# Dynamic Simulation of High Temperature Solar Heating and Cooling Systems

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## Abstract

This paper presents a dynamic model of a solar heating and cooling system (SHC) based on parabolic-through solar collectors (PTC). The SHC system under investigation is based on the coupling of PTC with a double single-stage LiBr-H2O absorption chiller; auxiliary energy for both heating and cooling is supplied by a biomass-fired heater, fed by wood chips. The SHC provides space heating and cooling and domestic hot water for a small university hall. The system was dynamically simulated in TRNSYS. A parametric analysis was also performed, varying climatic conditions as well as parameters regarding the design and the operating mode of the system.

## 1. Introduction

Solar Heating and Cooling (SHC) systems are well-known renewable energy plants which provide heating and cooling energy, by converting the solar irradiation incident on a solar collector field [1-3]. The first SHC prototypes were installed during 70s, showing scarce economic profitability [4]. Thus, SHC technology was abandoned for some decades. Then, in the last few years, a new impulse on this topic has been given, due to the increasing costs of fossil sources and to the enhanced interest in environmental issues. [5]. Nowadays, SHC systems are particularly promising for reducing the non-renewable energy consumption of buildings due to space heating and cooling and to domestic hot water production [3, 6]. The majority of the operating SHC are based on the combination of evacuated solar collectors and absorption chillers [3]. Similarly, papers published in literature are mainly focused on the combination of low-temperature solar collectors and single-stage absorption chillers [1-2, 6-15]. However, a possible alternative layout may be based on the use of concentrating solar collectors driving a double effect (DE) absorption chiller (ACH). Such configuration may be attractive due to the higher chiller Coefficient of Performance (COP ) [16] but may be penalized by the lower solar gain [17]. In fact, concentrating solar collectors can convert only the direct part of solar radiation, although this disadvantage may me partially compensated by their tracking device [4, 17]. Therefore, concentrating solar heating and cooling systems (CSHC) may be competitive in climates where the direct to total radiation ratio is sufficiently high. As for solar collectors, the most promising technology for CSHC applications is represented by Parabolic Trough Collectors (PTC) [4, 16-21]. Such technology was developed for solar power production [22]. In the last few years, several new companies entered PTC market, producing smaller collectors (e.g.: Solitem PTC 1800, with aperture and lengths of 1.8 m and 5.09 m, respectively), also aiming at reducing the capital cost and to increasing the efficiency of the collectors [23]. The final goal of many projects is also to produce solar collectors compatible for integration in buildings. Literature regarding CSHC systems is very scarce.

Tierney [20] investigated several high and low temperature solar heating and cooling systems for different climates. A similar simulation analysis was performed by Mazloumi et al. [19] for Iranian climates. Lokurlu and Richarts [17] presented a prototype of a novel CSHC integrating both PTC and double-effect absorption chiller technology. The combination of PTC and adsorbing chillers was investigated by El Fadar et al. [18]. Finally, a recent study was by Qu et al. [4] presented a numerical and experimental model of the smallest CSHC system in the world. The systems consists of 52 m2 PTC and a 16 kW double effect absorption chiller. The academic research on this topic can be enriched and improved by analysing advanced CSHC systems, with more complex layouts, and including: accurate building models, directly coupled with the CSHC simulator, detailed economic analyses, parametric studies for different climatic areas and parametric optimizations of the main design/operation parameters. The present paper aimed at covering some of these lacks by presenting a detailed dynamic simulation model of an innovative fully-renewable CSHC system, used to provide space heating and cooling and domestic hot water to a 7-zones university building, which was already selected as the test case in previous studies [6]. The system is claimed to be fully renewable since the auxiliary heat is provided by a biomass heater powered by wood chips. The only consumption of non-renewable energy is due to the remaining auxiliary equipment (pumps, cooling tower fans, etc).

## 2. System Layout

In this paper, a novel CSHC arrangement is here proposed. In fact, the solar field consists of small new-generation horizontal PTC collectors (e.g. Solitem PTC 1800 [23]), equipped with a single axis (NS oriented) mechanism, and using diathermic oil as High Temperature Fluid (HTF). In addition, the auxiliary heat is provided by a biomass heater, powered by wood chips. Finally, the solar loop is equipped with a variable volume storage system which is capable to vary discretely the capacity in winter and in summer. For a better comparison with the SHC systems considered in previous studies, the same building (area and volume, respectively of 2250 m$^2$ and 10125 m$^3$) layout of reference [6] was here considered. The system is schematically shown in Figure 1. Five different loops are present, for: Solar Collector Fluid, SCF (diathermic oil); High Temperature Fluid, HF (diathermic oil); Hot Water, HW; Cooling Water, CW; Domestic Hot Water, DHW; Chilled/Hot Water, CHW. The system includes the following main components: a Solar Collector field, SC, with Parabolic Trough Collectors, PTC; two hot storage tanks (TK1); a LiBr-H2O double-effect absorption chiller (ACH), activated by the thermal energy provided by the solar field; a wood chips-fired Auxiliary Heater (AHB); a closed-circuit Cooling Tower (CT), providing cooled water to the condenser and absorber of ACH; a fixed-volume pump (P1) for the HW loop; a variable-speed pump (P2) for the SCF loop; a fixed-volume pump (P3) for the CW loop; a fixed-volume pump (P4) for the CHW loop; an inertial chilled/hot water storage tank (TK2); an hydraulic separator (HS), balancing the flows between the primary and secondary hydraulic circuits; a plate-fin heat exchanger

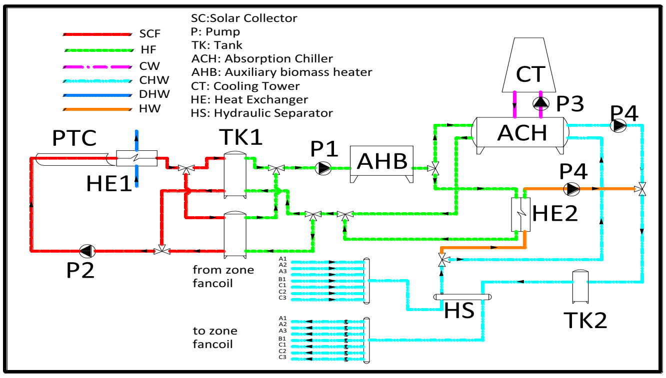

**Figure 1 - SHC simplified Layout**

on the solar loop, used to produce Domestic Hot Water (HE1); a plate-fin heat exchanger (HE2) on the HW loop, producing hot water to be supplied to the fan-coils during the winter; pipes connecting the HS with the fan-coils. The CSHC was dynamically simulated in TRNSYS, also including several additional components (not displayed in Figure 1), such as: mixers, diverters, controllers (feedback, proportional and on/off), schedulers (daily and seasonal), weather databases, printers, integrators, etc. The basic operating principle of the CSHC system is derived from the layout considered in previous studies [1, 6, 10-11], with the following modifications specified. The solar collector field is based on Parabolic Trough Collector technology. A single axis (NS) tracking systems is used to follow the solar azimuth angle. Due to the high operating temperature of the collectors, a diathermic oil was selected as the High Temperature Fluid (specific heat and density respectively of 2.8 kJ/kg K and 900 kg/m3). The hot storage TK1 consists of a system of two tanks, equipped with a group of mixing and diverting valves. By using two tanks, two different storage volumes can be used for summer and winter operation, respectively. The auxiliary heat is provided by a biomass (wood chip) boiler, heating the HW fluid (diathermic oil) drawn from the top of the TK1 tanks. During the summer, the fluid exiting the AHB goes to the generator of the double-effect absorption chiller. The management of the system, and details of the building are discussed in previous studies [6].

## 2. Simulation model

The CSHC described in the previous section was dynamically simulated in TRNSYS, which is a well-known software diffusely adopted for both commercial and academic purposes.

*Table 1 - SC, AHC, TK, HE parameters*

| ***Par.*** | ***Value*** | ***Unit*** | ***Par.*** | ***Value*** | ***Unit*** | ***Par.*** | ***Value*** | ***Unit*** |
|---|---|---|---|---|---|---|---|---|
| $A_{SC}$ | 800 | $m^2$ | $T_{SC,set,s}$ | 170 | °C | UTK1 | 3.0 | $kJ/(h\ m^2\ K)$ |
| $c_{SC}$ | 35 | / | $T_{SC,set,w}$ | 55 | °C | TK1 nodes | 5 | - |
| $\left(\tau\alpha\right)_n$ | 0.7 | - | $T_{SC,max}$ | 300 | °C | TK1 height | 2.0 | m |
| $F_R U_L$ | 4.17 | $W/\ (m^2\ K)$ | $\lambda_{pump}$ | 5 | % | TK2 Volume | 2.0 | $m^3$ |
| $c_f$ | 2.8 | kJ/(kg K) | $UA_{HE2}$ | $1.0\ 10^6$ | kJ/ (h K) | $\varphi_{P2}$ | 50 | $kg/h/m^2$ |
| $N_{series}$ | 10 | / | $Q_{P4}$ | 42990 | kg/h | $COP_{AHC,n}$ | 1.21 | / |
| $N_{parallel}$ | 15 | / | vTK1 | 20 | L/m2 | $T_{set,out,ACH}$ | 6.5 | °C |

The software includes a large library of built-in components, often validated by experimental data [24]. As mentioned above, the CSHC layout investigated in this paper was originated from the layout developed in a previous work [6], where the models of both built-in and user-developed components are diffusely discussed. Here, for sake of brevity, only the simulation models of some of the new components (defined types, in TRNSYS) are presented, not included in the above cited work [6]. Such types are: Parabolic Trough Solar Collectors (SC), Biomass-fired Auxiliary Heater (AHB), Winter Heat Exchanger (HE2), Primary energy calculator, economic cost calculator. The design/operating parameters of these types are summarized in Table 1.

***Parabolic-Trough Collectors.*** The simulation of Parabolic-Trough Collectors is developed on the basis of the equations given in [25]. Here, a modified collector loss coefficient (F'UL) is developed, based on the standard collector loss coefficient ($F_R U_L$, provided by manufacturer), corrected in function of the actual flow rate of the working fluid. The concentration ratio, $c_{SC}$, is the ratio of the aperture area to the receiver area ($c_{SC} = A_{SC} / A_{reciever}$). Thus, the useful energy gain from SC is:

$$Q_{SC} = R_1 R_2 A_{SC} N_{parallel} \left[ F_R IAM \left( \tau\alpha \right)_n I_b - \frac{F_R U_L \left( t_{in,SC} - t_{ext} \right)}{c_{SC}} \right] \qquad (1)$$

In equations (1), three corrections factors are considered: R1 accounts for flow rates different from test conditions; R2 takes into account the series/parallel arrangement of the collectors; IAM, Incidence Angle Modifier, provides data regarding the variation of the transmittance and absorbance of the receiver as a function of the solar incidence angle. R1 and R2 can be calculated analytically[25]. IAM values are provided by the manufacturers. Finally, the tracking system of the SC under investigation was equipped with a maximum temperature control management. This device defocuses the receiver when the temperature of the HTF overcomes a fixed set-point ($t_{SC,max}$), depending on the maximum temperature allowed for both SC materials and HTF.This circumstance should be avoided by the HE1 which is activated in DHW production, when SC outlet temperature overcomes the fixed set-point.

***Biomass-Fired Auxiliary Heater.*** The simulation model of this component is based on manufacturers data, providing both boiler and combustion efficiencies at different inlet temperatures and part-load ratios. The model lies in the algorithm previously developed for gas fired burners [24].

***Energy and cost savings.*** In order to calculate the energy saving of the CSHC system under investigation, a traditional HVAC system was assumed as a reference (RS). To this scope, for each SHC system under evaluation, a reference system was also implemented and simulated in TRNSYS, using the same building and climatic data. In the case study presented in the paper, an air-to-water electric driven heat pump (EHP) was considered as the reference system, producing hot water during the winter and cold water during the summer. The reference system is also equipped with a gas-fired heater, producing the same amount of DHW supplied by the CSHC. Thus, the primary energy consumed by the EHP and its annual operating costs, calculated by the TRNSYS simulation, were used in SHC simulation in order to evaluate CSHC relative primary energy savings and operating cost savings. The primary energy consumed by the reference system is mainly due to the electrical energy used for the EHP and for the pumps of the primary and secondary water loops. Furthermore, the RS also uses an additional amount of a primary energy (natural gas) to produce the same amount of DHW supplied by the CSHC. The total primary energy consumption is calculated as follows:

$$PE_{RS} = \frac{E_{el,RS}}{\eta_{el}} + \frac{Q_{DWH}}{\eta_c} \qquad (2)$$

The conventional thermoelectric efficiency was assumed equal to 0.46, representing the current Italian value, but also a good average value for all UE members. The conventional boiler efficiency, included in the RS and producing DHW, was assumed equal to 0.80.

Considering that biomass (wood chip) is a renewable energy source, the primary energy required by the SHC system, in terms of non-renewable sources, is only due to: i) electricity consumed by pumps; ii) electricity used by ACH, CT and SC tracking system. Therefore the SHC primary energy consumption is $PE_{CSHC} = E_{el} / \eta_{el}$ . The parameter adopted for the evaluation of the energy performance is the Primary Energy Saving (PES=1-$PE_{CSHC}$/$PE_{RS}$). In the case study, a PES' was also evaluated for an alternative CSHC layout, in which the Auxiliary Heater is fuelled by natural gas.

The model also calculates greenhouse gases emissions and savings, by assuming a $CO_2$ emission factor of 0.20 $kgCO_2$ per 1 kWh of primary energy for natural gas combustion and 0.60 $kgCO_2$ per 1 kWh for electric energy. According to this assumptions, the $CO_2$ emission saving (tons/year) was calculated as:

$$\Delta CO_2 = \frac{1}{3.6 \cdot 10^6}\left(0.20\frac{Q_{DWH}}{\eta_c} + 0.60E_{p,RS} - 0.60E_{p,CSHC}\right) \qquad (3)$$

A detailed cost model was implemented in the simulation tool, relating the cost of each component to the main design parameters. In addition, the operating costs due to natural gas and electrical energy consumption were evaluated, whereas maintenance costs were neglected. The total cost of owning and operating the SHC plant was expressed as:

$$C_{tot} = \frac{\sum_i J_i}{AF} + C_{op} = \frac{\sum_i J_i}{AF} + \frac{\sum_i E_{el,i}c_{el}}{3600} + m_{chips}c_{chips} \quad (4)$$

The price of wood chip ($c_{chips}$) was considered equal to 0.06 €/kg (with a LHV of 3.7 kWh/kg).

Capital costs were estimated by introducing a cost function for each component, obtained by regression of manufacturers data, as described in [11]. For the PTC, a cost of 200 €/$m^2$ was assumed [23]. As for the reference system, the capital cost was calculated with the same approach, and the operating costs were calculated as:

$$C_{op,RS} = \frac{\sum_i E_{el,RS,i}c_{el}}{3600} + \frac{c_{NG}Q_{DHW}}{\eta_c LHV_{NG}} \qquad (5)$$

The annual savings of the CSHC were calculated as the sum of two contributions: i) the actual saving, that is the difference between RS and CSHC operating costs; ii) a possible public funding, $\psi$:

$$\Delta C = \left(\frac{\sum_i E_{el,RS,i}c_{el}}{3600} + \frac{c_{NG}Q_{DHW}}{\eta_c LHV_{NG}}\right) - \left(\frac{\sum_i E_{el,i}c_{el}}{3600} + m_{chips}c_{chips}\right) + \psi \qquad (6)$$

Usually, in case of feed-in tariff incentive, the benefit is given only for the amount of primary energy saved for space heating and cooling (innovative) and not for DHW (conventional). Therefore, in case of feed-in tariff incentive, $\psi$ was calculated as:

$$\psi = C_{ft} = \left(PE_{RS} - PE_{CSHC}\right)c_{ft} \qquad (7)$$

Alternatively, the incentive could be proportional to the savings of $CO_2$ emissions:

$$\psi = C_{CO_2} = \left(m_{CO2,RS} - m_{CO2,CSHC}\right)c_{CO2} \qquad (8)$$

According to the typical values adopted in UE, in this study $c_{ft}$ was assumed equal to 0.231 € per each kWh of primary energy, whereas for $c_{CO2}$ a value equal to 20 € per ton of $CO_2$ was assumed. The incentive could also be represented by a contribution on the capital costs ($\phi$). In conclusions, four different economic criteria were used in the economic analysis: i) SPB, no incentive: $\phi = 0$ and $\psi = 0$; ii) $SPB_{ft}$, feed-in tariff, related to energy saving: $\phi = 0$ and $\psi = C_{ft}$; iii) $SPB_{CO2}$, feed-in tariff, related to $CO_2$ emission savings: $\phi = 0$ and $\psi = C_{CO2}$; iv) $SPB_{55}$, capital cost incentive: $\phi = 0.55$ and $\psi = 0$.

## 3. Results and discussion

The simulation software calculates the dynamic trends of temperatures and heat flows for all the components of the system. The software allows one to perform the analysis using different time periods (weeks, months, seasons, etc).

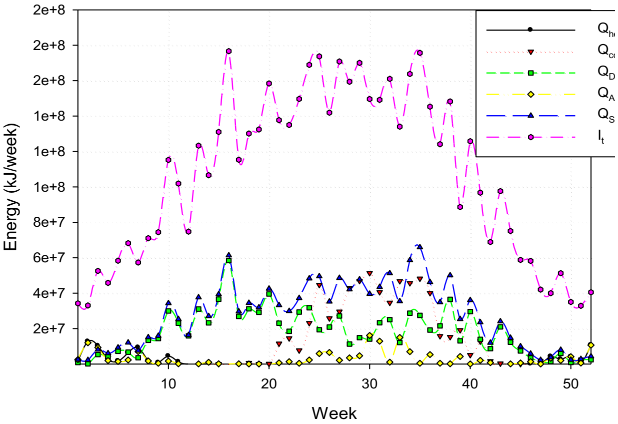

*Figure 2 - Energy flows (weekly)*

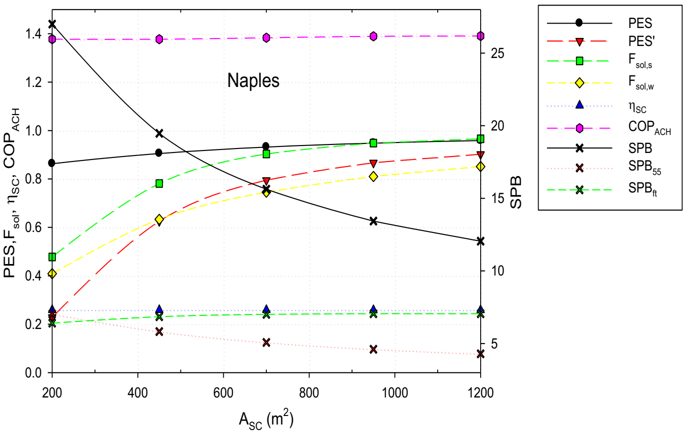

*Figure 3 – Sensitivity analysis ($A_{SC}$, Naples).*

As an example, in Figure 2 the main energy flows are shown on a weekly basis, in order to emphasize their trend during the year. Here, it is clearly shown that total solar irradiation ($I_t$) and SC useful gain ($Q_{SC}$) dramatically decrease during the winter. This trivial result is due not only to the well-known variation of solar irradiation during the year, but also to orientation of the PTC and to the tracking strategy. Using a WE orientation and a solar zenith tracking system, this trend would result more mitigated, but such configuration would be less profitable on a yearly basis. As a consequence, DHW is produced prevalently during the summer. In addition, during the intermediate seasons, solar energy is mostly employed for DWH production, whereas in the hottest summer weeks the amount of solar energy available for DHW production is scarce. Note also that the AHB must provide auxiliary energy mostly in the summer, due to high values of the cooling load. The results of the simulation, on a yearly basis, are resumed in Table 2. For Naples, such results are very encouraging, from all economic, environmental and energy points of view. The useful energy produced by SC is 1.43 TJ/year, which (integrated by 0.158 TJ/year of energy produced by biomass) is used to produce 0.930 TJ/year of DHW, 74.8 GJ/year of space heating and 0.625 TJ/year of space cooling. The summer solar fraction is 92.8 %, whereas the winter one is 77.5%. Thus, in the summer the space cooling demand is covered prevalently by solar energy, whereas during the winter the contribution of biomass is more significant. The total primary energy consumption of the CSHC, as for non-renewable resources, is 0.286 TJ/year, which is significantly lower (93.9 %) than the corresponding Reference System consumption. This result was largely expected, since the CSHC under evaluation consumes non-renewable energy only for secondary auxiliary devices. This remarkable energy saving also allows to save 136 tons of CO2 per year. The average SC efficiency (calculated on the total radiation) is good (25.9 %). A very good performance is also achieved by the ACH, whose average COP is 1.39. Regarding economic indexes, SPB (without incentives) is about 15 years. However, considering the possible funding strategies, this values can decrease down to 4.87 years. Such value would result in a investment profitable for the final user. The results are much more encouraging if compared with the those achieved in references [1, 6, 11] for similar low temperature SHC systems, in which the pay-back periods, with or without incentives, were significantly higher.

*Table 2 - Yearly results*

| PAR. | UNIT | TRIESTE | MARESEILE | ISTAMBUL | NAPLES | ATHENS | ALMERIA | CAIRO |
|---|---|---|---|---|---|---|---|---|
| $Q_{heat}$ | kJ/y | 2.23E+08 | 1.61E+08 | 2.11E+08 | 7.48E+07 | 5.94E+07 | 5.61E+06 | 0.00E+00 |
| $Q_{cool}$ | kJ/y | 4.26E+08 | 4.37E+08 | 4.73E+08 | 6.25E+08 | 7.47E+08 | 7.62E+08 | 1.34E+09 |
| $Q_{AH}$ | kJ/y | 3.00E+08 | 1.66E+08 | 1.92E+08 | 1.58E+08 | 1.67E+08 | 1.55E+08 | 2.37E+08 |
| $Q_{SC}$ | kJ/y | 1.08E+09 | 1.49E+09 | 1.55E+09 | 1.43E+09 | 1.25E+09 | 1.68E+09 | 1.83E+09 |
| $I_t$ | kJ/y | 4.75E+09 | 5.71E+09 | 5.83E+09 | 5.54E+09 | 5.37E+09 | 6.22E+09 | 6.86E+09 |
| $Q_{DWH}$ | kJ/y | 7.37E+08 | 1.06E+09 | 1.05E+09 | 9.30E+08 | 6.97E+08 | 1.13E+09 | 9.64E+08 |
| $E_{el}$ | kJ/y | 3.53E+07 | 3.63E+07 | 5.03E+07 | 4.75E+07 | 5.47E+07 | 6.17E+07 | 9.25E+07 |
| $PE_{CSHC}$ | kJ/y | 4.25E+08 | 2.73E+08 | 3.33E+08 | 2.86E+08 | 3.14E+08 | 3.14E+08 | 4.77E+08 |
| $PE_{RS}$ | kJ/y | 1.36E+09 | 1.75E+09 | 1.64E+09 | 1.70E+09 | 1.44E+09 | 1.97E+09 | 2.18E+09 |
| $m_{chip}$ | kg/y | 2.61E+04 | 1.46E+04 | 1.68E+04 | 1.38E+04 | 1.46E+04 | 1.35E+04 | 2.07E+04 |
| $\Delta CO_2$ | t/y | 1.12E+02 | 1.32E+02 | 1.10E+02 | 1.36E+02 | 1.23E+02 | 1.49E+02 | 1.97E+02 |
| $C_{tot}$ | € | 4.55E+05 | 4.55E+05 | 4.81E+05 | 4.55E+05 | 4.81E+05 | 4.81E+05 | 4.91E+05 |
| $C_{tot,55}$ | € | 2.05E+05 | 2.05E+05 | 2.17E+05 | 2.05E+05 | 2.16E+05 | 2.16E+05 | 2.21E+05 |
| $C_{tot,RS}$ | € | 8.00E+04 | 8.00E+04 | 9.00E+04 | 8.00E+04 | 9.00E+04 | 9.00E+04 | 1.00E+05 |
| $C_{el,SHC}$ | €/y | 1.27E+03 | 1.31E+03 | 1.82E+03 | 1.71E+03 | 1.98E+03 | 2.23E+03 | 3.34E+03 |
| $C_{op,SHC}$ | €/y | 2.84E+03 | 2.18E+03 | 2.83E+03 | 2.54E+03 | 2.86E+03 | 3.04E+03 | 4.58E+03 |
| $C_{DHW}$ | €/y | 1.54E+04 | 2.20E+04 | 2.19E+04 | 1.94E+04 | 1.45E+04 | 2.35E+04 | 2.01E+04 |
| $C_{ft}$ | €/y | 2.35E+04 | 2.23E+04 | 1.41E+04 | 2.77E+04 | 2.92E+04 | 2.73E+04 | 4.96E+04 |
| $C_{CO2}$ | €/y | 2.23E+03 | 2.64E+03 | 2.19E+03 | 2.73E+03 | 2.47E+03 | 2.99E+03 | 3.93E+03 |
| $C_{op,RS}$ | €/y | 7.37E+03 | 7.09E+03 | 5.47E+03 | 8.82E+03 | 9.53E+03 | 9.30E+03 | 1.62E+04 |
| SPB | y | 1.88E+01 | 1.39E+01 | 1.60E+01 | 1.46E+01 | 1.85E+01 | 1.31E+01 | 1.23E+01 |
| $SPB_{ft}$ | y | 8.65E+00 | 7.63E+00 | 1.01E+01 | 7.04E+00 | 7.77E+00 | 6.86E+00 | 4.81E+00 |
| $SPB_{55}$ | y | 6.26E+00 | 4.64E+00 | 5.16E+00 | 4.87E+00 | 5.97E+00 | 4.25E+00 | 3.82E+00 |
| $SPB_{CO2}$ | y | 1.69E+01 | 1.27E+01 | 1.46E+01 | 1.32E+01 | 1.65E+01 | 1.19E+01 | 1.10E+01 |
| $\eta_{SC}$ | / | 2.28E-01 | 2.61E-01 | 2.65E-01 | 2.59E-01 | 2.32E-01 | 2.70E-01 | 2.67E-01 |
| PES | / | 9.44E-01 | 9.55E-01 | 9.34E-01 | 9.39E-01 | 9.18E-01 | 9.32E-01 | 9.08E-01 |
| PES' | / | 6.88E-01 | 8.44E-01 | 7.97E-01 | 8.31E-01 | 7.82E-01 | 8.41E-01 | 7.81E-01 |
| $COP_{ACH}$ | / | 1.42E+00 | 1.43E+00 | 1.42E+00 | 1.39E+00 | 1.45E+00 | 1.37E+00 | 1.43E+00 |
| $F_{sol,s}$ | / | 8.84E-01 | 9.66E-01 | 9.87E-01 | 9.28E-01 | 9.04E-01 | 9.07E-01 | 8.85E-01 |
| $F_{sol,w}$ | / | 5.94E-01 | 7.14E-01 | 3.68E-01 | 7.75E-01 | 7.30E-01 | 9.71E-01 | 0.00E+00 |

For the base case of Naples, a parametric analysis was also performed, in order to assess the influence of the main design and operating variables on the performance of the CSHC system. The results are summarized in Figure 3. If the solar field area increases, both SPB and SPB55 decrease, whereas the SPBft becomes slightly higher. In fact, a capital cost funding strategy would encourage solar technology rather than biomass. Conversely, the feed-in tariff would slightly promote biomass heat production rather than solar. This is due to the higher specific cost of solar technology with respect to the biomass-fired heater. Regarding the energy performance, a larger solar field area would determine larger PES and Fsol values. However, the PES is scarcely sensitive to ASC, since the CSHC considered is nearly a fully-renewable system. Therefore, the PES is always close to 100%. It also noteworthy that all the considered economic and energy parameters do not depend linearly on the solar field area. The study was completed by a comprehensive sensitivity analysis, aiming at evaluating the potential energy savings and economic profitability of the CSHC considered when used in different cities of the Mediterranean Area. The following cities were selected: Trieste, Marseille, Istanbul, Naples, Athens, Almeria and Cairo. For all of them, the CSHC was re-sized on the basis of the local meteorological data and consequently both RS and CSHC were re-simulated in TRSNSYS. The coldest city is Trieste, in which the heating request is maximum (0.223 TJ/year) and the cooling energy is minimum (0.426 TJ/year). Conversely, the hottest city is Cairo, where 1.34 TJ/year are required for cooling and there is no heating load. Note that there is not a proportional relationship between the solar radiation incident on the collectors (It) and other results (PES, SPB, etc.), since they dramatically depend on the ration between cooling to heating energy demands. For example, in Istanbul the total radiation is very high (5.83 TJ/year; only for Almeria and Cairo the value is higher), but economic and energy results are not so good. This is due to the cold winter typical of this site, that determines very low winter solar fraction (36.8 %). Conversely, the summer is very hot and the corresponding solar fraction is close to 100%. In general, from the results shown in Table 2 the following conclusions can be drawn. The SC efficiency is scarcely sensitive to the climate, in fact the typical high operating temperature of the PTC makes the temperature difference less sensitive to the average external temperature. Conversely, the SC efficiency is much more influenced by the ratio of beam to total radiation, showing lower values in cities such as Trieste and Athens. The SPB values are significantly affected by the value of solar radiation. In fact, for Cairo and Almeria the SPB are 13 and 12 years, respectively, i.e. the system could be profitable even without any public funding. Eventual feed-in tariffs or capital cost funding would make this application very profitable for whatever city, specially for the hottest ones, such as Cairo and Almeria.Winter solar fraction dramatically varies with climates. In fact, the SC useful gain may significantly decrease in climates where at the same time direct radiation is scarce and building space heating demand is high (e.g. Trieste and Istanbul). Conversely, during the summer, in climates where the direct radiation is scarce, the cooling energy demand is low, too. Therefore, the summer solar fraction is always around 90%, for whatever climate. The consumption of wood chip is lower in intermediate climates, such Naples or Marseille. Conversely, in Trieste it is very high, mainly due to the winter operation of the AHB, and in Cairo is also high as a consequence of the large amount of cooling energy required by the building.

## 4. Conclusions

The system evaluated is based on the combination of solar and biomass renewable energy sources; concentration solar collectors are considered, based on the PTC technology. In addition, the use of a double-effect absorption chiller is proposed, due to the availability of a high-temperature hot fluid. In all cases evaluated, primary energy savings higher than 80% were achieved, and very interesting economic results were also obtained. In fact, the pay-back periods, even without any incentive, are largely lower than the system operating life, and become very low in case of public funding (commonly recognized for many renewable energy system, in UE). This result is very important, especially if compared with the corresponding results of low-temperature SHC systems whose SPB are comparable with the operating life, even in case of public funding. This is due in part to the higher COP of the double-effect ACH, which require half of the thermal input of that required by a single-effect ACH. So, the good profitability of the system analyzed is mainly due to the following reasons: i) the PTC capital cost is significantly lower with respect to evacuated tubes; ii) the heat integration by wood chips is very cheap, since the specific cost of such fuel (approximately 0.016 €/kWh) is significantly lower with respect to fossil fuels. The study was extended to several cities of the Mediterranean Area and the results showed that the CSHC considered is generally profitable, especially in hot climates (Cairo, Almeria, etc.).

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