

## PERFORMANCE ASSESSMENT OF A SOLAR ASSISTED DESICCANT COOLING SYSTEM

by

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*The use of solar energy to drive cooling applications, such as air-conditioning is attractive since the cooling load has a high coincidence with the availability of solar irradiation. Combination of solar thermal and cooling has a high potential to reduce the electricity consumption of conventional air-conditioning. This work delivers a description of solar desiccant solid system (DEC) and presents results of tests and performance analysis. The overall cooling efficiency is evaluated using simulation data typical of Mediterranean Region. In this context the autonomous operations both of a solar desiccant system (DEC) and an absorption refrigerant chillers powered by direct-flow vacuum-tube collectors are investigated. It is found out that the DEC system can achieve a primary energy saving of around 40%, compared to an absorption refrigerant and of around 150% compared to a conventional vapour compressor refrigerator.*

Key words: *desiccant cooling, solar energy, energy saving, absorption refrigerant*

### Introduction

Energy demand for cooling and air-conditioning is growing worldwide. The high summer temperatures recorded in southern European countries, also occurred due to climate change taking place, annually contribute to the achievement high-peaks of energy for air conditioners [1].

Moreover, the rising demand for air-conditioning in buildings involves unfavorable fossil fuel consumptions as well as upcoming stability problems in the electricity supply; which in turn requires a costly upgrading of the grids to handle peaks power of electricity demand.

Chillers powered by thermal energy can be an alternative to chillers powered by electricity. Especially, they are an alternative when they are driven by solar thermal energy or waste heat sources [2]. Air-conditioning plants driven by renewable sources can also achieve substantial fossil energy saving. The solar cooling (SC) is an interesting option to coverage the cooling demands of buildings because the supply of solar energy and the demand for cooling are greatest during the same season.

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Different thermally driven cooling technologies exist: desiccant cooling, absorption, and adsorption chillers. They are all sorption cycles where a sorption material, either liquid or solid, acts as a chemical compressor for the refrigerant. They can further be divided into open and closed cycles. The best known open cycle process is the desiccant evaporative cycle (DEC). During the past decades many efforts have been made to develop the DEC [3, 4]. Through different system configuration and integration, DEC can facilitate effective temperature and humidity control for buildings with the most stringent ventilation requirements in a vast domain of climatic conditions [5-7]. This paper aims to analyze the thermodynamic performance of a solar assisted hybrid desiccant cooling system in the context of the South Regions of Italy. The performances of the DEC system were compared, at the same operating conditions, with other two widely used conventional systems: absorption refrigerator (AR) and vapor-compression refrigerator (VCR).

### Material and methods

The dominating technology of thermally driven chillers is based on absorption. The basic physical process consists of at least two chemical components, one of them serving as refrigerant and the other as the sorbent. Adsorption technology is very similar to absorption, the difference is that the sorbing agent is not a liquid but a solid. The sorption material adsorbs the refrigerant, while it releases the refrigerant under heat input. A quasi-continuous operation requires for at least two compartments with sorption material. While closed cycle produce chilled water, which can be supplied to any type of air-conditioning equipment, open cycles produce directly conditioned air.

### Absorption refrigerator system

The two widespread absorption cycles currently in use are the lithium bromide (LiBr) cycle and the ammonia-water ( $\text{NH}_3/\text{H}_2\text{O}$ ) cycle [8-10]. In the former, water acts as the refrigerant and LiBr acts as the absorbent. In the latter, the ammonia water solution is the refrigerant and water is the absorbent. For chilled water above 0 °C, as it is used in air-conditioning, the LiBr cycle tends to be more common. The cooling effect is based on the evaporation of the refrigerant (water) in the evaporator at very low pressure. The vaporised refrigerant is absorbed in the absorber, thereby diluting the  $\text{H}_2\text{O}/\text{LiBr}$  solution. To make the absorption process efficient, the process has to be cooled. The solution is continuously pumped into the generator, where the regeneration of the solution is achieved by applying the driving heat such as from hot water supplied by a solar collector. The refrigerant leaving the generator by this process condenses through the application of cooling water in the condenser and circulates by means of an expansion valve again into the evaporator. The thermal efficiency of the system is defined by the ratio of cold produced in the evaporator and driving heat used in the generator and is called coefficient of performance:

$$COP_{AR} = \frac{Q_L}{Q_G} \quad (1)$$

LiBr machines come as single-effect (single stage) or double-effect (two stage) cycles. The double-effect machines are thermally more efficient but are more complex and require higher grade heat input. Typical characteristics of single-stage absorption chillers are [11-13]:  $COP_{AR}$  of about 0.6 to 0.8; driving temperatures of 75 to 95 °C; heat rejection temperatures of around 27 °C; chilled water temperatures in the range of 7 to 15 °C.

### ***Desiccant evaporative cooling systems***

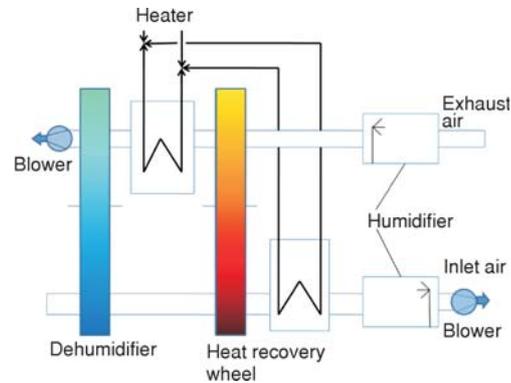
Desiccant evaporative cooling systems produces the cooling effect by evaporative cooling however, the potential of using evaporative cooling is increased due to the dehumidification of air by the desiccant. As desiccants can be either solid or liquid, desiccant air-conditioning systems can be classified into two categories, namely, solid desiccant air-conditioning systems, which consist of fixed bed type and rotary wheel type, and liquid desiccant air-conditioning systems. The standard cycle (fig. 1) which is mostly applied today uses rotating desiccant wheels, equipped either with silica gel or lithium-chloride (LiCl) as sorption material [14].

Desiccant wheel is an air to air heat and mass exchanger, with a relatively low rotation speed. The wheel consists of a matrix made of metal coated with molecular sieves or silica gel or paper impregnated with lithium chloride. Typically, the desiccant is loaded into a rotating tray or impregnated into a honeycomb-form wheel, which rotates slowly between the dry air stream (process) and the heated air stream (reactivation). The wheel is split into the adsorption and desorption sections and the area of the two sections are not necessarily equal. Water vapor of the process air is adsorbed and stored in the matrix. The rotation of the wheel causes periodic reactivation of the adsorption part. The desiccant wheel performance and the dehumidification achieved is influenced by several operating parameters such as rotation speed, regeneration temperature, volumetric air-flow rate and inlet process air humidity and temperature [15, 16]. DEC are quite efficient in dealing with the latent load, but considerably less so with regard to the sensible load [17], so they are advantageous particularly for use in hot and humid climates.

There are two possible options to overcome such limit:

- a combination with a conventional vapour compression chiller. This combination allows the removal of the latent load mainly with a thermal driven sorption wheel and removal of the sensible load mainly with the chiller. Since the chiller works at higher evaporator temperatures, if dehumidification is done by sorption, it works with higher efficiency, and
- to improve the performance of the desiccant wheel and make the process air sufficient dry so that the sensible heat can be removed entirely by means of evaporative cooling.

Desiccant materials have been playing a crucial role in the development of desiccant air-conditioning. The characteristics of the desiccant material being utilized impact the performance of the desiccant air-conditioning systems significantly [18]. The widely used desiccant materials include activated carbon, activated alumina, molecular sieve, silica gel, lithium chloride, calcium chloride, natural and synthetic zeolites, and synthetic polymers. The differences in the desiccant properties are clearly defined by the static adsorption isotherm curves, a measure of the desiccant's ability to adsorb moisture under constant static conditions. In general, the adsorbent used in desiccant wheel should have both high hygroscopic capacity and saturated adsorption rate.



**Figure 1. Scheme of desiccant cooling systems**

Two key principles for selecting appropriate desiccant materials are: the desiccant materials should possess large saturated adsorption amount and can be reactivated easily; the adsorption performance of the desiccant materials should approach the Type 1M material [19], which represents the optimum shape for air-conditioning application. Recent investigations on solid desiccant generally consists of four aspects, namely, modification of conventional desiccant [20, 21], natural rock-based desiccant [22], bio-desiccant [23] and composite desiccant [24-26]. For rotary desiccant dehumidification, researchers are under way to find desiccant materials that approach the type 1M material in its sorption performance [19], and composites formed by confining salt to porous host adsorbent have been identified to be an effective way [27].

Many research on silica gel-based composite desiccants have been performed. New technology, close to market introduction, are desiccant cooling systems using a liquid water-lithium chloride solution as sorption material. This technology is a promising option for a further increase in exploitation of solar thermal systems for air-conditioning [28].

The desiccant can be regenerated with various low grade thermal energy resources, such as solar energy, waste heat, extra heat produced by existing processes, *i. e.* heat coming from combined heating and power (CHP) process, thus resulting in a combined cooling, heating and power (CCHP) system. Several studies show that desiccant cooling system have a limited dehumidification potential for given features of the desiccant rotor, the regeneration temperature, the supply air flow rates and so on [5]. The dehumidification process has two purposes: dehumidification in order to match indoor comfort criteria and an “extra” dehumidification in order to allow for subsequent humidification to produce the cooling effect. Therefore an auxiliary cooling power for dehumidification is required to fulfill the desired supply air conditions.

#### *DEC performances indexes*

The following energy performance indexes are usually evaluated:

- the cooling energy fraction  $SF_{DEC}$  covered by the desiccant cycle to the total cooling energy ( $Q_{AHU}$ ) delivered by the air handling unit (AHU) [29], calculated by eq. (2),
- the solar heat efficiency  $\eta_{sh}$ , that is the ratio between the useful energy output from the HC and the total incident solar energy on the surface of the solar collectors irradiation, calculated by eq. (3),
- the solar cooling  $COP_{DEC}$  that is the ratio between the useful cooling output of the desiccant cycle and the regeneration heat delivered by the solar heating coil (HC), calculated by eq. (4),
- the primary energy  $PE$  consumption calculated by eq. (5), and
- the primary energy ratio  $PER$  calculated by the formula (6):

$$SF_{DEC} = 1 - \frac{(Q_{CC})_{VCR}}{Q_{AHU}} \quad (2)$$

$$\eta_{SH} = \frac{\sum Q_{HC}}{\sum I_{\beta} A_c} \quad (3)$$

$$COP_{DEC} = \frac{Q_{DEC}}{Q_{HC}} \quad (4)$$

$$PE = \frac{Q_{el}}{\eta_{el}} + \frac{Q_{reheating}}{\eta_{fossil}} \quad (5)$$

$$PER = \frac{Q_{AHU}}{PE} \quad (6)$$

The  $COP_{DEC}$  of a desiccant cooling system depends strongly on the conditions of ambient air and supply air. The  $PER$  is a benchmark for the energy efficiency.

### Case study

Analyzed AC consists of an AHU equipped with a hybrid desiccant cooling that integrates desiccant, evaporative and conventional cooling technologies. The desiccant is regenerated by solar thermal energy produced by vacuum tube collectors. The choice of the type of collector depends by the selected cooling technology and the site weather climatic conditions.

The proposed configuration of the DEC includes an auxiliary cooling coil (fig. 2), so if the humidity ratio and/or temperature set point of supply air is not reached, then further decrease of dehumidification and temperature is achieved by means the auxiliary cooling coil powered by a vapour compressor refrigerator.

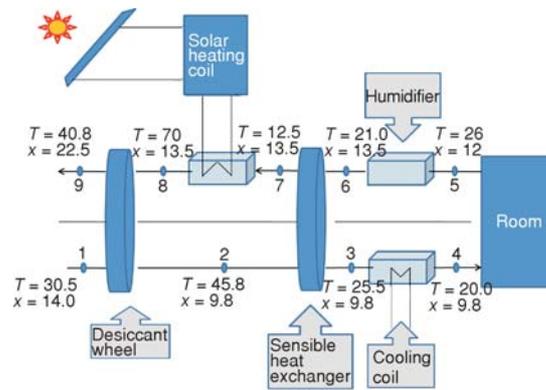


Figure 2. DEC system configuration

### Design conditions

The performance of the air-conditioning system has been evaluated for the climatic conditions of Catania city (longitude 37°15'; latitude 15°04'), which is placed in the southern part of Italy (Sicily).

The given design conditions are the following:

- temperature of indoor space 26 °C, humidity ratio of 50%,
- sensible heat factor of the indoor space load of 0.7,
- operating time from 9,00 a.m. to 6,00 p. m.,
- supply air ( $m_{SA}$ ):  $T_{SA} = 20$  °C and  $x_{SA} = 9.8$  g/kg of dry air,
- return air ( $m_{RA}$ ):  $T_{RA} = 26$  °C and  $x_{RA} = 12$  g/kg of dry air,
- outside air:  $T_E = 30.5$  °C and  $x_E = 14$  g/kg (state 1), and
- the heat recovery efficiency (0.85) and the humidifier efficiency (0.90) are assumed constant.

Thermodynamic transformations are depicted in fig. 2:

- the ambient air, point 1, flows through a rotary desiccant wheel and becomes hot and dry, point 2,
- from point 2 the air flows through a sensible heat exchanger, where it is cooled, point 3,
- from point 3, the dry and cooler air flows through the cooling coil, where it is cooled up to the request design temperature, point 4,
- the exhaust air from the point 5, flows through the humidifier, where it is cooled to point 6,
- from the point 6 the air flows through the sensible heat exchanger, where it exchanges heat with processed air point 7,

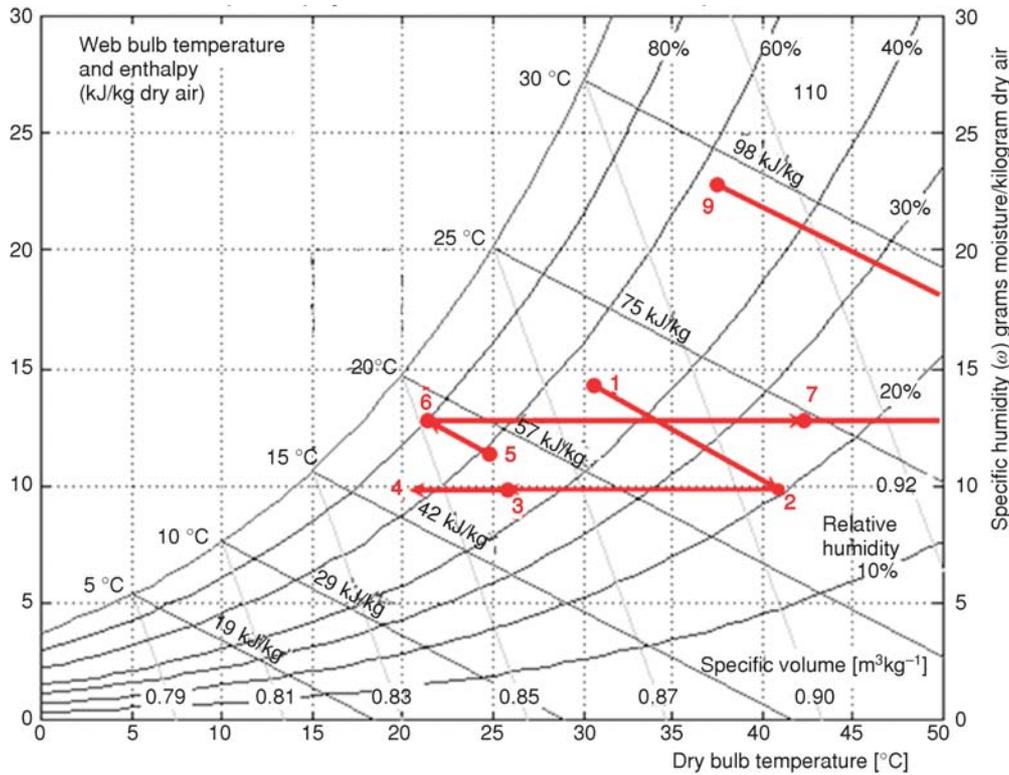


Figure 3. Thermodynamic transformations

- the exhaust air from the heat exchanger, point 7, flows through the heating coil to elevate its temperature to point 8, and
- the hot exhaust air is used to regenerate the desiccant dehumidifier.

Figure 3 shows the thermodynamic transformations in the psychrometric chart. It can be noted that no additional dehumidification is required to reach the required humidity ratio, thanks to the low humidity ratio of the outside air (about 14 g/kg). The final supply air temperature is reached by means of the auxiliary cooling coil (CC).

The heat flow rates for processes of: regeneration  $Q_{HC}$  (7-8), desiccant cooling  $Q_{DEC}$  (2-3), and auxiliary cooling  $Q_{CC}$  (3-4), have been calculated with the following relations.

$$Q_{HC} = m_{RA}(h_8 - h_7) \quad (7)$$

$$Q_{DEC} = m_{SA}(h_3 - h_2) \quad (8)$$

$$(Q_{CC})_{VCR} = m_{SA}(h_4 - h_3) \quad (9)$$

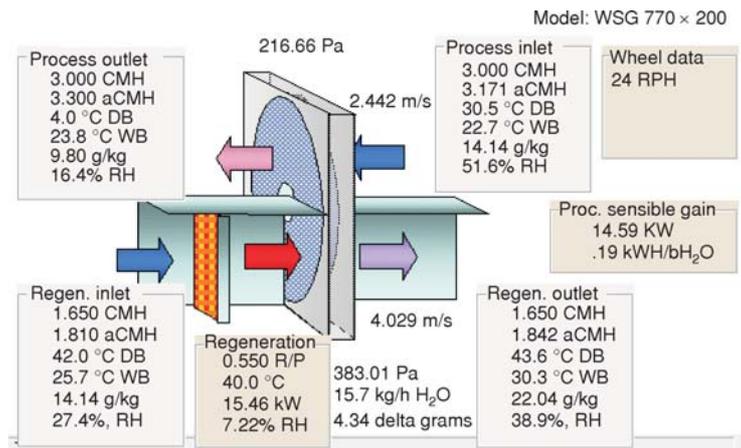
The post-cooling process requires a temperature source of the cold fluid of about 12 °C, that is higher than the temperature required to obtain the process of dehumidification for a conventional AHU. So, it is possible to obtain high COP of the vapour compression chiller. The cycle does not require post-heat process, which is another energetic advantage of the DEC system.

This system has been studied considering three volumetric flow rates typical of small size air-conditioning system (AC): 1500, 2000, and 3000 m<sup>3</sup>/h. The heat flow rates,  $Q_{HC}$ ,  $Q_{DEC}$ , and  $Q_{CC}$  calculated by means eqs. 7, 8, and 9 are reported in tab. 1.

**Table 1. Desiccant wheel and heat flow rates in function of supply air**

Supply air $V_{SA}$ [m <sup>3</sup> h <sup>-1</sup> ]	Return air $V_{RA}$ [m <sup>3</sup> h <sup>-1</sup> ]	Desiccant wheel dimension [mm]	$Q_{HC}$ [kW]	$Q_{CC}$ [kW]	$Q_{DEC}$ [kW]
1500	750	500 × 200	7.1	3.02	7.39
2000	1000	770 × 200	10.2	4.02	9.57
3000	1500	770 × 200	15.4	6.03	14.36

The desiccant wheels have been chosen in function of the design data previously defined using the manufacturer's performance software (see fig. 4) [30].



**Figure 4. Desiccant wheel data and thermodynamic state**

The chosen commercial desiccant system operates as:

- desiccant wheel with equally split the process and regeneration air streams (50/50),
- wheel speed rotation varies from 18 to 24 rph, that is the optimum or near-optimum speed,
- the process air velocity is maintained between 2.8 m/s and 3.0 m/s at standard conditions (15 °C and 101.039 kPa), and
- the desiccant is assumed to be regenerated at 70 °C, which representing the regeneration air temperature downstream of the heating coil.

### **Solar energy supply**

The AC plant utilizes a hot water storage tank to balance the heat furnished by solar collector and the heat supplied to the heating coil (HC). A thermal level higher than 70 °C has to

maintained to fulfil the regeneration of the desiccant rotor. The temperature of water within the storage tank has been calculated by means the hourly energy balance, eq. (10):

$$\frac{mC_{\text{H}_2\text{O}}(T_{i+1} - T_i)}{\Delta\tau} = Q_{\text{SS}} - Q_{\text{D}} - Q_{\text{HC}} \quad (10)$$

$$Q_{\text{SS}} = I_{\beta}\eta_{\text{coll}}h_{\text{HS}} \quad (11)$$

The efficiency of the evacuated heat-pipe solar collectors has been calculated utilising by [31]:

$$\eta_{\text{coll}} = 0.82 - 2.19 \frac{T_{\text{m}} - T_{\text{a}}}{I_{\beta}} \quad (12)$$

Solar radiation ( $I_{\beta}$ ) and the outside temperature have been carried out from the database of the energy plus [32]. The solar collectors area ( $A_{\text{ST}}$ ) has been calculated according to the formula provided by Henning [33]:

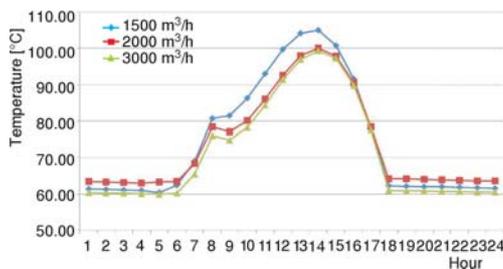
$$A_{\text{ST}} = \frac{Q_{\text{HC}}}{I_{\beta}COP_{\text{DEC}}\eta_{\text{coll}}} \quad (13)$$

**Table 2. Area of solar collector and mass of storage water tank**

$V_{\text{SA}}$ [ $\text{m}^3\text{h}^{-1}$ ]	1500	2000	3000
Area of solar collector $A_{\text{ST}}$ [ $\text{m}^2$ ]	15	22	32
Mass of storage water tank [kg]	400	550	800

Average solar radiation value ( $I_{\beta}$ ) has been set equal to  $0.8 \text{ kW/m}^2$ , the  $COP_{\text{DEC}}$  is assumed equal to 0.90, and efficiency of the evacuated solar collectors has been set equal to 0.50.

The capacity of storage tank has been calculated considering a volume of about 25 litres of water per square meter of solar collector. Table 2 reports the surface of solar collectors and the volume of storage tank foreseeing for the three AC size.



**Figure 5. Thermal storage temperatures**

The chosen solar collector areas guarantee that the storage tank could supply hot water to the DEC system for about 8.5-9.0 hours, which is, more or less, the daily period of functioning of AC plants.

Therefore under the specific assumption, it is possible to assert the solar collector areas, reported in tab. 2, are sufficient to satisfy the need of thermal energy of DEC system without auxiliary source of heating. It has also be noted a constant ratio of about 2.0 between the volumetric air flow and the regeneration heat power ( $Q_{\text{HC}}$ ) provided through solar collectors. This result means that are necessary about  $1.00 \text{ m}^2$  of solar collectors per  $100 \text{ m}^3/\text{h}$  of supply air.

As well-known solar plants are usually "assisted" by a conventional energy source that provides an aliquot of the requested energy ( $Q_{\text{AUX}}$ ) during the peak period, so it is possible reduce the area of solar collectors requested. Consequently, it can be defined the  $SF$  as the ratio of

Using the previous design conditions the daily temperatures in the storage tank have been calculated by eq. (10) (see fig. 5). DEC system runs when the temperature in the upper part of the storage water tank exceed  $70 \text{ }^\circ\text{C}$ , this thermal level is sufficient to feed the HC and regenerate the desiccant material.

The calculated temperatures are always higher than the request regeneration temperature, within the forecasted operating time (from 8:00 to 18:00), during sunny days. So the chosen solar collector areas guarantee that the storage tank could supply hot water to the DEC system for about 8.5-9.0 hours, which is, more or less, the daily period of functioning of AC plants.

the solar power to the total thermal power which, in turn, is given by the sum of the thermal power provided to the DEC system by solar collectors and auxiliary sources:

$$SF = \frac{Q_{ss}}{Q_{ss} + Q_{AUX}} \quad (14)$$

With the aim to consider different fractions of the heating power covered by solar thermal energy the  $SF$  has been calculated varying the area of the solar collectors ( $A_{Si}$ ). The area of solar collectors, reported in tab. 1, provides a value of  $SF = 1$ . In fig. 6 are depicted the values of  $SF$  varying the ratio ( $A_{Si}/A_{ST}$ ).

This analysis shows that there is proportionality between the area of solar collector area and the solar fraction. It is possible to assert that for areas of solar collector higher than 0.75 AST the values of  $SF$  increase swiftly up to 1. Thereby, it is possible to assert that for  $A_{Si}/A_{ST} > 0.75$  DEC system requires very low amount of energy provided by auxiliary source.

#### Energy system performance

The energy performance indices have been calculated under the assumption:

- the DEC systems are equipped with the solar collector area reported in tab. 2,
- the electric energy power necessary for the fan operations has been calculated considering a pressure drop of 750 Pa, along the process side, and of 850 Pa, along the regeneration side,
- the energy power supplied to the cooling coil ( $Q_{CC}$ )<sub>VCR</sub> has been calculated considering the use of a vapour compressor refrigeration with a coefficient of performance  $COP_{VCR} = 3.0$ ,
- the working day period of the DEC system, in accordance with the simulation results, goes from 9:00 to 18:00, and
- for the efficiency of thermoelectric plants  $\eta_{EL}$ , the value of 0.37 has been used, which is the current efficiency of the Italian thermoelectric plants.

Figure 7 shows the performance indices of the DEC systems powered by the solar collector area reported in tab. 2, for the given operative condition. It is possible to notice that:

- the system is able to supply about 70% of the total cooling required,
- the efficiency of solar collectors maintains values more than 60%, and
- a primary energy ratio of 2.0 kWh per kWhPE of primary energy supplied.

Under the chosen design conditions the achieved  $COP_{DEC}$  is about 1.0 and the cooling power lies in the range of about 4,5-5 kW per 1000 m<sup>3</sup>/h of supply air. These results indicate that reducing the regeneration air flow (50% of the supply air) without a significant reduction in the dehumidification efficiency, enabling desiccant cooling systems to run with high  $COP_{DEC}$ . Globally these results confirm that the solar energy is an excellent, practical heat source for desiccant regeneration.

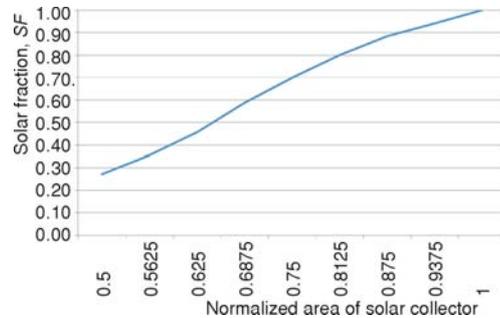


Figure 6.  $SF$  vs. normalised area of solar collector

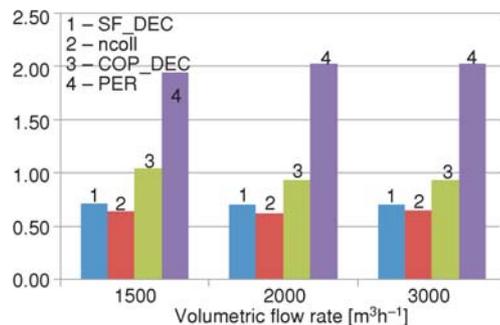


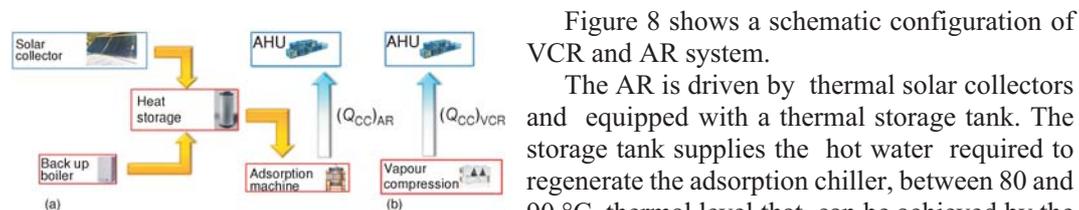
Figure 7. Performances indexes of DEC

### Cooling technologies comparison

The energy performances of the DEC system have been compared with a conventional AHU, for which the input power of the chilled water cooling coil is supplied by: a vapour compression refrigerator system (VCR), or by a single effect absorption refrigerator system.

The performance of the vapor compressor refrigerators is expressed in terms of coefficient of performance ( $COP_{VCR}$ ), defined as:

$$COP_{VCR} = \frac{Q_L}{W_i} \quad (14)$$



**Figure 8. (a) AR configuration; (b) VCR configuration**

Figure 8 shows a schematic configuration of VCR and AR system.

The AR is driven by thermal solar collectors and equipped with a thermal storage tank. The storage tank supplies the hot water required to regenerate the adsorption chiller, between 80 and 90 °C, thermal level that can be achieved by the solar water heating system. To allow proper comparison between DEC and AR the following assumptions have been made:

- absorption chiller is single stage LiBr machines with efficiency equal to 0.7,
- energy required for the post-heating coil is supplied by the thermal solar collectors,
- the electricity consumption ( $Q_{el}$ ) of the “conventional” AHU has been considered to be the 70% of the one required by the DEC [13], to consider the higher pressure losses in the desiccant wheel and the pressure losses in the additional cooling coils,
- AR stops running when water temperature in the upper part of the hot water tank falls below 80 °C. When the AR does not operate the cooling energy requested by the AHU is furnished by a VCR system ( $Q_{cc}$ )VCR, and
- additional electricity input ( $Q_{el}$ ), required to allow the heat rejection in the cooling tower, has been calculated considering a value of the ratio cooling capacity/additional electricity input equal to 4.0.

The comparison has been developed considering only a supply air demand of 3000 m<sup>3</sup>/h. The processes of sensible cooling and dehumidification, performed in the chilled water cooling coil, require an input cooling power  $Q_{AHU} = 27.20$  kW both for the VCR and the AR. This value of cooling power is about 35% more high than which one required by the AHU equipped with DEC system (20.03 kW) to obtain the same processes. Moreover for the DEC system only 6.03 kW are supplied by chilled water cooling coil.

### Results of the comparative analysis

The comparison between DEC and AR has been performed considering two scenarios characterised by different surfaces of solar collectors. The first scenario, called (AR<sub>32</sub>), is characterized by 32 m<sup>2</sup> of solar collectors used to feed the AR generator. This area is not sufficient to satisfy the thermal energy demand of the generator of the AR, so the remaining fraction of cooling power ( $Q_{cc}$ )VCR is supplied by an auxiliary VCR. The second scenario, called (AR<sub>66</sub>), is characterized by a surface of 66 m<sup>2</sup> of solar collectors used to feed the AR generator. This solar collector area permits to compare the AR with the DEC maintaining the same cooling fraction covered by solar energy ( $SF_{DEC} = 0.7$ ). Also in this case, as in the previously scenario, the area

of solar collectors is not sufficient to satisfy the thermal energy demand of the generator of the AR, so the remaining fraction of cooling power ( $Q_{cc}$ )VCR is supplied by an auxiliary VCR. The coefficient of performances of the VCR is the same used for the DEC system ( $COP_{VCR} = 3.0$ ). In tab. 3 are reported the heat flow rates and the electricity consumption required by three refrigerator systems.

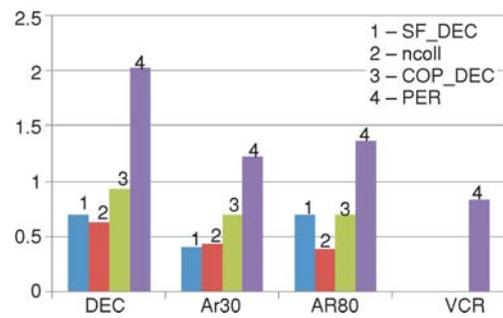
**Table 3. Heat flow rates and electricity consumption required by three refrigerator systems**

	$Q_{DEC}$ [kW]	$(Q_{cc})_{AR}$ [kW]	$(Q_{cc})_{VCR}$ [kW]	Electricity consumption [kW]
DEC	20.03	–	6.03	1.70
(AR <sub>32</sub> )	–	10.78	16.42	2.70
(AR <sub>66</sub> )	–	19.0	8.16	4.75
VCR	–	–	27.20	1.25

Figure 9 reports the energy performance indices calculated using the set of equations from 7 to 11 for the analysed cooling system.

The primary energy ratio, PER, for the desiccant system, is  $2.03 \text{ kWh}_{cold}/\text{kWh}_{PE}$ , whereas the one for the vapour compression system amounts to  $0.83 \text{ kWh}_{cold}/\text{kWh}_{PE}$ , while for the absorption chiller is  $1.23 \text{ kWh}_{cold}/\text{kWh}_{PE}$  or  $1.36 \text{ kWh}_{cold}/\text{kWh}_{PE}$  in function of the utilised area of solar collectors.

The primary energy saving of the desiccant system is about 40%, or 33,5 %, compared to the AR system and more than 150% compared to the VCR system. Further the financial and emission savings of the DEC system in comparison with the VCR have been calculated. In tab. 4, the electric energy saving has been calculated considering an operating period of 800 hours per years, the monetary saving has been calculated considering a price for electric energy of 0.13 €/kWh and the emission saving considering a factor of  $0.524 \text{ kg CO}_2/\text{kWh}$  of electric energy [34].



**Figure 9. Comparison of energy performance indices**

**Table 4. Energy, monetary, and emission savings of DEC system compared to a VCR system**

	1500 [m <sup>3</sup> h <sup>-1</sup> ]	2000 [m <sup>3</sup> h <sup>-1</sup> ]	3000 [m <sup>3</sup> h <sup>-1</sup> ]
Supply air demand	1500 [m <sup>3</sup> h <sup>-1</sup> ]	2000 [m <sup>3</sup> h <sup>-1</sup> ]	3000 [m <sup>3</sup> h <sup>-1</sup> ]
Electric energy saving, [kWh]	2540.0	3520.71	5281.06
Monetary saving, [Euro]	330.27	457.69	686.54
Emission saving, [kgCO <sub>2</sub> ]	1331.24	1844.85	2767.28

Previous analysis indicates that the attractiveness of DEC system may increase, also thanks the possibility to exploit the thermal energy produced by solar collectors both for space heating and yearly production of DHW. Another advantageous of implementation of such cool-

ing technologies is represented by the significant downsizing of the power of a traditional refrigerator system.

Despite the fact that the adoption of solar energy technology is recognized as a realistic response to the energy and environmental problems, evaluations are often unfavourable. There is still a need for subsidy to ensure their effective penetration into the cooling market. One of the difficulties for a larger diffusion of the solar cooling technologies is their initial investment cost. This limit may be removed thanks the introduction of incentive grants. For example in Italy, recent legislation named "Conto Termico" [35] has introduced financial framework for SE investments. Specifically for solar cooling systems an incentive of 510 €/m<sup>2</sup> (2 × 255 €/m<sup>2</sup>) to purchase the solar collectors has been foreseen. So both the proposed DEC and AR system, equipped with 32 m<sup>2</sup> solar collector can benefit, for two year of an incentive of: 2 × 255 (€/m<sup>2</sup>) AST(m<sup>2</sup>) = 16.320,00 €. These incentive can significantly increase the diffusion and competitiveness of DEC compared to standard cooling systems.

## Conclusions

In this study the performance of a desiccant cooling system solar assisted under typical climate conditions has been evaluated. Obtained results confirm that solar energy is an excellent, practical heat source for desiccant regeneration. The proposed system configuration, that foresees the use of an auxiliary cooling source, fulfils standard comfort criteria during the cooling season.

It has been found out that the DEC system can lead a primary energy saving of around 40%, respect both to AR and around 150% respect to a conventional VCR system. The energy-saving advantage over conventional systems confirm that DEC systems are feasible from an energetic point of view. In addition DEC reduces energy operating costs significantly when peak electric utility demand charges are high especially in warm humid climate. Moreover these system are useful to reduce grid congestion, energy price volatility, and offer an important contribution to environmental protection. Critical factors that will ensure the spreading of DEC systems are technological maturity and economic viability. The latter depends on many factors, including the legislation, the guidelines and the energy policy.

The study can also contribute to give standardised procedures for the design and sizing of solar cooling systems in order to reduce costs, installation mistakes and operation faults.

The replacement of compressor cooling systems by solar cooling systems may contribute to replacement of fossil fuel demand by solar energy and by this, contributing to the European policy targets on the increased use of renewable energies.

## Nomenclature

AST	– surface of solar collectors, [m <sup>2</sup> ]	PER	– primary energy ratio
C <sub>H<sub>2</sub>O</sub>	– specific heat of water, [kJ(kg°C) <sup>-1</sup> ]	Q <sub>AUX</sub>	– heat flow rate supplied by auxiliary source, [kW]
COP <sub>AR</sub>	– coefficient of performance of AR	Q <sub>AHU</sub>	– cooling of AHU, [kW]
COP <sub>DEC</sub>	– coefficient of thermal performance of DEC	(Q <sub>CC</sub> ) <sub>VCR</sub>	– auxiliary cooling supplied by VCR, [kW]
COP <sub>VCR</sub>	– coefficient of performance of VCR	Q <sub>D</sub>	– storage tank thermal loss, [kW]
h <sub>1</sub>	– specific enthalpy, [kJkg <sup>-1</sup> ]	Q <sub>DEC</sub>	– desiccant cooling, [kW]
I <sub>β</sub>	– solar irradiance, [Wm <sup>-2</sup> ]	Q <sub>el</sub>	– electrical power, [kW]
m	– mass of storage water, [kg]	Q <sub>G</sub>	– heat flow rate for generator, [kW]
m <sub>RA, SA</sub>	– regeneration (supply) air mass rate, [kgs <sup>-1</sup> ]	Q <sub>HC</sub>	– heat flow rate for desiccant regeneration, [kW]
PE	– primary energy, [kWh]		

$Q_L$	– cooling effect, [kW]	$V_{RA, SA}$	– regeneration (supply) air flow rate, [ $m^3h^{-1}$ ]
$Q_{re-heating}$	– heat flow rate for post heating process, [kW]	$W^1$	– electric power for VCR compressor, [kW]
$Q_{SS}$	– heat flow rate supplied to storage tank, [kW]	$x_E$	– design specific humidity, [ $kgvkgair^{-1}$ ]
$SF$	– solar fraction	$x_{RA, SA}$	– specific humidity of regeneration (supply) air, [ $gvkgair^{-1}$ ]
$SF_{DEC}$	– energy fraction covered by DEC	<i>Greek symbols</i>	
$T$	– temperature of supply air, [ $^{\circ}C$ ]	$\Delta\tau$	– time interval, [hour]
$T_a$	– ambient air temperature, [ $^{\circ}C$ ]	$\eta_{coll}$	– solar collectors efficiency
$T_E$	– design outdoor temperature, [ $^{\circ}C$ ]	$\eta_{el}$	– efficiency of thermoelectric plants
$T_i$	– storage temperature at time $i$ , [ $^{\circ}C$ ]	$\eta_{fossil}$	– boiler efficiency
$T_m$	– mean collector temperature, [ $^{\circ}C$ ]	$\eta_{HS}$	– solar heat exchanger efficiency
$T_{RA, SA}$	– temperature of regeneration (supply) air, [ $^{\circ}C$ ]		

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