systems may result in a complex tuning exercise due to the number of components involved. Therefore, alternative approaches have been investigated in which a subset of components are operated steadily according to some rules [10-11] and only a limited number of components use feedback control simultaneously. The particular operating condition is associated to a well defined 'operation mode' (e.g., free cooling, indirect evaporative cooling, active cooling). The transition from one operation mode to another one is managed by a selector which can be based on simple heuristic rules. Actuators use can be reduced by adopting a finite state machine approach [9], i.e. delaying the transition until the current operation mode has reached saturation for a predefined time interval. In the following, the analysis of the system operation modes is presented and the adopted control logic is described.

The DEC system operates at constant and balanced flow rates (6000 m3 h^{-1}), which are the minimum required, at the given application, for pollution control requirements. Neutral primary air shall be supplied to the conditioned space, i.e., $T_{SA,min}$ = 20 °C, $T_{SA,max}$= 24 °C. The maximum supply air humidity $\omega_{SA,max}$ is fixed at 10.5 g kg^{-1}. According to these requirements, the possible operation modes can be defined as follows. A graphical presentation is given in Fig. 3.

3.1. Cooling modes

The simplest operation mode is free cooling (FC), in which only the supply and return fans are on, the flow rates are fixed and the bypasses BP_1 BP_2 are open (see Fig. 1). Considering the fan heat, outdoor air limits (T_B, T_C) which lead to the allowed supply air temperature range are easily calculated (see Fig. 3).

Above T_C, the indirect evaporative cooling mode is activated (IEC). RA temperature is heated by the fan and adiabatically cooled by the humidifier, thus capable to provide cooling. The sensible heat exchanger bypass (BP_2) is PI controlled in order to maintain $T_{SA,max}$. Assuming RA at 26 °C and 60% RH and considering the HX effectiveness, the limit temperature T_D can be calculated. Above T_D, active cooling (AC) is needed. The electrical heat pump is turned on in cooling mode, along with the associated pumps (P_4, P_5) and the regeneration fan. The control valve CV_2 is PI controlled in order to keep constant $T_{SA,max}$.

3.2. Heating modes

Below T_B, little margin exists for heat recovery (HR) since return temperature is not expected to be higher than 21°C. Nevertheless, this operation mode is introduced in order to assure stability. The sensible heat exchanger bypass (BP_2) is PI controlled in order to keep $T_{SA,min}$ constant. Considering fans heat and sensible heat exchanger effectiveness, the limit outdoor temperature T_A is calculated. Below T_A, active heating (AH) is necessary. The pump P_5 is turned on and the control valve CV_2 is PI controlled in order to maintain $T_{SA,min}$. Heat can be supplied in three ways, depending on the available temperature (T_{HWS}) in the hot water storage. If T_{HWS} is above 35°C with 10 °C hysteresis, direct solar heating (DSH) is possible. If T_{HWS} is above 15°C with 5 °C hysteresis, solar and heat pump heating (SHP) are allowed. In this mode, EHP is turned on in heating mode and P_4 is also operated. If none of the previous modes can be operated, the backup heating (BUH) is used.

3.3. Dehumidification mode

When the outdoor air (OA) humidity ratio is higher than the maximum allowed, dehumidification (DH) is needed. If T_{HWS} is above 50°C with 10 °C hysteresis, the regeneration fan is turned on along with the wheel motor and the pump P_3. The bypass around the wheel (BP_1) is closed and the control valve CV_1 is PI controlled having as target $\omega_{SA,max}$. Due to the fact that humidity drop is associated to temperature increase across the desiccant wheel, the vertical lines T_A, T_B, T_C, T_D bend to the left in dehumidification mode (see Fig. 3).

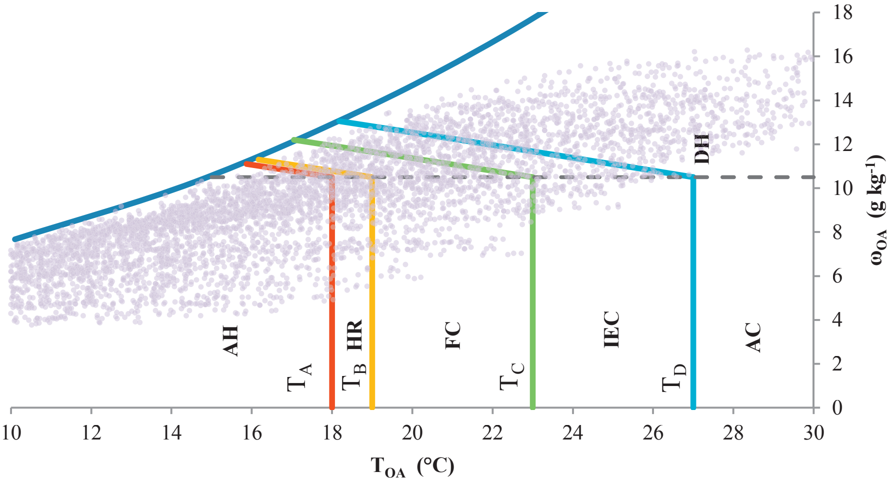

Fig. 3. Operation modes Vs outdoor air conditions

Having defined the OA temperature and humidity conditions associated to the main operations mode, it is possible to verify how often the different operation modes are used for the given application (typology of end-user and considered climate). It can be already said that AC is useful only when also DH is needed and for the majority of the DH hours. Moreover, AH is not used when DH is needed. Hence, the power consumed for the regeneration fan is well used.

3.4. Control logic

The solar subsystem, comprising pumps P_1 and P_2, is independently controlled according to the collector temperature and the storage temperatures, as usual. The heat pump compressors are internally controlled on the basis of the feed water temperature, 10 °C in cooling mode and 35 °C in heating mode. All other actuators are controlled by a programmable automation controller according to the selected operation mode. The logic used to manage the transition from one mode to the other is illustrated in Fig. 4. Climatic control is performed on the basis of outdoor temperature. $T_{SA,max}$ is proportionally decreased when OA temperature exceeds 26 °C. Similarly, $T_{SA,min}$ is proportionally increased when OA temperature falls below 15 °C. The maximum absolute increase (in heating) and decrease (in cooling) are set to 2 and 2 °C, respectively (accordingly, the climatic control configuration is denominated CC 2/2).

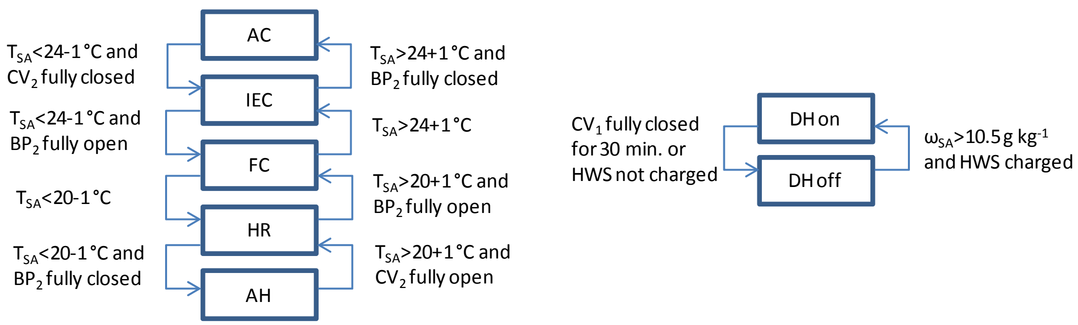

Fig. 4. Logic of operation mode selector

4. Simulation results and optimization

The energy performance of the system for the different operation modes is shown in Table 1. About cooling, it is clear that IEC is seldom useful alone, as expected. However, IEC is also used in DH mode, together with AC. Regarding heating modes, DSH and SHP together cover more than 50% of the heating demand, thus greatly contributing to the reduction of the BUH mode, the mode with highest total PE consumption.

Table 1. Operation mode hours and energy performance, CC 2/2

Mode	Time (hrs)	Q_{tot} (kWh)	Q_{lat} (kWh)	PE_{fans} (kWh_{PE})	PE_{aux} (kWh_{PE})	PE_{sol} (kWh_{PE})	PE_{EHP} (kWh_{PE})	PE_{BUH} (kWh_{PE})	PE_{tot} (kWh_{PE})
DH+AC+IEC	752	-13,101	-9,031	6,868	1,081	309	2,508		10,765
IEC	324	-292		2, 468	165	93			2,726
FC	1,006	2,111		5,399	84	146			5,630
HR	1,520	16,130		7,626	17	125			7,767
DSH	2,295	55,029		11,984	647	534			13,165
SHP	941	31,930		4,910	439	290	3,761		9,399
BUH	1,922	78,144		10,030	542	235		31,656	42,463

The primary energy saving of the plant (SDEC) with respect to a conventional AHU using a gas boiler and an air-condensed chiller (C-AHU) is of about 45,500 kWh_{PE} (33% of total PE consumption of C-AHU). Savings are achieved during every month (see Fig. 5).

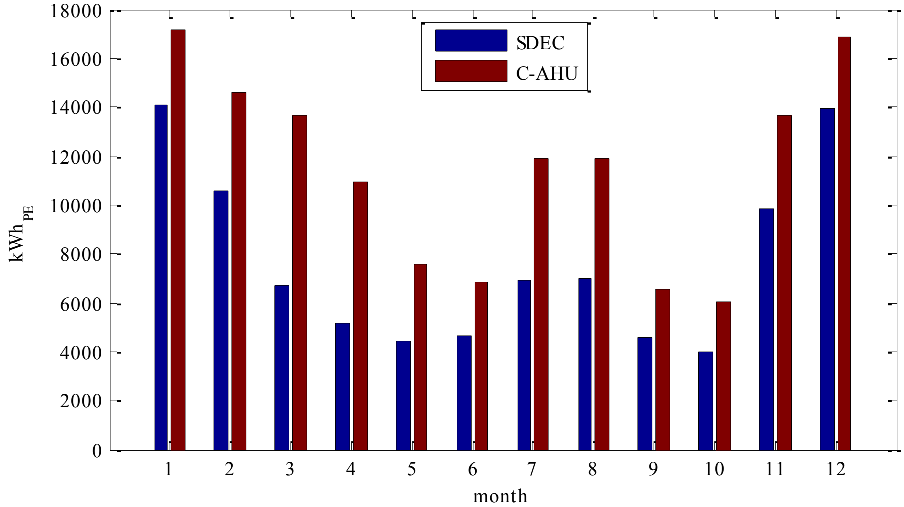

Fig. 5. Monthly primary energy consumption of the system (SDEC) and a reference conventional AHU (C-AHU)

The achieved SA temperature and humidity conditions during cooling and heating are shown in Fig. 6. The effect of climatic control was also investigated by varying the maximum temperature decrease in summer. The influence on SA temperature and humidity for the settings CC 2/0, CC 2/4 is also shown. Increasing the target temperature in summer (CC 2/0) leads to -9.4 % dehumidification capacity, whereas decreasing it (CC 2/4) provides +8.2 % dehumidification capacity. The overall primary energy savings with respect to the reference conventional system are varying less markedly (+/- 0.6%).

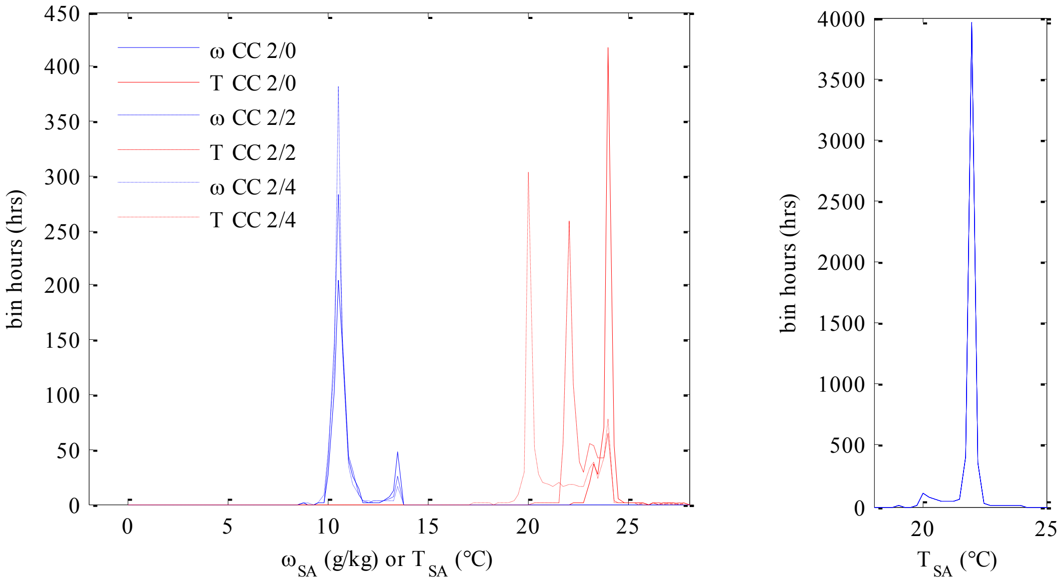

Fig. 6. Histograms for SA temperature and humidity (a) in heating and (b) in cooling (bins size is 0.25 K for temperature and 0.25 g/kg for humidity)

Finally, the configuration of the system was changed adding a second sensible heat exchanger in series with the first one and of the same type and efficiency. An increase in PE savings of about 10% was achieved with respect to the base case (CC 2/2), despite of an increase of about 12% in electricity consumption of the fans. During summer, the recovery of sensible cool increases while, consequently, the contribution of the heat pump decreases. On the other hand, the latent energy removed from SA reduces of about 8.5% due to lower pre heating of the regeneration air. During winter the operating hours of the HR mode increase with a significant reduction of the PE consumption in BUH mode (60%).

5. Conclusion

The predicted energy performance of the system is promising. Total primary energy consumption amounts to nearly 92 MWh, saving some 45.5 MWh with respect to a conventional air handling unit using a gas boiler and an air-condensed chiller. Savings are equivalently achieved in winter and summer.

The analysis has shown that, in the considered climate, supply air is sufficiently well controlled, although little margin exists towards both severe winter temperatures and very high summer humidity levels.

One interesting feature of the system is that, when the heat pump is active, supply air is cooled and simultaneously, if dehumidification is also needed, the desiccant wheel regeneration air is pre-heated. Therefore, adjusting the supply air target temperature has some influence on the dehumidification capacity of the system, at constant solar collector area and storage size.

A second heat exchanger would lead to additional 10% primary energy savings but would lower the dehumidification capacity of the system. This fact is another consequence of the interrelation existing between the sensible cooling effect provided by the heat pump and the associated pre heating effect of the regeneration air.

The developed simulation tool will be very useful also during the commissioning phase of the plant, in order to quickly detect possible system failures and control faults.

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