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Performance assessment of a solar trigeneration system for residential applications by means of a modelling study

Luca Cioccolanti[a], Mauro Villarini[b,]*, Roberto Tascioni[c], Enrico Bocci[c]

[a] Università Telematica e-Campus, Via Isimbardi 10, 22060 - Novedrate, CO, Italy
[b] Università degli Studi della Tuscia, Largo dell'Università, 01100 - Viterbo, Italy
[c] Università degli Studi Guglielmo Marconi, Via Plinio 44, 00193 - Roma, Italy

Abstract

Concentrated solar technologies coupled with ORC system is a well-known topic in temperature ranges lower than 200°C. However, the integration to an efficient and economic working system is still a challenge especially at small scale. Efforts exist to achieve higher overall efficiencies but they are solely focused on thermal and electrical production while few of these is encompassing small-scale solar trigeneration systems. In the present article, the potential of a small scale concentrated solar Organic Rankine Cycle plant coupled with an absorber is investigated using a simulation analysis of a small scale 50 m^2 CPC solar field, a 3.5 kWe ORC and a 17.6 kWc absorption chiller to satisfy respectively heating, electricity and cooling needs of a residential user. The simulation analysis of the overall system has been carried out in TRNSYS and an own model of the ORC system has been developed by the authors in Matlab thus improving the previous general model. The final aim of the proposed work is indeed the performance assessment of the small scale integrated system in order to evaluate the potential feasibility of such a system for residential applications.

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Keywords: concentrated solar power; Organic Rankine Cycle plant; absorption chiller; simulation analysis; renewable energy production; polygeneration

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* Corresponding author. Tel.: +39-340-2266196.
E-mail address: mauro.villarini@unitus.it

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1. Introduction

Despite the challenges they are facing, including integration and regulatory barriers, renewable energy technologies provided an estimated 19.2% of global final energy consumptions in 2014 and an estimated 147 GW of new power capacity in 2015 [1].

Among renewable energy technologies [2] Concentrated Solar Power (CSP) is one of the viable options because it is considered a valuable alternative to substitute generation from fossil-fueled plants thanks to its lower environmental impact in terms of carbon dioxide and pollution emissions [3]. Among CSP technologies, the Compound Parabolic Collector (CPC) is a suitable option due to its low cost and good thermal performance at low and medium temperature ranges [4]. CPC indeed is able to collect both direct and diffuse solar radiation without a tracking system. Therefore, one of its potential and very promising application is in combination with Organic Rankine Cycles (ORC) as already addressed by several studies [5,6]. For example, Antonelli et al. [6] already investigated the integration of small size compound parabolic collectors with ORC for electricity distributed production using the simulation tool AMESim.

An Organic Rankine Cycle plant works similarly to a Rankine steam power plant but it makes use of organic working fluids which are able to condense and evaporate at acceptable temperatures [7]. Moreover, such system exhibits great flexibility, high safety and low maintenance requirements in recovering low temperature heat [8] even at small scale. A significant number of studies is focusing on this field. For example, Li et al. [9] evaluated the influence of heat source temperature and ORC pump speed on the performance of a small-scale ORC system using R245fa as working fluid. Al Jubori et al. [10] instead focused on the influence of several turbine design features on turbine performance in ORC systems. Pie et al. [11] experimentally investigated the performance of a specially designed radial-axial turbine using R123 as working fluid.

However, in order to achieve higher conversion efficiencies and annual performance of small scale ORC systems the modeling of the different subsystems and their integration is of paramount importance. For example, He et.al [12] developed a transient simulation model of a typical PTC system coupled with an Organic Rankine Cycle focusing on the effects of several key parameters. In particular, the authors evaluated the incidence of different size of the thermal storage tank on the performance of the system with seasonality. Instead, Borunda et al [13] evaluated the potential of PTC-ORC system as cogeneration unit in a textile industrial process using TRNSYS to emulate the real operating conditions of the user. On the contrary, Calise et al. [14] developed a dynamic simulation model of a 6 kWe Organic Rankine Cycle coupled with innovative flat-plate evacuated solar collectors.

Despite micro cogeneration has a very interesting potential for household applications [15] only a small number of research papers focus on small scale solar Organic Rankine Cycle plants and none of these is encompassing a small-scale solar trigeneration solution to satisfy all the energy needs of a household. In this work the authors further develop the modelling of a solar trigeneration plant presented in a previous paper [16] consisting of the same 50 m^2 CPC solar field coupled with a 3.5 kWe ORC plant and a 17, instead of 8, kWc absorber to better satisfy the energy demand of a small residential user. Among the novelties of the work are the improved ORC model, the increased flow rate of the pumps and a monthly analysis of the system. Therefore, the paper is organized as follows: after the Introduction, Section 2 describes the whole prototype plant; Section 3 reports a detailed description of the improved numerical model; Section 4 presents and discusses the main results of the work while Section 5 reports the conclusions.

2. Plant Description

The integrated plant consists of: (i) a 35 kWth CPC solar plant developed and patented by K-Engineering and Kloben Sud [17]; (ii) a 3.5 kWe regenerative Organic Rankine Cycle unit produced by Newcomen with a declared efficiency in the range 8%-10% [18]; (iii) a 17.6 kWc absorption chiller by Yazaki Energy Systems [19]. Other components of the system are also the heat storage tanks (two 3000 l tanks) and an evaporative cooling tower to reject heat from the absorber.

Figure 1a-c shows the key components of the prototype plant that has been built in the city of Orte near Rome (Italy).

Fig. 1. (a) the solar collector; (b) the ORC unit; (c) the absorber

With respect to the solar plant, it is able to reach heat fluid temperatures up to 150°C thanks to the use of copper tubes for high vacuum applications. The absorbing surface consists of an Al–N/Al selective material with an absorptance coefficient > 0.92 and an emittance coefficient $\varepsilon < 0.065$. The expander of the ORC unit is a three radial cylinders alternative engine using R245fa as working fluid. This fluid has low specific volume ratio, high molecular weight, zero Ozone Depletion Potential, it is inexpensive, non-corrosive and non-flammable. Moreover, its critical temperature is above the maximum operating temperature of the system which is in the range 100-150°C depending on seasonality.

The released heat by the ORC flows to the low temperature heat storage tank which feeds the house heating and cooling (via absorber) loads. Two fluid loops separate the collected heat from the solar plant to the ORC unit using therminol 62 as thermal vector thanks to its high thermal stability up to 325°C and low vapor pressure [20].

Finally, the absorber has a nominal Coefficient of Performance (COP) of 0.7 with 88 °C inlet hot water temperature and 7 °C chilled water output temperature but it is able to work with acceptable performance up to 70 °C.

For the sake of clarity, the consequent heating and cooling temperatures of the system are adequate for radiant panel floors with the lowest heating temperature set at 30 °C and the highest cooling temperature at 15 °C.

Table 1 reports the characteristics of the main power plant components:

Table 1. Characteristics of the main components.

Component	Value	Producer	Further specifications
Solar Collectors	50 m^2	Kloben	CPC heat pipes
ORC System	3 kWe	Newcomen	Piglet
Absorber	17.6 kWc	Yazaki Energy Systems	-
Pumps	30-120 l/min; 10 m*	Wilo	TOP-S 40/10 EM
HT and LT Storage Tanks	3 m^3; 4W/K**	Kloben	no heat exchangers
Temperature @ Terminals	W:30°C; S:15°C	Kloben	Klimaboden

*pressure head; **heat losses

W: winter; S: summer

3. Model description

Starting from the prototype plant installed in the city of Orte a simulation model of the whole system has been developed in TRNSYS 17. TRNSYS is a powerful software tool which allows to simulate complex energy flows varying with time [21]. Hence, it allows to include in the model also the fluctuant and variable radiation of the sun with regards to the site of location of the plant and to analyse and monitor the behaviour of the integrated system.

Despite TRNSYS library has a wide range of tested types for the simulation of many components, a specific subroutine for the ORC unit has been developed by the authors in Matlab [22]. Therefore, the system represented in the model mainly consists as follows: Type 71 for the CPC solar field; Type 4 for the diathermic oil (Hot Temperature storage Tank, HTT) and hot water (Low Temperature storage Tank, LTT) storage tanks, Type 155 for calling Matlab, Type 107 for the absorber and Type 510 for the evaporative cooling tower.

In particular, the useful power, P_u, from the solar field is assessed by means of Eq. 1:

$$P_u = A \cdot (\eta_0 \cdot (G_b \cdot K_\theta - G_d \cdot K_d) - a_1 \cdot (T_m - T_a) - a_1 \cdot (T_m - T_a)^2) \qquad (1)$$

where A is the collector area, G_b and G_d the direct and diffuse radiation on collector plane, K_θ and K_d the Incident Angle Modifier for direct and diffuse radiation respectively, T_m the mean temperature of the fluid in the collector, T_a the ambient air temperature and η_0 the maximum optical efficiency. With reference to the ORC unit, the electric power produced is:

$$P_{el} = \dot{m}_f \cdot [\eta_m \cdot \eta_{el} \cdot \Delta h_e - \Delta h_p/(\eta_m \cdot \eta_{el})] \qquad (2)$$

with $\dot{m}_f$ the organic fluid flow rate, η_m the mechanical efficiency, η_{el} the electrical efficiency, Δh_e and Δh_p the actual specific enthalpy difference across the expander and the pump. The variation of the parameters within equation (2) has been determined by the ORC model described below. Finally, the cooling power of the absorption chiller is equal to Eq. 3:

$$P_c = P_t \cdot COP \qquad (3)$$

where P_t is the inlet thermal power and COP depends on the operating conditions according to the absorber technical specifications.

Since the main objective of this analysis was to evaluate the performance of the integrated system in terms of overall efficiency and energy production, the final user thermal demand was not taken into account in the following evaluations. Therefore, two Type 4 (load and load-2) were used to collect the heating and cooling energy production by the integrated system as much as possible, i.e. large tanks able to always accept the heating and cooling power.

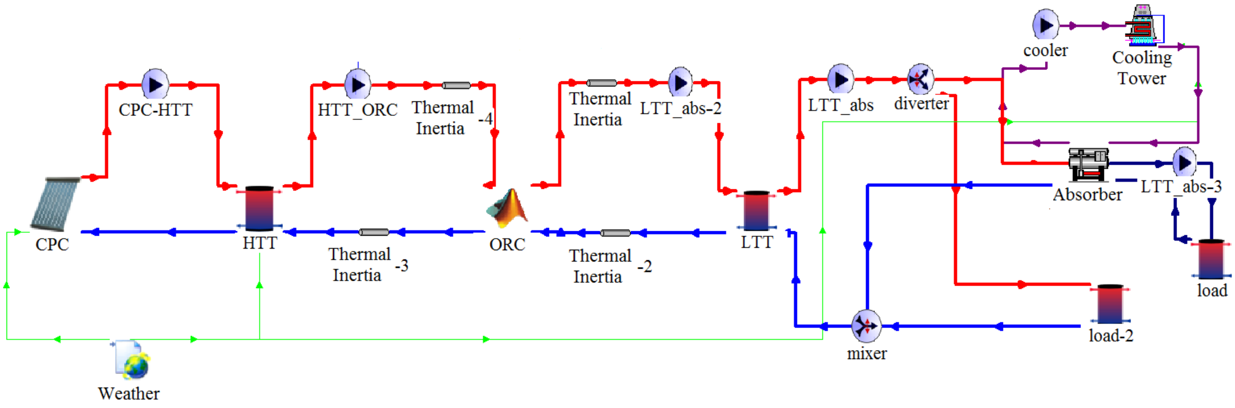

Fig. 2: a scheme of the simulation model

The Thermal Energy Storage (TES) tanks were used to decouple: (HTT) the thermal energy production by the solar field and the energy supply to the ORC and (LTT) the ORC thermal output and the absorber. Indeed, both the ORC and the absorber need that the inlet temperature of the heating fluid is within a certain range to achieve good performance. As a consequence, the performances of the ORC unit are expected to be lower in summer due to the sensibly higher condenser temperature with respect to the winter season.

As regards the ORC subsystem, the following assumptions have been considered into the model according to the specifications of the manufacturer:

- no pressure drops across the components;
- no thermal capacity of the components;
- thermal losses in the storage tanks only;

- minimum driving temperature difference between the evaporator and the condenser and pressure ratio at the expander equal to 50°C and 1.66;
- maximum inlet pressure at the expander 25 bar;
- constant isentropic efficiency of the pump (70%) and the expander (60%);
- constant heat exchangers efficiencies;
- steady state conditions.

In addition, a mechanical efficiency of 95% and an electrical efficiency of 90% both for the pump electric motor and the expander generator have been fixed. The heat transfer rate in the heat exchangers is assessed by means of the Number of Transfer Units (NTU) method. The organic working fluid flow rate varies with ambient conditions and is calculated according to an iterative procedure in Matlab considering a fixed overheating of 5 °C and a maximum evaporation temperature equal to 150 °C. In particular, the temperature difference at the evaporator has been varied accordingly up to a minimum value of 34°C between inlet diathermic oil temperature and evaporating temperature. Finally, R245fa has been considered as working fluid in a non-regenerative cycle and the values of its thermodynamics properties based on the open source library Coolprop [23].

At very low-part load conditions, the ORC power output is similar to the absorbed power by the auxiliaries. Therefore, a minimum 50°C temperature difference between the heat source and the sink has been assumed to run the ORC unit conveniently. In order to reduce the thermal losses in the HTT storage tank, the diathermic oil flows from the CPC solar field to the HTT storage tank if its outlet temperature is at least 5°C higher than the average temperature of the tank (T_{av}). The HTT_ORC and LTT_abs-2 pumps, shown in figure 2, are turned on as soon as the average temperature of the HTT storage tank is > 150°C while they are switched off when this temperature decreases to less than 90°C. Accordingly to the power available at the solar field, flow rates of these pumps have been fixed equal to 1800 kg/h and 3600 kg/h respectively. With respect to the flow rates considered in the previous paper [16], i.e. 7000 kg/h, the flow has been decreased in order to reach higher temperatures with the consequence of CPC yield reduction.

As regards the LTT_abs pump, it operates with water flow rate of 4320 kg/h at temperatures in the range 28-33°C in winter and 65-75°C in summer to supply adequate thermal power to the absorption chiller. Finally, an evaporative cooling tower extracts heat when the absorber is in operation at a constant flow rate of about 9180 kg/h according to the specifications of the chiller.

Hence, with respect to the design configuration of the prototype plant the sensitivity of the system performance to the operating conditions in terms of energy production, conversion efficiencies and operating hours has been analyzed on a daily, monthly and annual basis.

4. Results and discussion

In order to better appreciate the impact of seasonality on the performance of the system and on the behavior of each subsystem, results of the analysis have been determined on a monthly basis throughout a whole year. Table 2 reports the main results of the simulation in terms of temperature, conversion efficiency, energy, mean power and operating hours.

Table 2. Annual performance of the integrated system

Month	Energy to CPC [kWh]	η CPC	T_{av} HTT [°C]	η_{el} ORC	P_{el} ORC [kW]	Operation ORC [h]	T_{av} LTT [°C]	COP absorber	P_c absorber [kW]	Operation absorber [h]
January	5027.78	32%	106	6.6%	2.51	122	28			-
February	5416.67	35%	111	6.8%	2.67	139	29			-
March	7444.44	38%	128	5.3%	2.07	214	46	0.75	18.11	122
April	8069.44	33%	133	3.2%	1.05	259	68	0.73	17.77	349
May	8791.67	35%	146	3.2%	1.11	304	74	0.67	16.46	419
June	8736.11	34%	138	3.2%	1.14	301	71	0.65	16.08	425
July	9583.33	41%	150	3.2%	1.16	343	76	0.65	15.90	485

August	9458.33	38%	144	3.3%	1.13	335	73	0.64	15.88	464
September	8361.11	38%	129	4.1%	1.57	275	59	0.66	16.17	298
October	7027.78	45%	114	6.9%	2.82	205	31			-
November	5208.33	33%	105	6.5%	2.48	128	28			-
December	4680.56	39%	118	6.8%	2.74	110	30			-
Tot/Average	87805.56	37%	127	5%	2	2734	51	0.68	16.62	2561

In general, the annual incident energy to the CPC plant is almost 88'000 kWh which represents an interesting amount of energy for small scale solar trigeneration systems. The conversion efficiencies of the CPC plant range from 32% in January to 45% in October. As expected performances of the solar collector are lower than ones provided by the manufacturer due to the higher temperature difference at which the CPC operates in such application. On the other hand, higher temperature differences allow to reach interesting average temperatures of the HTT storage tank thus permitting the operation of the ORC unit throughout a year.

In terms of conversion efficiencies, the ORC unit has higher performance during the winter due to the lower temperatures at the condenser compared to the summer season when the absorber is in operation. Therefore, it reaches a peak electrical efficiency of about 6.8% in December and January while it operates at about 3.2% electrical efficiency from April to August when a relevant component of the available thermal power is reserved to the cooling load. On the contrary, the operating hours in summer are almost 3 times higher than in the cold months thus reaching an higher electrical energy production.

As regards the absorber, despite the low average temperatures of the LTT storage tank it is able to operate at COP in the range 0.64-0.75 with a mean cooling power of 16.62 kWc when in operation.

The presence of the HTT and LTT storage tanks of 3000 l each allows to assure a longer operation of the ORC and absorber. In particular, since the ORC thermal power output is higher than the absorption chiller thermal power input the surplus is accumulated in the LTT contributing to obtain a number of operating hours of the absorber in summer (2561) higher than that of the ORC unit (2031).

Compared to the results of the previous work [16], the improved model allows to obtain a higher cooling energy production due both to the higher capacity of the absorber and the different control strategy. In the former model, indeed, not only the absorber was half of the size (8 instead of 17 kWc), but also the absorber here, can remain active also if the ORC is deactivated, owing to the reach of the minimum temperature difference of 50 °C. On the contrary, in the previous paper the control system deactivated the absorber in order to guarantee higher temperature differences at the expander sending the thermal power to the domestic hot water or to the cooling tower). Thus, the efficiencies obtained with the present model are more realistic than in the former. More precisely, the mean conversion efficiency of the CPC is about 37% which is lower than the 45% of the previous model. The electrical efficiency of the ORC unit varies sensibly with seasonality and the mean conversion efficiency of 5% is obtained rather than the nominal 13% of the former which is consistent with the data of the literature for such small scale units.

In addition to the monthly data, the trend of the performance of the system has been evaluated also in a daily basis. Figures 3a-b show the temperatures and electrical power trend during a typical winter and summer week for the above described system. Due to the lower total radiation and ambient temperature in winter, several days are necessary to activate the ORC unit. Moreover, the mean average temperature of the HTT tank is well under the upper limit temperature of the storage. Nevertheless, the significant capacity of the storage allows to extend the operation of the ORC also when solar radiation is off or very low. The ORC indeed is switched on when the HTT average temperature reaches the upper bound and continues to work till the HTT average temperature decreases down to the lower bound of 90°C. On the contrary in summer, the average temperatures of the HTT are far higher than in winter and do not go down 120°C. Since the temperature of the cold sink has been set to about 70°C in summer and a minimum temperature difference of 50°C is requested to run the ORC properly, when the HTT average temperature reaches about 120°C the ORC unit is switched off thus reducing the potential operating hours of the ORC in summer. Due to the high temperature at the condenser in summer the maximum electrical power output of the ORC remains lower than 2 kWe while in winter despite the lower inlet temperatures at the expander the ORC reaches the nominal power output.

With respect to the LTT storage tank, the high thermal power output of the ORC increases the average temperature of the tank but its significant capacity allows to keep it under 100°C. In this way, it is possible to satisfy the cooling demand also at night when the ORC is off. Finally, since thermal losses in the circuit have been neglected in the model the HTT-ORC pump is on until the average temperature of the HTT reaches the lower limit temperature of 90°C. This means that in summer it is continuously on while in winter due to the lower temperature of the storage tank it runs intermittently.

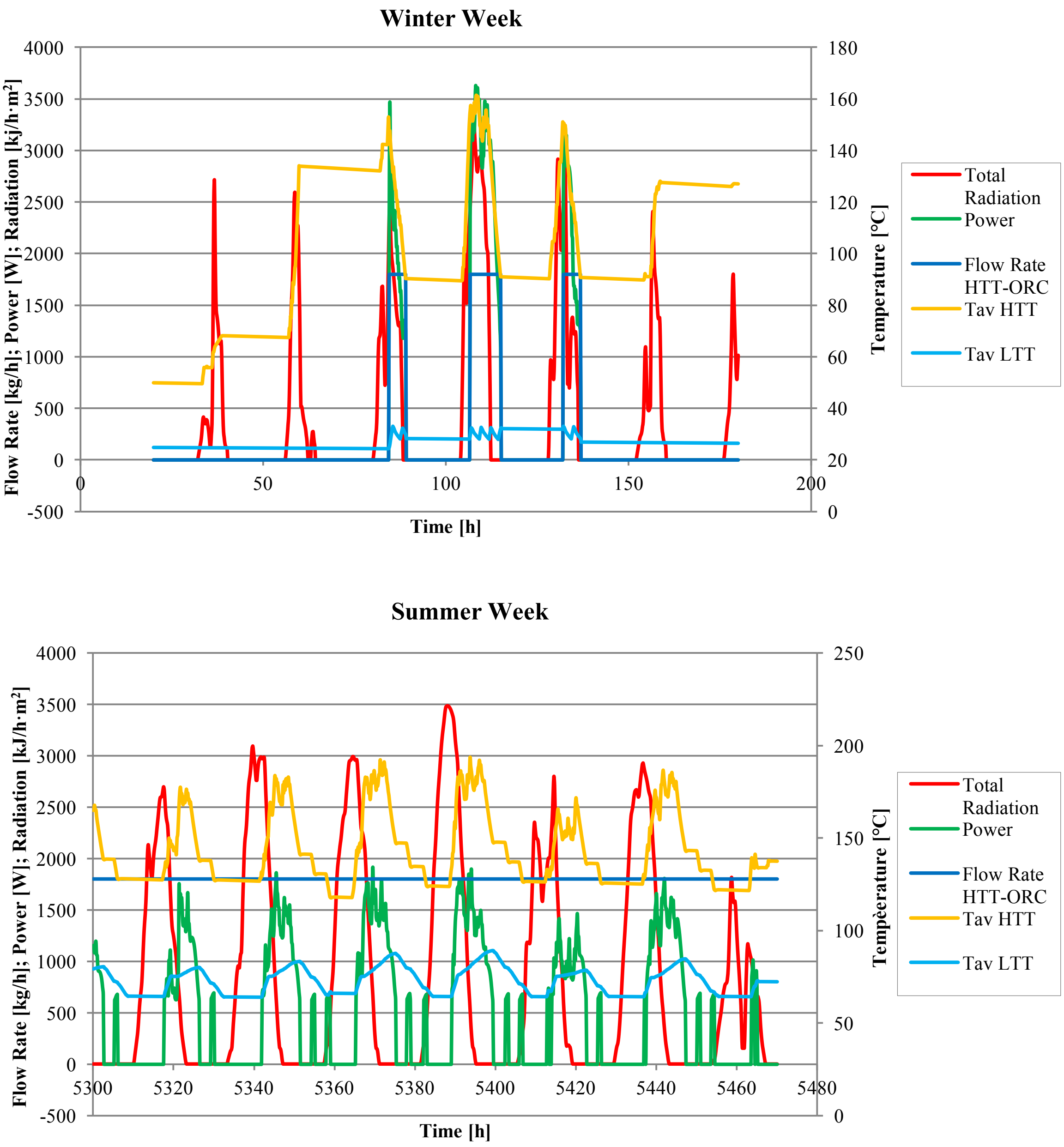

Fig. 3a-b: trend of the daily performance of the integrated system with seasonality

5. Conclusions

In this paper an improved model of a solar trigeneration plant has been analyzed and its performance compared with those of a previous model. The dynamic behavior of the plant has been obtained by means of a simulation model and it has been evaluated in an annual, monthly and daily basis. The analysis shows that the integrated system is able to operate for > 2500 hours/year thus generating a significant amount of electrical and thermal energy. Due to the high operating temperatures, the CPC conversion efficiency is low but on the other hand it allows to obtain good electrical efficiency of the ORC also in winter. Despite the higher inlet temperature at the expander, in summer the electrical efficiency of the ORC unit is < 3.2% because of the absorber operation. The significant capacity of the HTT and LTT storage tanks allows to sensibly extend the operation of the ORC unit and the absorber and to enable their operation also when solar radiation is low. In general, compared to the previous work more realistic performance of each subsystem have been obtained although a real user as not considered in the model. With respect to the prototype unit, the analysis shows that the design of the integrated system is adequate but room of maneuvers exists to improve the performance of the system. Such integrated trigeneration systems are indeed complex and too expensive at small scale. Therefore, it is fundamental that the systems operate as long as possible with good overall conversion efficiencies. For this reason, the authors are planning to investigate further in the next future the influence of different control strategies and operating parameters to provide interesting contributions to the optimization of the real prototype system.

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