Comparison of two hygroscopic materials for a solar-assisted desiccant-based air handling unit

C Roselli¹, M Sasso¹ and F Tariello²,*

¹ Department of Engineering, Università degli Studi del Sannio, Piazza Roma 21, 82100 Benevento, Italy
² Department of Medicine and Health Sciences “Vincenzo Tiberio”, Università del Molise, Campobasso, Italy
* Corresponding author e-mail: francesco.tariello@unimol.it

Abstract. Solar-driven air-conditioning systems are an interesting solution to reduce greenhouse gas emissions as they are essentially powered with a renewable energy source. Among the solar heating and cooling systems, desiccant-based air handling units (AHUs) are a very promising alternative as they require low temperature thermal energy for operation in cooling mode. In this paper the experimental plant of University of Sannio (Benevento, Southern Italy), consisting of an AHU with a desiccant wheel using as hygroscopic material silica gel is coupled with solar thermal collectors. This AHU has been also simulated with a different material for the desiccant wheel, the MIL101@GO-6 (MILGO). It is a composite material, consisting of graphite oxide dispersed in the MIL101 metal organic framework network structure, that shows a better dehumidification effectiveness. The energy and environmental performance of a desiccant-based solar heating and cooling system with this new material have been evaluated with respect to the conventional AHU based on cooling dehumidification. Yearly based dynamic simulations of the plant serving a university classroom located in Benevento have been performed. Furthermore, a comparison with the air-conditioning system equipped the currently adopted material (silica gel) has been carried out too. The optimal plant configuration varying collector type (flat plate and evacuated tube), tilt angle and surface has been found.

1. Introduction

The replacement of electric-driven air-conditioning systems with solar activated ones could make an important contributions in terms of buildings primary energy demand saving, GHG emissions reduction and summer peak electric load cut. Desiccant-based air handling units operate with a thermally-driven cycles as they require heat to regenerate the desiccant material. This thermal energy can be supplied by various conventional means like waste heat, gas burner or electric heater. However, the best coupling is with solar energy technologies because solar energy availability and cooling load are approximately in phase. During 2016, in European Union up to 40% of the total energy consumption derives by the building sector, and at the same time also 40% of the total greenhouse gas emissions are released in this ambit [1]. The energy demand for air-conditioning in the residential sector is estimated to be about the 65% of the total energy consumption [2]. In addition to this, it could be observed that cooling energy demand is showing a growing trend both in developing and developed Countries in the last two decades. In such a scenario, it seems interesting to investigate the benefits deriving by solar heating and cooling systems, in particular by those solutions based on AHU equipped with desiccant wheel (DW). In [3] a solar-driven AHU equipped with a silica gel DW and a heat wheel was investigated. A Maisotsenko cycle cooler is also integrated in the AHU to reduce the process air temperature and so to meet the sensible load of the building. The solar field consists of 7.2 m² flat plate solar collectors and 4.7 m² evacuated tube collectors. On the basis of the experimental results the Authors verified a cooling capacity in the range 3.4 – 4.28 kW and a COP variable between 0.65 and 1.17. The dehumidification effectiveness was significantly affected by the inlet air temperature (reduction) and by the inlet air humidity (increase), varying from 29% to 49%. The solar fraction reaches the maximum value of 70%.
The Authors of [4] analysed from an energy, exergy and economic point of view seven different configurations of desiccant-based AHU using TRNSYS software. The air handling units are diversified by:
- the adoption or not of a heat recovery wheel, which allows to transfer thermal energy between the warm exhausted regeneration air and the same regeneration air (pre heating) before it is heated with solar energy;
- the number of dehumidification stages (five solutions provide one stage while two are with two stages);
- the amount of recirculated process air.

The best COP and exergy efficiency take place in Dunckle configuration and ventilation mode. They are 0.6 and 35%, respectively. The heat recovery systems improve the performance of AHUs. From an economic point of view there is 50% in electric energy saving of the Uckan and Dunckle configurations compared to air conditioning heat pumps, leading to a payback period of about 4 years.

In [5], a mathematical model of a desiccant-based AHU is developed. It is used to predict the plant behaviour considering the typical climatic conditions of the Algerian coast and the effect of operating parameters. Temperature and humidity ratio of the process air supplied to the room decrease when the DW rotation speed passes from 20 rpm to 10 rpm, ensuring better performance.

The most widely used material for DW is the silica gel, but many alternative substances have been analysed too. In [6], a synthesized metal silicate DW is adopted in an AHU together with a sensible heat wheel and an electric heat pump. The COP varying air stream flow rates, regeneration temperature and ambient conditions have been experimentally evaluated. The results are used to a TRNSYS model that demonstrates to efficiently simulate the cooling system.

In [7], two alternative desiccant materials (Silica-Gel and Titanium Dioxide) have been compared. A solar-desiccant cooling system is numerically investigated through a validated TRNSYS model in three East Asian climatic conditions (temperate, subtropical and tropical). Titanium Dioxide has proven to be a good alternative material as it can reach lower indoor temperature and humidity ratio with higher cooling performance than the Silica-Gel, considering the same specification of the solar thermal system and desiccant cooling system. The system coefficient of performance (SCOP) is within range 1.5–3 while the solar fraction is between 65% and 90%.

In this work an AHU equipped with a DW filled with an alternative hygroscopic material, named MILGO, is considered and compared with a conventional HVAC system for the air-conditioning service of an university classroom located in Benevento. Flat plate and evacuated tube solar thermal collectors are considered as heat source. The aperture area of the solar field is varied as well as the tilt angle. Furthermore the possibility to recover different percentage of the solar thermal energy surplus for further low temperature uses is assessed too. Finally, the energy, environmental and economic results obtained by the innovative AHU, with respect to the conventional air-conditioning system, are compared with the same results achieved by the plant incorporating a silica gel DW.

2. Plant description
The alternative plant analysed in this paper is a HVAC system mainly driven by solar thermal energy. Such thermal energy is used:
- to regenerate, during the cooling operation (Figure 1), the desiccant wheel (DW) of the air handling unit;
- to realize the pre and post heating processes in the heating period (Figure 2).

More in detail, the alternative system (AS) consists of a solar subsystem and a desiccant-based air handling unit. A solar field, with different collectors configurations, is connected to a 1000 l thermal energy storage tank (TS); to prevent the solar circuit from high temperature level (>100°C) a heat exchanger that dissipates thermal energy surplus or produces hot water for sanitary use or for other purposes (HW-HX) is installed. For the solar field the analyses developed later will consider two solar thermal collectors (SC) types (flat plate and evacuated tube) and three aperture areas: about 20, 27 and 34 m². The solar field layouts are described in Table 1 [8].
The proposed AHU is arranged as the experimental system installed at the University of Sannio laboratory [9]. It is a hybrid desiccant-based air handling unit because air cooling is also controlled with an electric chiller (CH). It has three air channels: one for process air, that is the air supplied to the conditioned space after dehumidification (1-2) and cooling (2-3-4); one for the cooling air, that is outdoor air cooled down by humidification (1-7); the last one for regeneration air, that is outdoor air heated up (1-5) by solar thermal energy or if necessary by the heat supplied by a natural gas fired boiler (B) to regenerate (5-6) the hygroscopic material of the desiccant wheel. The adsorption process realized on the surface of the lower part of the desiccant wheel allows a nearly isenthalpic dehumidification process, (see Figure 1). The DW actually installed has a matrix composed of alternating layers of smooth and corrugated silica gel and metal silicates sheets, chemically incorporated into a support of inorganic fibers. This honeycomb structure maximizes the contact surface with air, reduces the pressure drop and the weight, and increases the structural strength. The wheel has a weight of 50 kg and its dimensions are 700 and 200 mm (diameter and thickness). The frontal area of the wheel exposed to process and regeneration air flows is characterized by a diameter of about 600 mm, because a circular crown of the total area is obstructed by the metallic frame of the wheel cassette. The rated rotation speed is 12 revolutions per hour. Sixty percent of the cross section of the DW is crossed by process air while the remaining 40% by regeneration air.

Hereinafter, it will be considered to replace the silica gel with an alternative hygroscopic material, the MIL101@GO-6 (MILGO), numerically analyzed in a previous paper [10]. It is a composite material, consisting of 6% w/w graphite oxide dispersed in the MIL101 metal organic framework network structure. On the basis of previous evaluations, the material shows a dehumidification efficiency 30% greater than silica gel, so it is a promising substitute of the conventional desiccant material in the solar driven system. The performances of the AS, equipped with a DW filled of this new material, are compared with those of a conventional system (CS). In addition, the energy, environmental and economic indices are compared with the results reported in [8] for Benevento.

In the CS the AHU realizes dehumidification by cooling down the air flow below the dew point (cooling energy is removed by an electric chiller), subsequently it is heated up to supply to the conditioned space air not too cold.

In heating mode the innovative AHU is configured as in Figure 2, it is arranged as in the standard configuration that provides pre-heating (1-2-3), humidification (3-4) and post-heating (4-5), but thermal energy is supplied by the solar subsystem and if not sufficient by the boiler. In the conventional plant the boiler is the only heat source during the winter period. More detailed information about the components rated characteristics are described in the following Table 2.
Table 1. Solar collectors configurations [8].

| Collector Types    | Aperture Area [m²] | Arrangements                                                   | Solar Loop Pump Power [W] |
|--------------------|--------------------|----------------------------------------------------------------|---------------------------|
| Flat-Plate         | 9x2.25=20.3        | 1 row of 4 collectors + 1 row of 5 collectors                 | 200                       |
|                    | 12x2.25=27.0       | 3 rows of 4 collectors                                         | 300                       |
|                    | 15x2.25=33.8       | 3 rows of 5 collectors                                         | 375                       |
|                    | 6x3.43=20.6        | 2 rows of 3 collectors                                         | 200                       |
| Evacuated Tube     | 8x3.43=27.4        | 4 rows of 2 collectors                                         | 300                       |
|                    | 10x3.43=34.3       | 2 rows of 3 collectors + 1 row of 4 collectors                 | 330                       |

The operation of the AHU is determined by the user demand. It is a university classroom located in Benevento (Southern Italy) with a floor surface of 63.5 m² and 30 seats occupied in the weekdays from 9:00 to 18:00. The air conditioning system is switched on at 08:30 in the morning, half an hour before the opening of the classroom, and it is turned off at 18:00 in the afternoon, when the classroom is closed. The indoor air set-point temperatures in winter and summer operation are 20 °C and 26 °C, respectively, while the relative humidity is constantly maintained at 50%. A dead band of 0.5 °C and 10% is assumed in the control strategy respectively for temperature and humidity.

The circulation pump in the solar circuit starts working when the fluid temperature in the solar collectors is higher than that in the TS; thermal energy is taken from the tank to feed the heating coils (HC, HC2) when the AHU is ON, the boiler operates as a back-up system if the temperature of the hot water is not high enough. The cooling coil is fed by the electric chiller during the cooling operation if the process air has not been sufficiently cooled by the cooling air in the cross flow heat exchanger (CF).

Table 2. AS main components rated characteristics.

| Devices   | Rated main characteristics | Values |
|-----------|----------------------------|--------|
| AHU fans  | Volumetric flow rate (m³/h) | 800    |
|           | Power (W)                  | 300    |
|           | Net volume (l)             | 971    |
| TS        | Tank loss coefficient (W/(m²·K)) | 1.37  |
|           | Heating capacity (kW)      | 24.0   |
| B         | Efficiency (%)             | 90.2   |
| CH        | Cooling capacity (kW)      | 8.50   |
|           | COP                        | 2.98   |
The main characteristics of the building envelope are listed in Table 3.

Table 3. Classroom envelope characteristics.

| Opaque Components | Transparent Components |
|-------------------|------------------------|
| Roof              | North                  |
|                   | South                  |
|                   | East/West              |
| U (W/m²K)         | 2.30                   |
| Area (m²)         | 63.5                   |
| g (-)             | -                      |
| External walls (N/S) | 1.11                   |
| External walls (E/W) | 1.11                   |
| On the ground floor | 0.297                  |
| North             | 2.83                   |
| South             | 2.83                   |
| East/West         | 2.83                   |
| East/West         | 63.5                   |
| Area (m²)         | 8.53                   |
| g (-)             | 0.755                  |
| External walls (N/S) | 15.87                  |
| External walls (E/W) | 63.5                   |
| North             | 9.40                   |
| South             | 0.976                  |
| East/West         | 0.976                  |

3. Models and analyses
The software TRNSYS 17 [11] integrated with the “TESS” libraries [12] has been used to dynamically simulate the AS and CS. Simulations with a time-step of 1.5 min have been performed. In TRNSYS the system components, the controls and the building are reproduced by types, each of which implements a mathematical model. In order to simplify the discussion and to concentrate the attention on the DW, only this mathematical models will be described, the others have been already reported in [8], [9] and [13].

3.1 Desiccant Wheel model
The DW with silica gel adsorbent material is modelled with the type 1716 of TESS library. Its mathematical model calculates the performance using the simplified Maclaine-Cross and Banks method [14], that models the dehumidification process, a combination of mass and heat transfer, similarly to the simple thermal energy transfer in a heat exchanger. The coupled equations, that describe the two processes, are reduced to two uncoupled differential equations with two independent variables, called characteristic potentials $F_1$ and $F_2$, [15], [16]. In the psychrometric chart, the isopotential $F_1$ lines approximately represent constant specific enthalpy lines while the isopotential $F_2$ lines approximately describe constant relative humidity curves. The potential functions depend on the thermohygrometric properties of the air and on the thermophysical properties of the adsorbent material [17]. These relations have been expressed for the pair silica gel-air by Jurinak [16], and they are:

\[
F_{1,j} = \frac{-2865}{(t_j+273.15)^{1.49}} + 4.344(\omega_j/1000)^{0.8624}, \quad (1)
\]

\[
F_{2,j} = \frac{(t_j+273.15)^{1.49}}{6360} - 1.127(\omega_j/1000)^{0.07969}, \quad (2)
\]

where the subscript “$j$” refers to the generic thermo-hygrometric condition of the air at which the two potentials are evaluated, whereas $t$ and $\omega$ are the air temperature (°C) and the humidity ratio (g/kg), respectively. The result of the intersection of the isopotential lines are the output conditions of the process air in the ideal case, when both the adsorption and the desorption processes are isoenthalpic. The Jurinak’s model provides that the conditions of real output are estimated using two efficiency indices of the wheel, $\eta_{F1}$ and $\eta_{F2}$, calculated in analogy with the efficiency of a heat exchanger as:

\[
\eta_{F1} = (F_{1,2} - F_{1,1})/(F_{1,5} - F_{1,1}), \quad (3)
\]

\[
\eta_{F2} = (F_{2,2} - F_{2,1})/(F_{2,5} - F_{2,1}), \quad (4)
\]

where potentials $F_1$ and $F_2$ must be evaluated in the states 1, 2 and 5 of Figure 1.

The parameter $\eta_{F1}$ and $\eta_{F2}$ for the silica gel DW of the experimental plant were validated and calibrate in [18] and their values are 0.207 and 0.717 respectively. Instead, for MILGO the two effectiveness parameters are
calculated as the average values obtained by 48 simulated conditions and they are equal to 0.029 and 0.904, respectively [10].

3.2 Energy, environmental and economic analysis

The results of dynamic simulation have been elaborated in order to perform a comparison between the AS and CS. The analyses consist in the evaluation of the:

- primary energy saving (PES) achieved by AS with respect to CS;
- equivalent CO$_2$ emissions avoided by AS with respect to CS;
- simple pay back period (SPB) related to the investment in the new plant.

In addition, these results obtained by the proposed plat with MILGO DW are compared with the same indices assessed for the AS with silica gel based DW and reported in [8].

From the energy point of view the comparison of AS an CS is carried out considering the primary energy requirements related to fossil fuels, in fact the Primary Energy Saving of no-renewable energy sources is:

$$PES = \left(1 - \frac{E_p^{AS}}{E_p^{CS}}\right) \times 100$$

where:

$$E_p^{AS/CS} = \left(\frac{E_{el,CH}^{AS/CS} + E_{el,aux}^{AS/CS}}{\eta_{EG}} + \frac{E_{th,B}^{AS/CS}}{\eta_B}\right)$$

the primary energy of the alternative and conventional system ($E_p^{AS/CS}$) is evaluated taking into account that the energy efficiency of the Italian national electric system ($\eta_{EG}$), including transmission and distribution losses, is 42% [8] and using the boiler efficiency reported in Table 2, 90.2%. In addition, it is assumed that there is no primary energy associated to solar energy because it is a renewable energy source.

To assess the positive effects on the environment of the AS installation, equivalent CO$_2$ emissions of the two systems have been calculated and the equivalent CO$_2$ avoided emissions have been derived:

$$\Delta CO_2 = \left(1 - \frac{ECO_2^{AS}}{ECO_2^{CS}}\right) \times 100$$

where:

$$ECO_2^{AS/CS} = \left(\frac{E_{el,CH}^{AS/CS} + E_{el,aux}^{AS/CS}}{\eta_{EG}} + \frac{E_{th,B}^{AS/CS}}{\eta_B}\right) \cdot \alpha + \frac{E_{th,B}^{AS/CS}}{\eta_B} \cdot \beta$$

$\beta$ is the specific emission factor of primary energy related to natural gas combustion, equal to 0.207 kgCO$_2$/kWh$_{ep}$ [8] and $\alpha$ is the specific emission factor of electricity drawn from the Italian grid, equal to 0.573 kgCO$_2$/kWh$_{el}$ [8].

Concerning the economic analysis, the feasibility of the AS can be estimated by means of the Simple Pay Back (SPB) period:

$$SPB = \frac{EC}{\sum_{k=1}^{N} F_k}$$

where $N$ is the number of years to payback the investment, i.e. the number of years for which the equation is verified, $EC$ is the extra cost of the AS (desiccant-based AHU, storage tank and collectors) with respect to the CS, $F_k$ is the cash flow for the generic year $k$:

$$F_k = OC^{CS} - OC^{AS}_k$$

with $OC^{CS}_k$ and $OC^{AS}_k$ are indicated the operating costs of the AS and CS; the former is given by:

$$OC^{AS}_k = \sum_{i} V_{NG,i} c_{NG,i} + \left(E_{el,CH}^{AS} + E_{el,aux}^{AS}\right) c_{el} + MC_B + MC^{AS}_C + MC^{CS}_C - I_{a,tot}$$

where the following assumptions, according to Italian conditions and specific components used, were considered:

- Lower Heating Value ($LHV$) of natural gas equal to 9.52 kWh/Sm$^3$;
- the total volume of natural gas $V_{NG,tot} = \sum_i V_{NG,i} = \frac{E_{th,B}}{\eta_B LHV}$ should be divided according to the brackets reported in [8], to which different unitary costs ($c_{NG,i}$) are associated;
- unitary cost of electricity ($c_{el}$) equal to 0.211 €/kWh;
- extra cost of desiccant-based AHU with respect to the conventional one equal to 10,000 €, it also includes maintenance costs, as provided by the purchase contracts;
- investment cost of storage tank equal to 3,000 €;
- investment cost of chiller: 3,000 € for the AS, 6,000 € for the CS;
- specific cost of collectors ($c_{SC}$): 360 €/m$^2$ for flat-plate collectors; 600 €/m$^2$ for evacuated collectors;
- maintenance cost of the boiler ($MC_b$) equal to 80 €/y;
- maintenance cost of the chiller in the AS ($MC_{CS} = mc_{col} \cdot S \cdot c_{SC}$) with specific maintenance cost $mc_{col}$ equal to 2% and $S$ gross solar collectors area [19];
- total annual incentive ($I_{a,io}$) evaluated on the basis of the Italian mechanism so-called “Conto Termico”.

4. Results and description
In terms of primary energy saving (PES) the AS shows a certain reduction of the energy demands for all the configurations considered, Figure 3. However, great benefits do not derive from the aperture area growth when flat plate collectors and the new desiccant material is considered, in fact the best PES takes place for 27 m$^2$ of aperture area (Figure 3-a). With evacuated tube collectors greater savings appear with the widest solar field but the second better option is that of 21 m$^2$ (Figure 3-b). On the contrary, with silica gel DW an improvement in the PES always corresponds to the increase of the solar field aperture area (Figure 3-c-d). Evacuated tube collectors with both desiccant materials determine better energy performance, only the configuration with 20° tilt angle, flat plate collectors and MILGO DW shows worse PES than all the solutions with silica gel desiccant wheel.

The reason of the irregular trends, obtained with the new material, can be explained by considering the solar fraction indices:
- Total solar fraction

\[ SF_{Total} = \frac{E_{th}^{TS-DWreg} + E_{th}^{TS-preheat} + E_{th}^{TS-postheat}}{E_{th}^{DWreg} + E_{th}^{preheat} + E_{th}^{postheat}} \]  

(12)

- Cooling mode solar fraction

\[ SF_{Cooling} = \frac{E_{th}^{TS-DWreg}}{E_{th}^{DWreg}} \]  

(13)

- Heating mode solar fraction

\[ SF_{Heating} = \frac{E_{th}^{TS-preheat} + E_{th}^{TS-postheat}}{E_{th}^{preheat} + E_{th}^{postheat}} \]  

(14)

Where $E_{th}^{TS-DWreg}$ is the thermal energy contribution of the solar subsystem to the regeneration of the DW; $E_{th}^{TS-preheat}$ and $E_{th}^{TS-postheat}$ are the thermal energy shares of the solar subsystem respectively to the pre-heating and post-heating phases; $E_{th}^{DWreg}$ is the regeneration energy for the DW; $E_{th}^{preheat}$ and $E_{th}^{postheat}$ are the thermal energy for the pre and post heating processes.

For all the configuration a decreasing trends with the tilt angle are recorded for the SF in cooling mode, whereas a the SFs grow with solar collectors inclination in heating mode (Figure 4). Furthermore, in cooling operation, SFs are always greater than 85% except for the configuration with 20 m$^2$ of flat plate collectors (Figure 4-a): this implies that not significant benefits derive from the bigger collecting surfaces, contrariwise, the solar thermal energy dissipated increases.
Figure 3. Primary energy savings as a function of the tilt angle and the solar field aperture area for: (a) MILGO DW and flat plate collectors, (b) MILGO DW and evacuated tube collectors, (c) silica gel DW and flat plate collectors and (d) silica gel DW and evacuated tube collectors.

Within the range of variation of the tilt angle (20°-55°) greater improvements are observed in the winter solar fractions; instead, the total SFs have a quite flat trends for each solar field surface and collector type, the optimal tilt angle appear at about 46°. In the plants with MILGO DW the maximum mean increase (about 7 percentage points) takes place between 20 and 27 m² (only 5 percentage point between 27 and 34 m²).

Taking into consideration the SFs reached in the plants with silica gel DW (Figure 5) the same trends can be observed (SFs in cooling decrease with tilt angle whereas SFs in heating increase) but wider variations take place in the solar fraction evaluated in cooling mode rather than in heating especially with the solar fields of 20 and 27 m² and with flat plate collectors (Figure 5-a-b-d-e). These results highlight that there is a higher possibility to exploit solar radiation both in heating and in cooling operation when the desiccant wheel filled with silica gel is considered; as expected the new material allows to reduce the regeneration energy in summer and therefore small collector fields are necessary. Bigger surfaces are justified only by the need to better meet the thermal energy demand during the heating operation. Finally, looking at the total solar fractions with MILGO hygroscopic material one can see that, unlike the PES, they increase with the solar field aperture area, this because SF does not consider the electric energy used to run the fan of a dry cooler to dissipate solar thermal energy surplus (it replaces the HW-HX when hot water production is not considered).

As concern the environmental analysis, Figure 6 shows the calculated ΔCO₂. In terms of equivalent CO₂ emissions it is emphasized the condition for which better results take place with smaller collecting surfaces when the air handling unit is equipped with MILGO DW (Figure 6-a-b). Also for this index it is preferable to enlarge the solar field that drives the innovative AHU in its initial configuration (Figure 6-c-d).
Figure 4. Solar fraction for the plant equipped with MILGO DW in cooling operation, heating operation and total considering: (a) 20 m² of flat plat solar collectors, (b) 27 m² of flat plat solar collectors, (c) 34 m² of flat plat solar collectors, (d) 20 m² of evacuated tube solar collectors, (e) 27 m² of evacuated tube solar collectors and (f) 34 m² of evacuated tube collectors.

Despite the new desiccant material achieves better energy (and environmental) results with respect to the silica gel the investment costs are too high and the simple pay back period is much longer that the useful life of the plant. Interesting SPB periods take place only if the solar thermal energy over-productions are considered for other thermal uses at low temperature. The results for the partial (50%) and total surplus exploitation are reported in Table 4 and Table 5.

The use of solar thermal energy extra-production has sensible effects also in improving the PES and ΔCO₂. In Figure 7, Figure 8, Figure 9 and Figure 10 the diagrams of PES and the ΔCO₂ indices when the solar thermal energy surplus is recovered for half or totally, are reported. It is evident that a strong improvement in the exploitation of the renewable energy source happens: the best results are achieved with evacuated tube solar collectors and the largest solar field surface.
Figure 5. Solar fraction for the plant equipped with silica gel DW in cooling operation, heating operation and total considering: (a) 20 m$^2$ of flat plat solar collectors, (b) 27 m$^2$ of flat plat solar collectors, (c) 34 m$^2$ of flat plat solar collectors, (d) 20 m$^2$ of evacuated tube solar collectors, (e) 27 m$^2$ of evacuated tube solar collectors and (f) 34 m$^2$ of evacuated tube collectors.
Figure 6. Equivalent CO₂ emissions avoided as a function of the tilt angle the solar field aperture area for (a) MILGO DW and flat plate collectors, (b) MILGO DW and evacuated tube collectors, (c) silica gel DW and flat plate collectors and (d) silica gel DW and evacuated tube collectors.

Table 4. SPB period for the AS in different configurations when 50% of the solar thermal energy surplus is used for additional low temperature purposes.

| Aperture area (m²) | Flat plate | Evacuated tube |
|-------------------|------------|----------------|
|                   | MILGO (y)  | Silica gel (y) | MILGO (y)  | Silica gel (y) |
| 20                | 15         | 17             | 17         | 16             |
| 27                | 12         | 11             | 14         | 12             |
| 34                | 9          | 8              | 12         | 10             |

Table 5. SPB period for the AS in different configurations when the solar thermal energy surplus is totally used for additional low temperature purposes.

| Aperture area (m²) | Flat plate | Evacuated tube |
|-------------------|------------|----------------|
|                   | MILGO (y)  | Silica gel (y) | MILGO (y)  | Silica gel (y) |
| 20                | 10         | 12             | 10         | 11             |
| 27                | 7          | 8              | 8          | 9              |
| 34                | 5          | 6              | 6          | 7              |
Figure 7. Primary energy savings as a function of the tilt angle and the solar field aperture area when half of the solar thermal energy surplus is used, in the following configuration: (a) MILGO DW and flat plate collectors, (b) MILGO DW and evacuated tube collectors, (c) silica gel DW and flat plate collectors and (d) silica gel DW and evacuated tube collectors.

Figure 8. Primary energy savings as a function of the tilt angle and the solar field aperture area when the solar thermal energy surplus is totally used, in the following configuration: (a) MILGO DW and flat plate collectors, (b) MILGO DW and evacuated tube collectors, (c) silica gel DW and flat plate collectors and (d) silica gel DW and evacuated tube collectors.
Figure 9. Equivalent CO₂ emissions avoided as a function of the tilt angle and the solar field aperture area when half of the solar thermal energy surplus is used, in the following configuration: (a) MILGO DW and flat plate collectors, (b) MILGO DW and evacuated tube collectors, (c) silica gel DW and flat plate collectors and (d) silica gel DW and evacuated tube collectors.

Figure 10. Equivalent CO₂ emissions avoided as a function of the tilt angle and the solar field aperture area when the solar thermal energy surplus is totally used, in the following configuration: (a) MILGO DW and flat plate collectors, (b) MILGO DW and evacuated tube collectors, (c) silica gel DW and flat plate collectors and (d) silica gel DW and evacuated tube collectors.
5. Conclusion
In this paper an innovative solar-driven hybrid AHU equipped with a desiccant wheel is compared with a conventional HVAC system that serves a university classroom located in Benevento (Southern Italy). Two alternative materials for the DW are considered and both show positive results in terms of primary energy saving and equivalent CO₂ emissions reduction. A significant improvement in the performance takes place with the new MILGO material instead of the silica gel, actually used. Furthermore, a parametric analysis was performed considering two solar collector types, three solar field area and finally tilt angles variable in the range 20-55°. The best result is achieved for the MILGO-based AHU: the percent primary energy saving reaches the value of about 23% and 29% with 27 m² of flat plate solar collectors and 34 m² of evacuated tube solar collectors, respectively. As concern the environmental analysis similar considerations can be derived, the maximum percent reduction in the CO₂ equivalent emission are about 21% and 26% with flat plate and evacuated tube collectors but in both cases with 20 m² of aperture area. Simple pay back period less than the useful life of the plant can be achieved only if other low temperature energy uses of the solar thermal energy over-productions are considered; in this case the best pay back periods are 5 and 6 years with 34 m² of flat plate collectors for MILGO and silica gel DW, respectively.

6. Nomenclature
\begin{align*}
c & \text{ Unitary Cost (€/Sm}^3), (€/kWh), (€/m}^2) \\
\text{CO}_2 & \text{ Equivalent CO}_2 \text{ emission (kg/y)} \\
\text{E} & \text{ Energy (MWh/y)} \\
\text{EC} & \text{ Extra Cost (€)} \\
\text{F}_1, \text{F}_2 & \text{ Isopotential lines} \\
\text{F}_k & \text{ Cash Flow (€/y)} \\
\text{I}_a & \text{ Annual Incentive (€/y)} \\
\text{mc} & \text{ Specific Maintenance Cost (%) } \\
\text{MC} & \text{ Maintenance Cost (€/y)} \\
\text{N} & \text{ Number of year (y)} \\
\text{OC} & \text{ Operating Costs (€/y)} \\
\text{PES} & \text{ Primary Energy Saving (%)} \\
\text{S} & \text{ Gross solar collectors area (m}^2) \\
\text{SF} & \text{ Solar Fraction (-)} \\
\text{SPB} & \text{ Simple Pay Back period (y)} \\
\text{t} & \text{ Temperature (°C)} \\
\text{V} & \text{ Volume (Sm}^3) \\
\end{align*}

\textit{Greek symbols}
\begin{align*}
\alpha & \text{ Specific emission factor of electricity drawn from the grid (kgCO}_2/kWh_{el}) \\
\beta & \text{ Specific emission factor of primary energy related to natural gas combustion (kgCO}_2/kWh_{ep}) \\
\Delta\text{CO}_2 & \text{ Equivalent CO}_2 \text{ avoided emission (%)} \\
\eta & \text{ Efficiency (-)} \\
\omega & \text{ Humidity ratio (g/kg)} \\
\end{align*}

\textit{Superscripts}
\begin{align*}
\text{AS} & \text{ Alternative System} \\
\text{CS} & \text{ Conventional System} \\
\text{postheat} & \text{ Post-heating phase} \\
\text{PP} & \text{ Power Plant} \\
\text{Preheat} & \text{ Pre-heating phase} \\
\text{TS} & \text{ Thermal Storage} \\
\text{DWreg} & \text{ Desiccant Wheel regeneration} \\
\end{align*}

\textit{Subscripts}
\begin{align*}
\text{aux} & \text{ Auxiliaries} \\
\text{B} & \text{ Boiler} \\
\text{CH} & \text{ Chiller} \\
\text{Cooling} & \text{ Cooling mode} \\
\end{align*}
EG Electric Grid
el Electric
F₁, F₂ Isopotential lines
Heating Heating mode
NG Natural Gas
p Primary
SC Solar thermal Collector
th Thermal
tot, Total Total

Acronyms

AS Alternative System
AHU Air Handling Unit
B Boiler
CC Cooling Coil
CF Cross-Flow heat exchanger
CH Chiller
COP Coefficient Of Performance
CS Conventional System
DW Desiccant Wheel
EC Evaporative Cooler
GHG Greenhouse Gas
HC, HC2 Heating Coils
HVAC Heating, Ventilation and Air Conditioning
HW-HX Hot Water Heat exchanger
LHV Lower Heating Value
MILGO Hygroscopic material, consisting graphite oxide dispersed in the MIL101 metal organic framework network structure
SCOP System Coefficient Of Performance
SC Solar thermal Collector
SF Solar Fraction
SPB Simple Pay Back period
TS Thermal Storage

7. References

[1] IEA, International Energy Agency. Energy Efficiency, 2017. Available online: http://www.iea.org/publications/freepublications/publication/Energy_Efficiency_2017.pdf [last access: 5 June 2018].

[2] Eurostat. Energy balance, January 2017. Available online: http://ec.europa.eu/eurostat/web/energy/data/energy-balances. [last access: 05 June 2018].

[3] Chaudhary G Q, Ali M, Sheikh N A, Gilani S I H and Khushnood S, Integration of solar assisted solid desiccant cooling system with efficient evaporative cooling technique for separate load handling Appl Therm Eng 140 pp 696–706.

[4] Abbassi Y, Baniasadi E and Ahmadikia H, Comparative performance analysis of different solar desiccant dehumidification systems 2017 Energ Buildings 150 pp 37–51.

[5] Merabti L, Merzouk M, Merzouk N K and Taane W, Performance study of solar driven solid desiccant cooling system under Algerian coastal climate 2017 Int J Hydrogen Energ 42 pp 28997–29005.

[6] Jani D B, Mishra M and Sahoo P K, Performance analysis of a solid desiccant assisted hybrid space cooling system using TRNSYS 2018 J Building Engineering 19 pp 26–35.

[7] Enteria N, Yoshino H, Mochida A, Satake A, Yoshie R, Takaki R, Yonekura H, Mitamura T and Tanaka Y, Performance of solar-desiccant cooling system with silica–gel (SiO2) and titanium dioxide (TiO2) desiccant wheel applied in East Asian climates 2012 Sol Energy 86 pp 1261–1279.

[8] Angrisani G, Roselli C, Sasso M and Tariello F, Dynamic performance assessment of a solar-assisted desiccant-based air handling unit in two Italian cities 2016 Energ Convers Manage 113 pp 331–345.
[9] Angrisani G, Roselli C, Sasso M and Tariello F, Dynamic performance assessment of a micro-trigeneration system with a desiccant-based air handling unit in Southern Italy climatic conditions 2014 Energ Convers Manage 80 pp 188–201.

[10] Bareschino P, Diglio G, Pepe F, Angrisani G, Roselli C and Sasso M, Numerical study of a MIL101 metal organic framework based desiccant cooling system for air conditioning application 2017 Appl Therm Eng 124 pp 641–651.

[11] Solar Energy Laboratory, TRNSYS 17, a TRaNsient System Simulation program 2010 University of Wisconsin, Madison.

[12] T.E.S.S. Component Libraries v.17.01 for TRNSYS v.17.0 and the TRNSYS Simulation Studio, Parameter/Input/Output Reference Manual 2004 Thermal Energy System Specialists, LLC.

[13] Calise C, Dentice d’Accadia M, Roselli C, Sasso M and Tariello F, Desiccant-based AHU interacting with a CPVT collector: Simulation of energy and environmental performance 2014 Sol Energy 103 pp 574–594.

[14] Maclaine-Cross I L, Banks P J., Coupled heat and mass transfer in regenerators – predictions using an analogy with heat transfer 1972 Int J Heat Mass Tran 15 (6) pp 1225–1242.

[15] Howe R R, Model and Performance Characteristics of a Conditioning System Which Utilizes a Rotary Desiccant Dehumidifier 1983 MS thesis, University of Wisconsin, Madison, USA.

[16] Jurinak J J, Open cycle solid desiccant cooling: component models and system Simulations 1982 PhD thesis, University of Wisconsin, Madison, USA.

[17] Banks P J, Prediction of Heat and Mass Regenerator performance using nonlinear analogy method: part 1 – basis 1985 ASME J Heat Transf 107 pp 222–229.

[18] Angrisani G, Roselli C and Sasso M, Experimental validation of constant efficiency models for the subsystems of an unconventional desiccant-based Air Handling Unit and investigation of its performance 2012 Appl Therm Eng 33–34 pp 100–108.

[19] Sabiha MA, Saidur R, Mekhlife S, Mahian O, Progress and latest developments of evacuated tube solar collectors 2015 Renew Sust Energ Rev 51 pp 1038–1054.