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Analysis of a low concentration solar plant with compound parabolic collectors and a rotary expander for electricity generation

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Abstract

In the last decade many studies have been carried out on low temperature solar compound parabolic collectors (CPC) that are able to collect solar direct and diffuse radiation without the need of a tracking system, especially if coupled with small scale Organic Rankine Cycles for electricity and heat combined production.

This paper presents a study on a thermal power plant that uses an expansion device driven with pressurized vapor generated with the heat collected by a CPC solar field. The numerical model of the expansion device was developed with the simulation tool AMESim v.12 and allowed the simulation of the indicated cycle of this machine for the evaluation of delivered power, isentropic efficiency and specific working fluid consumption.

At the same time in this paper an analytical model of the evacuated solar CPC is presented for the evaluation of the collected heat as a function of sun incidence angle, external temperature, inlet carrier fluid temperature and mass flow rate.

These two models were used in conjunction for the analysis of the electricity that may be generated as a function of the ambient and working conditions. This study was performed in steady state conditions and with several working fluids to evaluate the power that would be delivered by a given displacement expander rotating at a fixed speed.

The results are given in terms of delivered power, thermal efficiency and amount of collecting surface needed for a given displacement expansion machine, calculated for several working fluids and different operating conditions.

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

1. Introduction

Organic Rankine Cycles (ORCs) are quite attractive since they are able to employ low temperature heat for electricity generation while maintaining relatively high efficiencies. They furthermore allow the construction of low and medium scale power plant that may be suited to a great variety of employments and sizes.

One field of employment is the coupling with low temperature solar collectors [1] for combined electricity and heat production [2,3]. High efficiency parabolic through collectors (PTC) allow in facts the direct utilization of solar heat even at very high temperatures for electricity generation or industrial processes. However a tracking system is needed

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to focus the direct radiation into the collectors and as a result such systems are particularly suited to large power plants. When taking into account smaller plants, stationary collectors are desirable to avoid the cost of tracking. In this case flat plate collectors or evacuated collectors have to be used, however concentration in this case is not possible, giving in the other hand the possibility of partly collecting diffuse radiation.

When a solar plant is used to collect hot water for domestic purposes it is undoubtely more convenient the solution of the flat plate collectors except the cases in which ambient air is particularly cold or insulation is much reduced.

Flat plate collectors in one hand have a higher optical efficiency than evacuated tubes as well as they use in a more efficient way the occupied area. On the other hand evacuated collectors are able to reach a higher global efficiency when the difference of temperature between the carrier fluid and ambient air is greater than 25^{o}C [4].

However when a relatively high temperature has to be reached by the carrier fluid, a certain degree of concentration is desirable. Compound Parabolic Collectors (CPCs) are an interesting solution since they avoid the cost for the tracking system and at the same time they allow a moderate concentration.

Low concentration solar systems have since many years analyzed both analytically and practically [5–11]. For moderate concentrations non-imaging reflectors have long be known to be a suitable choice.

Nomenclature

A	Area (m^2)
C	Concentration
r	Radius (m)
I	Incident solar radiation (kW)
$\dot{E}$	Energy flux (W)
E	Energy amount (J)
$\dot{Q}$	Heat flux (W)
T	Temperature (K)
N	Number of reflections
U	Convective heat transfer (W/m^2K)
h	enthalpy
D	Direct radiation (W/m^2)
H	Diffuse radiation (W/m^2)
R	Factor of inclination
c	Specific heat
V	Displacement (cm^3)
e	Eccentricity (mm)
b	Axial depth of the rotor (mm)
$\dot{W}$	Mechanical power (kW)

Subscripts

c	collector
a	ambient
r	receiver
ac	acceptance
in	incoming
out	outgoing
u	useful
av	average
min	minimum
max	maximum
l	lost
cg	cover glass
rg	receiver glass
s	solar
op	optical
n	index day
d	daily
con	convective
rad	radiative
di	direct
df	diffuse
r	reflected
vf	carrier fluid
wf	working fluid
ed	expansion device
fc	fan coil
aux	auxiliaries
tc	thermal cycle
is	isentropic
cd	condensation

Greek

θ	Angular coordinate (deg)
ρ	Radial coordinate (m)
η	Efficiency
τ	Transmission coefficient
φ	Reflection efficiency
α	Absorption efficiency
ξ	Lost rays percentage
σ	Stefan-Boltzmann constant (W/m^2K^4)
ϵ	Emissivity
β	Collectors tilt angle (deg)
ξ	Lost rays percentage (deg)
γ	Collectors azimuthal angle (deg)

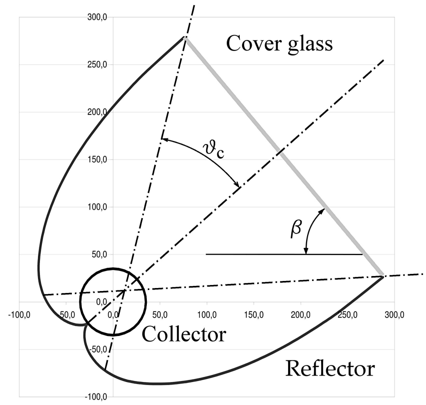

Fig. 1. Elementary scheme of a CPC with C = 1.5

The principles of non-imaging reflectors for solar concentration have been presented by Winston [5,6] and Rabl [7,8], which showed that CPC collectors are able to concentrate sunlight onto a flat absorber over many hours in a day up to $C = 10$ without tracking.

In this work the combination of evacuated solar collectors and CPCs was analyzed since, as shown in literature, this solution has a great potential since it is able to couple the high collecting efficiency of evacuated pipes with the high area utilization that is typical of flat collectors [9].

As for the power plant conception, it is widely recognized that turbines are the most suitable technical solution for electrical powers greater than 0.5-1 MW. Though the use of organic fluids allows the choice of a turbo-expander also for a much smaller plant size, volumetric expanders still are a suitable solution around 50 kW.

This kind of plant show global efficiencies that may be lower than photovoltaic (PV) panels; however they easily allow cogeneration and don't require expensive materials for their construction. Moreover their electrical production is less affected by temporary or partial shadowing than PV systems.

The presented work therefore focused on the application of the concept of the CPCs to a thermal power plant using an Organic Rankine Cycle with a volumetric engine. More in detail, as shown by other researchers [12,13], the use of a Wankel rotary engine converted to an expansion device was taken into account as already presented in [14].

The analysis of the global electrical energy delivered will be presented as a function of both the thermodynamic parameters of the plant and the geometrical parameters of the collectors. This first analysis was carried out at the steady state; a transient study is currently in progress and it will be the subject of a future paper.

2. Modeling

2.1. CPCs' modeling

In this analysis the absorber axis was hypothesized to be aligned in east-west direction to yield the higher output, as recommended by Mills et al. [10] for non-tracking through-type collectors with operating temperature over 100^{o}C. Since the highest amount of yearly collected energy is yielded by the best compromise between concentration and solar acceptance ($2 \cdot \theta_{ac}$), it must be recalled that concentration and acceptance angle run contrary. The effective concentration in many cases is slightlty reduced respect to the theoretical value due to the use of a truncated collector, for bulk and cost reduction sake [1].

The trouble of the rays lost in the gap between absorber and concentrator was solved by Rabl et al. [8] providing different shapes for the reflector itself and particularly for the cavity in the lower part of it. In the present work, the geometry of the lower part of the collector was obtained in the form of a circle involute.

Thus the coordinates of the profile representing the compound parabolic collector were obtained by applying the well-known relationships [11] for tubular receivers. As an example, Fig. 1 reports an elementary scheme of an untruncated reflector with C = 1.5.

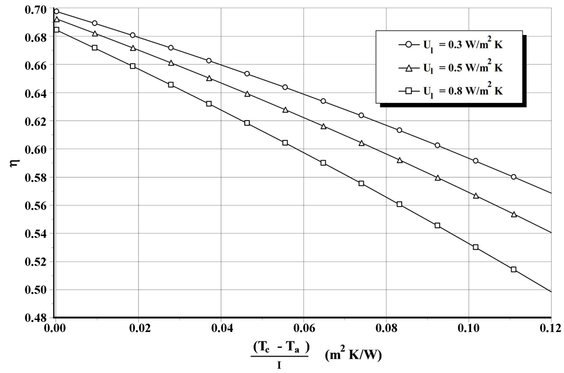

Fig. 2. Calculated efficiency of an evacuated tube for several values of U_l

As for the useful collectable energy, the analysis was carried out at the steady state. Thus the energy balance of the investigated collector was calculated as follows, balancing the incoming solar energy captured and the outgoing energy fluxes (the sum of the useful heat flux and the thermal losses), and neglecting the effect of the thermal capacity of both the collector and the carrier fluid:

$$\dot{E}_{s,in} = \dot{Q}_u + \dot{Q}_l \tag{1}$$

The incoming energy $\dot{E}_{s,in}$ into the collector was evaluated as in the following expression, in which F_f is a correction factor that takes into account the non-constant temperature of the fluid within the collector.

$$\dot{E}_{s,in} = I \cdot A \cdot \eta_{op} \cdot F_f \tag{2}$$

The optical efficiency was evaluated with the commonly used relationship usually reported in literature [1]:

$$\eta_{op} = \tau_{cg} \cdot \tau_{ag} \cdot \varphi^N \cdot \alpha \cdot (1 - \xi) \tag{3}$$

in which the following values were used: $\tau_{cg} = 0.92$, $\tau_{ag} = 0.94$, $\varphi = 0.96$, $N = 0.60$, $\alpha = 0.91$, $\xi = 0.02$.
The total heat loss was accounted as the sum of a convective and a radiating loss:

$$\dot{Q}_u = \dot{Q}_{l,con} + \dot{Q}_{l,rad} \tag{4}$$

The convective and radiating losses were respectively evaluated as:

$$\dot{Q}_{l,con} = U_l \cdot \frac{A_c}{C}(T_r - T_a) \qquad \dot{Q}_{l,rad} = \sigma\epsilon \cdot \frac{A_c}{C}(T_r^4 - T_a^4) \tag{5}$$

In literature the lost heat flux is often accounted by an equivalent convective heat transfer coefficient that takes into account of both the convective and the radiating heat losses and its value is usually the range $10 - 15\ W/m^2K$. In this work however the two losses were separately considered, therefore a much smaller value of U_l was found, since it accounted only the convective loss that is much reduced due to the presence of the glass envelope.

The evaluation of the convective heat transfer U_l was carried out by calculating the efficiency of the evacuated tube without the concentrator for several values of U_l (Fig.2 reports the results for $U_l = 0.3$, 0.5 and 0.8 W/m^2K as an example). The results were compared with the values found in literature [4,17–19] and quite a good agreement was found with $U_l = 0.5\ W/m^2K$. Although this value might be further reduced due to the presence of the concentrator that obstacolates the convective phenomena, this was kept unchanged, therefore the results obtained are to be treated as conservative.

As for the receiver temperature T_r, this was calculated as a function of the following quantities: the saturation temperature of the working fluid T_{sat}, the temperature rise across the solar field ΔT_{vf}, the temperature drops between the receiver and the carrier fluid $\Delta T_{r,vf}$ (pressurized water) and between the carrier fluid and the working fluid $\Delta T_{vf,wf}$.

$$T_r = T_{sat} + \frac{\Delta T_{vf}}{2} + \Delta T_{r,vf} + \Delta T_{vf,wf} \tag{6}$$

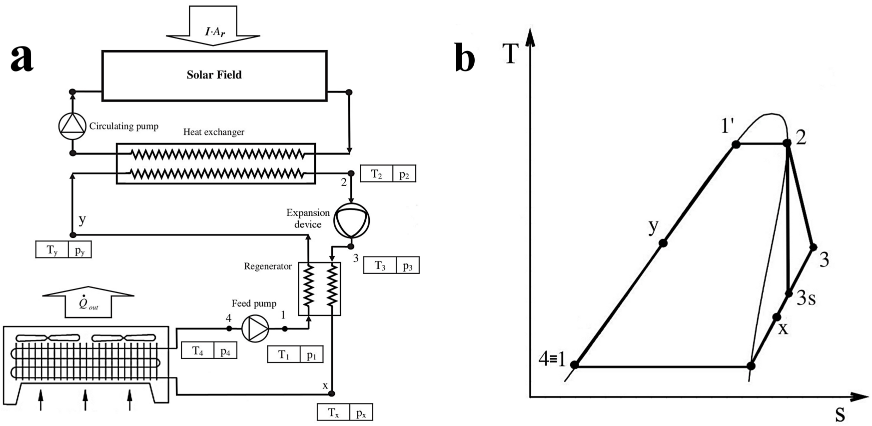

Fig. 3. Basic layout of a solar organic regenerative Rankine plant (a) and cycle on a $T - s$ diagram (b)

The solar intensity on the collector was evaluated with the model of Liu and Jordan [20] that takes into account the distribution of direct, diffuse and total solar radiation. The location for which computations are made was Pisa (4342'42"48 N, 10^o24'52"92 E). The source of the irradiation data was the Italian Rule UNI 10349-1994 [21].

The analysis of the collected energy was carried out by considering several concentration values, tilt angles and cycle saturation temperatures. The insulation at the latitude of the Central Italy (43^o) was used in calculations.

For a given concentration ratio, and hence for a given θ_{ac}, the maximum collectors tilt angle β_{max} was calculated in order to accept the solar rays at the maximum inclination of the sun:

$$\beta_{max}(C) = \alpha_{max} - (90^o + \theta_{ac}) \tag{7}$$

For a given value of β the minimum angle of sunrays acceptance was calculated as

$$\alpha_{min} = 90^o - (\beta + \theta_{ac}) \tag{8}$$

2.2. Thermal cycle modeling

As mentioned above, the proposed thermal plant receives the heat captured by the collectors by means of a carrier fluid (pressurized water), so that the organic vapor is generated in a heat exchanger and not in the collectors. The plant (Fig. 3) is therefore composed by the following apparatus: the solar field, a circulating pump, a vapor generator, an expansion device, a condenser and a feed pump.

As far as the condensation phase is concerned, in this first analysis the presence of cogeneration was not taken into account and the use of air condensing units was hypothesized, by imposing a fixed temperature drop between ambient air and condensing fluid (25^oC).

The analysed thermal plant uses an expansion machine derived from a modified Wankel engine. This expansion device has been extensively described in other papers [14–16] as regard of its features, isentropic efficiency, performances and operating maps and a prototype was built and tested to verify the feasibility of this solution as well as to validate the response of the numerical model used also in this paper. The displacement of this engine was $316\,cm^3$, with a dead volume of $27\,cm^3$ and an eccentricity equal to $12.05\,mm$.

The numerical model used in this work was developed with the simulation tool AMESim v.12.0. As already shown in previously published papers, this model simulates the behavior of temperature, pressure and other thermodynamic properties of the working fluid inside the chambers of the expansion device itself. This model was validated by comparing the numerical with the experimental results obtained at the engine test bench, by taking into account the delivered power, the inlet mass flow rate and the indicated cycle [14].

This numerical model calculates the thermodynamic properties of the working fluid by making reference to the available library of two-phase fluids, among which many organic fluids are present. This way several working fluids were taken into account to optimize the expansion machine isentropic efficiency for a given inlet thermal power. In this first analysis only subcritical cycles were considered and a further investigation on supercritical cycles will be carried out in a future work.

In this analysis the thermal power provided by the solar field was used as input of the simulations together with the saturation temperature of the working fluid. By means of this numerical model the isentropic efficiency of the machine and the thermal cycle efficiency were evaluated, as well as the rotating speed that was necessary to handle the available working fluid mass flow rate $\dot{m}_{wf}$.

The useful power delivered was calculated as the power delivered by the expansion device minus the power absorbed by the pump and the power absorbed by the auxiliaries. Among these last the fans of the condenser assumed a particular relevance; a survey on the commercially available air condenser showed the specific power absoption per unit of exchanged thermal power is in the range $5 - 15\ W/kW_{th}$. Conservatively, the highest value was considered. The efficiency of the thermal plant was calculated in both the cases of regenerative and non regenerative cycle.

3. Performance analysis

In the following lines the results of the analysis of thermal plant and solar field will be presented with a particular attention to the saturation temperature T_{sat} of the thermal cycle. In facts, in the previous paragraphs it was pointed out that T_{sat} influences the efficiency of both the solar field and the thermal cycle. As a consequence, the saturation temperature T_{sat} of the working fluid was considered as the main operating parameter. In this work several saturation temperatures in the range $80 - 130^oC$ were taken into account. Higher values were not considered due to low concentration of the collectors.

The operating conditions during the month of July were considered as the design point because this is the condition in which the working fluid mass flow rate is the highest. The captating surface was calculated in order to allow the engine to rotate at 3000 rpm at the design point.

3.1. Solar field analysis results

The analytical models used allowed the calculation of the monthly and yearly collectable energy per surface unit of panel. The considered surface was the one of the cover glass, so that from the point of view of the captating surface these collectors may be treated as flat ones. The calculation of the captured energy was carried out for several values of T_{sat} and C. The tilt angle was also considered as an input variable. As an example, Fig. 4 reports the results for the lower (80) and upper (130) investigated temperatures. The right-upper quarter of the graph is empty since for these couples of values (β - C) the tilt angle is greater than the maximum acceptable calculated with eqn. 7.

As expected, the increase in T_{sat} provided a reduction in the captured energy, due to increase of thermal losses, while for the investigated range of T_{sat}, there is no need for the increase of C above the value of 2-2.25.

As shown in Fig. 5-(a) and (b), for a given value of T_{sat} several combinations of C and β may provide the same amount of collected energy. They however provide a different distribution of the captured energy along the year, as reported in Fig. 5, that shows that a higher value of β gives some benefits in terms of a more constant energy collection along the year.

3.2. Thermal cycle analysis results

The performance of the thermal cycle were at first analysed from the point of view of the working fluid. Even for a given saturation temperature, the use of different working fluids provided quite different performances in terms of delivered power and efficiency. As an example, Fig.6(a) and (b) shows the variation in power delivered and cycle thermal efficiency by the expansion machine for the various fluids that were investigated. This diagram was traced assuming a rotating speed equal to 3000 rpm, an admission grade equal to 0.3 and $T_{cd} = 31.8^oC$.

While R-134a and R-152a provide interesting results in terms of delivered power, their use was limited to $T_{sat} \leq 100 - 110^oC$ due to their low critical temperature. R-600a (isobutane) provided a quite lower delivered power but also a good efficiency over a wider temperature range and it was therefore chosen for a more detailed analysis.

The influence of the admission grade was also analysed and finally a value of 0.2 was chosen as the best compromise between delivered power and thermal efficiency.

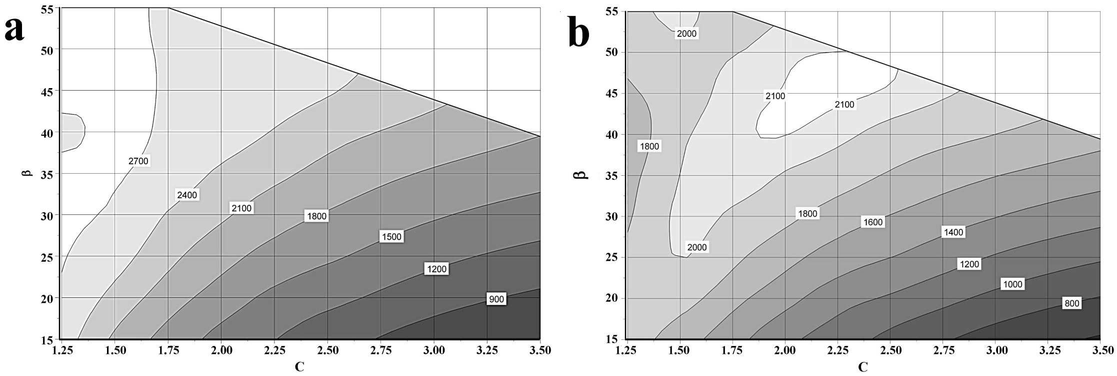

Fig. 4. Energy yearly captured per unit surface (MJ/m^2) as a function of C and β in the cases of $T_{sat} = 80$ (a) and 130^oC (b)

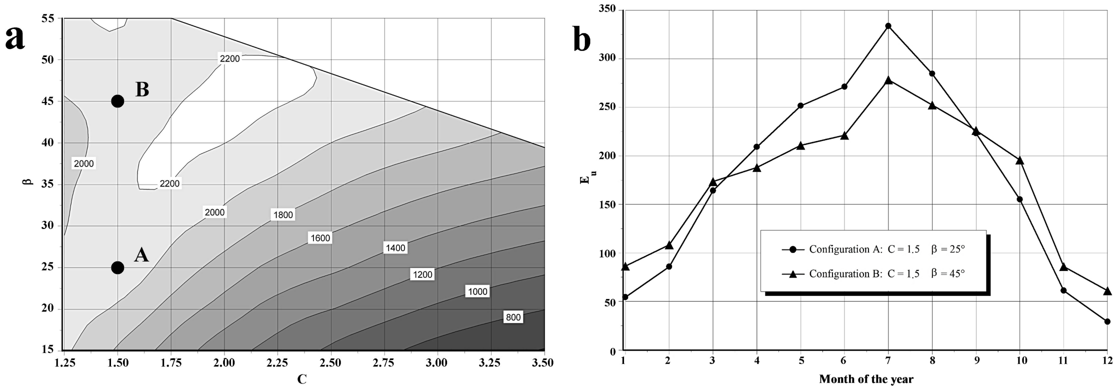

Fig. 5. Energy yearly captured per unit surface (MJ/m^2) as a function of C and β with $T_{sat} = 120^oC$ (a) and monthly distribution (b) for the two configurations A and B

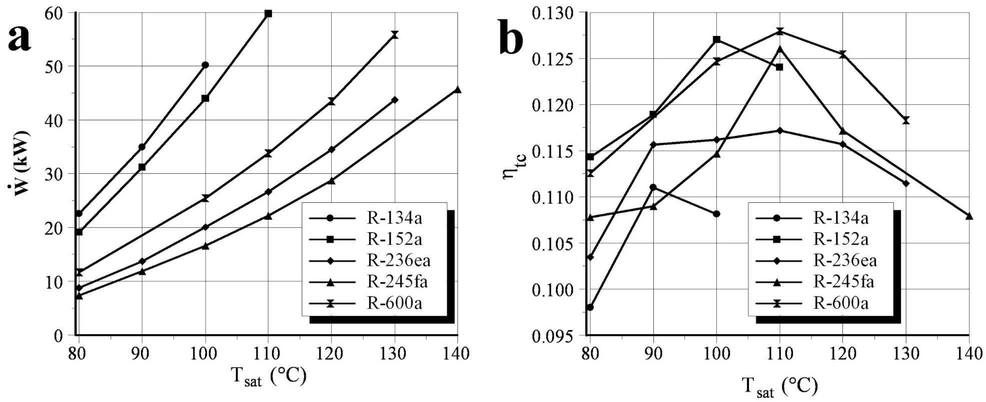

Fig. 6. Delivered power (a) and thermal cycle efficiency (b) with several working fluids

Fig. 7 reports the thermal efficiency of the cycle as a function of T_{sat} for the months of january and july and shows that, due to the limited expansion ratio of the device, it is not useful to go beyond a saturation temperature of 120^oC.

As mentioned above, the condensing temperature T_{cd} was not kept constant along the year since condensation was supposed to happen by thermal exchange with the ambient air. The efficiency of the thermal plant was therefore calculated by taking into account the variation of T_{cd} and values of T_{sat} ranging from 100 to 120^oC (Fig. 8). These results were obtained by taking into account the variation of $\dot{Q}_u$ and $\dot{m}_{wf}$. The rotating speed of the expansion device

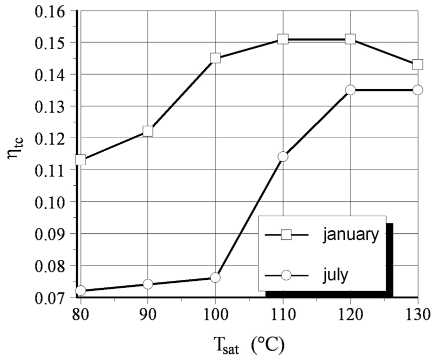

Fig. 7. Thermal cycle efficiency using R-600a as working fluid as a function of T_{sat} calculated in January and July

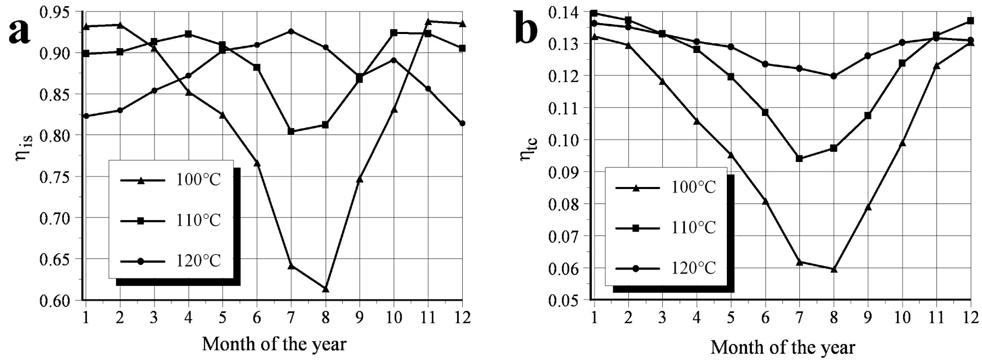

Fig. 8. Isentropic efficiency (a) and thermal cycle efficiency (b) calculated along the year using R600-a

was supposed to be accomodated to follow the variation of $\dot{m}_{wf}$ by using an inverter, as hypothesized in other published papers [22].

Fig 8 also shows that the both the isentropic and the thermal efficiency suffer quite heavy reduction during the summer months in the case of $T_{sat} = 100^{o}C$, while much more reduced variations can be observed at higher values of T_{sat}. The isentropic efficiency was comparable and in several cases even higher than a radial turbine of the same power range [23] or a volumetric expansion device of other type (Scroll as an example) [24–28].

The thermal efficiency of the plant shows smaller monthly fluctuations in the case of $T_{sat} = 120^{o}C$, however during winter season the efficiency of the plant results to be higher in the case of $T_{sat} = 110^{o}C$, due to the higher value of η_{is}.

3.3. Plant performance analysis

The analysis of the performance of the whole plant was carried out by combining the previously presented results. In order to establish which are the optimal values of T_{sat}, C and β, the yearly generated electricity was calculated. Fig. 9 (a)-(d) shows the results obtained with the aforementioned values of T_{sat}.

When the plant is operated at constant T_{sat} along the whole year, the most favourable case is $T_{sat} = 120^{o}C$; in this case the maximum yearly amount of energy is collected when concentration and tilt angle are respectively in the range $C = 1,75 - 2,25$ and $\beta = 35 - 50^{o}$. Lower C-values require smaller β-values and viceversa. It is then clear that the decrease in the captating efficiency of the collectors, due to the higher temperature of the receivers, may be adequately counterbalanced by the increase in the thermal efficiency of the cycle. Moreover, the higher is the thermal efficiency of the cycle, the smaller also is the mechanical power absorbed by the air condensing unit fans. When T_{sat} is increased from 110 to $120^{o}C$, as an example, the consumption of these auxiliaries in the month of July decreases from 9.1 to 7.0% of the net power.

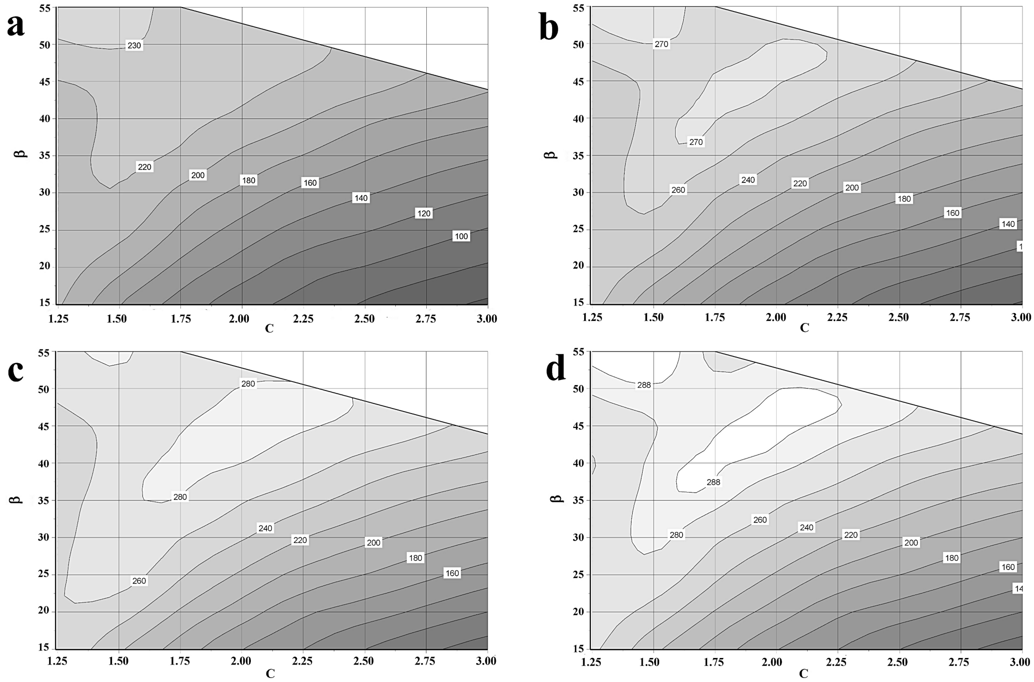

Fig. 9. Energy yearly generated per unit surface (MJ/m^2) as a function of C and β with T_{sat} = 100 (a), 110 (b), 120^oC (c)and with variable T_{sat} (d)

Conversely, when the plant may be operated at variable T_{sat}, there is the possibility of extending the range of optimal working conditions, also giving the possibility of using smaller values of C with some benefits in terms of compactness of the concentrators, yearly number of operating hours and sensitivity to cloudy wheather (capability of capturing the diffuse radiation). In this case, the efficiency of the collectors varies from a minimum in the month of december ($T_{sat} = 100^oC$, $\eta_c = 0.35$) to a maximum in the month of september ($T_{sat} = 120^oC$, $\eta_c = 0.51$).

The results of the calculations allowed to determine the main features of the plant operating in the optimal conditions; the optimal tilt angle resulted to be $\beta = 40^o$, the concentration $C = 1.75$ and the yearly electrical energy generated was nearly 60 MWh. The total collectors aperture was 750 m^2 and the ground occupied area was 1670 m^2. The maximum overall efficiency of the plant was calculated as 7.27%.

4. Conclusions

The presented analysis pointed out the fundamental role played by the saturation temperature of the working fluid. This parameter in facts not only influences the efficiency of the thermal cycle but also the captating efficiency of the solar plant, the isentropic efficiency of the expansion device and, as a consequence, the power absorbed by the auxiliaries.

The isentropic and the thermal cycle efficiencies were calculated by means of the numerical model developed with the simulation tools AMESim, that allowed the calculation of the indicated cycle of the expansion device, that was assumed to be a volumetric one, derived from a Wankel engine, as proposed in previously published papers. This device proved to be able to maintain a reltively high isentropic efficiency over a wide range of working fluids and operating conditions, due to the possibility of choosing the proper admission grade for every application.

This analysis allowed to determine the optimal conditions from the point of view of the energy produced per unit of collector aperture area, in terms of working fluid saturation temperature, tilt angle and concentration. This calculation was carried out by assuming that no tilt adjustment has to be done along the year.

The analysis pointed out that temperatures above 110-120^oC don't provide any noticeable advantage, especially if the expansion takes place in a single stage device and this is mainly due to the decrease of both the isentropic efficiency of the expansion device and the captating efficiency of the solar field. The use of these low saturation temperatures also allows the use of a limited receivers temperature and therefore there is no need for the increase of concentration above 1.75-2.25.

The energy produced may be lower than a well designed photovoltaic plant, however it must be pointed out that much work has to be done to analysis the full potentialities of this kind of systems, among which the use of non-symmetrical concentrators and multiple stage expansion in conjunction with supercritical cycles may be only some examples.

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