# Performance analysis of a small scale solar cooling plant based on experimental measurements

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### Abstract

The main scope of the paper is analyzing the experimental data and evaluating the techno-economic feasibility of a solar cooling plant with solar thermal concentrators for residential application in Central Italy. An extensive monitoring of a solar cooling system was carried out in Forlì (Italy) in the summer of 2014 and 2015. The solar cooling plant was remotely monitored and data on environmental conditions, fluids flowrate and temperatures and both heating and cooling power production were continuously acquired and stored. The results showed that the combination of solar thermal concentrator and absorption chiller can simultaneously satisfy the demand for domestic hot water production and space cooling. Nevertheless, the solar thermal concentrator proved limited efficacy due to suboptimal environmental framework conditions and to low reliability. A preliminary economic analysis has been carried out for a residential application through the Net Present Value method. The Net Present Value has been computed for different effectiveness of the solar cooling plant in meeting the demand for domestic hot water production and space cooling. The results show that i) an investment reduction in comparison to the costs of the experimental plant and ii) incentives are fundamental to make the solar cooling plant profitable.

*Keywords: Solar cooling, parabolic dish concentrator, experimental measurements, performance analysis.*

## 1. Introduction

Cooling applications like air-conditioning and refrigeration become as basics of everyday life. The conventional cooling systems use refrigerants with high ozone depletion potential (ODP) and greenhouse gas (GHG) impact and as a consequence the prohibition of hydrochlorofluorocarbons (HCFCs) has been programmed starting by 2015 (Benhadid-Dib and Benzaoui, 2012). Therefore, innovative solutions for renewable energies development and exploitation are needed to meet human needs while saving the environment. Furthermore, the reduction of electric energy consumption for buildings cooling can contribute in the limitation of electric peak demand in summertime.

Exploring the so-called "*solar cooling*" concept seems a fascinating idea since cooling needs coincide most of the time with the solar radiation availability. The main alternative routes (Montagnino, 2017; Lazzarin and Noro, 2018) from solar energy into cooling (and heating) are solar "thermal" or "electric" (photovoltaic). One of the most promising plant configuration is the one that combines solar thermal devices with absorption refrigeration chillers (Eicker and Pietruschka, 2009). In fact, the use of solar thermal energy in summertime for different purposes from domestic hot water (DHW) can increase the efficiency and efficacy of the solar thermal plant (Mugnier et al., 2017). In fact, solar cooling could be an effective way to increase the annual solar fraction of DHW production and prevent the solar system from overheating. Indeed, solar thermal plant is usually sized to meet all the hot water requirements in summer to reduce waste heat, thus doing without relevant solar heat contributions in the rest of the year. The integration with a solar cooling plant allows to oversize the solar thermal plant, which would increase the solar fraction throughout the year, and to use the surplus to power an absorption chiller for space cooling in summer.

Thermally driven cooling equipment based on the integration between solar thermal collectors and absorption chillers was commercially available since 70's. Nevertheless, such equipment still does not penetrate the market, due to both economic and performance issues. Several experimental plants have been realized, usually in insulated regions and with flat plate or evacuated tube solar thermal collectors (Toure at al., 2012; Hang et al., 2014; Lecuona et al., 2015; Aliane et al., 2016), including also the development of simulation tools (Puglisi et al., 2015; Delac et al., 2018).

The main objective of this paper is to measure and to assess the performance of a solar cooling plant in Central Italy


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(or, more in general, Central-South Europe). The solar cooling plant is hosted in the applied research centre on renewable energy called "HEnergia" (HE), located in Forlì (Italy), and which was designed to host different kinds of renewable energy plants (Bianchini et al., 2016, Bianchini et al., 2017). The originality of the paper lies in the fact that i) solar energy is produced by a concentrating solar dish device and ii) the solar cooling plant is located in a tempered zone. Starting from the experimental data analysis, a preliminary economic analysis has been carried out for a residential application in central Italy. Similar studies can be found in literature, but limited to sunny areas, in particular southern Mediterranean areas (Bouhal et al., 2018; Mugnier and Seleme, 2015).

## 2. Material and Methods

### *2.1. The solar cooling plant and monitoring system*

In the spring of 2013 eight different photovoltaic (PV) systems, one hybrid photovoltaic/thermal (PV/T) solar system and one solar cooling plant were installed at HE. The solar cooling plant consists of one concentrating solar dish device (the Solar Beam, manufactured by Solartron-Ghibli, 11.5 kW th peak), one absorption chiller (WFC-SC5 manufactured by Yazaki, 17.5 kW fr in nominal conditions), two storage tanks (800 l volume for hot water produced by the solar dish, 500 l volume for the cooled water produced by the absorption chiller), one cooling tower and all the necessary components for water feeding and control (pumps, valves, instruments). Since the solar cooling plant fed the fan-coils circuit of HE laboratories with chilled water, the plant was integrated with a compression chiller to ensure supply continuity. The Piping and Instrumentation Diagram (P&ID) of the solar cooling plant is shown in Figure 1, while in Figure 2 a picture of the solar parabolic dish concentrator is shown. Table 1 and Table 2 summarize the main characteristics of the solar parabolic dish and of the absorption chiller, respectively.

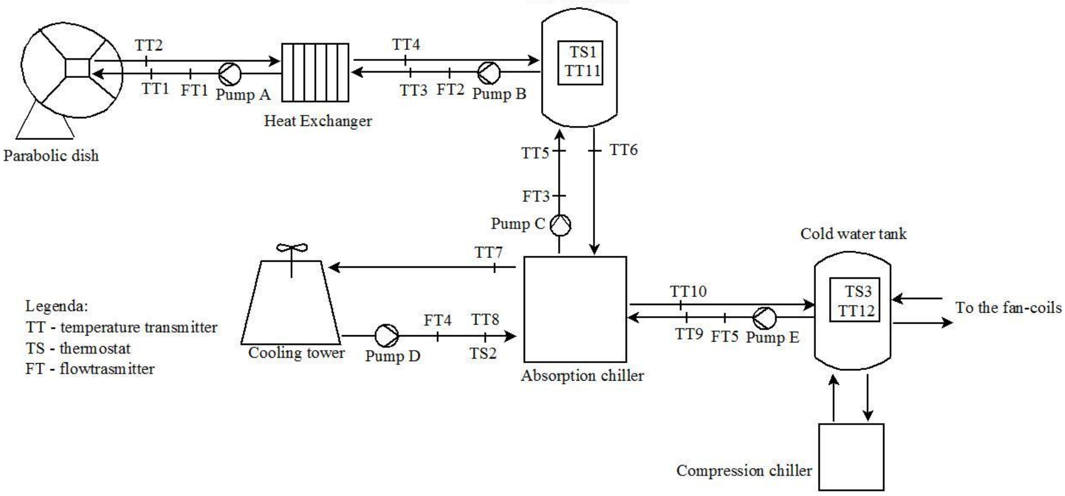


**Fig. 1: Solar cooling P&ID.**


**Fig. 2: Picture oft he solar parabolic dish concentrator (on the right).**

**Tab. 1: Main characteristics of the solar parabolic dish.**

| Item | Specification | Unit |
|---|---|---|
| Peak power production | 11.5 | kW |
| Optic efficiency | 0.86 | % |
| Peak beam thermal efficiency | 0.73 | % |
| Collector diameter | 4.5 | m |
| Gross collector area | 15.9 | $m^2$ |
| Gross absorber area | 0.065 | $m^2$ |
| Focal point distance | 2.2 | m |
| Mounting post height | 2.4 | m |
| Concentration ratio | 1:245 | - |
| Weight | 463 | kg |

**Tab. 2: Main characteristics of the absorption chiller.**

| Item | Specification | Unit |
|---|---|---|
| Cooling capacity at nominal condition | 17.5 | kW |
| Heating consumption at nominal condition | 25.1 | kW |
| Coefficient of Performance (COP) | 0.7 | - |
| Nominal inlet temperature of heating fluid | 88 | °C |
| Nominal outlet temperature of heating fluid | 83 | °C |
| Nominal flowrate of heating fluid | 1.2 | l/s |
| Nominal inlet temperature of cooled water | 12 | °C |
| Nominal outlet temperature of cooled water | 7 | °C |
| Nominal flowrate of cooled water | 0.77 | l/s |
| Nominal inlet temperature of cooling water | 31 | °C |
| Nominal outlet temperature of cooling water | 35 | °C |
| Nominal flowrate of cooling water | 2.55 | l/s |
| Size | 594 x 744 x 1,786 | mm |
| Weight (in operation) | 420 | kg |
| Noise level | 46 | dB(A) at 1 meter |

All the circuits are filled with water, with the exception of the primary circuit of the solar concentrator, which works with a 30% vol. mixture of water and propylenic glycol. All the pumps connected to the absorption chiller work at constant volume flow rate and have been set accordingly to nominal values (see Table 2). Table 3 summarizes the whole nominal installed electric power of the solar cooling plant.

**Tab. 3: Total installed electric power in the solar cooling plant. Pump items refer to Figure 1.**

| Item | Installed power [W] |
|---|---|
| Absorption chiller | 48 |
| Cooling tower fan | 550 |
| Pump A | 130 |
| Pump B | 410 |
| Pump C | 1,100 |
| Pump D | 1,100 |
| Pump E | 410 |
| **Total** | 3,748 |

The solar cooling plant is automatically driven and all the water flows (FT in Figure 1) and temperatures (TT in Figure 1) are measured to characterize each component performance. Furthermore, environmental conditions such as solar global radiation, ambient temperature and humidity, wind speed and direction are measured through a local meteorological station, installed on the HE roof. All the installed instruments have a maximum measuring error of 0.5%. The automatic operation logic of the solar cooling plant can be summarized as follows:

- Solar concentrator: the pump of the primary circuit (pump A in Figure 1) turns on when a solar radiation over 300 W/m$^2$ is measured, while the pump of the secondary circuit (pump B in Figure 1) is turned on if the temperature TT11 in the hot water storage tank is lower than the outlet temperature TT2 of the solar concentrator cooling fluid.
- Absorption chiller: the pump of the hot water circuit (pump C in Figure 1) turns on or off depending on the temperature measured by thermostat TS1. The temperatures levels at which the pump turns on and off can be remotely set.
- Cooling tower fan: turning on and off of the cooling tower fan is driven by the thermostat TS2, that measures the temperature of the cooling fluid at the absorption chiller inlet. TS2 is an internal instrument of the absorption chiller controller unit and was set with factory values.
- Compression chiller: the compression chiller works as a back-up or integration unit, depending on the end-user cooling demand and on the performance of the solar cooling plant. The compression chiller is turned on by the thermostat TS3 when the temperature in the cold water storage tank is too high, and it is turned off when a satisfactory temperature level is restored. Both temperature levels of thermostats can be remotely set.

The solar cooling plant has been realized for experimental purposes, and that's why some design aspects are not optimized as for a real case application. First of all, the matching between the solar concentrator and the absorption chiller is not optimal, since the peak production of the solar concentrator is lower than the nominal heating demand of the absorption chiller. Due to budget limitations, it was not possible to install one or two additional solar concentrators or other solar thermal devices.

## 2.2. Solar cooling plant performance indexes

The performance of the solar cooling plant is a combination of the efficiency of both solar device and absorption chiller, and it is strongly affected by the effectiveness of storage design and control strategy. The ISO 9806:2017 covers performance, durability and reliability testing of almost all solar thermal collector types available in the market, including concentrating solar thermal collectors. The ANSI/ARI Standard 560:2000 defines test and rating requirements of water-cooled single effect steam and hot water operated water chilling units, water-cooled double-effect steam and hot water operated water chilling units, and double-effect Direct-Fired water chilling/heating units.

The ISO 9806:2017 quasi-dynamic test method is normally applied to evaluate concentrating solar thermal collector performance, i.e. the heating power $P_{th}$. The quasi-dynamic method is basically the same as the steady-state model, but with some extra correction terms (including, among others, the impact of wind speed, dependence on direct and diffuse radiation, thermal capacitance) that make the mathematical formulation of $P_{th}$ complex and consequently complicate the evaluation of the impacts of the single parameter on the performance of the solar device. Nevertheless, $P_{th}$ [kW] produced by concentrating solar thermal collectors can be computed by Eq. 1, once mass flowrate $m_F$ [kg/s] and heat capacity $c_F$ [kJ/kgK] of the cooling fluid are known, and the solar thermal collector inlet $T_{in}$ and outlet $T_{out}$ temperatures of the cooling fluid [K] are measured.

$$P_{th} = m_F \cdot c_F \cdot (T_{out} - T_{in}) \quad \text{(eq. 1)}$$

Furthermore, the Reference Yield $Y_R$ has been identified as a relevant parameter to describe the environmental working condition. $Y_R$ is measured in Wh/W and it is defined as the ratio between the global solar radiation energy per surface unit I evaluated in the considered time interval (expressed in Wh/m$^2$) and $G_{STC}$, that is the Standard Test Condition (STC) global solar radiation (1,000 W/m$^2$, as in standard IEC 61724). This parameter is usually related to photovoltaic performance assessment, but it can be effectively adopted also in the field of solar thermal collector to indicate the number of peak sun-hours in a certain period (i.e. day, month, year).

The ANSI/ARI Standard 560:2000 defines the Coefficient of Performance (COP) as the ratio of cooling capacity [W] the power input values [W] at any given set of rating conditions. So, the COP of the absorption chiller can be computed once inlet and outlet temperatures and fluid flowrate in the generator and in the evaporator have been measured. The COP can be also referred to a certain period (i.e. day, month, year) and can be a useful instrument to measure the efficiency of the absorption chiller over the time.

## 2.3. Economics considerations

The evaluation of the economics of a solar cooling plant is not simple since the system can produce both hot and cold water over the year, and it is usually compared with electric fed devices. So, it is difficult to make a comparison based on the energy production costs (i.e. levelized cost of energy as for electric energy production). The economic analysis of the solar cooling plant studied was carried out through the application of the Net Present Value (NPV) methods in comparison with a reference system. The chosen reference system is a reversible heat pump for seasonal cooling and domestic hot water (DHW) production all over the year. The formula of NPV can be expressed as in Eq. 2:

$$\text{NPV} = -F_0 + \sum_{t=1}^{n} \frac{R-C}{(1+i)^t} \quad \text{(eq. 2)}$$

where t (years) is time, n (number of years) is the time period considered for the investment evaluation (which will be assumed equal to both depreciation and technical life time of plants for treatment simplicity purpose), *i* (%) is the discount rate (differentiated in household and industry investment), R is the annual revenue [€], C the annual cost [€]. The net cash flow $F_0$ at t=0 corresponds to the starting investment [€]: the simplifying hypotheses of full investment payment and plant operation start in the same year (t=0) are also assumed. The net cash flow F=(R-C) for period t>0 was computed taking into account the main costs and revenues components.

The investment has been initially computed similar to the one present in HE on the basis of real HE costs and on market quotations (Table 4): the result is an investment of about 70,000€. Nevertheless, it should be noted that at least two further solar concentrators should be installed to satisfy the demand of the absorption chiller, meaning that the real cost of an effective installation would be considerably higher, i.e. 120,000-130,000€. The result is an estimated investment cost of about 7,000-7,500 €/kWfr, that is quite high if compared with literature data (Eicker and Pietrusckha, 2009; Eicker et al., 2014). This fact can be justified i) by the small size of the plant, ii) by the experimental nature of the installation, and iii) by the high cost of the solar devices.

A yearly cost to be considered is the operating and maintenance cost, which is evaluated on the basis of the experimental campaign findings. In particular, maintenance cost has been computed to be around 1,000€ per year, and includes actions like cooling tower seasonal cleaning, parabolic dish cleaning, sun-tracking system maintenance, yearly check of pumps, valves and instruments. The yearly measured maintenance cost is consistent with literature data (Jakob, 2015) The decreasing cost of maintenance activities on the reference plant is not considered, since the reference plant can be considered as a back-up and integration system, and so no relevant cost decreasing can be expected. The yearly operation cost is substantially produced by electric consumption for pumping and for the feeding of the cooling tower fan. The total installed electric power is equal to about 3.8 kW (see Table 3). Nevertheless, the solar cooling plant is supposed to have a lower mean electric consumption, since the pumps consumption is not equal to the nominal installed power, while the cooling tower fan is not always working. The NPV will be computed by considering 1 kW of mean electric energy consumption, which has been estimated by considering the nominal pressure losses and a non-continuous operation of the cooling tower fan.

Yearly revenues are produced by the lower electric energy consumption of the reference system, and they are strongly affected by the effectiveness of the solar cooling plant, i.e. the capacity of satisfying the energy demand of the end-user for space cooling and DHW production. The NPV has been computed by assuming an increasing effectiveness (25%, 50%, 75% and 100%) of the solar cooling plant for both space cooling and DHW production. Electricity price varies considerably, depending also upon the country of installation, and so it can have a huge effect on the NPV computation. In this paper, since the solar cooling plant was realized in Italy, averaged figures are used, being representative for the Italian residential energy market. No feed-in tariff or tax deduction have been considered. Table 4 summarizes the main assumptions about economic assessment.

**Tab. 4: Assumptions about the economic assessment.**

| Item | Symbol | Value |
|---|---|---|
| Evaluation period | n | 20 years |
| Discount rate | *i* | 4.0% |
| Maintenance cost of the solar cooling plant | - | 1,000 €/year |
| Electricity price (Italy) | - | 0.229 €/kWh |
| Absorption chiller | $F_0$ | 18,620€ |
| Cooling tower | $F_0$ | 3,100€ |
| Parabolic dish solar concentrator | $F_0$ | 25,452€ |
| Other solar cooling components (including tanks, piping, valves, fittings, instruments, pumps) and installation | $F_0$ | 10,000€ |
| PLC and control software | $F_0$ | 6,000€ |
| Solar cooling plant design | $F_0$ | 6,500€ |

## 3. Results and discussion

### 3.1. Solar cooling plant performance

Figure 3 and Figure 4 represent an example of a daily data plotting; the reference day is 18th August 2014, that has been selected as a representative of clear sky and sunny day. Figure 3 shows solar global radiation G, ambient temperature $T_a$ and wind velocity variation throughout the day.

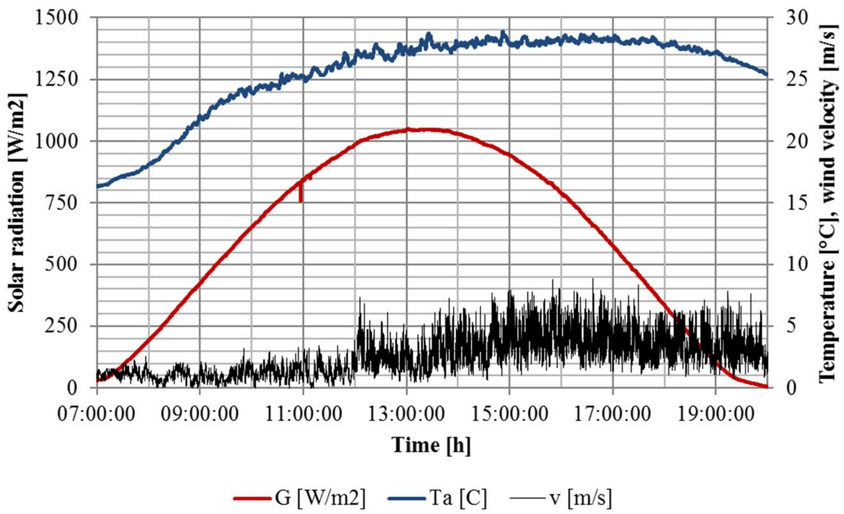


**Fig. 3: Example of daily available dataset regarding environmental conditions (global solar radiation, ambient temperature and wind velocity) – reference day 18th August 2014.**

Figure 4 and Figure 5 show an example of the daily performance of solar concentrator and absorption chiller: in fact, the measured thermal power $P_{th}$ and the outlet temperature $T_{out\ conc}$ produced by the solar concentrator, plus the measured cooling power $P_{fr}$, the heat consumed by the absorption chiller $P_{gen}$, the inlet hot water temperature (from the hot water tank), inlet cooling water temperature (from the cooling tower) and the outlet cooled water temperature are represented.

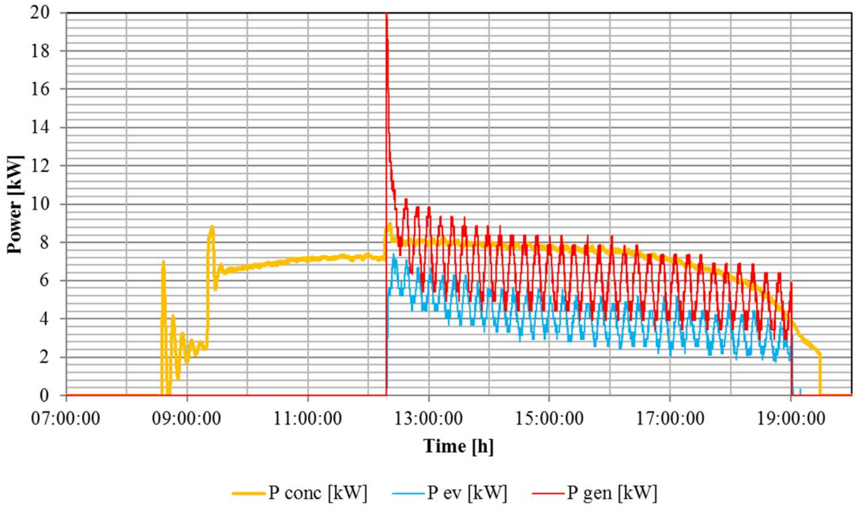


**Fig. 4: Example of daily available dataset regarding thermal energy flows produced by the solar cooling plant: heat produced by the solar concentrator (P conc), cooling power produced by the absorption chiller evaporator (P ev), heat consumed by the absorption chiller generator (P gen) – reference day 18th August 2014.**

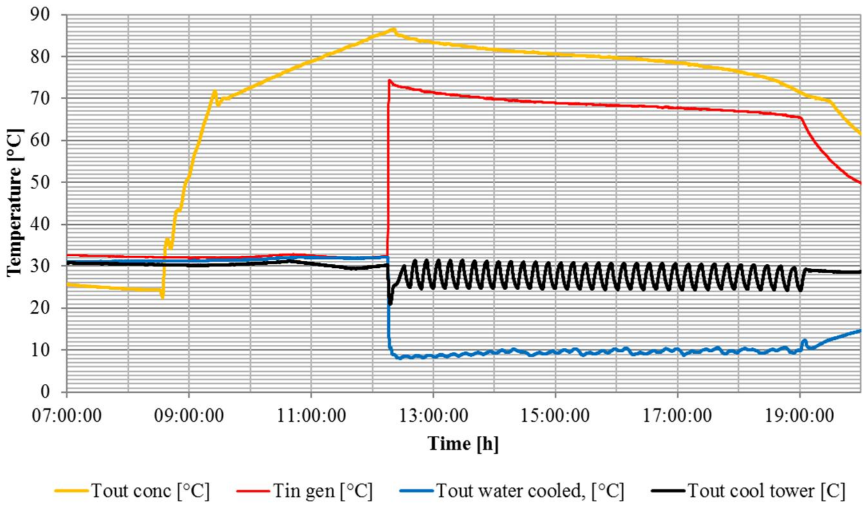


**Fig. 5: Example of daily available dataset regarding fluid temperatures of the solar cooling plant – reference day 18th August 2014.**

Figure 4 shows an interesting operation behavior of the absorption chiller, i.e. the oscillating production of cooled water and consumption of hot water. The same behavior was observed in all the monitored days. It is also relevant to note that both solar concentrator and absorption chiller are characterized by a full operation delay due to transient time: in the case of the solar concentrator the transient time is about one hour long, while the absorption chiller takes about 15 minutes to start operating. The cooling power expressed by the absorption chiller in Figure 4 is considerably lower than the one expected in nominal condition: nevertheless, the absorption chiller was still far from working at nominal condition, since the inlet temperature of the hot water in the generator (see Figure 5) is about 70°C (i.e. 18°C less than the nominal value). It should be underlined that the solar cooling plant, and the absorption chiller in particular, was tested with different inlet temperature of the hot water feeding the generator. Such an option was made possible due to the presence of the thermostat TS1 (see Figure 1), which allows the hot water to circulate in the generator only if a set temperature was reached. The experimental test carried out on 18th August 2014 was set at 75°C to turns on the circulating pump and 65°C to turns it off. Different set up were tested, and it was found that, due to the undersize of the solar concentrator device, an upper limit of about 80°C should be defined, since if the set of thermostat TS1 would set at higher temperature, the system would not be able to reach that temperature in the hot tank.

Another important comment is about the 8-10°C temperature difference measured between the solar concentrator outlet and at the generator inlet. The temperature difference is due to i) the presence of a heat exchanger between the solar concentrator and the hot water tank and ii) the volume of the hot water tank. The installation of the heat exchanger was justified by the experimental nature of the installation, that requires the highest versatility.

Nevertheless, the heat exchange penalizes the solar section efficacy, since the temperature that can be transferred to the hot water tank is lower than in the case of direct connection between the solar concentrator and the tank. Secondly, also the volume of the tank can negatively influence the solar section efficacy, since if the volume is oversized it may be difficult to satisfy the nominal temperature value at the generator inlet. So, the size of the tank is a key aspect in the design of the solar cooling plant. A possible solution to optimize the system is to foresee the opportunity to by-pass the hot water tank when the performance of the solar devices allows to reach high temperatures, or to develop sophisticated control strategies including variable-volume storage systems (Buonomano et al., 2014).

The oscillating operation of the absorption chiller (see Figure 4) can be explained if Figure 5 is analyzed, since the temperature at the cooling tower outlet shows the same behavior. The oscillating temperature at the cooling tower outlet is produced by the control strategy of the absorption chiller: when the temperature of the fluid coming from the cooling tower enters the absorption chiller at the temperature higher than 31°C, the absorption chiller control unit directly turns on the fan of the cooling tower through the thermostat TS2 (see Figure 1). The cooling tower is switched off once the temperature goes down at 25°C. The absorption chiller control unit can be accessed only through an external panel that was not included in the furniture. So, that control strategy cannot be modified. The result of this on/off control of the cooling tower fan is a wide variation of the cooling water temperature (between 25°C and 31°C) that highly influence the absorption chiller yield, especially at low inlet generator temperatures. In fact, a variation of about 6-7 $kW_{fr}$ can be observed in each oscillating period. Moreover, such a control strategy is the responsible of a high number of start/stop cycles of the cooling tower fan, which may have negative impact on the reliability of the system on the long term. Further analysis is needed to verify the relation (if any) between waves' frequency and amplitude of cooling tower outlet and of absorption chiller yield, and how this relation can affect the control system efficacy.

Different possible actions were identified to improve the control strategy: i) changing the controller strategy by adding an external thermostat, preferably very close to the cooling tower outlet to reduce the response time of the system, thus by-passing the absorption chiller internal control; ii) implementing the control strategy by adding an inverter to control the cooling tower fan rounds per minute; iii) adding a 3-ways thermostatic valve by-pass from the cooling tower inlet to the cooling tower outlet, controlled by the mixed temperature inlet to the absorption chiller, to minimize the temperature inlet to the absorption chiller. In the latter configuration, the cooling tower fan is always running. Further analysis is needed to verify the effectiveness and efficiency of such control strategies and will be included in a following paper.

The first phase of data analysis was carried out by focusing on the performance of the solar parabolic dish. The performance analysis of the solar device shows a high dependence of thermal yield on solar radiation and temperature difference between the cooling fluid and the ambient temperature (Bianchini et al., 2019). Figure 6 shows the monthly thermal yield production (i.e. the whole heat produced in one month per square meter of solar collector), including the monthly average outlet and mean temperature of the cooling fluid and the monthly average reference yield. Data about May and September 2015 are not taken into consideration due to limited number of days of thermal energy production caused by solar parabolic dish malfunctioning. Data about June and July 2015 are missing since a PV/T was tested instead of the solar thermal concentrator (Bianchini et al., 2017).

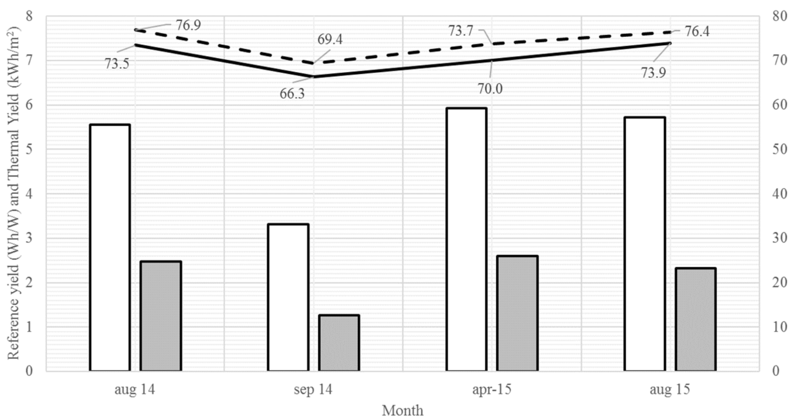


**Fig. 6: Average daily thermal yield per square meter for each month of the solar concentratot; also monthly average values of daily reference yield and outlet and mean temperature of the cooling fluid are included (adapted from Bianchini et al., 2019).**

Another important factor in the analysis of parabolic dish concentrator is the mean temperature $T_m$ of the cooling fluid [K]. In Figure 7 $T_m$ and $T_a$ are analyzed together taking into account global solar radiation: the values are averaged over 30 minutes. When ambient temperature $T_a$ rises up to 15 °C an increasing trend can be identified, after which a constant temperature $T_m$ seems to be achieved, independently from $T_a$. Therefore, the solar concentrator can produce hot water for the absorption chiller feeding in summertime, as well as hot water for DHW production in wintertime.

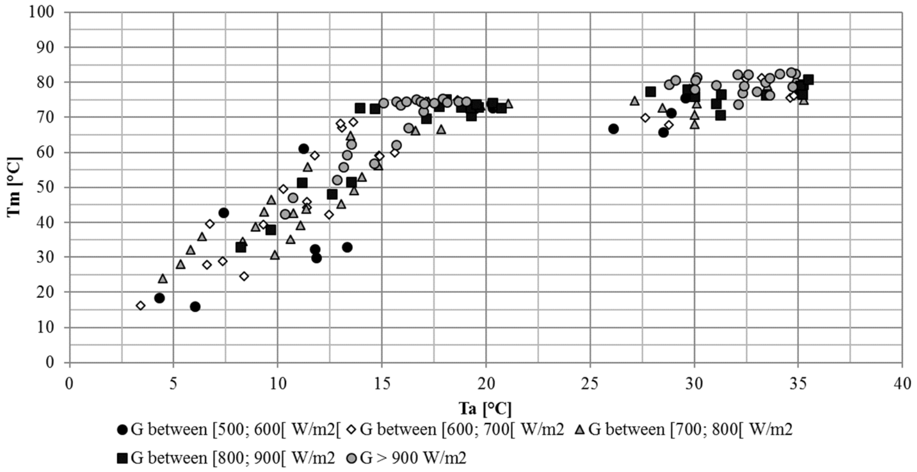


**Fig. 7: Correlation between mean temperature $T_m$ of the solar device cooling fluid and ambient temperature $T_a$ and solar global radiation G.**

The recorded working days of the parabolic dish is equal to 279 days: since the potential working days is equal to 301, it means that the parabolic dish didn't work for 22 days, which mean 7.3% of potential working days lost due to the reliability concerns (Bianchini et al., 2019). The main problems observed were: mechanical failure on the solar tracker (solved in the first months) and not stopped alarms that require manual cancellation, thus leaving the optical concentrator in the rest position until an operator manually restarts the plant on-site. These potential failures should be considered when comparing a solar thermal concentrator with a fixed solar thermal collector.

The recorded working days of the absorption chiller is equal to 46 days. The instantaneous performance of the absorption chiller is not easy to compute due to the oscillating operation of the system (see Figure 4). Data has been averaged over a period of 10 minutes. Figure 8 shows the instantaneous COP of the absorption chiller computed for different temperatures at the generator inlet. The presence of low COP values (i.e. under 0.6) can be justified by the fact that absorption chiller starts up are included in the analysis. It is interesting how over 70°C of inlet generator temperature the COP seems to stabilize around a mean value of 0.7, thus suggesting that the efficiency of the absorption chiller is not affected by inlet generator temperature variation up to about 20°C from nominal conditions. Once again, it seems appropriate to favor the most direct connection between the solar thermal source and the absorption chiller, since even significant variations in the inlet generator temperature do not penalize the efficiency of the system, but only the yield.

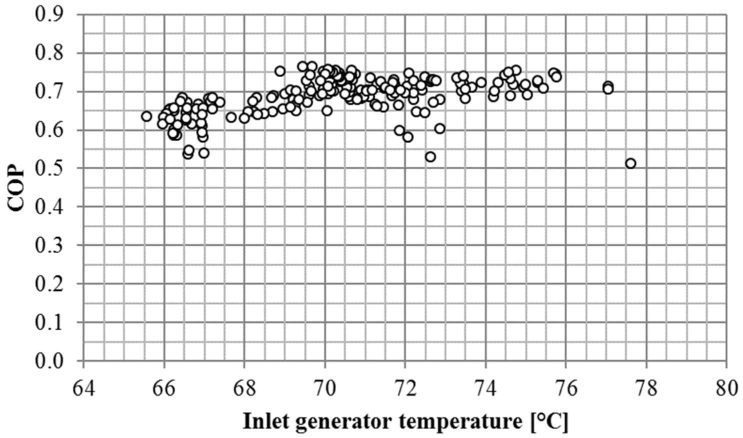


**Fig. 8: Correlation between absorption chiller COP and temperatures at the generator inlet.**

### *3.2. Economic assessment*

Due to the experimental nature of the installation, it was not possible to directly assess the techno-economic suitability of the solar cooling plant based only on the measurement carried out in HE. The paper aims to evaluate the sustainability of the investment in a solar cooling plant through a comparison with a reversible liquid-air heat pump – considered as the reference technology – for space cooling and DHW production. To do that, an end-user should be defined. A complex of 5 apartments of about 60 $m^2$ each of conditioned area can be considered as suitable for a 17 $kW_{fr}$ cooling plant, resulting in a yearly energy demand for space cooling of approximately 22,500 kWh (i.e. 1,500 cooling hours per year). In the hypothesis of about 15 people living in that residential complex, more or less 45 kWh/day of energy should be used for DHW production. The reversible heat pump seasonal efficiencies are estimated in a mean COP of 3.0 in heating mode and a mean energy efficiency ratio (EER) of about 3.0 in cooling mode.

Figure 9 shows the result of the economic analysis carried out by considering different effectiveness of the same plant in reaching the demand for DHW and cooling: the NPV has been computed by considering different level of savings produced by higher meeting demand for DHW and cooling. Based on the assumptions about the residential user characteristics, it is interesting to note that the solar cooling plant should be designed to reach as much as possible the space cooling demand, being the capacity of meeting DHW demand less impacting. Nevertheless, a solar cooling plant can hardly satisfy a percentage higher than 75% of space cooling demand in residential application, since cooling demand may be present also on late afternoon or night. So, the capacity to satisfy DHW production all over the year becomes essential, i.e. solar panels characteristics regarding hot water production in wintertime. The result is that a solar cooling plant can be feasible only if mutual benefits occur between energy production for DHW and space cooling.

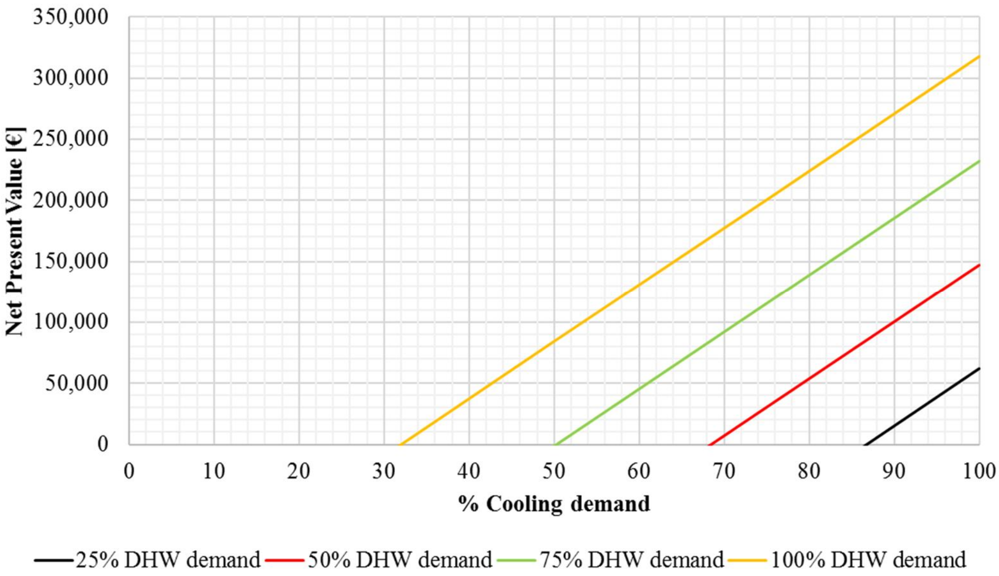


**Fig. 9: Net Present Value variation as a function of the effectiveness of the solar cooling plant in meeting the demand for DHW and space cooling.**

In an optimistic case of being able to satisfy 50% of space cooling and 75% of DHW demands, the NPV would be zero at 20 years. A way to increase the NPV is to reduce the initial investment and/or to increase annual revenues. The first objective can be reached by considering lower cost and/or higher performance solar thermal collectors. Other solar cooling plant costs seem to be not diminishing. A 35% reduction of solar collectors' costs would reduce the investment to about 5,900 €/kW $_{fr}$. In this condition, the NPV in the case 50% cooling and 75% DHW demands would increase up to 23.830€, with a pay-back time of 15 years. Revenues increasing can be achieved only through incentives and tax benefits. In Italy, the "*Conto Termico*" program pays for two years a variable amount of 0.39-0.43 c€ per kWh of thermal energy produced by solar thermal collectors with an area lower than 50 $m^2$ and that are coupled with an absorption chiller. The subsidies value is considerably lower if the solar thermal collector area is higher than 50 $m^2$. In our application, this means an annual revenue of about 10.000€ for two years. The combination of lower investment (35% reduction of solar thermal collectors) and of incentives reduces the pay-back time of the investment to 12 years (NPV near 44.000€).

## 4. Conclusion

The residential application of combined parabolic dish and absorption chiller to produce DHW and guarantee space cooling in central Italy can be effective if: i) 50% of space cooling and 75% DHW production demand are covered by the solar cooling plant, ii) a substantial reduction (up to 35%) in the cost of parabolic dish can be reached and iii) incentives are taken into consideration in the economic assessment.

So, the solar cooling plant can be more attractive from both technical and economic point of views if evacuated tube solar thermal collectors should be coupled with the absorption chiller, since it is fundamental to guarantee system reliability and to produce heat at relatively high temperature (over 70°C) also in cloudy and relatively cold days. Moreover, an effective design of the plant (i.e. storage tank size) and of the control strategy (with a particular focus on the cooling tower operation) are fundamental to gain the highest energy production from the sun. Nevertheless, solar cooling plant application in non-residential frameworks (i.e. office heating and cooling) can be more attractive, since heat and cold demand is usually contextual to sun presence. A further investigation is needed to verify the opportunity to use solar cooling plant for offices heating and cooling, including a system simulation and a sensitive analysis of the techno-economic assessment.

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