DESIGN AND SIMULATION OF A SOLAR ASSISTED DESICCANT-BASED AIR HANDLING UNIT

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ABSTRACT

Desiccant-based Air Handling Units (AHU) can guarantee significant technical and energy/environmental advantages related to the use of traditional ones (with dehumidification by cooling). For these reasons, a test facility has been located in Benevento (Southern Italy), in which a silica-gel desiccant wheel is inserted in an AHU which treats outside air only. For this wheel, the regeneration temperature can be as low as 65 °C, therefore energy savings and emissions reductions are more consistent when the regeneration of the desiccant material is obtained by means of available low grade thermal energy, such as that from solar collectors or cogenerators.

In the actual configuration, regeneration is obtained by means of thermal energy recovered from a micro-cogenerator (MCHP, Micro Combined Heat and Power) based on a natural gas-fired reciprocating internal combustion engine, eventually integrated through a natural gas-fired boiler.

Future activity aims to reduce the regeneration fossil energy requirements by introducing a solar collector system that substitutes or integrates thermal energy supplied by the MHCP.

To this aim, a commercial software has been used to design the solar collector system (collectors type, absorber area, water flow rate…) considering the thermal power and temperature requirements of the regeneration process.

The existing AHU and the designed solar collector system have been successively simulated by means of TRNSYS software, in order to evaluate operational data and performance parameters of the system in a typical week of operation, e.g. thermal-hygrometric conditions of air in the mean sections of the AHU, solar collectors efficiency and solar fraction.

INTRODUCTION

In order to obtain dehumidification, in a conventional air conditioning system, air is usually deeply cooled, below the dew point temperature, by means of a coil interacting with an electric vapour compression chiller. Subsequently, it is heated up (e.g. electrically) to the desired supply temperature.

In a desiccant cooling system, moist air is dehumidified by means of a Desiccant Wheel (DW), increasing overall system energy efficiency by avoiding overcooling air and reheating [1]. The process air stream flows through the desiccant material (such as silica gel, activated alumina, lithium chloride salt, or molecular sieves) that retains the moisture of the air. The desiccant capacity of this material can be restored through its regeneration via a warm air stream, usually heated by a gas-fired boiler. The process air stream, exiting the wheel, is then cooled to desired supply temperature, for example by the cooling coil of an electric chiller.

Some of the main advantages of these systems, in comparison with conventional ones, are [2]:

- sensible and latent loads can be controlled separately;
- the chiller is smaller and operates at a small temperature lift with a greater COP;
- reduced electric energy use;
- consistent energy savings can be obtained;
- reduced environmental impact.

Moreover, desiccant cooling systems are an interesting technology for sustainable building air conditioning, as the main energy required is low temperature heat, which can be supplied by solar thermal energy or waste heat.

The use of solar energy, being a renewable energy source, has several advantages, e.g.:

- reduction of fossil fuel demand;
- energy source differentiation;
- reduction of the environmental impact.

In particular, the use of solar energy for space cooling requirements (solar cooling) is highly desirable, because its availability coincides with the need for cooling, therefore the summer peak demand of electricity due to extensive use of electric air conditioners, that matches with the peak solar irradiance, can be lowered [3].

The aim of this paper, after describing the actual configuration of a test facility in which a desiccant-based Air Handling Unit (AHU) is experimentally tested, is to design a Solar Collectors (SC) system that provides the required regeneration thermal energy.

The designed solar assisted desiccant cooling system is successively simulated by means of TRNSYS software, in order to evaluate operational data and performance parameters of the system in a typical day of operation, e.g. thermal-hygrometric conditions of air in the mean sections of the AHU and the Solar Fraction (SF).

NOMENCLATURE

A	[m^2]	Collectors Area
COP	[-]	Coefficient Of Performance
G	[W/m^2]	Solar radiation
$\dot{m}$	[kg/s]	Mass flow rate
P	[kW]	Power
SF	[-]	Solar Fraction
t	[°C]	Temperature

Special characters

η	[-]	Efficiency
ω	[g/kg]	Humidity Ratio

Subscripts

amb	Ambient
B	Boiler
co	Cooling
el	Electric
th	Thermal

SOLAR ASSISTED DESICCANT-BASED AHU

At Sannio University, in Benevento (Southern Italy), a desiccant AHU coupled to a micro-cogenerator based on a natural gas-fired reciprocating internal combustion engine, an electric chiller and a natural gas-fired boiler, is experimentally tested [4].

Nominal characteristics of the devices are:

- cogenerator: P_{el} = 6.00 kW, P_{th} = 11.7 kW, η_{el} = 28.8%, η_{th} = 56.2%;
- air-cooled water chiller: P_{co} = 8.50 kW, COP = 3.00;
- boiler: P_{th} = 24.1 kW, η_B = 90.2%;
- desiccant based AHU: 800 m^3/h of process air.

The MCHP (Micro Combined Heat and Power) unit supplies thermal power for the desiccant wheel regeneration and electric energy for AHU electric loads (fans, pumps, desiccant wheel, etc.), chiller and further external electric devices (computers, lights, etc.).

The boiler can be used to integrate thermal power supplied by the micro-cogenerator, e.g. when very humid outdoor air conditions require a higher regeneration temperature.

The DW is filled with silica-gel, a desiccant material that can be effectively regenerated at temperatures as low as 60-70 °C. The rotor matrix is composed of alternate layers (smooth and wavy) of silica-gel sheets and metallic silicate, chemically bound into an inorganic fiber frame.

The so realized "honeycomb" frame has several advantages, such as the maximization of the superficial contact area, low pressure drops, low weight but high structural durability.

60% of the rotor area is crossed by the process air, while the remaining 40% by the regeneration air. This configuration is often used when low regeneration temperature are used.

The weight of the DW is 50 kg; its diameter and thickness are 700 and 200 mm, respectively.

Several experimental tests have been carried out with the actual test facility configuration [1, 2, 4, 5, 6]; future research activity aims to investigate the technical feasibility of a solar collector system that provides regeneration thermal energy replacing the MCHP.

The future layout of the solar assisted desiccant based AHU in shown in Figure 1.

The only difference with the current test facility configuration is the presence of the solar collector system and the storage tank; in particular, the latter is used to store thermal energy provided by both the solar collectors and the boiler; then, when there is a demand for cooling energy for space conditioning, thermal energy is taken from the tank in order to regenerate the desiccant wheel.

As regards the AHU, three outdoor air streams (t_1, ω_1) are processed:

- *process air*, dehumidified by the desiccant wheel (1-2), pre-cooled by the cooling air stream in an air-to-air cross flow heat exchanger (HEX, 2-3) and finally cooled to the supply temperature by a cooling coil (CC) interacting with the chiller (3-4);
- *regeneration air*, heated by the heating coil interacting with the storage tank (1-6), in order to regenerate the desiccant material (6-7);
- *cooling air*, cooled by a Direct Evaporative Cooler (DEC, 1-8) and then used to pre-cool the process air (8-9).

As regards the solar collector system, it has been designed, through a commercial software, considering the typical operating requirements of the regeneration coil, as reported in Table 1.

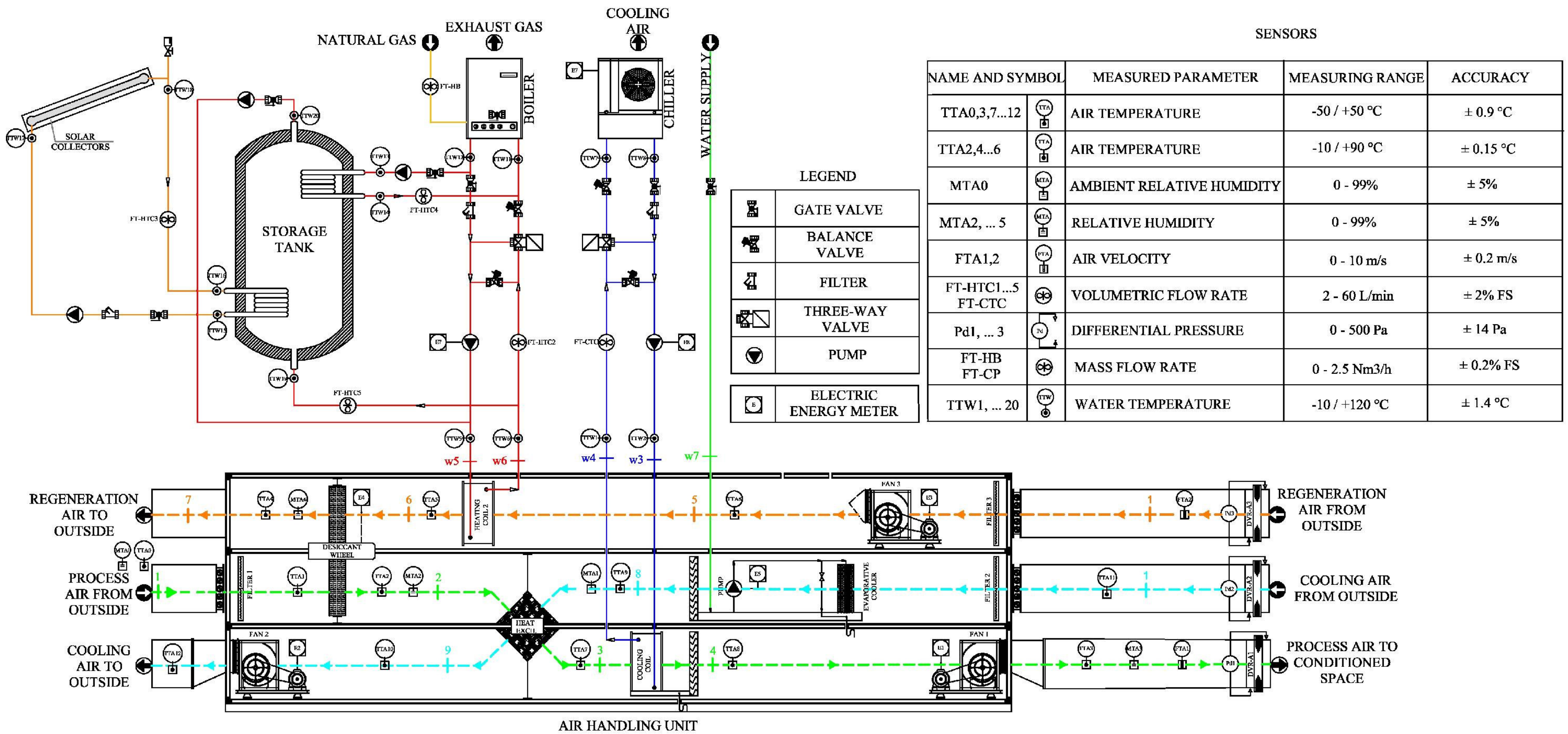
Regeneration thermal power [kW]	12.0
Regeneration temperature [°C]	65.0
Regeneration mass flow rate [kg/s]	0.262
Outdoor air temperature [°C]	25.0

Table 1 Typical operating requirements of the regeneration coil

Considering the quite low needed temperature, flat plate collectors have been chosen; the characteristic of the solar field and collectors have been summarized in Table 2.

Number of collectors	7
Collectors area [m^2]	14
Azimuth [°]	0
Slope [°]	20

Table 2 Characteristics of the solar field and collectors



SENSORS

NAME AND SYMBOL	MEASURED PARAMETER	MEASURING RANGE	ACCURACY
TTA0,3,7...12	AIR TEMPERATURE	-50 / +50 °C	± 0.9 °C
TTA2,4...6	AIR TEMPERATURE	-10 / +90 °C	± 0.15 °C
MTA0	AMBIENT RELATIVE HUMIDITY	0 - 99%	± 5%
MTA2, ... 5	RELATIVE HUMIDITY	0 - 99%	± 5%
FTA1,2	AIR VELOCITY	0 - 10 m/s	± 0.2 m/s
FT-HTC1...5 FT-CTC	VOLUMETRIC FLOW RATE	2 - 60 L/min	± 2% FS
Pd1, ... 3	DIFFERENTIAL PRESSURE	0 - 500 Pa	± 14 Pa
FT-HB FT-CP	MASS FLOW RATE	0 - 2.5 Nm3/h	± 0.2% FS
TTW1, ... 20	WATER TEMPERATURE	-10 / +120 °C	± 1.4 °C

Figure 1 Layout of the solar assisted desiccant-based AHU

Table 3 Main models used for the simulation and their main parameters

Component	Type no.	Library	Main parameters	Value	Unit
Desiccant Wheel	1716a	TESS	F1 Effectiveness	0.207	-
			F2 Effectiveness	0.717	-
Solar Collectors	1b	Standard	Tested Flow Rate	0.0306	kg/(sm^2)
			Intercept Efficiency	0.712	-
			Efficiency Slope	3.56	W/(m^{2}K)
			Efficiency Curvature	0.0086	W/(m^{2}K^2)
Thermal Storage	60d	Standard	Tank Volume	0.855	m^3
			Tank Height	2.18	m
			Tank Loss Coefficient	2.30	W/(m^{2}K)
			Number of Internal Heat Exchanger	2	-
Cross Flow Heat Exchanger	91	Standard	Effectiveness	0.446	-
DEC	506c	TESS	Parasitic Power	40 W	W
			Saturation Efficiency	0.551	-
Cooling Coil	508f	TESS	Coil Bypass Fraction	0.177	-
Heating Coil	670	TESS	Effectiveness	0.868	-
Process Air Fan	744	TESS	Rated Flow Rate	0.226	kg/s
			Rated Power	310	W
			Power Coefficients	Eq. 6	-
			Motor Efficiency	0.80	-
Boiler	6	Standard	Maximum Heating Rate	26.7	kW
			Efficiency	0.902	-
Air-Cooled Chiller	655	TESS	Rated Capacity	8.45	kW
			Rated COP	2.93	-
			Performance Data	Manufacturer data	-

Moreover, it is assumed that the collectors are positioned on the roof of the four-storey building that hosts the test facility.

TRNSYS SIMULATION

In order to simulate the solar assisted desiccant-based AHU, a simulation model was built by using TRNSYS 17 [7] and the component library TESS [8].

A list of the main models used in the simulations is reported in Table 3.

In particular, for the Desiccant Wheel model, TRNSYS uses the simplified approach of Maclaine-Cross and Banks [9]. This approach models the dehumidification process, a combined heat and mass transfer process, in analogy with a simple heat transfer process. Equations for coupled heat and mass transfer are reduced in two uncoupled differential equations of two independent variables called characteristic potentials, F_1 and F_2.

The discussion about the nature of these potentials is quite difficult and it is out of the scope of this work, but it can be stated that constant F_1 lines coincide with constant enthalpy lines, while constant F_2 lines coincide with constant relative humidity lines on the psychrometric chart, [10].

The potential functions depend on thermohygrometric properties of air and on the thermo-physical properties of the rotor, especially the desiccant material, [11].

Jurinak, in [10], has expressed such a relation for the working pair air-silica gel, defining the model presented below:

$$F_{1,i} = \frac{-2865}{(t_i + 273.15)^{1.49}} + 4.344(\omega_i/1000)^{0.8624} \quad (1)$$

$$F_{2,i} = \frac{(t_i + 273.15)^{1.49}}{6360} - 1.127(\omega_i/1000)^{0.07969} \quad (2)$$

The intersection of constant potential lines gives the outlet conditions of process air in the ideal case, i.e. assuming that both the adsorption and the desorption process are isenthalpical.

Then actual outlet conditions are estimated using two effectiveness indices of the wheel, η_{F_1} and η_{F_2}, calculated in analogy to the efficiency of a heat exchanger:

$$\eta_{F_1} = \frac{F_{1,2} - F_{1,1}}{F_{1,6} - F_{1,1}} \quad (3)$$

$$\eta_{F_2} = \frac{F_{2,2} - F_{2,1}}{F_{2,6} - F_{2,1}} \quad (4)$$

In particular, η_{F_1} represents the degree to which the process approximates the adiabatic one, while η_{F_2} represents the degree of dehumidification, [10]. If $\eta_{F_1} = 0$ and $\eta_{F_2} = 1$, the dehumidification process is ideal, i.e. it is adiabatic and there is a maximum dehumidification level for assigned geometry and flow conditions.

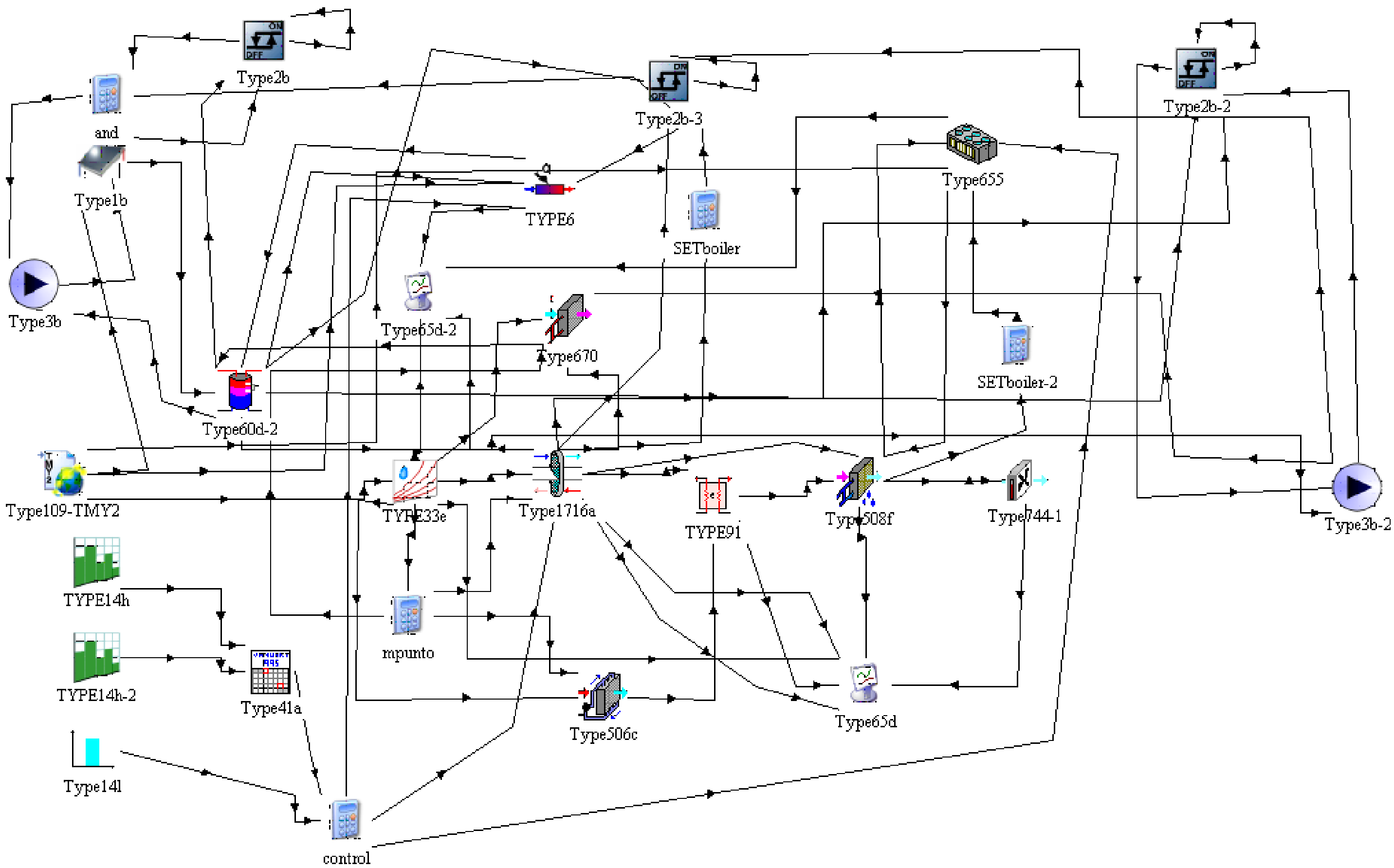


Figure 2 TRNSYS simulation studio project of solar assisted desiccant-based AHU showing components and their connections

If the values of η_{F_1} and η_{F_2} are known, then temperature and humidity ratio of the processed air exiting the wheel can be evaluated.

The model of the thermal solar collectors is described by the following equation:

$$\eta = \eta_0 - a_1 \frac{(t - t_{amb})}{G} - a_2 \frac{(t - t_{amb})^2}{G} \tag{5}$$

where the parameters η_0 (Intercept Efficiency), a_1 (Efficiency Slope) and a_2 (Efficiency Curvature) are the characteristics coefficients of performance of the flat plate collectors.

The motor drawn by the fan is given by:

$$\frac{P}{P_{rated}} = b_0 + \sum_{i=1}^{6} b_i \left(\dot{m} / \dot{m}_{rated}\right)^i \tag{6}$$

where $b_0 = 8.08$, $b_1 = -68.0$, $b_2 = 253.1$, $b_3 = -488.2$, $b_4 = 518.2$, $b_5 = -287.4$, $b_6 = 65.3$.

The effectiveness values of the Desiccant Wheel, of the Cross Flow Heat Exchanger and of the Heating Coil, the efficiency of the DEC, the Bypass Fraction of the cooling coil and the power coefficients of Eq. 6 have been calibrated by means of over than 200 hours of experimental tests [1, 2, 4, 5, 6].

Manufacturer data have been used for the performance of the solar collectors, of the thermal storage, of the chiller and of the boiler.

A view of the complete simulation studio project is shown in Figure 2.

RESULTS

The simulation was run for a time period of one week, from the 1st to the 7th of August, assuming the Meteonorm climatic conditions for Naples.

Moreover, it was assumed that the AHU provides conditioned air to an office, whose working hours are from 9.00 a.m. to 6 p.m., from Monday to Friday.

It was assumed that the AHU is controlled in order to obtain a supply humidity ratio (point 4 in Figure 1) of 9 g/kg; to this aim, the solar collectors and the boiler interact with the thermal storage in order to supply hot water at the regeneration temperature required by the DW.

Finally, the chiller is controlled to provide chilled water to the cooling coil in order to obtain a supply temperature of 18 °C.

In Figure 3, the temperature of process air at various section of the AHU and the required regeneration temperature are shown, while in Figure 4 humidity ratio of outdoor air and of process and regeneration air exiting the Desiccant Wheel are shown.

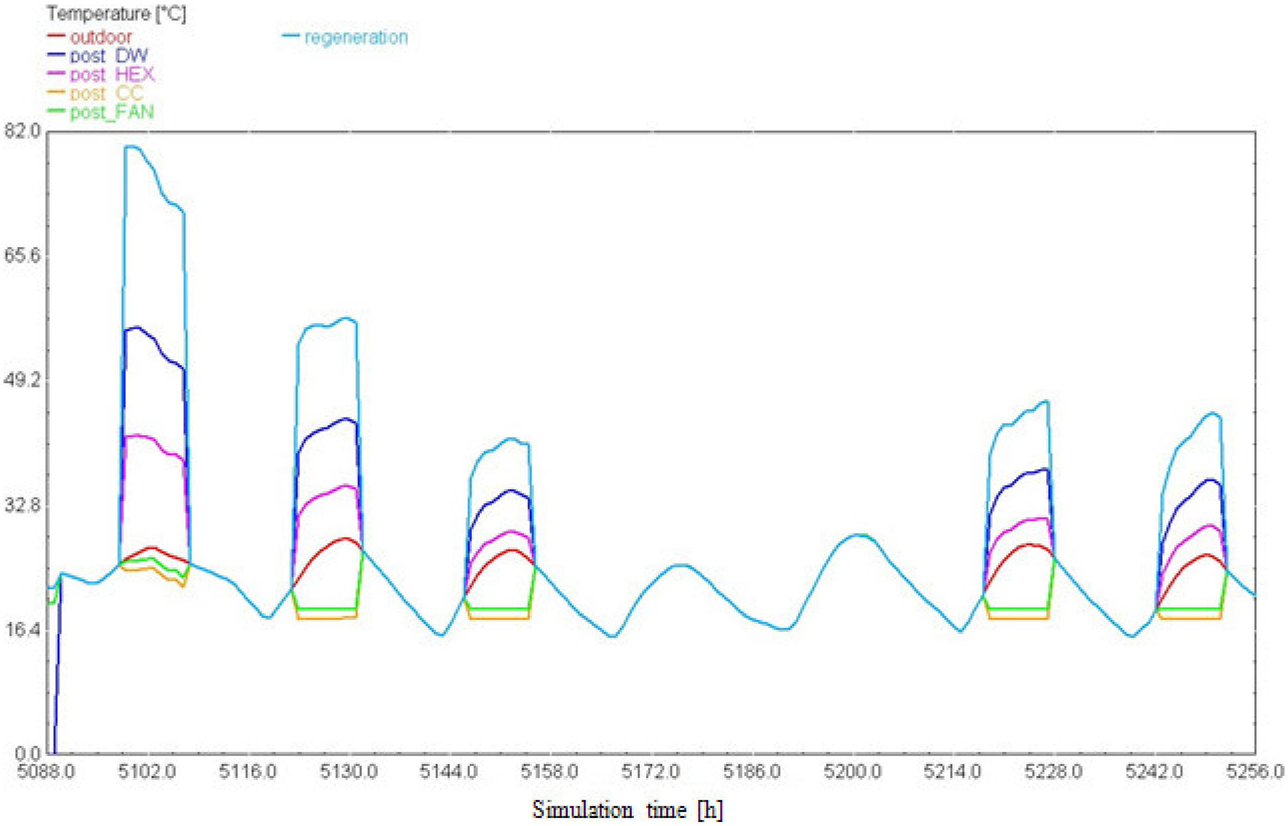


Figure 3 Temperatures of process air at various section of the AHU and required regeneration temperature

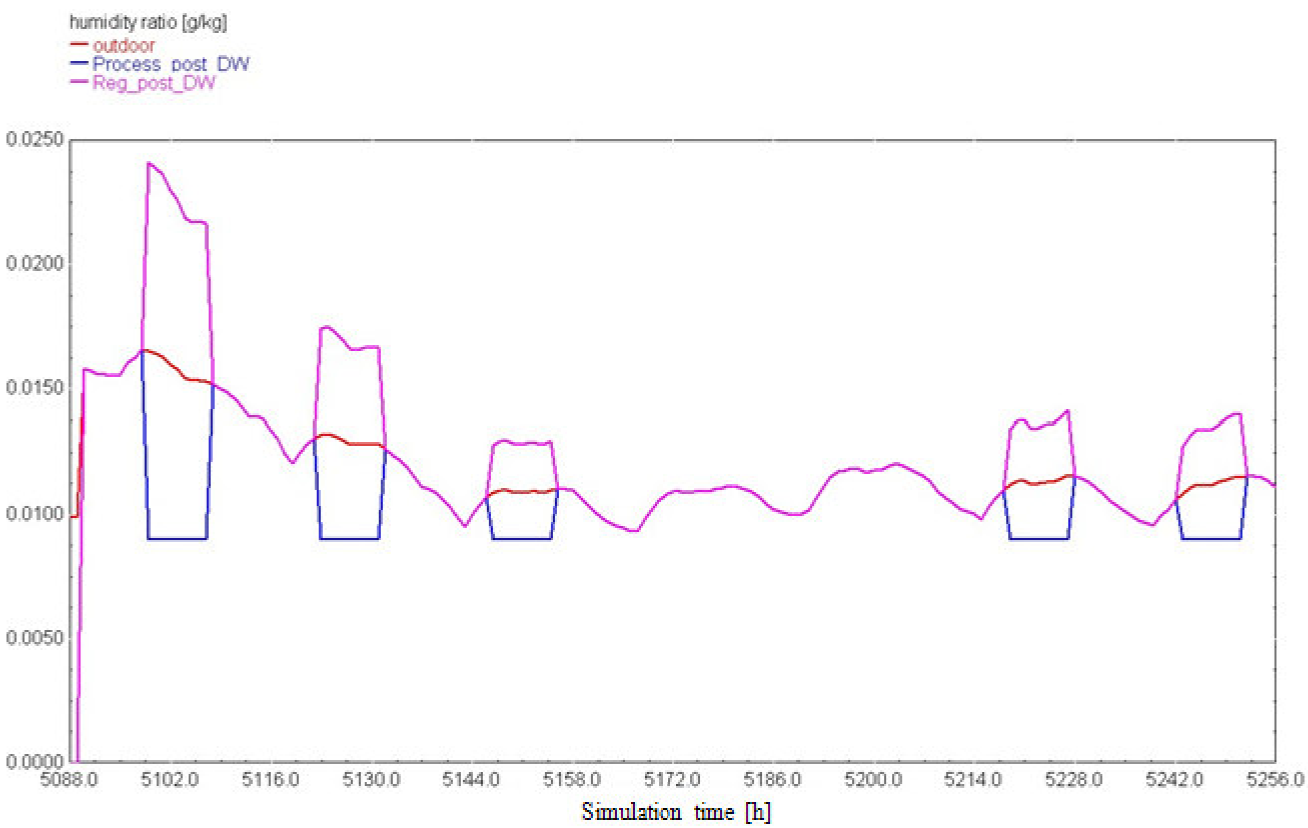


Figure 4 Humidity ratio of outdoor air and of process air and regeneration air exiting the DW

Obviously, the higher the outdoor air humidity ratio (Figure 4), the higher are both the regeneration temperature required to obtain the fixed value of supply humidity and the temperature of process air exiting the DW (Figure 3).

Moreover, during the first day of simulation, the chiller cooling power is less than required and it is not able to guarantee the desired supply temperature; in fact, process air temperature exiting the cooling coil is higher than 18 °C.

Finally, the fan determines an increase of process air temperature of about 1 °C.

TRNSYS simulation also allows to evaluate typical performance parameters of a single component or of the whole system.

For example, the efficiency of the solar collectors can be evaluated. It is defined as, [12]:

$$\eta_{SC} = \frac{\int P_{th,SC} d\tau}{\int G A d\tau} \tag{7}$$

where $P_{th,SC}$ is thermal energy provided by the solar collector system for regeneration, G is the solar radiation, A is collectors area and the integration is carried out for the hours of working of the AHU.

For the week of simulation, the efficiency of the solar collectors is 0.48.

Another critical parameter for solar cooling systems is the Solar Fraction, [13], that shows the contribution of thermal gain from solar collectors within the total heat input for regeneration. It is defined as:

$$SF = \frac{\int P_{th,SC} d\tau}{\int P_{th,SC} d\tau + \int P_{th,B} d\tau} \tag{8}$$

where $P_{th,B}$ is thermal energy for regeneration provided by the boiler. The simulation provides on a weekly basis a SF = 0.603, therefore a large amount of thermal energy for the regeneration of the Desiccant Wheel is effectively provided by the solar collectors.

CONCLUSIONS

In this paper, a test facility located in Benevento (Southern Italy) consisting in a desiccant-based AHU is experimentally described. The AHU contains a silica gel desiccant wheel that is regenerated by means of thermal energy provided by a micro-cogenerator and a boiler. An electric chiller is used to cool the hot and dehumidified process air exiting the wheel.

Future research activity aims to investigate the technical feasibility of a solar collector system that provides regeneration thermal energy replacing the MCHP.

To this aim, a commercial software has been used to design the solar collector system.

Afterwards, the simulation software TRNSYS has been used in order to simulate the operation of the whole system, in which the solar collectors and the boiler interact with a tank where thermal energy for regeneration is stored.

Both Standard and TESS libraries have been used to models the main component of the solar assisted desiccant-based AHU, that provides conditioned air to an office in Naples.

Results show that the solar collectors achieve a reasonable efficiency and can provide a significative amount (about 60%) of the thermal energy required for the regeneration of the desiccant wheel.

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