Electrical Production of a Small Size Concentrated Solar Power Plant with Compound Parabolic Collectors

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Abstract

The use of the solar energy for electricity or useful heat generation has been extensively investigated as an alternative to fossil fired energy conversion. Particularly in the last decade, many studies have been carried out on Concentrated Solar Power (CSP) which was developed worldwide with Spain acting as the leading country in this field. Concentrating solar energy requires complex mirror systems which continuously move to track the sun. In comparison with flat mirrors, Parabolic Through Collectors (PTCs) have allowed to reduce costs, but they still remain quite an expensive solution. Instead, compound parabolic collectors (CPCs) are able to collect a higher fraction of both the direct and the diffuse radiation, although they have a lower efficiency at high temperature. Moreover, at least within certain limits, they do not require a tracking system. Their employment is therefore suited for the collection of medium temperature heat (up to 200 °C) and is useful for the reduction of the installation cost of Concentrated Solar Power (CSP) heating/cooling and energy generation systems. Small size plants (10 - 50 kW) were studied in this paper since they are more likely to be realized due to their smaller initial investment cost and to the capability of being installed

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on the roof of existing buildings. While the Organic Rankine Cycle (ORC) solution is well established to be the optimal for small size, distributed generation plants, the technology of the expansion device is still to be defined for the investigated installed power range. Accordingly to previous studies, an expansion device based on the Wankel mechanism was employed.

Based on these considerations and prior to more detailed analyses, a study of the annual energy production of a small scale ORC power plant using CPCs as a heat source and a volumetric machine as an expansion device was carried out. The influence of the thermodynamic cycle parameters, the working fluid, the concentration and the tilt angle of the collectors on the electrical energy production were taken into account. The thermal module delivered power, the expansion device isentropic efficiency and the overall efficiency were evaluated by means of a numerical model developed within the simulation tool AMESim v. 12.0.

The aim of this work is to provide a contribution in the assessment of the optimal configuration of such kind of plants in terms of collectors concentration and tilt angle on one hand, and thermodynamic parameter of the thermal module on the other. The annual electricity production was used as a criterion of comparison among the various parameters combinations. The number of operating hours per year was also taken into account for the sake of ensuring a regular production of energy. A selection of commercial solar tubes for the realization of the solar field was carried out and the optimal configuration for both the solar field and the thermal module was found. The results of this study are encouraging and constitute the basis for the development of future analyses.

Key words:
solar energy; CPC; ORC; Wankel; volumetric expansion device; renewables; evacuated collectors

1. Introduction

The attractiveness of the Organic Rankine Cycles (ORCs) mainly resides in that they are able to use low temperature heat sources while operating at relatively high efficiencies, thus enabling the construction of low and medium scale power plants that may be suited to a large variety of applications. Most of these advantages may also fit for solar applications, especially for small-size power plants, in combination with low/medium temperature solar collectors [1–3], where the integration with other resources is always an interesting option (with biomass or geothermal energy for example [4]).

The nature of the working fluid has also been the object of several studies: in the first research works [5, 6] high Ozone Depleting Potential (ODP) refrigerants such as R11 or R13 were used. In more recent studies other newly developed refrigerants were used, such as R245fa [7]. The optimization of the fluid selection for different cycle architectures and collectors' temperatures was treated in more recent studies [8–13]. However, no single fluid has been identified as optimal for the ORC, due to the strong interdependence between the working fluid features, the operating conditions and the cycle architecture. Most of the above mentioned studies show that the ORC efficiency is significantly improved by inclusion of a recuperator, of cascaded cycles, or of reheating [9, 14, 15].

At present, only one commercial solar ORC power plant is reported in the technical literature: the 1 MWe Saguaro Solar ORC plant in Arizona, which uses n-pentane as working fluid and shows an overall efficiency of 12.1%, with a collector efficiency of 59% [16]. The relatively high efficiency of this plant is due to the employment of high concentration tracking parabolic trough collectors.

The lowest efficiencies were in facts obtained with stationary collectors. Some authors [7] reported a 3.2% overall efficiency in a 1.6 kWe solar ORC with flat-plate collectors and 4.2% with evacuated tube collectors. A similar efficiency (lower than 4%) was obtained in a 2 kWe low-temperature solar ORC with R134a as working fluid and evacuated tube collectors [17]. In both those experiences, however, the collectors were used without any prior optimization process concerning concentration, tilt angle and collectors alignment. The collectors were aligned in the north-south direction and the originally built-in concentrator was used. For the sake of comparison of the previously mentioned solutions with those with a tracking system, a 7.7% efficiency was reported in a 9 kWe ORC employing a linear Fresnel Collector (collector efficiency of 57%).

Although solar ORCs feature lower efficiencies than photovoltaic (PV) systems, the presence of a thermal storage and even the thermal inertia itself of these plants provide a more stable electrical production, which make their power generation more predictable and easy to dispatch than PV systems. In addition, this technology does not require the employment of advanced or rare materials such as pure silicon. Finally, the employment of commonly available and reusable or recyclable materials (steel, plastics, aluminum, copper, etc.) makes the end-life disposal of the plants easier than for PV panels.

Focusing the attention on mini and micro (up to 50 kW) solar applications, the absence of a tracking system and the use of compact design collectors are useful for the reduction of the installation and maintenance costs. In fact, if a maximum cycle temperature of 200 °C is considered, Compound Parabolic Collectors (CPCs) can be used since they do not require a tracking system and they allow a moderate concentration. These concentrators have been studied for many years, both analytically and practically [18–24] as well as solar ORCs, which reported overall efficiencies varying between 2.5% and 7% [5, 25, 26].

The aim of this work is to fill the gap observed in the related literature about the analysis of the optimal combination of the operating parameters of both the solar field (concentration, collectors tilt angle) and the thermal module (thermodynamic parameters, plant configuration). The preliminary study presented about the feasibility of such a system [27] was further extended in the present work through the investigation of the thermal cycle optimal layout, the characterization of collectors built on the basis of commercially available components and a more detailed analysis of the solar field performance.

The optimal solution, to which type of expander is most suited, has not been found yet: some studies proposed the use of vane expanders [5, 6], others proposed a rolling piston expander [7] or a machine derived from a Scroll compressor [28]. In the present work the authors propose to use a specifically designed unit, based on the Wankel capsulism, which was described in a previous publication [29–32] where they showed that such device is an effective solution in the 10-50 kW size range. Such an expander, moreover, is more compact than reciprocating devices and is able to rotate at higher speeds with lower vibrations.

This first analysis was carried out at steady state, whereas a study of transient operation is currently in progress and will be the subject of a future paper.

Nomenclature

a	Solar tube thermal loss coefficient
A	Area (m^2)
C	Concentration
D	Direct radiation ($\mathrm{kW\,m^{-2}}$)
I	Incident solar radiation (kW)
E	Energy amount (kJ)
$\dot{E}$	Energy flux (kW)
h	enthalpy ($\mathrm{kJ\,kg^{-1}}$)
H	Diffuse radiation ($\mathrm{kW\,m^{-2}}$)
K	Proportionality constant
N	Number of reflections
$\dot{Q}$	Heat flux (kW)
r	Radius (m)
R	Factor of inclination
T	Temperature (K)
U	Convective heat transfer ($\mathrm{kW\,m^{-2}}$)
V	Displacement (cm^3)
$\dot{W}$	Mechanical power (kW)
Z	Number of hours

subscripts

a	ambient
ab	aerocondenser blower
ac	acceptance
ap	approach point
av	average
aux	auxiliaries
c	collector
cd	condensation
cg	cover glass
con	convective
d	daily
df	diffuse
di	direct
ed	expansion device
hf	heat transfer fluid
in	incoming
is	isentropic
l	lost
max	maximum
min	minimum
op	optical
out	outgoing
p	pump

pp	pinch point
r	receiver
ref	reflected
rad	radiative
rem	removal
s	solar
tc	thermal cycle
th	thermal
u	useful
wf	working fluid

Greek

α	Sun elevation
β	Collectors tilt angle (°)
ϵ	Emissivity
η	Efficiency
φ	Reflection efficiency
Φ	Heat removal factor
γ	Collectors azimuthal angle (°)
ρ	Radial coordinate (m)
σ	Stefan-Boltzmann constant ($\mathrm{kW\,m^{-2}\,K^{-4}}$)
θ	Angular coordinate (°)

2. Method

The plant layout which has been taken as a reference is typical of the small-scale solar systems (fig. 1). Here the solar field and the thermal module were connected via a heat transfer fluid circulation (water in this case). The working fluid condensation was supposed to happen through air-cooled condensers with induced-draft fans.

INSERT FIG1 ABOUT HERE

The thermal module included the preheating section, the evaporator, the eventual superheater, the expansion device, the recuperator and the condenser. Its annual electricity production was calculated by means of a numerical model which is described hereinafter. The transient behavior of the solar source was not taken into account here and an average insolation was employed.

Since steady state conditions were investigated, the storage tank, which is usually employed in solar systems, was not modeled. The cogeneration was not taken into account as well since the aim of this study was evaluating the optimal conditions for the electricity generation.

Under the hypothesis of steady state, averaged working conditions, the annual electricity production was calculated as:

$$E_{an} = \sum_{i=1}^{12} \overline{I_i \cdot Z_i \cdot \eta_{s,i} \cdot \eta_{th,i}} \tag{1}$$

In order to separate the effects of variation of the solar field and the thermal module parameters, the previous relationship was approximated as

$$E_{an} =\simeq \sum_{i=1}^{12} \bar{I}_i \cdot \bar{Z}_i \cdot \bar{\eta}_{s,i} \cdot \bar{\eta}_{th,i} \tag{2}$$

in which $\bar{I}_i$ denotes the solar radiation averaged over the generic i-th month, Z_i is the number of operating hours during the i-th month, $\bar{\eta}_{si}$ the solar field average efficiency and $\bar{\eta}_{th,i}$ the thermal cycle average efficiency. For the sake of brevity, in the following lines the superscripts denoting the operation of averaging will be omitted.

2.1. Solar intensity and operating hours

Since the investigated temperature range exceeded 100 °C, based on literature [23] the axis of the absorbers was aligned in east-west direction. The disposition used in the present work enabled the employment of various concentrations reflectors, differently from other papers found in literature in which the absorbers were aligned in nord-south direction [7, 17].

The solar intensity on the collector was calculated at the latitude of the Central Italy (43^o) through the model of Liu and Jordan [33] which takes into account the distribution of direct, diffuse and reflected solar radiation:

$$I = R_{di} \cdot D + R_{df} \cdot H + R_{ref} \cdot (D + H) \tag{3}$$

The average number of operating hours per month was calculated by considering the sunrise and the sunset time relative to a surface tilted by β (fig. 2) with respect to the horizontal and oriented toward the south. The operating hours are furthermore limited by the angle of the collectors.

INSERT FIG2 ABOUT HERE

In order to collect the solar radiation at noon, for each value of θ_{ac}, the maximum collectors tilt angle β_{max} was calculated as:

$$\beta_{max}(C) \;=\; \alpha_{max} - (90^o + \theta_{ac}) \tag{4}$$

At the same time, for each value of β, the minimum angle α_{min} at which the sun radiation was collected by the collectors was calculated as:

$$\alpha_{min} = 90^o - (\beta + \theta_{ac}) \quad (5)$$

2.2. Collectors average efficiency

Focusing the attention on small-scale power systems and aiming at the maximum reduction of the installation costs, widely commercially available components such as the U-pipe evacuated tubes were considered (fig. 3).

INSERT FIG3 ABOUT HERE

The efficiency of the collectors was calculated by taking into account the performances of commercially available components. Four types of tubular U-pipe collectors were considered. In order to avoid any form of commercialism, the brand name of these components will not be explicitly mentioned and the the various types will be identified by capital letters.

The efficiency of the collectors was calculated through a balance between the incoming and the outgoing energy, namely the solar radiation on one hand, the useful and the lost heat on the other.

$$\dot{E}_{s,in} = \dot{Q}_u + \dot{Q}_l \quad (6)$$

The energy collected by the solar tube $\dot{E}_{s,in}$ was evaluated as the product of the solar radiation by the optical efficiency and the heat removal factor Φ that takes into account the non-constant temperature of the receiver.

$$\dot{E}_{s,in} = \dot{E}_s \cdot \eta_{op} \cdot \Phi = I \cdot C \cdot A_c \cdot \eta_{op} \cdot \Phi \quad (7)$$

The solar field was discretized in a series of collectors in which the temperature variation effect was negligible from the point of view of the collector efficiency. The factor Φ consequently had a unit value.

As far as the collector's efficiency is concerned, the technical documentation reports the coefficients η_0 (optical efficiency), a_1 and a_2 (linear and the quadratic terms coefficients, respectively), according to EN 12975 [34] (Tab. 1). Those coefficients however include the effect of the built-in reflector which usually has a concentration in the range C = 0.6 - 0.8.

INSERT TAB1 ABOUT HERE

In order to take into account the influence of C on the optical efficiency, a simplified approach, with respect to other models described in literature [35], was employed. The effect of the reflections number N, which is usually provided by well-known datasets for both untruncated and truncated reflectors [1], was accounted here by considering that a generic sunray entering the collector may directly impinge the glass tube or be reflected one time by the reflector and attenuated by the factor φ. In addition, since a U-pipe solar tube was employed instead of a trough collector, the sunrays pass through the glass two times and therefore the relative loss was accounted twice.

The values of the optical efficiency obtained with this approach by applying the original values of C were compared with the values of η_0 reported in tab. 1, showing quite a good agreement (tab. 2).

INSERT TAB2 ABOUT HERE

This approach also accounted for a slight decrease of the optical efficiency with C, with a certain correspondence with theory (fig. 4).

INSERT FIG. 4 ABOUT HERE

The efficiency of the solar collector was evaluated by calculating the convective and the radiating losses to the ambient air:

$$\dot{Q}_l = \dot{Q}_{l,con} + \dot{Q}_{l,rad} = \\ = U_l \cdot \frac{A_c}{C}(T_r - T_a) + \sigma\epsilon_r \cdot \frac{A_c}{C}(T_r^4 - T_a^4) \tag{8}$$

The tuning of the numerical model coefficients was carried out by varying the convective heat transfer coefficient U_l and the emissivity of the receiver ϵ_r (tab. 3).

INSERT TAB3 ABOUT HERE

The resulting efficiencies of the various collectors were consistent with the ones declared by the companies (fig. 5) and with other data reported in literature [36–39]. The deviation between the declared and the recalculated efficiency is lower than 1% for all but model C.

INSERT FIG. 5 ABOUT HERE

All these relationships were finally summarized to evaluate the collectors efficiency as a function of the solar radiation I, the concentration C, the inlet $T_{r,in}$ and outlet $T_{r,out}$ temperature of the heat transfer fluid and the ambient air temperature T_a, thus allowing the model of the solar collector to be included into the model of the whole plant.

As for the inlet and outlet collectors temperature $T_{hf,in}$ and $T_{hf,out}$, their evaluation was carried out by considering (fig. 2) a fixed value for the temperature difference at the pinch ΔT_{pp} and at the approach point ΔT_{ap} (5 and 10K, respectively).

2.3. Thermal cycle modeling

The thermal plant efficiency η_{th} was calculated as the ratio of the net power over the heat flux provided by the solar field.

$$\eta_{th} = \frac{\dot{W}_{net}}{\dot{Q}_u} = \frac{\dot{W}_{net}}{\eta_s\, I} \tag{9}$$

The net power delivered by the plant was evaluated as the difference between the power generated by the expander minus the power employed by the auxiliaries (pump, condenser blowers).

$$\dot{W}_{net} = \dot{W}_{ed} - \dot{W}_p - \dot{W}_{ab} \tag{10}$$

The isentropic efficiency of the pump was assumed to be constant and equal to $\eta_p = 0.7$, from which the power required by the pump was calculated as:

$$\dot{W}_p = \frac{\dot{m}_{wf}(p_{vap} - p_{cd})}{\eta_p\, \rho_{wf}} \tag{11}$$

The power required by the aerocondenser blowers was evaluated as proportional to the rejected heat. The proportionality constant K_{ab} was evaluated through a survey of the commercially available components and its value was set at 25 W per thermal kW of rejected thermal power $\dot{Q}_{rj}$ (at the design point, that is to say ΔT between condensing fluid and ambient air equal to 15 °C).

$$\dot{W}_{ac} = K_{ab}\, \dot{Q}_{rj} = K_{ab} \left(\dot{Q}_u - \dot{W}_{ed} \right) \tag{12}$$

As for the expansion device, the proposed machine was already analyzed in previously published papers [31, 32, 40]. Although this machine is a rotary device, the thermodynamic limit cycle is the same of a reciprocating one (fig. 6).

INSERT FIG6 ABOUT HERE

The numerical analysis was carried out using a numerical model built using the simulation tools AMESim v.12.0, simulating the in-chamber pressure as a function of the crank angle. The numerical model of the Wankel expansion machine was developed in previous works [31], but in this analysis the two-phase fluids library of the code was used to analyze this device operated with organic fluids. The volume variation of the chamber was calculated by the crank-conrod model, whose mathematical formulation was modified in accordance with the variation of volume of a Wankel engine using the AMESet utility. The pressure drop across the intake and the exhaust valves was accounted by modeling the valves themselves as variable area orifices. The numerical model of the device was validated by comparing the results with experimental data [40].

Since the available solar heat was obviously variable along the year, the working fluid mass flow rate was supposed to be variable too by means of the expansion device rotating speed variation. This regulation strategy in facts proved to be effective for this kind of machines, although a certain decrease of η_{is} with the rotating speed was observed [40, 41]. Conservatively, η_{is} was supposed to depend on p_2/p_3 and T_2 only and, for every combination of these two parameters, the value assumed at the maximum rotating speed (3000 rpm) was considered.

3. Numerical results and discussion

As pointed out in the literature [42], the choice of the optimal temperature in the receivers is a tradeoff between collector efficiency and ORC efficiency. Increasing the temperature leads to higher thermal losses but also to a higher thermal module conversion efficiency. The optimal conditions may furthermore vary along the year due to the different insolation and ambient temperature. The optimal operating conditions were therefore evaluated respect to the annual electricity production.

3.1. Working fluid and introduction grade

Based on previously published work [27], R-600a was considered as working fluid because, in in one hand it provided a somewhat lower delivered power than other fluids (like R-134a and R-152a), in the other it also yielded a better efficiency over a wider temperature range.

Since the performance of a volumetric expansion device is affected by the introduction grade (eq.13), an appropriate value has to be chosen as a tradeoff between isentropic efficiency and delivered power.

$$\sigma = \frac{V_b - V_a}{V_{ud}} \tag{13}$$

Based on the performed analysis, an introduction grade equal to 0.2 enabled the device isentropic efficiency to be equal or above 0.8 over the entire range of the expansion grades p_3/p_4 (ratio of the upstream over the downstream pressure) included between 3 to 8 (fig. 7).

INSERT FIG7 ABOUT HERE

In these conditions the expansion was almost complete (fig. 8, dashed line). Lower pressure ratios led to over-expanded cycles (continuous line), with a loss represented by the counter-clock wise area, and higher pressure ratios obviously led to under-expanded cycles (dotted line).

INSERT FIG8 ABOUT HERE

The resulting isentropic efficiency was comparable and in several cases even higher than a radial turbine of the same power range [43] or than a volumetric expansion device of other type (Scroll as an example) [44–48] over the most part of the investigated working conditions.

As for the delivered power, at 3000 rpm it was included within the range 10-50 kW, which is interesting for this research (fig. 9). Based on these considerations, the value of 0.2 was therefore retained acceptable.

INSERT FIG9 ABOUT HERE

3.2. Collectors concentration and tilt angle

As it is well known, a high concentration of the collectors improves the direct irradiation captating efficiency at high temperature, however the employment of a low concentration also implies the capability of captating more diffuse radiation and enables the plant to operate for a larger number of hours. In the present work concentrations up to 3 were considered, since for higher values at least one seasonal replacement is needed.

The optimal annual electrical production conditions were practically the same whatever the thermodynamic parameters of the plant (fig. 10 and 11). This trend was also found when changing the tube type, although with different values of the value of the annual generated energy (fig. 12 and 13). In every case, values of the generated energy greater than 90% of the maximum were found in a region which is delimited at the left by C $\simeq$ 1.2 and at the right by a slanted line intersecting the first in a point whose ordinate is β = 15-20°. On the right side of the graph the annual production is nearly proportional to the number of operating hours, while on the left side it depends on the concentration.

INSERT FIG10 ABOUT HERE

INSERT FIG11 ABOUT HERE

INSERT FIG12 ABOUT HERE

INSERT FIG13 ABOUT HERE

The largest annual production per unit surface of panel was yield when the tilt angle of the collectors is larger than 20-25°. However when the tilt angle is further increased, as widely recognized, the employed ground surface becomes greater and greater. The choice of the optimal tilt angle is therefore a tradeoff between annual energy income per unit surface of panel and ground occupation.

Tilt angle and concentration also affected the plant average specific (per panel unit surface) power even when the annual generated energy was the same, because the number of operating hours was different (fig. 14). Therefore it was possible to reduce the average delivered power without reducing the generated energy, with significant implications on the size of the components and hence on the initial investment costs.

INSERT FIG14 ABOUT HERE

3.3. Thermal cycle layout

The employment of saturated or superheated cycles was deeply studied in the literature. Some authors [49] expressed some doubts about superheating a dry fluid, but they did not take into account the regeneration of the residual heat at the outlet of the expansion device. Moreover the use of regeneration enables the partial recovery of the energy loss due to the under-expansion of the working fluid which is typical of volumetric machines.

As expected, the increase of recuperated heat produced different effects on the thermal cycle and solar field efficiency (positive effect on the first and negative on the second), since the heat transfer fluid entered the solar field at a higher temperature (fig. 15). Since the first effect was prevailing on the second, the overall efficiency was increased even in the less favourable conditions (winter season). In addition, the partial removal of the residual sensible heat by the recuperator reduced the condenser thermal load and consequently a lower energetical consumption was required by the blowers.

INSERT FIG15 ABOUT HERE

In order to compare saturated and superheated cycles, the results were related to the collectors efficiency degradation with temperature: although the collector model "A" had the highest optical efficiency (fig. 7), the annual production yield was lower than collector "D", even at a very low saturation pressure and without superheating. The second model not only allowed the global efficiency (solar + thermal) to be higher, but also made convenient the use of superheating (fig. 16 and 17). For these two tube models, the optimal conditions respectively were $p_3 = 20$ bar and saturated conditions and $p_3 = 30$ bar and $T_3 = 160\,°C$.

INSERT FIG16 and FIG 17 ABOUT HERE

4. Conclusions

This paper summarizes the results of a research carried out to evaluate the optimal average operating conditions for a small-size solar power plant that employs stationary Compound Parabolic Collectors and a volumetric rotary expansion machine.

The proposed Wankel expansion machine allowed the use of the most suited introduction grade by means of a proper choice of the intake valves timing choice. This feature allowed to keep the isentropic efficiency equal or higher than 0.8 over the majority of the assumed working conditions and namely when the expansion ratio was within the range 3-8.

The features of the solar tubes played a fundamental role in determining the annual energy yield, since the collectors efficiency decrease rate with the temperature not only affected the amount of energy collected, but also changed the optimal thermal cycle features (saturated or superheated) and operating conditions (saturation pressure and eventual superheating temperature). The best performance was attained with $p_{sat} \simeq 30$ bar and $T_{sh} \simeq 160\,°C$ and employing the tube model D, which showed the highest insulation properties amongst the investigated commercial types. Based on the comparison between some commercial models, in facts the annual energy production was more affected by the insulation properties than by optical efficiency, which appeared to play a secondary role when temperatures above 80 - 90 °C were needed.

As for the solar field optimal parameters, the best performance was yielded with a concentration ensuring at least 3000 operating hours per year: the energy production was maximized when concentration was in the range 1.1 - 1.4, which in fact allowed the plant to be operated for 3000 - 3500 hours per year. Although an amount of energy close to the maximum may be collected with different combinations of C and β, a reasonable tradeoff between energy yield per collectors' unit surface and occupied ground surface may be achieved by using a tilt angle smaller than 20-25°.

The importance of C and β was not limited only to the amount of generated energy. A proper choice of these two parameters maximized the number of operating hours without an appreciable reduction of the energy specific production. This feature enables the reduction of the initial investment cost because the size of the various components (heat exchangers, pumps, aero-condensers and relative blowers, connection pipes) can be decreased.

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