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The potential role of trans-critical CO₂ heat pumps within a solar cooling system for building services: The hybridised system energy analysis by a dynamic simulation model

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ABSTRACT

The rotary desiccant wheels application in the air conditioning systems are used for the air dehumidification by means of hygroscopic layers for water vapor adsorption. Nevertheless, external heat sources are required for water desorption to close the air treatment cycle. This paper investigates on the possibility to integrate in that cycle a new component, such as the trans-critical CO₂ heat pump, to reduce the contribution of external thermal sources. In so doing, the high temperature waste heat discharged by the heat pump hot sink can be fruitfully exploited. Additionally, a PV array has been added to the typical layout based on the solar collectors, in order to assure the heat pump electrical driving. The energy analysis is carried out by calculating the energy performance indicators of the whole cooling system, simulating it by a dynamic model built in the MATLAB SIMULINK environment. Specifically, an air handling unit has been properly sized to supply cooling load to a reference conference hall of 1200 m³, with changes in boundary conditions (i.e. solar radiation, daily temperature and relative humidity variations). Indeed, three different cities representing the most typical Italian climatic zones, have been considered for assessing the proposed technical option suitability.

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1. Introduction

It is well known how the building sector is responsible for approximately 21.6% of the global primary energy consumption over the European Countries [1]. For that reason, interventions on the building envelope allow reducing the energy need of existing buildings by enhancing their energy performance. In this framework, increasing the energy efficiency of existing buildings is a great challenge to cope with, by investigating on the best suitable measures to match the restrictive environmental limitations called for the national and European regulations [2]. The renewable energy sources integration is not always allowed because of landscaping or architectural constraints, even if such plants installations are feasible from a technical point of view. Yet, the use of hybrid systems integrating the renewables together with the traditional thermodynamic cycles is a viable option to get the NZEB (Nearly Zero Energy Building) qualification for the existing building [3–5]. Several research projects have been recently carried out, but the hybridization of cooling plants has not been widely addressed. The growing demand of green energy, especially in the residential sector, implies huge efforts in searching for new technical options and economic models [6,7]. Indeed, buildings consume large amount of energy (e.g. 50% in the tertiary sector) to provide the indoor environmental comfort conditions over the hot season. Since the RES (Renewable Energy Sources) have to be applied as much as possible in the next-generation buildings, the solar thermal power can be fruitfully exploited to provide the so-called thermal cooling. By that technique, it is possible to have the cooling effect without mechanical work and electricity consumption once alternative materials have been used. Moreover, the low-grade heat hailing from the sun can be useful to drive totally or partially such devices. On the basis of that working principle, various technical solutions and plant layouts have been developed in several international research projects. Reviewing the scientific literature on that topic, it emerges how those systems generally allow a reduction of 30%–40% of rated chiller power, improving also the economics in operation (from 30% to 70% depending on the

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Among all those developed technical options, it is noteworthy to mention the so-called solar adsorption cooling systems as well. In those plants the solar collectors structure plays the key role to get energy savings and the cycle time optimization can reduce the collector size [9,10].

As a matter of fact, in this process the solar collectors operate as adsorption and desorption device directly heating up the refrigerant in the HVAC systems for buildings that approach is not used since the air has to be treated from thermo-hygrometric point of view.

The air dehumidification process is generally energy expensive when traditional cooling systems are applied. An alternative solution for that purpose consists of physically removing moisture by adsorbent substances able to entrap the air water content within their structures.

Having said this, heating ventilation and air conditioning systems equipped with the desiccant wheels (DW) for solar cooling application are energy efficient, with a low environmental impact, and can be profitable if compared to traditional systems. In the last years, new perspectives in the desiccant cooling system (DCS) have been derived, which are helpful to achieve an energy-efficient design by the heat-driven cooling equipment [11]. Thus, several studies analysed the impacts of the main operating parameters on systems performance [12], including dimensionless indicators, chemical solution properties and inlet air conditions, so as to improve the technologies deployment and their readiness [13,14].

Recently, Alahmer and Ajib reported in their impressive literature review which are the main barriers, challenges, and perspectives of solar cooling technologies wide deployment [15]. They stated that in the last 10 years, the solar cooling market registered an annual growth rate ranging in 40–70%, although the initial costs for installation are 2 or 2.5 times higher than the traditional systems, approximately. Nevertheless, it is expected by 2050 that solar cooling will have a potential market of 417 TWh/y, representing about 17% of energy used for cooling purpose, in accordance with the International Energy Agency (IEA) roadmap [16,17]. Currently, only few thousands of solar cooling plants have been installed worldwide; 59% of those systems integrate absorption chillers for the closed cycle configuration and only 12% of the establishments use an adsorption chiller. When the open cycle configuration was chosen, a plant fraction equal to 23% applied the desiccant cooling framework utilizing a sorption wheel. Just about 4% of all installations utilize liquid desiccant innovation, which demonstrates this technical option is still less produced on a commercial level.

The idea of a liquid desiccant materials application came out to develop a more compact design. Besides those technical and energy aspects, the liquid desiccant materials received much attentions, due to their effectiveness in improving the IAQ (Indoor Air Quality). Indeed, they are able to remove organic compounds and to either kill or deactivate bacteria and virus proliferation [18,19]. It is worth of highlighting that, up to date, this latter characteristic can be considered the crucial driver for further developments, especially after the COVID-19 era. Anyway, from literature review it emerged several research projects on that topic. For instance, Gandhidasan [20] developed a simplified model of a packed bed regeneration process to assess the evaporation rate when the solution is heated up by solar collectors or by a line heater. Stevens et al. [21] presented in their work a computationally efficient model for packed-bed, liquid-desiccant heat and mass exchangers. That model is derived from an effectiveness model of a cooling tower and it was validated by experimental measurements. Seeing that, model complexity and the required computational time usually do not

**Nomenclature**

- **COP** Coefficient Of Performance [%]
- **EEF** Energy Efficiency Ratio [%]
- **P** Power [kW]
- **q** Specific heat [kJ/kg]
- **h** Specific enthalpy [kJ/kg]
- **m** Mass flow rate [kg/s]
- **c_p** Specific heat capacity at constant pressure [kJ/kg*K]
- **s** Specific entropy [kJ/kg*K]
- **w** Specific work [kJ/kg]
- **T** Temperature [°C]
- **η** Efficiency [%]
- **K** Heat transfer coefficient [W/m²*K]
- **RH** Relative Humidity [%]
- **X** Absolute humidity [g/kg]
- **ρ** Density [kg/m³]
- **V** Volume [m³]
- **A** Surface [m²]
- **G** Specific normal irradiation [W/m²]

**Subscripts**

- **th** thermal
- **reg** regeneration
- **ext** external
- **el** electrical
- **ev** evaporator
- **cw** cold water
- **ihex** internal heat exchanger

**Abbreviations**

- **PV** Photovoltaic system
- **HVAC** Heating, Ventilation and Air Conditioning
- **DW** Desiccant Wheel
- **DCS** Desiccant Cooling System
- **HP** Heat Pump
- **DEC** Desiccant Evaporative Cooling
- **EH** Electric Heater
- **ES** Energy Storage
- **EC** Evaporative Cooler
- **REG** Regeneration coil
- **IEC** Indirect Evaporative Cooling
- **SC** Solar Collector
- **NZEB** Nearly Zero Energy Building
permit the use of FEM (Finite Element Modelling), Liu et al. [22] proposed empirical correlations for estimating the dehumidifier heat and mass transfer performance. Similarly, Wang et al. [23] developed a more accurate hybrid model to predict the heat and mass transfer processes in a packed column liquid desiccant dehumidifier. In the authors’ intentions, that model can be applied in operational optimization, performance assessment, fault detection and diagnosis as well. Koronaki et al. [24] carried out a thermodynamic analysis by modelling a counter flow liquid desiccant dehumidifier. The device was analysed and experimentally investigated using three different liquid desiccant solutions, namely LiCl, LiBr and CaCl₂. From their work it emerged that high absorber efficiency and system efficiency can be accomplished under humid conditions, low air mass flow rates and LiCl as the desiccant solution. Finally, Chen et al. [25], by their review article, effectively provided the most recent progresses in liquid desiccant materials application within HVAC systems. From their analysis, it is possible to notice that LiCl is the wider employed solution due to the fact that it offers the better dehumidification effectiveness compared with other salts. Conversely, even though LiBr and CaCl₂ are less expensive, they are less stable and are characterised by poorer dehumidification performance. Thereafter, potassium formate (KCOOH) solution emerged as another candidate for dehumidification system, owing to its environmentally friendly nature as well as its low corrosive properties.

1.1. Solar cooling systems layout: a quick literature overview and scope of the article

In the field of refrigeration, alternative options to the mechanical vapor compression systems for cooling production consists of sorption processes. Such a refrigeration technology can be mainly divided in two groups: closed and open cycle systems. The former generally utilize several types of absorption or adsorption cooling, ejectors, and solar assisted electric heat pumps. The latter provide a direct air treatment, in order to keep under control air flows thermo-hygrometric conditions by using of desiccant materials. In that case, the air handling units undergo a strong redesigning process since new system components must be integrated. For that reason, in the solar cooling sector, closed cycles are generally preferred to open ones, owing to their higher installation easiness over refurbishment interventions in existing buildings. Yet, some authors such as Fong et al. [26] explored the possibility to use both cycles in the same building, along with radiant ceilings installation, in order to efficiently figure out the hygrometric issues associated to subtropical regions. In the last decade, several authors investigated on which were the best solar technologies to be coupled with sorption chillers. Noro and Lazzarin [27,28] considered CPC (Flat Plate Collectors), ETC (Evacuated Tube collectors) PTC (Parabolic Trough Collectors) for thermal driving of single and double stage absorption chillers. Additionally, they compared those systems to PV-driven electric heat pumps. They concluded that PV solutions using silicon cells were generally better than thermal ones. Only the parabolic trough collectors showed similar performance when coupled to double effect LiBr absorption chillers. Similarly, Papoutsis et al. [29] carried out a parametric study, by numerical simulations, of different plant configurations under the Mediterranean climatic conditions. They analysed also a hybrid scenario where PV/T were installed. Furthermore, the cooling power was partly generated by an absorption chiller and a vapor compression one, in order to increase the overall system COP. Anyway, those data confirmed that PV driving led to the best energy and environmental performance. A similar plant concept was developed by the ZEOSOL project, which was funded by HORIZON 2020 program in 2017 [30]. The main objective of the ZEOSOL project was to develop a new advanced solar cooling and heating product, using advanced heat exchanger technology, and integrating a heat pump to meet peak power demand. It used synergies between the technologies of thermal chillers (heat to cooling technology) and heat pump (electricity to cooling technology). The main innovation consisted of constructing an adsorption chiller unit based on Fahrenheit patented zeolite coating technology, lessening both the unit’s volume and its capital cost by 50%. The scientific outcomes dissemination related to that international research activity can be found in Ref. [31,32]. Rad et al. [33] provided a thermo-economic analysis on an Iranian case study using dynamic simulation tools on the basis of First and Second Law of thermodynamics. Thus, Almohammadi et al. [34] proposed an innovative SDACS (solar powered adsorption cooling system) with three axial finned tubes heat exchangers connected in parallel. They tested on field a demo version able to accomplish a COP equal to 0.52 after the operating parameters optimization by the multi-objective genetic algorithm application.

Finally, other authors investigated on the possibility to integrate within solar cooling closed cycles PCMs (Phase Change Materials) for storing thermal energy [35]. On that topic, a recent work of Louhani et al. [36] proposed an interesting option to figure out the challenge of free cooling for humid climates at night-time. Indeed, they integrated moderate melting point PCM within rooms ceiling, as a way to use the evaporative cooling system and desiccant wheel. The most original aspect of that work lies in the use of Trombe Wall for storing solar energy over the day to regenerate the desiccant wheel. Finally, it is worth of highlighting the solar chimney concept for increasing the closed cycles efficiency. By that technique it is possible to cool down the PV modules for driving an electric water source heat pump, and to guarantee high coefficient of performance of that machine [37]. The working principle consists of spraying water parallel to the downward airflow, within a channel which is created by modifying the PV module back shield. In that channel, part of sprayed water evaporates cooling down the air; the remaining water drops down to the channel bottom side and it is stored in a small vessel. Since the PV module receives solar radiation, its surface temperature increases heating up the air in the back shield, so that the air circulation is favoured. At the end of that process, the PV module is cooled down by the air and the stored water can be delivered to the HP condenser.

As regards the open cycle systems, which typically include rotary desiccant wheels, they became increasingly attractive owing to their low electric energy consumptions. However, their efficiency rarely overcomes the threshold value of 0.5, inasmuch the energy performance is strictly correlated to the evaporative cooling section efficiency and to the saturator effectiveness. So that, such HVAC solutions are particularly suitable when the external air wet bulb temperature does not exceed 25 °C [38]. Many efforts have been made by research to improve the overall efficiency and to optimize the components designing process. For instance, the Dunkle layout [39], as well as the Uckan configuration [40] and the double stage systems [39,41] are characterised by a high complexity level, utilizing up to four rotary heat exchangers, two desiccant wheels and two additional in line heaters. Despite the integration of such devices the overall cooling efficiency does not exceed 0.65. It is important to point out that the evaporative cooling-based technologies are not able to precisely keep under control the supply air thermo-hygrometric conditions. As a matter of fact, the ASHRAE comfort zone for conventional air conditioning, referred to the inner space conditions, has to be enlarged. For that reason, an additional electric chiller can be installed in the desiccant cooling systems so as to improve the IAQ and to simplify the plant layouts.

Having said this, several research projects dealing with that topic can be found in literature. In Ref. [42,43] the authors performed numerical simulations on desiccant based air handling.
units in two different Italian cities and in low latitude isolated islands. In the first work [42] the authors evaluated the system energy and economic performance by varying the solar collectors technology. In the second one, using the same methodological approach, a plant hybrid version, including PV array, was proposed. In so doing, it was possible to face the big challenge to efficiently provide cooling power in all those remote areas characterised by extremely hot and humid conditions. The plant hybridization generally requires more available surface. As a consequence, several research projects dealt with the potential application in desiccant cooling system of both water-cooled and air-cooled PV/T technologies [44–46]. Unfortunately, those devices are still expensive, and they are not yet beneficiary of incentive schemes, so that their adoption can be not cost-effective. Finally, it is worth of noticing that other technical options are suitable for thermally regenerating the desiccant materials.

In accordance with literature, Fig. 1 depicts the typical layout of the hybrid desiccant-based air handling unit, where the external heat for the wheel regeneration can be provided by CHP (Combined Heat and Power) [47], HPs (e.g. ground source or air source) [48,49] conventional solar collectors or hybrid versions [44], and thermal cascades hailing from other processes. However, other interesting and attractive solutions for that purpose have been addressed by some authors. As a matter of fact, fifteen years ago, Mei et al. [50] proposed to use the heated up air, flowing through the building ventilated PV-facades, to improve the PV modules efficiency and to partly regenerate the desiccant wheel, together with solar collectors. Notwithstanding, the average system COP was limited to 0.51. The same approach have been used, more recently, by Peci et al., where Unglazed Transpired Collectors have been chosen for their numerical simulations, accounting for building shapes, facades orientation and climate conditions [51]. The UTCs are inexpensive, easy to install and are made up of a perforated metal absorption layer, a plenum, an insulation layer, and several ducts to distribute the heated air directly to the building or to the desiccant cooling section. The main strength point of that technology consists of simultaneously combining facades refurbishment interventions with the HVAC systems upgrade.

In the end, referring to the recent progresses in the heat pumps sector, the trans-critical CO2 cycles have become of growing interest owing to their potential to produce high temperature heat or low-pressure steam [52]. Hence, the high-grade heat recovery from the hot sink can be fruitfully exploited for wheels regeneration purpose. Aprea et al. [53] explored the integration of such a small scale heat pump operating in split mode by experimental data. In detail, the internal unit met the room sensible cooling load, while an air handling unit equipped with desiccant wheel and only IEC module met the latent heat. The main findings demonstrated the hybridised version goodness, showing an average COP gain equal to 77%.

In the end, a new concept of CO2 heat pump integration was introduced by Liu et al. [54]. The authors addressed the issue only from a theoretical point of view, presenting a parametric numerical simulation. Specifically, they proposed a new desiccant wheel system with closed loop air regeneration. The air regeneration process occurred inside a closed loop and a trans-critical CO2 HP was utilized for the recyled air regeneration including dehumidification, cooling and heating process. Moreover, an air heat exchanger was inserted to pre-cool the recycled air regeneration, which decreased the evaporator cooling load. Having fixed the saturation temperature at the evaporator equal to 10 °C and the discharge pressure of 8 MPa, the COP rose up to 6.5. By adopting that solution solar collectors or other thermal cascades are unnecessary and only PV arrays can electrically drive the HP.

After this premise, the main aim of this paper is to assess the suitability of a hybridised version of the solar-assisted desiccant cooling plant. By integrating different well-proven and commercial technologies, the authors investigate on the possibility to substitute some system components so as to produce a fully-renewable chilled air. Referring to the most common desiccant-based air handling unit, which includes an indirect evaporative cooling module and a chiller [42], the integration of trans-critical CO2 HPs has been proposed. In so doing, it is possible to lessen, on one hand, the external heat sources contribution, on the other hand, to generate additional electricity by PV modules for the building needs. By the dynamic simulation modelling, the energy performance, referred to three different Italian cities, along with some key performance indicators useful for HVAC designer, have been presented and discussed. Finally, the main operating parameters values of a commercial
2. Trans-critical CO₂ heat pumps working principle and related issues

Carbon dioxide is a safe, economic and sustainable refrigerant which can be suitable for heat pumps and cooling systems. Due to its thermophysical properties CO₂ has good heat transfer characteristics, working within a wide operating temperature range. Indeed, starting from supercritical conditions, the CO₂ temperature can rapidly pass from 180 °C to almost 0 °C allowing to get high energy performance for heat pump systems. The trans-critical CO₂ HPs coefficient of performance can reach also very high levels, owing to the possibility to use small compressors, mitigating the energy losses associated to the working fluid pressurization [55,56]. Yet, there are two main issues to take into account when CO₂ is used as a refrigerant: the first one concerns the high operating pressure, (i.e. beyond the critical pressure of 73.75 bar), and the second one is the low critical temperature, since the carbon dioxide reaches its critical point at a temperature of 304.13 K (31.1 °C). Fig. 2 shows the typical cycle transformations on the P-h plane. Compared to the most common refrigerants, carbon dioxide-based equipment requires a carefully designed system to cope with the peculiar characteristics of operating temperature and pressure.

Referring to the thermodynamic cycle shown in Fig. 2, the basic layout of a trans-critical CO₂ HP is composed by an evaporator, a high pressure compressor, a gas cooler, an internal heat exchanger (IHEX) and the expansion valve (see Fig. 3). It is noteworthy that in the supercritical region, the temperature and pressure values are independent properties, hence the CO₂ temperature downstream the gas cooler it is not correlated to the inner pressure level. Notwithstanding, that temperature is one of the most important parameters affecting the HP efficiency, because it determines the heat release amount. Moreover, to get better energy performance, the optimization process aims at indicating what is the best gas cooler outlet temperature corresponding to the maximum operating pressure.

The heat transfer mechanism within that component is also very different from the traditional condensing process associated to the use of typical refrigerants. Indeed, in the supercritical region, the phase-change cannot be distinguished, so that the transformation is not isothermal. Consequently, the heat transfer occurs by sensible

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Fig. 2. P-h diagram Carbon Dioxide Refrigerant [R744] (Source: Emerson Climate Technologies [57]).

Fig. 3. Basic layout of trans-critical CO₂ HP.
cooling and the carbon dioxide undergoes a wide temperature difference.

For that reason, the trans-critical CO₂ HPs are very suitable for high temperature applications [58] and they can be also used for boosting low temperature distribution networks, as reported in Ref. [59]. Anyway, the best energy performance can be accomplished as the gas cooler outlet temperature is the lowest. That implies a wide temperature difference for the secondary fluid, which is the end-user’ energy carrier. Therefore, for building applications the gas cooler is partitioned very often. That technical scheme can be deeply modified mainly acting on evaporation, expansion and compression phases. Fig. 4 outlines some of the most interesting technical solution hailing from previous research projects.

From literature survey, it emerged that several studies focused on the possibility to make some cycle modifications in order to enhance the HP performance. It is important to point out that the cycle efficiency improvements generally require considerations on the whole system. Indeed, some modifications lead to isolated alterations related to a single component, while other interventions are closely tied to the components’ mutual interactions. The HP technical scheme can be deeply modified mainly acting on evaporation, expansion and compression phases. Fig. 4 outlines some of the most promising technical solutions hailing from previous research projects.

In detail, Fig. 4 a) depicts the auxiliary loop system. That solution is composed by two separate heat pump loops with two independent compressors. Here, the IHEX fulfills simultaneously the evaporator function for the internal loop and the main loop subcooling, upstream the expansion valve. Notwithstanding, good performance are achievable only for low intermediate operating pressure. Fig. 4 b) refers to the so-called dual-compression dual-expansion system. It can be noticed how the layout is more complicated, including two compression stages with intercooler and one additional IHEX; adopting that solution, the COP for cooling is 29% higher than the single compression system [62]. Finally, the last image deals with a heat pump with flash gas bypass [63]. Despite the double stage compression with intercooler that system represents also an indirect modification of the evaporation process. Here, the refrigerant flows through a throttling valve up to the evaporator pressure and then, it enters within the flash tank. Thereafter, from the flash tank the vapor phase is directly bypassed downstream the evaporator, while only the saturated liquid is sent-off to that component. Thence, the COP for cooling purpose can be improved of 7% more, owing to the larger heat transfer capacity and lower pressure drop.

In the trans-critical CO₂ cycle the pressure drops are generally very high, i.e. from 140 bar to 35 bar, in the major part of applications. That entails very large expansion losses, which could be mitigated by substituting throttle valves with turbo expanders. In such a way, the mechanical work recovery is more feasible and beneficial. Indeed, the expander integration leads to a reduced optimum gas cooler pressure as well as to an improved COP value of 33% more [64].

It is noteworthy that many devices, based on different technologies, can be potentially applied as an expander for mechanical work recovery. For instance, rotary, reciprocating, vane, screw and scroll are suitable option in addition to turbines. However, each design solution is characterised by its own advantages, drawbacks and limitations. Fig. 5 shows three different layouts integrating turbo expanders able to drive the compressors.

Yang et al. [64] investigated on the convenience of direct or indirect compressor driving solution. They compare each other the three options reported in Fig. 5, showing that the solution b) is characterised by the highest COP (i.e. 3.52 instead of 3.49 for a) and 3.16 for c) and the optimised maximum pressure equal to 101.3 bar. Additionally, they demonstrated that an internal heat exchanger could be incorporated but it is not so effective when work recovery is implemented.

In the end, it is possible to state that recent researches on those machines are showing the trans-critical CO₂ cycles suitability, since they can perform well for water and air heating applications. For that reason, the authors decided to integrate this technology within the solar cooling plants in order to evaluate the potential benefits hailing from the energy system hybridization.

3. Case studies description and methodology

The solar cooling HVAC systems can be considered as a special application of the DEC systems (Desiccant Evaporative Cooling), in which thermal renewables are involved to heat the air stream for water desorption from the desiccant layer. The operating principle of such plants has been depicted in Fig. 6, where EH indicates the electric heater, ES is the thermal energy storage, ECs are the evaporative cooler and REG represents the regeneration coil. The dehumidification process does not occur by water condensation, due to the air stream flowing through a cold heat exchanger. Moreover, the overall cooling effect can be accomplished combining the IEC (indirect evaporative cooling) system with the sensible cooling provided by an electric chiller. Given that the desiccant wheel has to be regenerated, the most common solar cooling plants are thermally driven by solar collectors.
In this work, the reference system for the comparative analysis has been weakly hybridised so as to consider carbon free all of the energy need.

The proposed technical option consists of integrating a new trans-critical CO₂ heat pump, which is able to contemporary generate chilled water and high temperature heat, up to 100 °C. In so doing, an internal energy recovery for the thermodynamic cycle is suitable and external energy sources can be reduced, favouring the conversion efficiency. The heat recovery from an electric heat pump was investigated by Beccali et al., in 2012 [65]. However, in their work the authors considered a common chiller equipped with an external condenser connected in series to the AHU recovery coil for wheel regeneration. In so doing, the refrigerant is directly used for the air stream heating up.

The unconventional AHU has been designed to meet the energy need of a reference room located in three different Italian cities. According to previous works, that approach is useful to identify criticalities and suitability of these plants when they are built in different climatic zones. In this way, it is possible to account for performance variations caused by low and high relative humidity as well as outdoor environmental temperature.

A wide room, characterised by only two dispersing surfaces, has been assumed as reference case for cooling load calculations. Specifically, it is a 300 m² conference hall of floor surface, with a height equal to 4 m, which is designed for 200 people inside.

On the basis of Italian standards, such as UNI 10339, and building data analysis [66] the air flow rate for ventilation has been set equal to 8000 m³/h and no recirculation is required. Consequently, the ventilation rate is equal to 6.66 vol/h. Since for that end—use the cooling load is mainly due to inner heat generation and partly to the solar irradiation, it has been considered almost constant over the simulation time.
As a consequence, 15.5 kW of sensible heat (hailing from people, from air infiltrations and walls transmission) and 8 kW of latent heat are assumed to characterize the conference hall. More in detail, 65 W/pers and 40 W/pers have been fixed for sensible heat and latent heat, respectively. In addition, the dynamic simulations have been performed starting from May up to the end of September, considering the HVAC operation over 10 h per day (i.e. 8:00–18:00) and 5 days per week.

In the end, Milan, Rome and Palermo have been chosen as case studies to simulate the effects of different environmental conditions, as shown in Fig. 7, where environmental data refers to the TRY (Test Reference Year) according to UNIENISO 15927-4.

By the dynamic simulations, the energy balance associated to each component have been made in terms of electricity, heat and cool. The analytic formulation of each model component has been widely presented and discussed within the Supplementary Material Section attached to this work.

In accordance with literature, the main indexes to compare the energy performance are the solar cooling thermal COP and the electrical COP. The first one (see Equation (1)) represents the required regeneration energy to produce the cooling effect, while the second one shows the needed electricity amount to generate that cooling power considering the system as a common chiller.

\[
\text{COP}_{\text{th}} = \frac{P_{\text{cooling}}}{P_{\text{reg}}} = \frac{m_{\text{air}} \cdot (h_{\text{ext}} - h_{\text{supply}})}{P_{\text{reg}}}
\]  

\[
\text{COP}_{\text{e}} = \frac{P_{\text{cooling}}}{\sum_{i} P_{\text{el,i}}}
\]

Finally, by Equation (3) the regeneration power can be evaluated distinguishing external sources and internal heat recovery.

\[
P_{\text{reg}} = P_{\text{external source}} + P_{\text{recovery}} = P_{\text{th,SC}} + P_{\text{th,EH}} + P_{\text{th,HP}}
\]

Thus, the specific receiving surface of solar collectors and PV, referred to the cooling power, along with the thermal solar fraction (TSF) have been considered as further indicators for discussing the outcomes. This latter can be easily computed by Equation [4]:

Fig. 7. Temperature (on the left) and absolute humidity (on the right) over the simulation time: Milan, Rome, Palermo (from the top to the bottom). The red lines represent the supply absolute humidity. (For interpretation of the references to colour in this figure legend, the reader is referred to the Web version of this article.)
such a way, the indirect evaporative cooling system contribution can grow. However, that choice penalises the heat pump COP values, affecting also the optimal discharge pressure as well as the CO2 outlet temperature from the gas cooler, as shown in Fig. 8.

It is worth of noticing that all of data depicted in Fig. 8 refer to the HP performance analysis associated to Rome climatic conditions. Additionally, since the HP cooling power decreases as the effectiveness values enhances, a lower heat amount can be discharged to the hot heat sink. Therefore, the required heat for regenerating the desiccant wheel must be provided by more solar collectors or by the electric heater alternatively.

By a qualitative energy analysis, it is possible to discuss the main thermodynamic effects on the whole solar cooling cycle deriving from the trans-critical CO2 heat pumps application. To do so, the overall system COP can be calculated combining Equation (1) and Equation (2), and it reads as:

$$\frac{1}{\text{COP}_{\text{HP}}} = \frac{1}{\text{COP}_{e}} + \frac{1}{\text{COP}_{\text{th}}}$$

(5)

Additionally, the Heat Recovery Fraction (HRF) has been introduced in order to account for the HP contribution to the regeneration heat by Equation (6).

$$\text{HRF} = \frac{P_{\text{th,HP}}}{P_{\text{reg}}} = \frac{\Delta h_{\text{air,HP}}}{\Delta h_{\text{reg}}} \Leftrightarrow \frac{T_{\text{air,HP}} - T_{\text{air,8}}}{T_{\text{air,reg}} - T_{\text{air,8}}}$$

(6)

where $\Delta h_{\text{air,HP}}$ denotes the air specific enthalpy difference caused by the hot coil supplied by a generic heat pump and $\Delta h_{\text{air,reg}}$ is the total required specific enthalpy difference. It is important to point out that sometimes the required regeneration heat can be provided by other external sources depending on the heat transfer temperature level to get the assumed regeneration temperature.

By substituting Equation (6) in Equation (5) and considering the HP coefficient of performance (COP_{HP}) when it operates for heating purpose, Equation (5) has been rewritten as follows:

$$\frac{1}{\text{COP}_{\text{HP}}} = \frac{1}{\text{COP}_{e}} \cdot \left[ 1 + \frac{\text{COP}_{\text{HP}}}{\text{HRF}} \cdot (1 - \text{HRF}) \right]$$

(7)

In such a way, the HP conversion efficiency has been directly correlated to the solar cooling system coefficient of performance. Notwithstanding, only those parameters are not enough to completely understand the role of different HP technologies. Indeed, the HRF value is strongly dependent on the condenser saturation temperature and consequently from the heat pump COP.

According to the Zhang et al. [68] modelling methodology, the heat pump COP can be evaluated starting from the Carnot ideal cycle, introducing a thermodynamic perfection factor $\mu$, which typically ranges in 0.4–0.6.

So that, the heat pump coefficient of performance reads as:

$$\text{COP}_{\text{HP}} = \mu \cdot \frac{T_{\text{cond}}}{T_{\text{cond}} - T_{\text{hv}}}$$

(8)

Thereafter, introducing the temperature difference between the HP refrigerant at the hot heat sink and the air flow as an additional technical parameter, the $T_{\text{air,HP}}$ value for evaluating the HRF can be computed in accordance with Equation (9) and Equation (10).

$$T_{\text{air,HP}} = T_{\text{cond}} + \Delta T_{r-a}$$

(9)

### Table 1

| Location  | $P_{\text{reg}}$ [kW] | $P_{\text{chill}}$ [kW]coiling |
|-----------|-----------------------|-------------------------------|
| Milan     | 90                    | 27.5                          |
| Rome      | 90                    | 35                            |
| Palermo   | 90                    | 41                            |
mon water-source CO2 HPs commercial versions are able to produce chilled water at 7°C with COP equal to 0.14/C14, reducing the exhaust air temperature difference in the regenerator. That is due to the lower value of external actual temperature over the plant operational hours, compared to the reference one. For that reason, to reach the required regeneration temperature more thermal power has to be supplied. Consequently, to analyse the actual hybrid plant behaviour, dynamic simulations has to be performed.

Referring to Milan environmental conditions registered in May and August (see Fig. 10 and Fig. 11) it is possible to note that the thermal power for regeneration is higher in May, even if the cooling power is almost absent. In that case, the required thermal power is just above 100 kW, while in August it ranges between 87 kW and 98 kW. From data it emerges that the desiccant wheel operates to reduce the absolute humidity to the design values. Thus, cooling power is generally high, and it is partially provided by the trans-critical CO2 HP.

Moving towards the middle regions, such as Rome, the average outdoor absolute humidity lessens, and the desiccant wheel operates only for a few hours (see Fig. 12). The required cooling power is almost null, and the ventilation systems runs in free cooling mode. Furthermore, the evaporative cooling systems are enough to keep under control the supply air temperature as well as the absolute humidity to the design values.

Conversely, during the hot season the higher outdoor temperature affects positively the heat recovery exchanger effectiveness, reducing the exhaust air temperature difference in the regenerator. Thus, cooling power is generally high, and it is partially provided by the trans-critical CO2 HP.

Indeed, since the CO2 saturation pressure in the HP evaporator has been set to 34.85 bar, implying the saturation temperature of 0°C, the maximum COP ranges between 2.37 and 2.43. Even though lower COP values entail higher recoverable thermal power, once the cooling power has been set, the main objective is to minimise the electrical input power which is provided by PV arrays.

In such a way, it is possible to save roof surface for installing the electrical renewable source. In order to size properly the PV array, the net metering option has been accounted for. In detail, only the equivalent hours related to the cooling season have been used in calculations for determining the PV receiving surface. As a consequence, a renewable electricity excess occurs over the building heating season, which could be exploited to partially meet the overall building electricity need.

The reference outdoor environmental conditions, provided by the current standards, lead to an underestimation of the required heat for desiccant wheel regeneration. That is due to the lower value of external actual temperature over the plant operational hours, compared to the reference one.

For that reason, to reach the required regeneration temperature, the selected layout for this work could be modified by moving backward the HP evaporator, before the desiccant wheel. In that case, since the inlet air temperature is lower, the HP could get better performance in terms of COP. Yet, the most common water-source CO2 HPs commercial versions are able to produce chilled water at 7°C, entailing a typical saturation temperature equal to 0°C. Therefore, for that purpose, HPs characterised by saturation temperature at the evaporator equal to 8–10°C are required. That interesting option has not been addressed here by the authors, but it will be reasonably explored in further developments of this project.

Refring to Fig. 9, it is possible to note that the best performance can be achieved when the gas cooler pressure is equal to 140 bar.
power related to the Rome case study, i.e. 100 kW, but the physical
dehumidification process occurs more often. Obviously, since
Palermo belongs to the hot climatic zone the cooling power in-
creases over the whole simulation time period, up to 30 kW. It is
worth of noticing that to simulate properly the solar cooling
operation over the middle season the control strategy for load
following (in terms of temperature and absolute humidity) must be
implemented. Indeed, if an adjusting system was not implemented,
simulations will run from a mathematical point of view, but results
will be not reliable, and they could not have any physical sense.

Depending on the inner space end use, the building cooling load
is strongly related to the solar irradiation and to the internal
sensible and latent heat generation. According to literature [72],
buildings may be chilled even if the outdoor environmental tem-
perature is lower than the indoor one. Having said this, three
different conditions can occur: the first one consists of an outdoor
temperature lower than the supply one; the second case is related
to the external absolute humidity value lower than the supply ab-
solute humidity; the last one is the contemporary occurrence of
both previous cases. In Fig. 16, the air transformations during the
control application have been reported, while bypasses and recir-
culation for hot and cold air have been depicted in Fig. 17. In detail,
referring to the first case, the hot and exhaust air from the inner
space have to bypass the recovery heat exchanger.
On the contrary, when the external absolute humidity decreases, the desiccant wheel would adsorb too much water so that the
bypass of both hot and cold air is required, and regeneration is not needed.

Finally, when temperature and absolute humidity are very low all the bypasses are open, an exhaust air fraction is recirculated so as to be mixed for enhancing the water content in the supply air.

The dynamic simulation model has been also useful to evaluate the control strategy effectiveness, in terms of energy saving for desiccant wheel regeneration and water consumption associated to both direct and indirect evaporative cooling units. The main outcomes related to the case studies have been summarised in a systemic overview in Table 2.

From data analysis it emerges how the control strategy leads to better energy performance in Rome while the water consumption is minimised in Milan.

That control strategy implementation in the MATLAB model leaded to the time series profile shown in Figs. 10–15, which influenced strongly the energy balance as well as the renewable sources and the backup size.

Having said this, the dynamic simulations have been performed for both the traditional and hybrid solar cooling systems over the whole time period, i.e. starting from May to September. Fig. 18 reports the thermal and electrical energy balance associated to the traditional solar cooling layout when it operates in three different climatic zones.

From data it emerges that the cooling load for the air treatment in Palermo is almost 80% higher than in Milan, while the enhancement in the required thermal power for the desiccant wheel regeneration is limited to 19.3%.

It is important to point out that the regeneration heat is independent of systems layout, but it is correlated to the outdoor environmental conditions. Furthermore, that heat, in the reference system, is provided only by solar collectors and by an electrical heater which is PV driven.

Referring to the energy balance, the sum of those amounts is higher than the regeneration heat due to the fact that the storage device energy losses has been taken into account. The same approach has been used to analyse the hybrid solar cooling system performance. Fig. 19 depicts clearly how the heat recovery hailing from the trans-critical CO₂ HP can contribute to increase the solar cooling thermal COP value, allowing to save the roof surface. The HP heat fraction for Milan, Rome and Palermo are equal to 15.9%, 23.52% and 24.57%, respectively. That entails a reduced number of solar collectors and consequently a lower thermal energy to provide by external source from the cycle. On the basis of those outcomes, it is possible to state that the hybridization option by integrating the trans-critical CO₂ HP is more suitable where high cooling loads for the air treatment are needed.

Specifically, when the evaporative cooling is not enough to keep under control both supply absolute humidity and temperature, higher sensible heat has to be subtracted from the air stream by means of the HP. Therefore, the larger the cooling load, the larger the recovered heat from the HP is. In such a way, owing to the leverage effect of the heat pump COP, even if more electricity is required, the solar cooling thermal COP increases. Since the heat pump is fed by the PV array the hybrid solar cooling system is still completely renewable. In Table 3 the plants performance comparison as well as the specific receiving surface are outlined in a systemic overview. As shown, the best energy gain has been accomplished in Palermo, i.e. COPth,sys enhances of 35.5% while the COPel,sys decreases of 38.4%.

Table 2
Thermal energy for regeneration and water consumption. Scenarios comparison with and without control strategy implementation. In round brackets the percentage reductions.

|          | E_regen [MWh] | V_we, dec + V_we, sec [m³] |
|----------|--------------|----------------------------|
| Milan no | 102.05       | 67.38                      |
| control  | 36.69 (64%)  | 29.90 (55.6%)              |
| Rome no  | 101.39       | 66.84                      |
| control  | 33.85 (66.6%)| 30.68 (54.1%)              |
| Palermo no | 98.40       | 58.58                      |
| control  | 50.20 (49%)  | 33.08 (43.5%)              |

Fig. 13. Rome case study: Regeneration power and cooling power over August.
3.18 m²/kW\textsubscript{cool}. It shows that the solar cooling system electrification leads to a better exploitation of the sun power owing to a non-proportional interchangeability of thermal and electrical renewable sources. Additionally, from the total receiving surface values it is possible to note how the highest percentage reduction (i.e. −21.63%) can be achieved in Rome. Table 4 reports some useful key performance indicators referred to the unit of air flow rate passing through the selected air handling unit while Fig. 20 depicts the thermal solar fraction reductions together with the required PV receiving surface. It is notable how the lowest energy contribution related to the electric heater has been registered in Rome case study. That result is caused by a better exploitation of heat recovery from the HP and from solar collectors seeing that Rome is characterised by less severe thermo-hygrometric environmental conditions.

Finally, Table 5 summarises the renewable electricity production occurring during the winter season which hails from the PV arrays. It is notable how Milan and Palermo are characterised by very close values owing to comparable electricity off-takes of heat pump and electrical heater. Conversely, only 11.06 MWh/y can be used in Rome in order to increase the building renewable electrical fraction. Nevertheless, since Rome case study shows the higher roof surface saving, it should be possible to oversize the PV array to increase its yearly capability for that purpose.

5. Conclusions

In this paper a preliminary energy analysis on the possibility to hybridise a solar cooling HVAC system has been presented. A MATLAB SIMULINK model has been built and dynamic simulations when outdoor environmental conditions change have been performed. The main findings can be outlined as follows:

- The trans-critical CO\textsubscript{2} HP maximum operating pressure has to be 140 bar approximately in order to maximize the COP value, i.e. 2.4. In that way it is possible to transfer heat from hot carbon dioxide to the exhaust air stream effectively;
- Dynamic simulations are the best solution to properly size the solar cooling system since the absolute humidity plays a key role. Indeed, the system design based on the reference outdoor
environmental conditions leads to the energy needs underestimation and the HVAC should not be able to keep under control the inner space comfort values;

- The solar cooling hybridization by the trans-critical CO2 HP and PV arrays shows the best results when it is applied in hot climatic zone, accomplishing energy gain up to 38% more and saving roof surface due to receiving surfaces reduction of 21% less, such as in the Rome case study.
- The control strategy implementation leads to significant energy saving for the regeneration process, reducing also, the water consumption related to the DEC and IEC units integrated in the power plant. In detail, for Rome and Milan case studies the achieved energy benefits are equal to 66.6% and 64% respectively, while the associated water demands can be reduced up to 54.1% and 55.6%. Palermo case study results less sensitive to the adjusting system operation compared to the other cases.

Anyway, further developments are required in order to experimentally validate the simulation outcomes. Specifically, the building dynamic behaviour, in terms of latent and sensible heat hourly

Fig. 16. Thermodynamic transformations on psychrometric diagram for solar cooling system control strategy.

Fig. 17. Control strategy scheme for Solar cooling plant.
variations, have to be implemented and integrated within the present SIMULINK model in order to assess more effectively the hybrid system time response. In the end, a graphical tool for a fast sizing process should have to be developed so as to help HVAC

Table 3
Reference solar cooling system compared to the hybrid solution: energy performance indicators and specific receiving surfaces associated to the overall cooling power.

| System      | COP th, sys | COP el, sys | PV specific surface [m²/kW cool] | SC specific surface [m²/kW cool] | Total receiving surface [m²] |
|-------------|-------------|-------------|----------------------------------|----------------------------------|-----------------------------|
| Milan       | Reference   | 0.3         | 1.24                             | 3.4                              | 8.38                        |
|             | Hybrid      | 0.36        | 0.9                              | 4.65                             | 5.72                        |
| Rome        | Reference   | 0.46        | 2.18                             | 4.66                             | 5.78                        |
|             | Hybrid      | 0.61        | 1.75                             | 2.05                             | 3.79                        |
| Palermo     | Reference   | 0.45        | 2.89                             | 1.48                             | 5.25                        |
|             | Hybrid      | 0.61        | 1.78                             | 2.4                              | 3.18                        |

Fig. 18. Energy balance of reference solar cooling system over five months simulation time-period.

Fig. 19. Energy balance of hybrid solar cooling system over five months simulation time-period.
designers during the technical feasibility analysis. Additionally, the hybrid solar cooling plant seems to be a promising alternative solution for heading towards the NZEB qualification for buildings and to totally produce green cooling.

CRediT authorship contribution statement

Gianluigi Lo Basso: Conceptualization, Methodology, Writing - review & editing. Livio de Santoli: Supervision, Project administration, Funding acquisition. Romano Paiolo: Resources, Visualization, Writing - original draft. Claudio Losi: Software, Formal analysis, Writing - original draft.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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Table 4
Required thermal and electric energy to drive components which provide heat. All of values are normalised by the unit of air flow rate.

| System   | Available Renewable electricity [MWh/y] |
|----------|----------------------------------------|
| Milan    | Reference: 10.37, Hybrid: 14.18        |
| Rome     | Reference: 8.89, Hybrid: 11.06          |
| Palermo  | Reference: 8.93, Hybrid: 14.49          |

Fig. 20. a) Thermal solar fraction values for three different cities b) required PV receiving surface normalised by unit of air flow rate.

Table 5
Available renewable electricity over the winter season which could be useful to meet the building electric load.

| City     | Reference | Hybrid | Reference [kWh/m²/h] | Hybrid [kWh/m²/h] |
|----------|-----------|--------|----------------------|-------------------|
| Milan    | 4.3       | 3.3 (–23.3%) | 1.05                | 0.69              |
| Rome     | 4.23      | 3.17 (–25%)  | 1.26                | 0.41              |
| Palermo  | 6.32      | 4.34 (–31.2%) | 2.03                | 0.34              |

| City     | Reference | Hybrid | Reference [kWh/m²/h] | Hybrid [kWh/m²/h] |
|----------|-----------|--------|----------------------|-------------------|
| Milan    | 0.69      | 0.35   | 0.60 (–72.2%)        | 0.60              |
| Rome     | 0.41      | 0.47   | 0.47                 | 0.47              |
| Palermo  | 0.34      | 0.29 (–16%)  | 1.17 (–78.8%)       | 0.32              |
