Solar cooling technologies

Review, performance analysis and application to case studies

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Abstract

The main solar cooling technologies currently investigated are solar electric cooling by means of a vapour compression cycle, absorption cooling and adsorption cooling. In this study, the performance, economic and environmental viability of the most promising solar cooling technologies for the coupling with photovoltaic-thermal panels are analysed for the case of a 4-star hotel located in Madrid. The analysis is based on the installation of 100 Ecomesh hybrid solar panels, patented by Endef Engineering, on the rooftop of the hotel. Four case studies are compared: initial case of the hotel, hotel with only the hybrid panels, installation of the hybrid panels coupled with a reversible heat pump and an absorption cooling system with the Ecomesh solar panels. The preferred option is the solar electric cooling system with the hybrid panels and the heat pump. The achieved annual saving is 27,909 € and the CO_2 emissions cut is about 58 tons of CO_2/year. Since PV panels are much cheaper than hybrid ones, two alternative options have been suggested with those. The option with the reversible heat pump has been also recommended in case of deciding for Ecomesh PV panels instead of the hybrid ones.

Keywords: *renewable energy; solar cooling; hybrid panel; F-Chart method; absorption; heat pump*

1 Introduction

The global energy demand is increasing due to the population growth and the industrialization process. Therefore, the indoor comfort demand is also increasing, which results in higher electricity consumption. According to the International Energy Agency (IEA) [1], the energy demand could increase by 35% from 2010 to 2035. An interesting alternative to reduce the peak electricity consumption is the use of waste heat or renewable energy sources.

The use of air-conditioning (AC) systems has experienced a noticeable growth in the last decades all over the world, mainly in residential and commercial buildings [2]. The energy consumption for these systems is estimated at 45% of the whole households and commercial buildings [3]. Thus, solar cooling technologies represent a unique opportunity, especially in southern countries such as Spain, due to the huge amount of solar radiation and the increasing cooling demand [4]. Moreover, the heat obtained from the solar panels can be used for cooling purposes in a refrigeration cycle during the summer, when the consumption of domestic hot water (DHW) is not as high as in winter.

Solar cooling systems can reduce the issues related to energy consumption and environmental impact caused by conventional AC systems [5]. Scientists have recently paid more attention to solar energy due to the extended development of solar photovoltaic (PV) technology, and consequently, lower cost.

The major advantage of solar cooling systems is the utilization of the high amount of heat generated in summer in a solar installation. This is the season of maximum energy generation but minimum demand of DHW and heating.

However, in some cases the performance of these systems is rather low. The real challenge is the selection of the most suitable and efficient technology that maximizes the use of solar energy to meet the energy demand [6].

EndeF Engineering S.L. develops a specific type of glazed photovoltaic-thermal (PVT) solar panels. The research on solar cooling technologies is especially interesting for the company in order to explore the suitability and profitability of coupling these panels with refrigeration cycles for solar cooling applications. Moreover, there are not enough data regarding PVT panels for solar cooling in the literature.

First of all, a study of the existing technologies that combine solar panels with cooling systems is conducted, paying special attention to cases with PVT panels since it is the main product developed and manufactured in EndeF Engineering, S.L. The most promising solar cooling technologies with PVT panels are analysed for the case of a 4-star hotel, used as reference system. A technical study including the description of the systems, yields and simulations is conducted. Then, a comparison of four different case studies for the hotel is undertaken, based on several performance parameters, environmental impact and a simplified economic analysis. In the discussion, two alternative cases are also suggested with the use of PV panels.

After this introduction, the basic concepts of each solar cooling technology and the state of the art are presented in section 2. In section 3, the case studies are presented as well as an explanation of the calculations used in the simulations. The results and discussion are provided in section 4, where the preferred solar cooling systems are also determined. At the end, after analysing the results obtained, conclusions and outlook with suitable recommendations are presented.

2 Solar Cooling: State of the art

Solar-driven cooling systems use electrical or thermal processes to convert solar radiation into cooling. Most of the suggested solar cooling technologies are able to reduce or eliminate the harmful effects of traditional equipment as well as allow relevant energy savings [7]. Ullah et al. [8] reviewed different absorption and adsorption systems to conclude that solar cooling technologies can achieve primary energy savings of 40 to 50%.

In 2011, around 750 solar cooling systems were installed worldwide, including small installations with a capacity lower than 20 kW [1]. Very large installations have already been completed or are under construction.

One remarkable example is the system at the headquarters of the CGD bank in Lisbon, Portugal, with 400 kW of cooling capacity and a collector field of 1560 m² [1].

Solar cooling systems may be classified into several different categories that also include subcategories, but this paper focuses on two groups that include the most important and extended technologies: solar electric and solar thermal cooling systems.

The first one is a PV-based system where solar energy converts to electrical energy and feeds the compressor of a heat pump (vapour compression cycle) for cooling in a similar way as conventional methods. The second one corresponds to heat-driven cooling technologies such as absorption powered by solar collectors [5].

Thermally-driven technologies can be also classified into two main groups: open sorption systems, also called open cycles (direct treatment of air temperature and humidity), and closed sorption systems or closed cycles [9]. In the latter, the most common sorption machines are absorption and adsorption chillers. Absorption systems can use water-lithium bromide (H_2O-LiBr) or ammonia-water (NH_3-H_2O) as working fluid pairs. Adsorption cycles mostly include the systems H_2O-silica gel or H_2O-zeolite.

Adsorption cooling has some disadvantages like the poor heat exchange between the solid adsorbent and the refrigerant or the intrinsic intermittence. However, despite their low values of Coefficient of Performance (COP), these systems can work with low-temperature heat input, around 60 to 70ºC [10]. This seems to be an advantage for solar-driven systems, especially if flat plate collectors (FPC) or, in general, non-concentrating ones are used. Nevertheless, adsorption systems are more expensive and bulkier than absorption ones [11].

A relevant comparison between thermal and electric cooling was performed by Hartmann et al. [12] for a small office building located in Freiburg and Madrid. One of the chillers was driven by PV modules whereas the other was an adsorption unit connected to advanced FPCs. Regarding both economical and primary energy savings, the PV system showed better performance. Otanicar et al. [13] studied the future prospects of solar cooling technologies in economic and environmental terms. Solar electric cooling systems were estimated to require the lowest capital investments by 2030 due to the high COPs of vapour compression cycles and the cost reduction forecast for PV technology.

More than 1000 solar cooling plants have been installed so far all over the world ranging from small- to large- scale units. More than 254 installations are located in Europe [7]. In the last decades, thermal systems cost has decreased due to the lower prices of FPCs and ETCs as well as those of sorption chillers. The specific cost of a solar system based on FPCs is around 150 to 200 €/m^2 while that of ETCs is in the range of 250 to 300 €/m^2. The cost of the collector is 50 to 80% of the overall system cost and depends on the operating temperature of the chiller to a high extent [9].

Solar cooling technologies present several economic barriers such as the high initial cost or the lack of subsidies in developing countries, as well as technological barriers since they are generally complex systems and there are only a few demonstration and pilot plants to assess the performance. Moreover, the standardisation of these systems prototypes is difficult due to the worldwide variability of climatic conditions. Tax reductions and other financial incentives would help to increase the use of these technologies [7].

2.1 Solar photovoltaic cooling systems

In solar electric cooling systems, PV solar panels are coupled with a conventional vapour compression cycle. The panels produce electricity from the solar radiation that is used to run the compressor of the system. In case of scarce solar radiation, the compression chiller can be powered by the grid [9].

Solar PV panels have lower efficiencies (usually between 15 and 18%), but the solar cooling system as a whole is simpler and involves few components compared to solar thermal systems [9]. The application of these systems was limited due to the high initial cost and the low efficiency of PV panels until recent years [10], [13], [14]. However, the PV modules price has recently undergone a large decrease and these systems have gained importance. In fact, PV air-conditioning (PVAC) systems are predicted to attract more attention in the future, taking into account the potential PV market and the massive cooling market [15].

The most common and promising PVAC systems are those connected to the electrical grid, the so called grid-connected PVAC systems, where the grid acts as a backup. When there is an excess of power from the PV system, it can be sent to the grid. Likewise, when the PV power is lower than what the compressor needs, power is supplied from the electrical grid [15].

It is important to note that the price of electricity for large-scale commercial buildings is much higher than in domestic ones. Thus, it is important to estimate the generation and the savings that these PV installations can bring for commercial buildings and decide for the better option before investment.[15].

For solar cooling applications, different refrigerants may be used in vapour compression cycles. Since chlorofluorocarbon (CFC) refrigerants such as R12 were prohibited due to their high Ozone Depletion Potential (ODP), hydrochlorofluorocarbons (HCFC) started to be used although they still have a relatively high ODP and were finally banned in 2010 [16]. Later on, hydrofluorocarbon (HFC) refrigerants such as R134a or R410A gained importance although they are greenhouse gases and have higher Global Warming Potential (GWP) than the former. Last generation low-GWP refrigerants include difluoromethane (R32), which is an HFC with zero ODP and a GWP of about one-third of that of R410A [17].

2.2 Solar absorption cooling systems

Absorption cooling is based on the physical or chemical affinity of two substances in different states. At low pressure and temperature, absorption of the refrigerant vapour takes place. At high pressure and temperature, the liquid absorbent releases the previously absorbed refrigerant. Before the absorption process takes place, the refrigerant needs to be evaporated since refrigerant vapour is absorbed. Then the cooling effect occurs when the water or ambient air provides heat to the evaporator [18], [19].

Absorption air-conditioning systems are compatible with solar energy since the heat input is required at temperatures that available FPCs can provide [19].

Nowadays, the dominating technology in the market of solar thermally-driven cooling systems is based on absorption [9]. It requires very low electric input and the physical dimensions of an absorption machine are smaller than those for adsorption machines of the same capacity, due to the high heat transfer coefficient of the absorbent [14].

Most of the solar cooling systems available in the market are based on the single-effect LiBr–H_2O absorption cycle provided by FPCs or evacuated tube collectors (ETC) [20].

The driving temperatures are between 80 and 100 ºC for water-cooled systems. For air-cooled systems, an increase of 30 K is needed [11]. The COP of these chillers ranges from 0.6 to 0.8. Syed et al. [21] studied an H_2O/LiBr absorption system with 49.9 m^2 of FPCs in Madrid. With generator temperatures of 65–90ºC and 35 kW of cooling capacity, the average collector efficiency was approximately 50%.

The double-effect absorption chiller was launched in 1956 [5], superposing an extra stage cycle to the single effect system [22]. These systems are only available for large cooling capacities of 100 kW and above [11]. Tierney [23] compared four different systems with 230 m^2 area of collector and summarised that the double-effect chiller coupled with a trough collector had the maximum potential savings (86%) among the studied systems for a 50 kW cooling load.

Non-concentrating FPCs or ETCs are able to achieve the required temperature for the generator.

Nevertheless, the COP is lower in comparison with other technologies such as multi-effect absorption systems, often driven by steam produced from concentrating solar collectors, which are generally more expensive [5]. These cycles utilise an additional generator and heat exchanger to desorb the refrigerant with lesser heat input [5].

Grossman [24] carried out a comparison between the single-effect, double-effect and triple-effect chillers for solar cooling applications, concluding that the economics of the systems are dominated by the solar system cost. FPCs are suitable for the single-effect cycle, whereas the multi-effect absorption cycles require high temperatures (above 85 ºC) that only ETC or concentrating-type collectors can provide. They are able to reach higher COPs but those collectors are also more expensive.

Regarding the working fluid pair, LiBr–H_2O systems present higher COPs than NH_3-H_2O ones at the same temperature range because H_2O has larger latent heat than NH_3. Moreover, the NH_3-H_2O cycle requires high temperature in the generator, in the range of 125 to 170 ºC [22]. Thus, it is not suitable for most of the solar applications. Only in case of using parabolic trough collectors, which are not convenient for the residential sector due to the high maintenance requirements [25].

2.3 Solar adsorption cooling systems

Adsorption consists of bonding a gas or another substance on a solid surface. The adsorbent (solid) and the refrigerant (gas) experience a surface interaction that can be physical or chemical. In a similar way as absorption cooling cycle, the cooling effect is achieved in the evaporator, where the refrigerant is vaporised at low pressure and temperature [26], [27].

Adsorption technology is not competitive in small- or medium-size solar cooling systems since power densities are much lower than those for absorption chillers. This makes it a expensive technology for solar cooling systems [14]. However, adsorption systems may be a suitable alternative to the absorption chillers, which still dominate the refrigerant market in Europe [28], when the hot water coming from the solar collectors is below 90ºC [29]. Nevertheless, the performance (COP) is lower than that of absorption systems.

Henning and Glaser [30] designed a pilot adsorption cooling system with silica gel-water working pair. This system was powered by the solar heat produced in vacuum tube collectors with a surface area of 170 m^2. They reported a COP between 0.2 and 0.3. El Fadar et al. [31] developed a solar adsorption system with a thermal sensible storage and two adsorbers, in order to overcome the intermittency of adsorption cycle. The system was tested in Tetouan (Morocco); it achieved a cycle COP of 0.43 and a cooling effect of 2515 kJ for a collector area of 0.8 m^2.

2.4 Solar cooling with PVT systems

Photovoltaic thermal (PVT) solar panels are those that can produce both electricity and heat from the incident solar radiation. They include a PV laminate at the front and a thermal absorber and collector tubes at the back, optimising the surface of the collector as it is critical in cases with scarce space on the rooftop of a building, e.g. in city centres.

PVT panels have the advantage of allowing a temperature decrease in the module and PV cells due to the thermal contribution that leads to a continuous heat removal. Therefore, the photovoltaic efficiency is increased since the thermal losses are reduced. This is one of the reasons why these panels were initially developed and further investigated.

Ecomesh PVT panels subject to analyse in this paper, are glazed collectors that include a Transparent Insulating Cover (TIC), which encloses a neutral gas (Argon) to enhance the thermal insulation and therefore, the heat production. This is done because the thermal efficiency of a PVT panel is always lower than that of a thermal collector due to the PV laminate. However, it leads to additional optical interfaces and reflections providing a lower electrical yield [32].

M. Alobaid et al. [4] analysed a solar driven absorption cooling system with PVT panels. They reported that 50% of primary energy was saved and the maximum electrical efficiency of PVT panels achieved was in the range of 10-35%. In this study, several systems with PVT panels were also presented as possible options with different performances. The thermal efficiency of a combined system with PVT panels and a LiBr-H_2O single effect absorption chiller is in the range of 23-35% for a collector area of 30-70 m^2. Another system with PVT panels and a heat pump was also presented, with a thermal efficiency of 15-21% for an area of 70 m^2 and a photovoltaic efficiency in the range of 15-17%. [4].

3 TECHNICAL ANALYSIS

3.1 Installation in a hotel. Description of the system

This study focuses on Ecomesh PVT solar panels, patented by the company EndeF Engineering, in combination with the main cooling technologies: vapour compression cycle and absorption. Different solar cooling configurations are analysed for the same application: a 4-star hotel with a capacity of 100 guests located in Madrid, Spain. Some relevant data for Madrid and the solar panels installation are presented in Table 1.

Table 1. Data for the solar installation in Madrid.

Latitude (φ)	40.4º
Longitude	3.7 W
Tilt angle (β)	40º
Orientation (θ)	0º (south)
Active area of collector (A_c)	1.56 m^2

The thermal and electrical specifications of the Ecomesh PVT panels are presented in Figure 3:

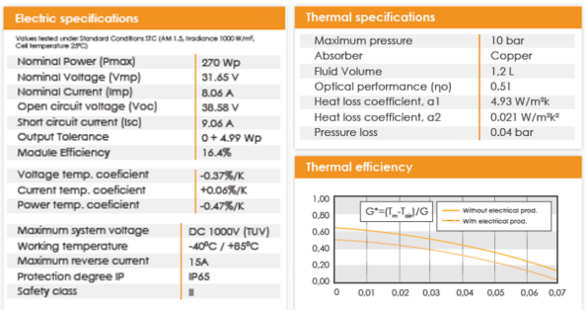

Figure 1. Extract of the technical sheet of the Ecomesh PVT panel.

The installation consists of 100 PVT panels, Ecomesh, on the rooftop of the hotel building. Two different options for the cooling cycle are analysed: first, a reversible heat pump (vapour compression cycle), which will generate heat during the winter for the heating system in the hotel and cooling in the summer for air conditioning; second, a single-effect LiBr-H_2O absorption chiller. As already explained, adsorption cooling systems present much lower performance than absorption units. This is why only the latter is considered in this study as solar thermal cooling technology.

3.2 The hotel case studies

The present work is based on the analysis and comparison of four different case studies for the hotel installation:

1) The hotel energy needs are fully covered by the common electrical grid and a gas boiler. This is the initial situation of the 4-star hotel considered.
2) The hotel is provided by 100 Ecomesh PVT panels. They supply heat to obtain DHW as well as electricity for the use in the hotel.

3) The hotel is provided by 100 Ecomesh PVT panels and an air-water reversible heat pump that works on heating mode during the winter and cooling mode in the summer. The panels are directly connected to the heat pump and provide the electricity needed to make it work. The thermal part of the hybrid panels generates heat to provide DHW for the hotel.
4) The hotel is provided by 100 Ecomesh PVT panels and an absorption chiller that only works during the summer to provide air conditioning. The electricity from the panels is used in the internal grid of the hotel during the whole year. In the winter, the thermal energy from the panels is used to produce DHW whereas during the summer months, this energy feeds the absorption unit, providing heat to the generator of refrigerant. However, an additional boiler is required in order to assure that the hot water that runs the cycle is at 88ºC (temperature at the generator of the absorption unit). During the summer, a conventional boiler of the hotel provides the DHW.

Winter has been considered to last from October until March, both included, and summer from June until September, also included. The spring months of April and May do not account for heating or cooling needs.

3.3 Performance indicators

3.3.1 Efficiency indicators

Solar performance:

- Photovoltaic part:

The photovoltaic annual performance, η_{PV} , is defined as follows:

$$\eta_{PV} = \frac{P_{DC}}{G \cdot A} \tag{1}$$

Where,

P_{DC} = total power produced in a year [W].
G = total solar irradiance in a year [W/m^2].
A = total surface area of panels exposed to the sun [m^2].

- Thermal part:

The thermal annual performance, η_t, is defined by the following expression:

$$\eta_t = \frac{Q_u}{E \cdot A} \tag{3}$$

Where,

Q_u = useful heat obtained from the solar collectors [MJ].
E = total irradiation on a tilted surface in a year [MJ/m^2].
A = total active area of collectors [m^2].

The efficiency of a thermal solar collector is defined by the following expression:

$$\eta_t = \eta o - a_1 \cdot \frac{Tm - Tamb}{G} - a_2 \cdot \frac{(Tm - Tamb)^2}{G} \tag{4}$$

Where ηo (optical efficiency), a_1 and a_2 (thermal losses coefficients) are thermal parameters defined for the particular solar collector. In this case, these values can be found in the technical sheet of the Ecomesh PVT panels. T_m is the mean operation temperature of the panel and T_{amb} the actual ambient temperature.

Solar fraction:

It defines the portion of solar energy contribution compared to the total energy required for the solar cooling system since an auxiliary energy source is usually needed.

- Photovoltaic part:

The solar fraction of solar electric cooling systems, SF_e, is defined as follows [33]:

$$SF_e = \frac{P_s}{P_s + P_{aux}} \tag{5}$$

Being P_s [kW] the solar electric power gain from the panels (corresponds to P_{AC} of a PV panel) and P_{aux} [kW] the auxiliary electrical power from the public grid.

- Thermal part:

This parameter measures the percentage of the energy demand covered by the solar collectors in a solar thermal installation or in this case, by the PVT panels.

Generally, the solar fraction of solar thermal cooling systems, SF_t, is defined by this expression:

$$SF_t = \frac{\dot{Q}_u}{\dot{Q}_u + \dot{Q}_{aux}} \tag{6}$$

Being $\dot{Q}_u$ [kW] the solar thermal power gain from the collectors and $\dot{Q}_{aux}$ [kW] the heat power required from an auxiliary source [33].

Energy Efficiency Ratio (EER):

It is a measure of the efficiency of cooling/refrigeration technologies such as vapour compression cycles and absorption chillers.

- Vapour compression cycle:

$$EER = \dot{Q}_r / P_c \tag{7}$$

Where,

$\dot{Q}_r$ = cooling or refrigeration power demand [kW].
P_c = power of the compressor [kW].

- Absorption chiller:

$$EER = \dot{Q}_r / \dot{Q}_g \tag{8}$$

Where,

$\dot{Q}_g$ = heat power supplied to the generator [kW].

The COP is similar to the EER but it is normally used for heating systems. Therefore, in this paper it is calculated for the winter mode of the reversible heat pump cycle, with $\dot{Q}_h$ instead of $\dot{Q}_r$.

Seasonal Performance Factor (SPF):

This parameter is a measure of the efficiency of heat pump systems over a year. Since the COP can vary a lot throughout the year, the operating performance of an electrical heat pump is better defined by the SPF. It includes the parasitic consumption of all the auxiliary equipment and allows to compare different heat pumps.

$$SPF = \frac{Total\ annual\ heat\ output}{Total\ annual\ input\ of\ electricity} \tag{9}$$

In this case, the heat pump is reversible and the COP is calculated for the winter mode whereas the EER is obtained for the summer, both refer to its performance.

However, the SPF can be also calculated to know if the heat pump is considered as a renewable energy source according to the European Directive 2009/28/EC (art. 5, Annex VII).

For this purpose, there is another expression that depends on the nominal COP and two coefficients, FP and FC:

$$SPF = COP_{nominal} \cdot FP \cdot FC \tag{10}$$

The FP coefficient depends on the climatic conditions in the particular location of study as well as the type of heat pump and installation.

The FC coefficient measures the difference between the distribution temperature and the testing temperature at which the nominal COP has been estimated. The heat pump is considered renewable when the SPF is higher than 2.5 [34].

Primary Energy Consumption (PEC):

In order to calculate the primary energy consumption (non-renewable), two conventional energy sources are considered: electricity and gas. Their energy efficiencies are presented below [35]:

- Energy efficiency for primary energy-electricity conversion: ε_{el} = 40%,
- Energy efficiency for primary energy (gas)-thermal energy conversion: ε_g = 90%.

Therefore, for solar electric cooling systems [33]:

$$PEC_{elect} = (W_{aux} + W_{parasitic})/\varepsilon_{el} \tag{11}$$

Where,

W_{aux} = auxiliary electrical energy from the common grid [kWh].
$W_{parasitic}$ = electrical energy consumption of parasitic equipment (valves, pumps, etc) [kWh].

For absorption cooling systems [33]:

$$PEC_{thermal} = Q_{aux}/\varepsilon_g + W_{parasitic}/\varepsilon_{el} \tag{12}$$

Where,

Q_{aux} = heat provided by the auxiliary (gas) heater [kWh].

Primary Energy Ratio (PER):

This parameter relates the energy output to the primary energy consumption as follows [36]:

$$PER = \frac{Q_{out}}{PEC} \tag{13}$$

Where,

Q_{out} = total energy output, usable energy [kWh].

3.3.2 Economic indicators

Capital cost or investment cost:

It refers to the total initial cost of the solar cooling installation subject to study.

Operational costs:

These refer to the variable costs, the ones derived from the energy consumption from conventional energy sources (fossil fuels), which are the electrical common grid and the natural gas boiler previously installed in the hotel. These sources provide auxiliary energy to the solar cooling installation.

Simple Payback Period (SPP):

It measures the time needed to recover the extra costs over a conventional system. The SPP does not consider the interest rate for the investment neither the energy prices inflation [37]. It is calculated as follows:

$$SPP = \Delta C_{inv}/\Delta C_{op} \tag{14}$$

Where,

ΔC_{inv} = extra costs over a conventional system [€].
ΔC_{op} = annual reduction of energy costs [€].

Cost of primary energy saved:

This parameter is defined by the formula [37]:

$$CPE,saved = \Delta C_{a,s}/PE_{saved} \tag{15}$$

Where,

$\Delta C_{a,s}$ = annual extra costs for solar technology [€].
PE_{saved} = primary energy saved [kWh_{PE}].

Net Present Value (NPV) and Internal Rate of Return (IRR):

These two parameters determine the economic viability of a project. Even though this is a preliminary study of solar cooling technologies, these values may be useful for the economic comparison of the case studies. The NPV should be positive for the project to be profitable. It is calculated from the following formula:

$$NPV = \sum_{i=1}^{n} \frac{CFi}{(1+r)^i} - I_0 \tag{16}$$

Where,

CF_i = cash flow of every year [€]. It is the net annual energy saving minus the maintenance cost (500 €).
r = interest rate (%). For this kind of installations, it is 2%.
n = number of years in operation, life time of the installation.
I_0 = initial investment cost [€].

The Internal Rate of Return (IRR) corresponds to the interest rate when the NPV is zero. For the project to be economically feasible, the IRR should be higher than the interest rate used to obtain the NPV.

3.3.3 Environmental indicators

CO_2 emissions cut:

It is the amount of CO_2 avoided when renewable energy sources are used for cooling (or heating) instead of conventional systems based on fossil fuels combustion.

Global Warming Potential (GWP):

This is an indicator of the contribution of certain greenhouse gases to the global warming effect. It is mainly measured for several refrigerants employed in vapour compression cycles [7].

Ozone Depletion Potential (ODP):

This indicator measures the impact of the degradation that a chemical compound can cause to the ozone layer [7]. The refrigerants used nowadays have low ODP values since previously used CFC and HCFC refrigerants have been banned in the EU for more than eight years due to the big damage of the ozone layer.

3.4 Simulation procedure

The software used for the simulation of the systems is Engineering Equation Solver (EES). This program can solve non-linear, differential and integral equations and includes a highly accurate thermodynamic and transport property database for a wide range of substances.

3.4.1 Data and assumptions

- The number of Ecomesh hybrid panels has been assumed as 100, regarding the capacity of the hotel and its standard.
- The cooling capacity or cooling power demand of the hotel is considered as $\dot{Q}_r = 64\ kW$. The heating demand of the hotel during the winter is $\dot{Q}_h = 72\ kW$ (data provided by the company EndeF Engineering).
- The thermal demand of DHW has been obtained from the consumption of water per person in a 4-star hotel in Spain, given in L in [38].

3.4.2 *Calculations and methods*

- The solar system:

Photovoltaic part:

The calculations for the photovoltaic part of the PVT panels have been made in an hourly basis since the photovoltaic dimensioning requires the global hourly radiation on a tilted surface [39].

In order to obtain the photovoltaic efficiency, the maximum electrical power provided by the PVT panels must be calculated by applying correction factors to the values of power for a conventional PV panel. These factors have been obtained from the values of energy production provided by the company EndeF Engineering for both PV and PVT panels in Madrid for every month of the year.

Thermal part:

The thermal calculations for the 4-star hotel have been carried out in a monthly basis. This allows to apply the F-Chart method, which is recommended for long time periods as considered here (yearly production).

The F-Chart method uses monthly average meteorological data to obtain the solar fraction, which is the percentage of the total thermal demand that is covered by solar energy, as well as the average yield in a long time period. The main equation used in the method is:

$$f = 1.029D_1 - 0.065D_2 - 0.245D_1^2 + 0.0018D_2^2 + 0.0215D_1^3 \quad (17)$$

Where f is the monthly fraction of the thermal load that is supplied from the PVT panels. Coefficient D_1 represents the ratio of the absorbed energy by the collector to the monthly thermal demand. Coefficient D_2 represents the ratio of the energetic losses in the collector to the monthly thermal demand.

The useful heat can be easily calculated from this expression:

$$Q_u = f \cdot Q_{DHW} \quad (18)$$

Thus, the annual solar fraction is obtained from this formula:

$$F = \frac{\sum_{i=1}^{i=12} Qu,i}{\sum_{i=1}^{i=12} Q_{DHW},i} \quad (19)$$

- The cooling system:

Vapour compression cycle

For real vapour compression cycles, the pressure drop in both heat exchangers and the connecting piping needs to be considered. Furthermore, the refrigerant leaving the evaporator and the condenser is superheated and subcooled, respectively [40].

These parameters are intrinsic and specific of the particular system or installation subject to study in a real case and different from other real systems or configurations. Therefore, since the given work focuses on a preliminary study for a real case of a 4-star hotel but the cooling installations do not exist yet, they are just suggested at this step, those parameters needed to know the degree of subcooling and superheating as well as the pressure drop are unknown. The approach for the simulation of a real system has been to apply a coefficient of 0.8 (correction factor) to the COP or EER of the ideal system obtained with EES.

In the following, the steps and calculations of an ideal vapour compression cycle are explained. In order to simulate the ideal cycle, it is necessary to declare some thermodynamic variables [16]:

- Evaporator temperature: T_{evap} = 2ºC
- Condenser temperature: T_{cond} = 45ºC

In the evaporator, heat is removed from the space or room that needs to be cooled. The mass flow rate of refrigerant, $\dot{m}_r$, necessary for a heat transfer rate of $\dot{Q}_r$ is defined as follows:

$$\dot{m}_r = \dot{Q}_r/(h_{\text{ve}} - h_{lc}) \quad (20)$$

Where h is the specific enthalpy and subscripts v and l refer to vapour and liquid states, respectively; c and e refer to condenser and evaporator pressures, respectively.

After the vaporisation process, the compression of the refrigerant from pressure P_e (evaporator pressure) to P_c' (condensing pressure) takes place. It requires work input from an external source. In general, the work of compression $\dot{W}_c$ is obtained from the expression [19]:

$$\dot{W}_c = \dot{m}_r\,(h_{\text{vd}} - h_{\text{ve}}) \quad (21)$$

Where h_{vd} is the specific enthalpy of superheated vapour. The compressor is usually assumed to be isentropic, when considering an ideal cycle.

The next step is the condensation of refrigerant. First, in the subprocess IVa, the vapour is cooled removing the sensible heat, at constant pressure from T_d to T_c. At T_c', the vapour is condensed at the saturation pressure and latent heat is removed:

$$\dot{Q}_c = \dot{m}_r(h_{\text{vd}} - h_{lc}) \quad (22)$$

This heat must be rejected into the environment, either to cooling water or to the atmosphere.

After the condenser, the hot liquid refrigerant expands in an isenthalpic way reducing its pressure to the evaporator pressure and the whole cycle starts again [19].

The overall performance of a cooling/refrigeration machine is usually expressed by means of the Energy Efficiency Ratio (EER) [19]:

$$EER = \dot{Q}_r / \dot{W}_c = (h_{\text{ve}} - h_{lc})/\,(h_{\text{vd}} - h_{\text{ve}}) \quad (23)$$

The maximum value of the EER for any given evaporator and condenser temperatures corresponds to that of the reversible Carnot cycle for the same system.

$$EER_{Carnot} = \frac{T_e}{T_d - T_e} \quad (24)$$

In real systems, frictional effects and irreversible heat losses reduce the performance much below this value [19].

Most of the commercially available reversible heat pumps that have been checked for the required capacity use conventional refrigerants such as R134a or R410A, which are commonly used in air-conditioning systems.

The ones that have been recently investigated (for being more environmentally friendly) for these kind of applications are not available yet for an installation of this size. Therefore, a commercial model of an air-water reversible heat pump that would be suitable for the studied system is suggested and considered for the economic analysis: ROOFTOP KRB-W 075, by Kosner.

Absorption cycle

In this case, as well as in the vapour compression cycle, the approach for a real system is based on the correction of an ideal cycle with a factor of 0.8.

For the simulation of the ideal cycle, it is necessary to declare the following thermodynamic variables [19]:

- Evaporator temperature: T_{evap} = 3ºC
- Absorber temperature: T_{ab} = 34ºC
- Condenser temperature: T_{cond} = 36ºC

- Generator temperature: T_{gen} = 88ºC

Figure 2 presents the ideal LiBr-H_2O absorption cycle subject to analysis. In the following, the main steps of the simulation of the system are explained.

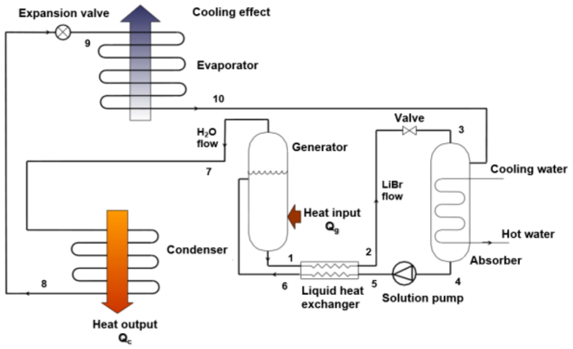

Figure 2. LiBr-H_2O absorption cooling cycle.

The first step is to obtain the high pressure and low pressure levels of the system. Then the corresponding temperatures, enthalpies and mass compositions can be sequentially calculated in EES.

The mass flow rate of refrigerant (water) is calculated from the equation:

$$\dot{Q}_r = \dot{m}_r \cdot (h_{10} - h_9) \tag{25}$$

And since $\dot{m}_r = \dot{m}_7$, the following mass balances can be easily solved:

$$\dot{m}_6 = \dot{m}_1 + \dot{m}_7 \tag{26}$$

$$\dot{m}_6 \cdot x_s = \dot{m}_1 \cdot x_{ab} \tag{27}$$

Being xs the concentration (in mass) of LiBr in the refrigerant-absorbent solution and $x$ab the mass fraction of LiBr in the absorbent solution, previously obtained.

The enthalpy of the solution at point 6 can be obtained from an energy balance in the heat exchanger:

$$\dot{m}_5 \cdot h_5 + \dot{m}_1 \cdot h_1 = \dot{m}_2 \cdot h_2 + \dot{m}_6 \cdot h_6 \tag{28}$$

Knowing that $\dot{m}_5 = \dot{m}_6$ and $\dot{m}_1 = \dot{m}_2$.

The heat rate that must be provided at the generator is calculated from Eq. (29):

$$\dot{Q}_g = \dot{m}_7 \cdot h_7 + \dot{m}_1 \cdot h_1 - \dot{m}_6 \cdot h_6 \tag{29}$$

The heat transfer rate at the condenser, which represents the heat rejected to the environment, is:

$$\dot{Q}_c = \dot{m}_7 \, (h_7 - h_8) \tag{30}$$

The energy efficiency ratio is calculated as follows:

$$EER = \dot{Q}_r \,/\, \dot{Q}_g \tag{31}$$

The commercial model of absorption chiller Yazaki WFC SC20 is selected according to the requirements of the installation and the technical specifications of these machines available in the market.

The values of the EER and COP for the real cycles calculated with a correction factor applied to the ideal values are presented in Table 2 for the reversible heat pump (two modes) and the absorption unit:

Table 2. EER and COP values of the real cycles subject to analysis.

Thermodynamic cycle	EER or COP
Heat pump, winter mode	4.48
Heat pump, summer mode	3.68
Absorption unit	0.63

4 RESULTS AND DISCUSSION

The results of this study are presented for the different configurations of solar cooling installations in the four case studies that are compared:

1) Hotel fully provided with gas and the common electrical grid
2) Hotel with Ecomesh hybrid solar panels
3) Hotel with Ecomesh hybrid solar panels and a reversible heat pump
4) Hotel with Ecomesh hybrid solar panels and absorption cooling

The motivation of this study is, among other aspects, the high electricity consumption in 4-star hotels, especially during the summer months, and the coincidence (in time) of this increase with the decrease in the DHW consumption. This may be observed in Figure 3 and 6, which present the electricity consumption and hot water demand of the hotel in a monthly basis.

The thermal and electrical contributions from the Ecomesh PVT panels are obtained with EES and Excel, respectively. Figure 5 presents both the electrical and thermal energy productions from the panels for every month of the year, for cases 2 and 3.

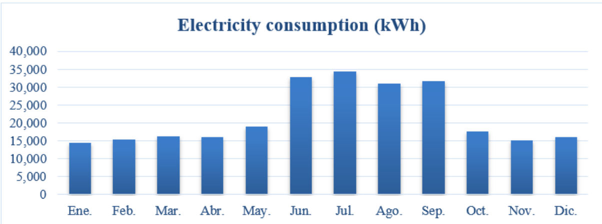

Figure 3. Monthly electricity consumption [kWh] in the hotel.

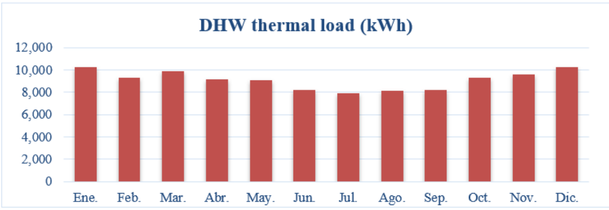

Figure 4. Domestic hot water thermal demand [kWh] along the year.

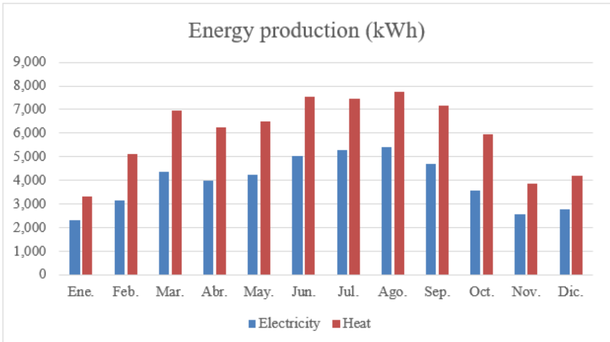

Figure 5. Energy production [kWh] from Ecomesh PVT panels.

A difference of 57% may be observed in the energy production (both thermal and electrical) between January and August.

For the fourth case with the absorption chiller, the thermal demand is much higher. Therefore, a different simulation is carried out in EES. The electrical contribution remains the same.

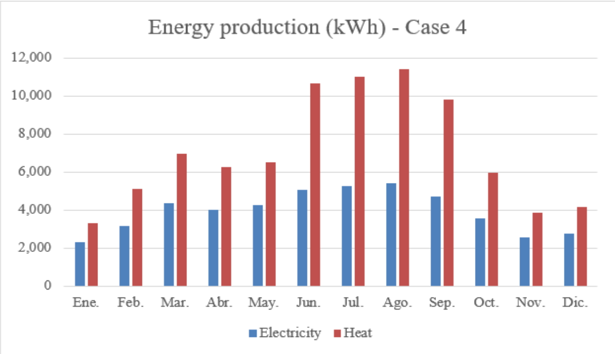

Figure 6. Energy production [kWh] from Ecomesh PVT panels in case 4.

The results obtained for the efficiency of the PVT panels and the solar fraction in each case study are presented in Table 3.

Table 3. Efficiency of the PVT panels and solar fraction for every case study.

		Efficiency (%)		**Solar fraction (%)**	
		Winter mode	**Summer mode**	**Winter mode**	**Summer mode**
CASE 2	**Thermal**	23.14		65.70	
	Electrical	16.00		18.20	
CASE 3	**Thermal**	23.14		65.70	
	Electrical	16.00		17.36	18.85
CASE 4	**Thermal**	23.96	31.66	54.54	34.49
	Electrical	16.00		21.80	

As it is expected, the thermal efficiency is generally lower than that of FPCs whereas the electrical is higher than that of PV panels. The results obtained are consistent with the literature review and there is a compromise between electricity and heat generation regarding the higher thermal insulation provided by the TIC [32]. In case 4, the thermal efficiency is quite higher during the summer since the demand is much higher than in other cases. Moreover, in the F-Chart method the energy production is very much influenced by the energy demand. The electrical solar fraction is the highest among all the studied cases because the demand for electricity is the lowest since there is no electrical AC equipment, but a thermally driven absorption chiller.

The SPF may be obtained in order to figure out if the installation can be considered renewable according to the Spanish and European regulations:

$$SPF = COP_{nominal} \cdot FP \cdot FC = 4.41 * 0.86 * 0.77 = 2.92$$

Since the obtained value, 2.92, is higher than 2.5, this heat pump can be considered renewable according to Directives 2009/28/EC and 2013/114/EU [41].

The efficiency indicators related to the use of primary energy are presented in Table 4.

Table 4. Annual PEC and PER for each of the case studies.

	CASE 1	**CASE 2**	**CASE 3**	**CASE 4**
PEC (kWh)	916,672	718,298	581,627	734,917
PER	0.584	0.745	0.888	0.728

The system that consumes the highest amount of primary energy, after the "base" case is the absorption cooling system of the fourth case study. It also has the second lowest primary energy ratio. Case 3 presents the most favourable results, with the lowest value of PEC and the highest PER. The results of PEC divided in thermal and electrical contributions are shown below:

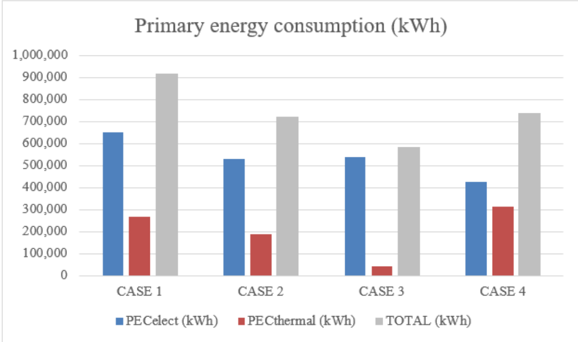

Figure 7. Comparison of the PEC [kWh] of the 4 case studies.

4.1 Economic comparison

The annual energy savings for the three case studies suggested as an alternative to case 1 are described in Figure 8.

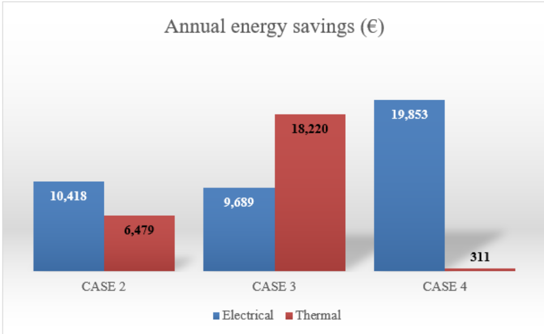

Figure 8. Comparison of the electrical and thermal annual energy savings.

It can be noticed that the electrical savings are higher in cases 2 and 4, being the fourth the highest value. However, this is probably due to the electrical space heating in case 3. Globally, the total annual saving of the third case study is the highest, considering both thermal and electrical contributions as a whole. The electrical savings during the summer are quite higher in case 4 than in 2 because the electricity used for the conventional AC system of the second case is saved. Moreover, in case 4 there is not thermal saving during the summer but during the winter due to the exceeding expense of the high thermal consumption of the absorption unit.

Both the investment and the operational costs are represented in Figure 9 for each of the case studies, now including the first case (current situation of the hotel), which has zero investment cost.

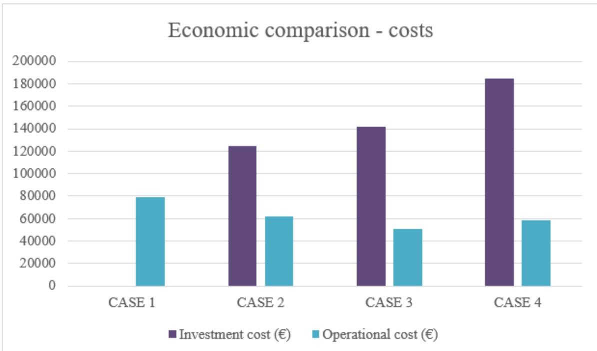

Figure 9. Investment and operational costs of each of the case studies.

It is observed that in case 4, the investment cost is the highest with a large difference (59,602 €) compared to case 2, whereas its annual operational cost is slightly lower, with a difference of 3,267 €. However, in the third case study, one can observe a bigger decrease in the annual operational cost (11,013 €), more than triple of the previously explained scenario.

Moreover, in this case there is a much lower increase (16,875 €) in the initial investment compared to the second case study.

The results of the SPP for each of the case studies are presented in Table 5.

Table 5. Simple Payback Period for every case study.

	CASE 2	CASE 3	CASE 4
Investment cost (€)	125,000	141,875	184,602
Annual energy savings (€/year)	16,897	27,909	20,163
SPP (years)	7.4	5.1	9.2

The lowest payback period corresponds to the third case study. It means that in five years and a half the investment would be recovered. Even though the investment cost in case 3 is higher than in case 2, it can be returned sooner due to the larger energy savings per year in case 3. However, case 4, absorption cooling system, results on both the highest investment and payback time, with more than 9 years.

Regarding the cost of primary energy saved, case 3 represents again the best option, the cheapest, with the lowest cost per kWh of primary energy saved, as presented in Table 6.

Table 6. Cost of primary energy saved for every case study.

	CASE 2	CASE 3	CASE 4
$\Delta C_{a.s}$ (€/year)	6,250	7,094	9,230
PE_{saved} (kWh)	198,374	335,045	181,755
CPE_{saved} (€/kWh$_{PE}$)	0.03	0.02	0.05

Considering that the annual increase of the electricity and gas prices is 3% and 4% respectively, the NPV and the IRR are calculated for every case and presented in Table 7:

Table 7. NPV (€) and IRR (%) results.

	CASE 2	CASE 3	CASE 4
NPV (€)	226,409	471,187	219,603
IRR (%)	12	20	9

Since in all of them the NPV is positive and the IRR higher than 2%, every case is economically feasible. However, the project suggested in case 3 would be much more profitable than the other two since the NPV is quite higher (more than two times than case 2), as well as the IRR.

4.2 Environmental impact

The amount of CO_2 avoided, expressed in tons of CO_2/year is presented in Table 8.

Table 8. Tons of CO_2 avoided for every case study.

	CASE 2	CASE 3	CASE 4
Tons of CO_2 avoided/year	33.1	58.3	35.9

It is observed that the preferred option is again case 3. It seems that the extra investment cost of installing a solar absorption cooling system in the hotel does not clearly have an environmental advantage.

Although case 4 leads to higher annual energy savings and CO_2 emissions cut, there is not such a large difference when comparing to the second case, whereas there is a big difference in the initial investment cost.

4.3 Discussion

All the analysed solar cooling technologies allow significant energy savings: 27.3% for the second case study, 54.8% for the third (heat pump) and 34.4% for the fourth case (absorption). The investment cost seems to be a bottle neck in these installations. The decrease of the initial cost of PVT collectors seems a key factor to reduce the cost of solar cooling systems, since the panels represent 88.1% of the total investment cost of the system in the third case study and 67.7% of this cost in the fourth case study (absorption).

In order to generalise a bit more this study for different situations or hotels, the case of a hotel that already has PVT panels and plans to install a cooling system coupled with them is also considered. For this, only cases 3 and 4 could be compared. Now the investment cost corresponds to the cost of the reversible heat pump and the total cost of the absorption installation (including the boiler), respectively. But it is clear that the latter is much higher and it is not profitable, as previously explained. Therefore, this analysis aims to observe the impact on adding a reversible heat pump to an already existing PVT installation in a hotel for solar cooling. Now a value of 0.60 is found for the SPP. This indicates that for those hotels that already have PVT panels, it is profitable to install such a heat pump since the investment would be recovered in less than a year.

4.4 Additional consideration

Moreover, one can observe that the PVT panels investment cost is the highest. A cheaper solar installation with PV panels may be considered either for the first (only solar panels) or the second (solar panels coupled with a reversible heat pump) case study of the ones previously analysed.

However, it is important to note that although the investment cost is lower, the electricity production of the same installation with solar PV panels is lower than the obtained from the PVT panels, mostly due to the losses from the temperature increase. Another important factor is the lack of thermal savings in this case, in comparison with the dual contribution of PVT panels. Ecomesh PV panels are considered, which actually have the same photovoltaic laminate as the already studied PVT panels.

In case 2, the installation of PV panels would lead to a decrease of 76% in the investment cost, whereas in case 3 the investment decreases by 67%. However, the efficiency obtained for this installation of PV panels is 15.4%, whereas the value for PVT panels was 16%, as presented in Table 3. In case 2, the installation of PV panels instead of PVT ones would lead to a decrease of 40.6% in the annual energy savings. In case 3, a value of 24.6% is observed, which is almost half of the decrease found in case 2. The ratio of the decreases in the savings between case 2 and case 3 is higher than that of the decreases in the investment costs. Therefore, it is generally more economical to install a solar electric system with a reversible heat pump coupled with 100 PV panels than just 100 PV panels on the rooftop for the general use of electricity since the annual savings will be higher in the former case.

With the installation of PV panels instead of PVT, the tons of CO_2 avoided decrease in 46.2% for case 2 and 26.2% for case 3. Therefore, once again the relative value (percentage) shows that the decrease is lower, proportionally, for the third case study, becoming the preferred option for an installation of PV panels.

5 CONCLUSION AND OUTLOOK

This paper is focused on the coupling of PVT panels with cooling systems since there are not enough data and results of calculations in the literature with the use of this type of panels. After a proper research of the state of the art, two different cooling technologies have been selected to study for the case of a 4-star hotel with 100 guests located in Madrid. The performance, economic and environmental feasibility of the installations are analysed by means of a comparative study of four different cases for the hotel. According to the results obtained, the preferred option regarding most of the indicators used in the analysis is a solar electric cooling system with 100 Ecomesh PVT panels and a reversible heat pump. The annual saving is 27,909 € and the CO_2 emissions cut is about 58 tons of CO_2/year.

For those 4-star hotels that already have installed the Ecomesh PVT panels, the only profitable option is a solar electric cooling system with a reversible heat pump. It has almost 72% lower investment cost than the absorption installation and around 38% higher annual savings during operation. For a new 4-star hotel, the investment cost becomes crucial. Therefore, two options are suggested: either installing only the PVT panels or installing the panels together with a reversible heat pump for solar cooling and heating. Despite the higher investment cost, the second of these options seems more beneficial from a long-term perspective.

Regarding the two alternative options suggested with PV panels, the investment cost is lower than in case of using PVT panels, but also the annual energy savings. The decrease on these savings is lower for the installation of the reversible heat pump; therefore this second option has been also recommended in case of deciding for Ecomesh PV panels instead of the PVT ones.

In all of the systems analysed in this work, the NPV is positive and the IRR higher than 2%; therefore, every case is economically feasible. However, the project suggested in the solar electric cooling system operated by a reversible heat pump would be much more profitable than the rest since the NPV is quite higher as well as the IRR. This occurs both for the systems with PVT panels and the ones with PV panels previously discussed.

It is important to note that this is a preliminary study performed in a simplified way with the example of a hotel in order to compare the possibilities and announce some conclusions. There was a lack of real data of the cooling part of the installation. Thus, it was necessary to make several assumptions for certain parameters. Moreover, the F-Chart method has several limitations.

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