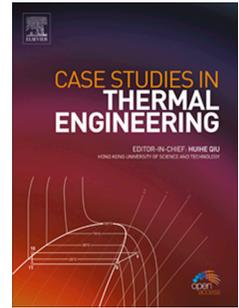# Journal Pre-proof

Direct steam generation solar systems with screw expanders and parabolic trough collectors: Energetic assessment at part-load operating conditions

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Paolo Iodice: Writing - Original Draft, Methodology, Formal analysis, Investigation, Software, Writing - Review & Editing

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Direct steam generation solar systems with screw expanders and parabolic trough collectors: energetic assessment at part-load operating conditions

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Abstract

This paper explains a numerical optimization of a novel screw expander-based solar thermal electricity plant to evaluate the energetic benefits in specific case studies. In the proposed solar electricity generation system, which is based on the steam Rankine cycle, water is used as working fluid and storage, parabolic trough collectors as a thermal source and screw expander as power machine. Such solar system offers major advantages over conventional power plants adopting steam turbines: low operating pressures, good exploitation of low temperature heat sources, acceptable efficiency in energy conversion with steam-liquid mixtures and reduced size. Since screw expanders can operate at off-design working conditions in several situations when installed in direct steam generation solar plants, the main aim of the present paper is to establish a thermodynamic model to study the energy performance of the proposed power system when off-design operating conditions and variable solar radiation befall. To assess best operating conditions and maximum efficiency of the whole power system at part-load operating

conditions, numerical optimization is then performed in a specific range of fluctuating evaporation temperatures under fixed condensation pressures.

Keywords: Steam screw expander; Solar thermal power efficiency; Direct steam generation; Part-load behavior; Polytropic expansion phase

1. Introduction

Recently much research has been carried out into harnessing renewable energy sources which, with their low heat values and low temperatures (such as geothermal energy and solar power), have attracted increasing interest worldwide owing to their high accessible quantities for exploitation [1]. In the last decades, in fact, extraordinary consumption of fossil fuels has caused many severe environmental problems all over the world, such as atmospheric pollution and global warming.

Being stimulated by such crucial questions, this paper shows a new solar electricity generation system (SEGS) with water used both as working fluid and storage. This specific SEGS, which is based on a steam Rankine cycle (SRC), uses parabolic trough collectors (PTCs) to produce dry steam (without a boiler) and screw expanders (SEs) to produce mechanical power. The satisfactory applicability in energy conversion with steam–liquid mixtures makes the screw expanders suitable for exploitation of low-medium temperature heat sources. In effect, direct steam generation (DSG) power plants adopting screw expanders as power machines are at present typically used for low-grade heat recovery and low temperature applications (waste heat recovery and geothermal energy), but now these new DSG power plants can represent an actual opportunity also for use in solar electricity generation systems [2].

In general, volumetric expanders are mainly appropriate for small scale applications characterized by low flow rates for high pressure ratios and low rotational speed, also accepting

fluid in two-phase conditions [3],[4]. In comparison with other volumetric power machines, SEs offer benefits of two-phase tolerance, simple structure and good efficiency under low working pressure [5]. Screw expander technology delivers some advantages also in comparison with steam turbines used in large-scale SEGSs. Indeed, SEs are power machines which can be used to convert heat to mechanical energy with low net power but without reducing the steam Rankine cycle efficiency (also while expanding liquid-steam blends) [6], whereas dynamic expanders are usually utilised in existing DSG solar systems providing higher power output but without admitting two-phase flows [7],[8]. The fluid velocities within steam turbines, in fact, are approximately one order of magnitude higher than those in screw expanders; thus, the risk of damage resulting from the admission of liquid/steam mixtures is very higher for turbo expanders. Taking all these features into account, screw expanders become a realistic choice for DSG solar systems with respect to both steam turbines and other sorts of volumetric machines [9]-[11].

In effect, the SE built-in pressure ratio can be much lower than the real working pressure ratio in most cases of (PTC)-based power plants with direct steam generation [1]. Consequently, under such operating conditions, the accessible enthalpy of high-temperature dry steam cannot be completely exploited when a single screw expander is used [12],[13]. In order to produce mechanical power with acceptable efficiency even under these off-design working conditions, this study adopts a plant solution based on two screw expanders joined in series in a DSG solar system [1].

The chief purpose of the present study is to explain a thermodynamic model of the whole power plant to explore the energy assessment of SE-based DSG solar plants in specific case studies. By using this mathematical model at part-load working conditions, parametric optimization of all variables is performed in order to attain maximum efficiency of the planned power system. The first aim of this study is to develop a mathematical model to describe the polytropic expansion phase in the SE. Subsequently, basic principles are established to assess the energy performance

of this SE-based DSG solar plant. In the numerical simulations used for this SE-based DSG solar system, the evaporation temperature of water is optimized with respect to solar thermal power generation efficiency. Hence, in the typical solar radiation range between 900 W/m^2 and 300 W/m^2, numerical optimization is developed assuming evaporation temperatures progressively rising from 170 °C to 320 °C, under condensation pressures of water assumed equal to 0.1 bar and 1 bar.

2. Mathematical models

Solar systems which employ parabolic trough collectors are a well-recognized and promising technology; such power systems account for nearly 95% of the worldwide capacity of actual solar power systems [14]-[16]. Furthermore, in the field of real SEGS, power systems with direct steam generation based on parabolic trough collectors represent a hopeful technology also for cost reduction [17],[18].

The solar power collector efficiency η_{PTC} of a parabolic trough collector (sole module) is evaluated by Equation (1), where T_a (K) is ambient temperature, T (K) is temperature at the PTC inlet and G_b (W/m^2) is beam solar radiation [5],[13]:

$$\eta_{PTC}(T) = 0.762 - 0.2125 \cdot \frac{T - T_a}{G_b} - 0.001672 \cdot \frac{(T - T_a)^2}{G_b} \tag{1}$$

In (PTC)-based power plants with DSG, water in parabolic trough collectors is both in binary phase and liquid phase regions. In liquid phase region, to obtain an outlet temperature T_{out} with a specific inlet temperature T_{in}, the necessary collector area A_l can be evaluated as in Equation (2), where the solar energy collection efficiency η_{PTC} is estimated with Eq. (1), $\dot{m}$ is the mass flow rate of water through the PTC and $C_p(T)$ is the heat capacity of water that can be calculated in Equation (3) by a first order approximation (T is the temperature of water in the liquid phase region ranging between T_{in} and T_{out}) [1].

$$A_l = \int_{T_{in}}^{T_{out}} \frac{\dot{m} \cdot C_p(T)}{\eta_{PTC}(T) \cdot G_b} dT \tag{2}$$

$$C_p(T) = C_{p,0} + \alpha(T - T_0) \tag{3}$$

Clearly, the solar energy collection efficiency $\eta_{PTC,b}$ in binary phase region can be estimated with Eq. (1) because water temperature remains constant in this phase. Consequently, in the binary phase region, the necessary collector area A_b can be calculated by Equation (4), in which Δh_b is the enthalpy increase of water [6]:

$$A_b = \frac{\dot{m} \cdot \Delta h_b}{\eta_{PTC,b} \cdot G_b} \tag{4}$$

For DSG power plants, as the solar field consists of liquid-steam mixture, the overall solar power collector efficiency can be assessed as in Equation (5), where $\dot{Q}$ represents the heat global transfer rate of water, A is the PTC global collector area (that is the sum of elementary modules), and h_2 and h_3 are the enthalpies of water at the PTC inlet and PTC outlet, respectively [19],[20].

$$\eta_{PTC}(T) = \frac{\dot{Q}}{G_b \cdot (A_l + A_b)} = \frac{\dot{m} \cdot (h_3 - h_2)}{G_b \cdot A} \tag{5}$$

The screw expander is a rotary-type positive displacement machine that can produce mechanical energy without high working fluid velocities, by using enthalpy in working fluid characterized by high temperature and pressure [21],[22]. Thus, screw expanders offer many advantages compared to dynamic expanders; indeed, these volumetric machines show good acceptance of steam-liquid mixtures, from superheated steam to saturated liquid [23],[24].

The polytropic and isentropic expansion processes are shown in the *p-V* diagrams of Figure 1. In ideal isentropic conditions, the expansion process of the working fluid follows the path *A-3-4_b-B*. However, since there are some energy losses which characterize screw expanders in real working situations (so reducing the ideal work output) [25],[26], the following four efficiencies can be introduced [1]:

- η_{Th} is the theoretical efficiency which contemplates losses owing to ill-matching of the real pressure ratio $r_p = p_3/p_4$ to the built-in pressure ratio $r_{p,b} = p_3/p_{4b}$ [2];

- η_L considers fluid leakage losses between the rotating helical screws;

- η_{ti} is the thermodynamic efficiency which considers energy losses due to thermodynamic irreversibility;

- η_m is the mechanical efficiency which represents mechanical losses owing to frictions from the moving rotors in the SE.

In Equation (6), the SE overall isentropic efficiency η_{SE} contemplates all the efficiencies itemized above [5],[8], with $\eta_D = \eta_L \cdot \eta_{ti}$ that is defined as the diagram efficiency. Hence, the influence of energy losses due to thermodynamic irreversibility on the overall screw expander efficiency η_{SE} are included in the diagram efficiency η_D.

$$\eta_{SE} = \eta_{Th} \cdot \eta_L \cdot \eta_{ti} \cdot \eta_m = \eta_{Th} \cdot \eta_D \cdot \eta_m \qquad (6)$$

All these energy losses can be evaluated by noticing the state points shown in Fig. 1. Losses during admission phase are represented in the path from A to $3'$ (in blue line) while losses during discharge phases are represented in the paths from $5'$ to B' and from $5''$ to B'' (in blue lines). Energy losses due to thermodynamic irreversibility are represented in the expansion process from $3'$ to $4'$. Obviously, in real operating situations, these paths (in blue lines) follow the equation $P \cdot v^n = const$, in which the polytropic index n replaces the index k of an ideal isentropic case to give the real expansion path of working fluid when it is correlated against the specified inlet and outlet conditions [6],[12].

Lastly, Fig. 1 also shows energy losses due to ill-matching of the SE built-in discharge pressure (that is equal to p_{4b} in ideal isentropic case and $p_{4'}$ in polytropic expansion process) to the operating discharge pressure p_4 (real exhaust pressure ranging between $p_{B'}$ and $p_{B''}$), that cause either a blowdown effect (patch from $4'$ to $5'$ in Fig. 1 (I)) or a blowback effect (patch from $4'$ to $5''$ in Fig. 1 (II)). In this regard, a chief characteristic of screw expanders is a small built-in volume ratio $r_{v,b} = v_{4,b}/v_3$. Generally, since a small built-in volume ratio is required to exploit the mass flow rate of working fluid before the high pressure port is closed, this parameter usually

ranges between 3 and 6 [12]. However, an excessively low built-in volume ratio means a small built-in expansion ratio ($r_{p,b}=r_{v,b}{}^{k}$), thereby causing under-expansion conditions when the actual pressure ratio r_p in real power systems exceeds the built-in expansion ratio $r_{p,b}$. Under such operating condition, the working fluid is exhausted at too high a pressure, thereby reducing the available energy and involving off-design operating conditions of the SE [1].
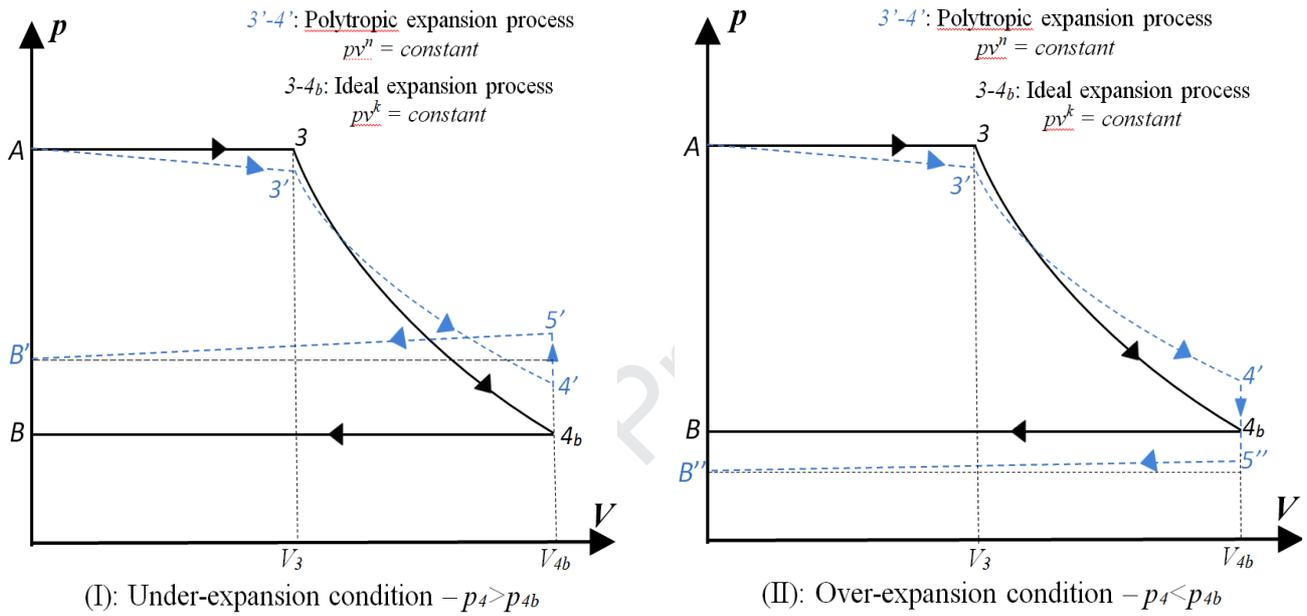

Fig. 1. (I): Polytropic expansion process with a blowdown effect during under-expansion conditions (patch in blue line: A-$3'$-$4'$-$5'$-B') and isentropic expansion process (path A-3-4_b-B)

(II): Polytropic expansion process with a blowback effect during over-expansion conditions (patch in blue line: A-$3'$-$4'$-$5''$-B'') and isentropic expansion process (path A-3-4_b-B)

In the isentropic expansion phase, the specific ideal work produced by a steady-flow screw expander can be evaluated by the area undergoing the ideal diagram $P{\cdot}v^{k}= const$ (shown in the path from 3 to 4_b in Figure 1). This ideal work is calculated as in Equation (7) and can be called theoretical isentropic work W_{Ti} [1].

$$W_{Ti} = -\int_{3}^{4b} v\,dp = -\int_{3}^{4b} [d(pv) - p\,dv] = p_3 v_3 - p_{4b} v_{4b} + \int_{3}^{4b} p\,dv$$

$$= \frac{k}{k-1} p_3 v_3 \left(1 - r_p^{\frac{1-k}{k}}\right) \tag{7}$$

In Equation (8), the theoretical efficiency η_{Th} is obtained by dividing the theoretical diagram work W_{Td} (that is the net work obtained when the real discharge pressure p_4 differs from the SE built-in expansion pressure p_{4b} relevant to an isentropic case) by the theoretical isentropic work W_{Ti} expressed in Equation (7). Hence η_{Th} represents losses owing to ill-matched real expansion ratio r_p and built-in expansion ratio $r_{p,b}$.

$$\eta_{Th} = \frac{W_{Td}}{W_{Ti}} = \frac{\frac{1}{k-1} p_3 v_3 \left(1 - r_{p,b}^{\frac{1-k}{k}}\right) + p_3 v_3 \left(1 - \frac{r_{p,b}^{1/k}}{r_p}\right)}{\frac{k}{k-1} p_3 v_3 \left(1 - r_p^{\frac{1-k}{k}}\right)}$$

$$= \frac{\left(1 - r_{v,b}^{1-k}\right) + (k-1)\left(1 - \frac{r_{v,b}}{r_p}\right)}{k\left(1 - r_p^{\frac{1-k}{k}}\right)} \tag{8}$$

In Equation (9) the diagram efficiency η_D is obtained by dividing the thermodynamic work W_{Tm} (that is the sum of the shaft work output W_S and the work for mechanical frictions W_{MF}) by the theoretical diagram work W_{Td}. Thus, this efficiency η_D contemplates losses due to both thermodynamic irreversibility and fluid leakage losses (during discharge and admission phases). In a previous scientific publication [23], the variation in diagram efficiency η_D with r_p and $r_{v,b}$ was exposed when saturated vapour is used as working fluid. It was shown that the diagram efficiency η_D can be reduced by high built-in volume ratio $r_{v,b}$, but the relevant reductions become insignificant under rising real pressure ratios r_p. In effect, when the operating expansion ratios are almost twice the built-in expansion ratio $r_{p,b}$, the values of the diagram efficiency become around constant with the operating pressure ratio r_p.

$$\eta_D = \frac{W_{Tm}}{W_{Td}} = \frac{W_S + W_{MF}}{p_3 v_3 \left[\left(\frac{1 - r_{v,b}{}^{1-k}}{k-1} \right) + \left(1 - \frac{r_{v,b}}{r_p} \right) \right]} \tag{9}$$

When the theoretical efficiency η_{Th} reaches its maximum $\eta_{Th,p}$, the isentropic efficiency η_{SE} reaches its peak value $\eta_{SE,p}$, as shown in Equation (10). With the assumption of the maximum theoretical efficiency $\eta_{Th,p}$ as unitary, the overall isentropic efficiency of the screw expander can be calculated as in Equation (11) [1]. Thus, the peak isentropic efficiency $\eta_{SE,p}$ comprises fluid leakage losses, losses due to mechanical frictions and energy losses due to thermodynamic irreversibility. Finally, as explained in Equation (11), once the built-in volume ratio $r_{v,b}$, the isentropic index k and the peak isentropic efficiency $\eta_{SE,p}$ are fixed, the overall isentropic efficiency η_{SE} depends on the operating pressure ratio r_p.

$$\eta_{SE,p} = \eta_{Th,p} \cdot \eta_L \cdot \eta_{ti} \cdot \eta_m = \eta_{Th,p} \cdot \eta_D \cdot \eta_m \tag{10}$$

$$\eta_{SE} = \eta_{SE,p} \cdot \eta_{Th} = \eta_{SE,p} \cdot \frac{\left(1 - r_{v,b}^{1-k} \right) + (k-1) \left(1 - \frac{r_{v,b}}{r_p} \right)}{k \left(1 - r_p^{\frac{1-k}{k}} \right)} \tag{11}$$

Equation (12) shows the steam Rankine cycle efficiency η_{SRC}, where P_{SRC} is the power of the Rankine cycle (difference between the SE power P_{SE} and the power absorbed by the pump P_p) and $\dot{Q}$ is the heat transfer rate of water in the binary phase and liquid phase regions [1],[19]. In such equation, η_{SE} is the global efficiency of the screw expander (already shown in Equation (11)), h_1 is the enthalpy value of water at the pump inlet, η_p is the pump overall efficiency which considers both the adiabatic and mechanical efficiency of the pump and $h_{2,is}$ is the isentropic enthalpy at the pump outlet, which can be calculated on the basis of water enthalpy properties by using REFPROP tables once the pressure increase is fixed. Besides, h_3 is the enthalpy value at the SE inlet and $h_{4,is}$ is the isentropic enthalpy value at the SE outlet. Clearly, all these enthalpy values depend on the specific steam Rankine cycle.

$$\eta_{SRC} = \frac{P_{SRC}}{\dot{Q}} = \frac{P_{SE} - P_P}{\dot{Q}} = \frac{(h_3 - h_{4,is}) \cdot \eta_{SE} - (h_{2,is} - h_1)/\eta_p}{(h_3 - h_2)} \tag{12}$$

In Equation (13) and Equation (14), η_G is the solar thermal power efficiency which describes the share of solar radiation efficiently converted into output power $P_{NET} = P_{SRC} \cdot \eta_{mec}$ (net power produced by the PTC-based power system), where η_{mec} is the mechanical efficiency of the PTC-based power system (also comprising generator efficiency) [12]. Therefore, this overall efficiency can be considered the global efficiency of the DSG power system as a whole.

$$\eta_G = \frac{P_{NET}}{G_b \cdot A} = \eta_{PTC} \cdot \eta_{SRC} \cdot \eta_{mec} \tag{13}$$

$$\eta_G = \eta_{PTC} \left[\frac{(h_3 - h_{4,is}) \cdot \eta_{SE,p} \cdot \frac{\left(1 - r_{v,b}^{1-k}\right) + (k-1)\left(1 - \frac{r_{v,b}}{r_p}\right)}{k\left(1 - r_p^{\frac{1-k}{k}}\right)} - \frac{(h_{2,is} - h_1)}{\eta_p}}{(h_3 - h_2)} \right] \eta_{mec} \tag{14}$$

3. Results and discussion

The particular SE-based DSG solar plant examined in this paper is presented in Fig. 2, whereas the *T-s* diagram of the steam Rankine cycle is shown in Fig. 3 under evaporation temperature and condensation pressure assumed equal to 200 °C and 1 bar, respectively [12]. As shown in the schematic diagram, the dry saturated steam flows into the screw expanders to provide mechanical power: the expansion phase begins in the first path from *3* to *3'* (in *SE₁*) and is then completed in the second path from *3'* to *4*. Once the saturated steam has been exhausted, it flows into condenser *C* (from *4* to *1*) to be condensed to saturated liquid. Finally, the saturated liquid is pressurized by pump *P₁* (from *1* to *2*) toward the parabolic trough collectors (*PTCs*) to

produce saturated steam which is then separated in the water storage unit, thereby obtaining dry saturated steam [1].

In the mathematical simulations adopted for this steam Rankine cycle, two SEs are coupled in series in order to decrease the actual expansion ratio r_p of each screw expander. Indeed, as was clarified in the previous section, SE isentropic efficiency decreases when the real expansion ratio r_p increases excessively in comparison with the SE built-in expansion ratio $r_{p,b}$, thus involving off-design operating conditions [6]. Therefore, with the plant configuration adopted in this study, the whole pressure ratio is shared on two power machines, such that each screw expander can benefit from lower actual expansion ratio r_p.
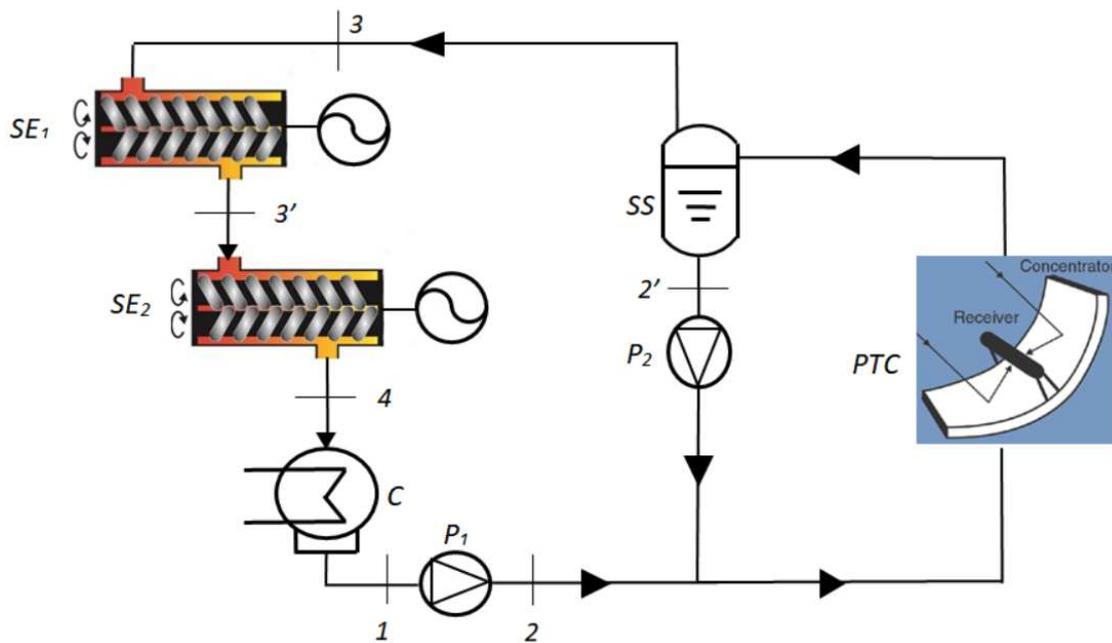

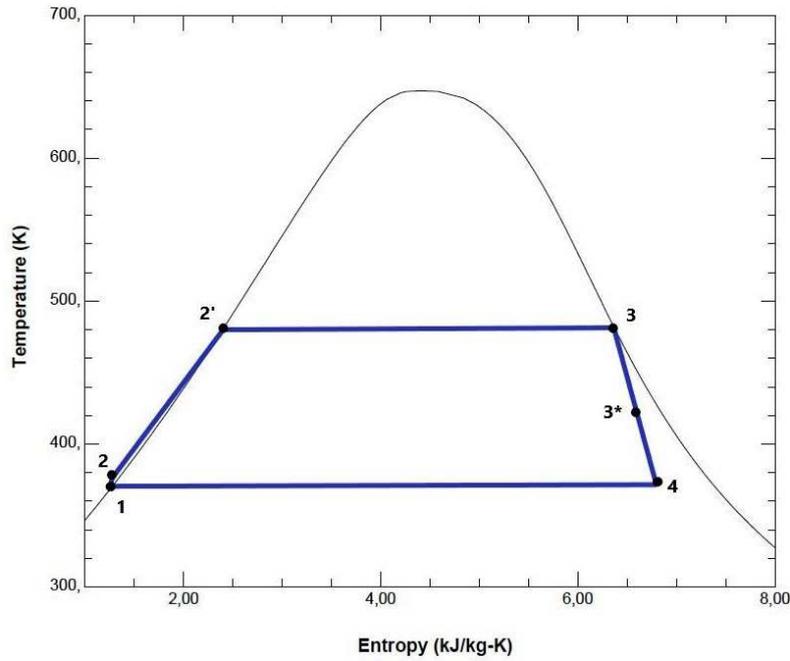Fig. 2. Plant configuration of the PTC-SRC power system

Fig. 3. T-s diagram cycle of the steam Rankine cycle when T_3 is 200 °C and p_4 is 1 bar [12]

Numerical optimization of this sort of SE-based DSG solar plant is improved by linking algorithms established for the energetic assessment of the whole power plant with thermodynamic formulas obtained for the SE part-load behavior [1]. In order to derive maximum solar thermal power efficiency for solar radiation G_b ranging between 900 W/m^2 and 300 W/m^2, numerical optimization considered specific case studies; condensation pressure p_4 is assumed equal to 0.1 bar and 1 bar and evaporation temperature T$_3$ of water (at the SE_1 inlet) is assumed to be fluctuating between 170 °C to 320 °C. In this section, using the mathematical model in question and once all key factors for numerical simulations are fixed (Table 1), the evaporation temperatures will be optimized and calculated with respect to steam Rankine cycle efficiency and solar thermal power generation efficiency for the fixed condensation pressures.

Using Equation (1), in Fig. 4 variations in the solar power collection efficiency with evaporation temperature are shown in the specified solar radiation range. It is evident that the heat collector efficiency η_{PTC} always decreases with evaporation temperature T_3 [1],[12].

Table 1. Set parameters

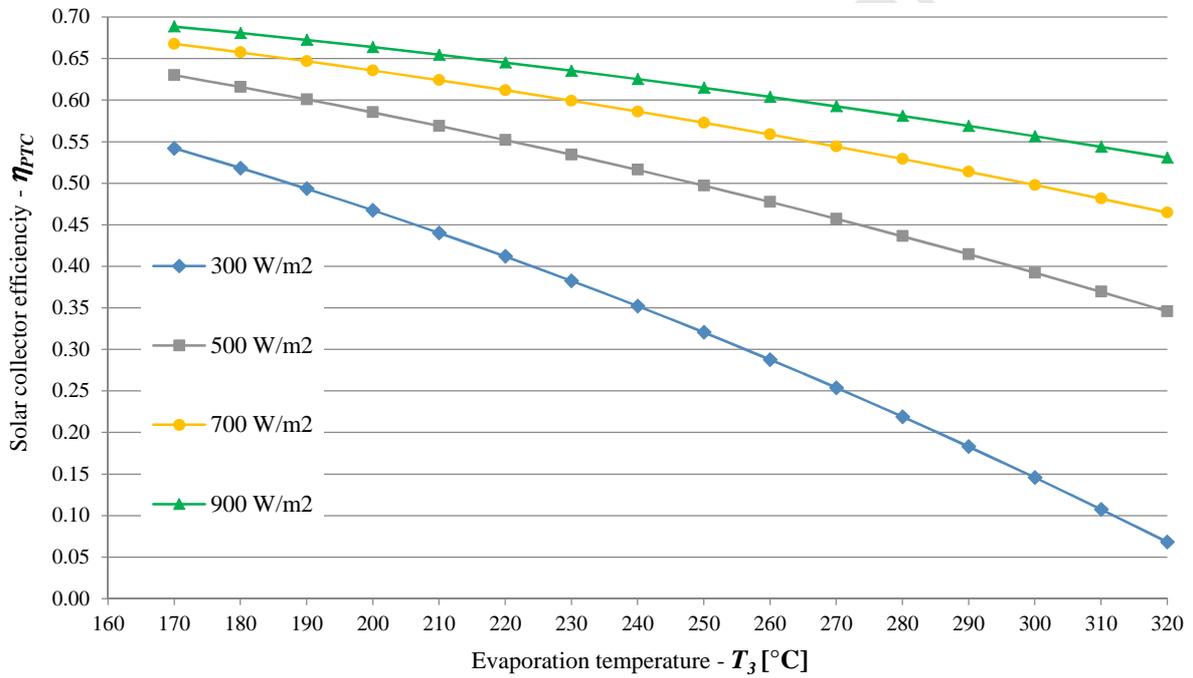| Terms	Value
T_a [C°]: ambient temperature	25
η_{mec}: mechanical efficiency	0.95
η_p: pump efficiency	0.80
*k:*isentropic index	1.13
$r_{v,b}$: SE built-in volume ratio	5
$\eta_{SE,p}$: SE peak isentropic efficiency	0.75

Fig. 4. Heat collection efficiency η_{PTC} with evaporation temperature under increasing solar

radiation

Assuming Equation (11) (which shows variations in SE isentropic efficiency η_{SE} with the

operating pressure ratio r_p), in Fig. 5, η_{SE} is plotted against temperature T_3 when condensation

pressure p_4 is assumed equal to 0.1 bar and 1 bar once the built-in volume ratio $r_{v,b}$ is set (as

shown in Table 1).

For the PTC-SRC power plant proposed in this paper, on analyzing in Fig. 5 the over-expansion

working conditions ($r_p > r_{p,b}$), a decrease in SE efficiency η_{SE} is manifest when actual

condensation pressure drops from 1 bar to 0.1 bar. In effect, with rising evaporation temperature T_3 and decreasing exhaust pressure p_4, the resultant actual expansion ratio r_p increases gradually (compared to the built-in pressure ratio), hence causing off-design working conditions (blowback effect) [12]. Moreover, in Fig. 5, similar decreases in the SE overall efficiency η_{SE} are also manifest when $r_p < r_{p,b}$, namely in under-expansion operating conditions. In fact, in this other case (when exhaust pressure p_4 is assumed equal to 1 bar and evaporation temperature T_3 drops), the resultant actual pressure ratio r_p becomes progressively lower than the built-in expansion ratio, thereby causing blowdown operating conditions [5]. In effect, the maximum SE isentropic efficiency falls when the real expansion ratios r_p of both screw expanders reach the built-in expansion ratio $r_{p,b}$, whereas the mismatch of the working expansion ratio r_p with the built-in expansion ratio involves two opposite conditions: a blowback effect during the over-expansion process and a blowdown effect during the under-expansion process [1].
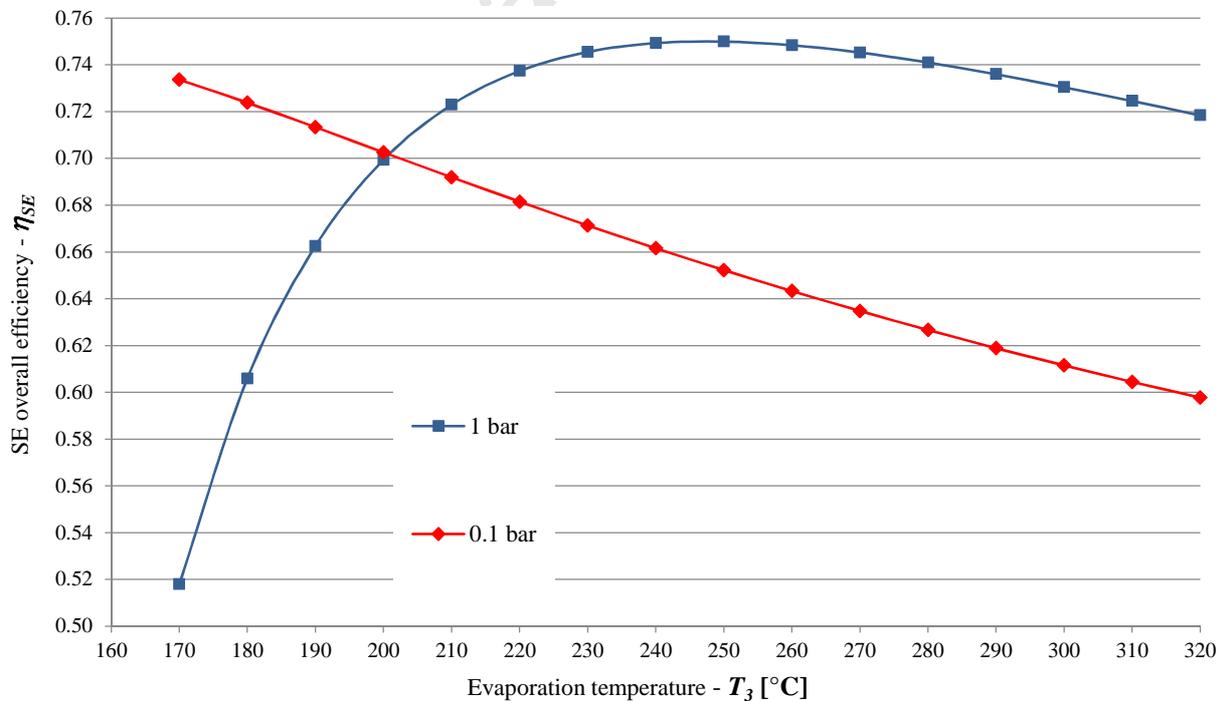

Fig. 5. Variations in SE overall efficiency with evaporation temperature under condensation

pressure assumed equal to 0.1 bar and 1 bar

Using equation (12), η_{SRC} (steam Rankine cycle efficiency) results from SE isentropic efficiency η_{SE} and real pressure ratio r_p. In Fig. 6, this heat-to-power conversion efficiency is shown against evaporation temperature under condensation pressure assumed equal to 0.1 bar and 1 bar. For low evaporation temperature and high condensation pressure, the steam Rankine cycle efficiency is clearly damaged both by a decrease in the available enthalpy of expansion and a decrease in SE isentropic efficiency η_{SE} due to extreme under-expansion conditions (the resultant real pressure ratio r_p decreases overall in comparison with the SE built-in expansion ratio, thereby causing the blowdown effect which induces poor SE performance) [1].

By contrast, steam Rankine cycle efficiency always increases with rising temperature T_3 when condensation pressure p_4 is assumed equal to 1 bar. Consequently, under such working conditions, a rise in the obtainable enthalpy of expansion with a rising evaporation temperature overcomes a reduction in overall efficiency of the screw expanders (due to under-expansion conditions). However, when condensation pressure p_4 is assumed equal to 0.1 bar, efficiency η_{SRC} becomes almost constant with rising temperature T_3 because a decrease in SE overall efficiency as evaporation temperature increases equipoises a rise in the available enthalpy of expansion [1]. Indeed, in this specific case, the heat-to-power conversion efficiency is damaged by extreme over-expansion conditions due to the blowback effect [12].
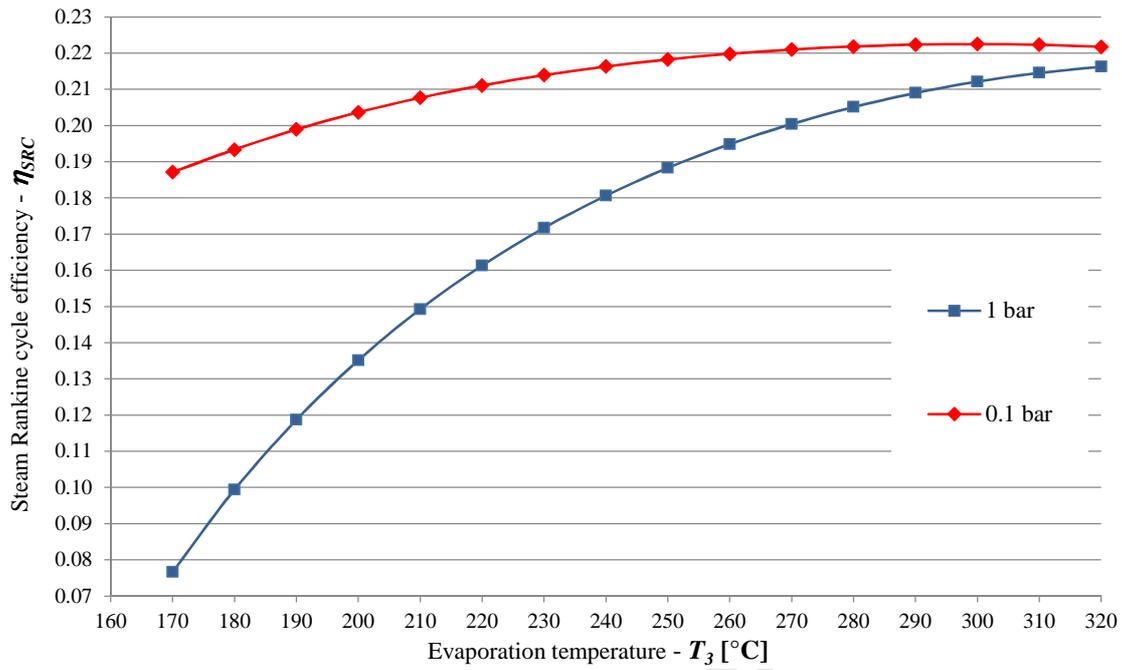
Fig. 6. Steam Rankine cycle efficiency versus evaporation temperature under condensation

pressure assumed equal to 0.1 bar and 1 bar

For the DSG power plant proposed in this study, by adopting Equation (14), variations in solar

thermal power generation efficiency with evaporation temperature are calculated in Fig. 7 for

condensation pressures assumed equal to 0.1 bar (red lines) and 1 bar (blu lines). As expounded

in Equation (13), this overall efficiency η_G depends on heat collection efficiency η_{PTC}, heat-to-

power conversion efficiency η_{SRC} and mechanical efficiency η_{mec}. Hence, once the SE built-in

volume ratio is fixed as in Table 1, the global efficiency η_G is a function of evaporation

temperature T_3, actual expansion ratio r_p and beam solar radiation G_b [27],[28].

When condensation pressure is assumed equal to 0.1 bar, Fig. 7 shows that low evaporation

temperatures produce a little lower or almost constant overall efficiencies η_G in comparison with

its maximum values. Instead, for high temperature T_3 (thus in excessive over-expansion

conditions), the global efficiency η_G decreases due to energy losses from the blowback effect.

On the other hand, when the condensation pressure is assumed equal to 1 bar and under beam

solar radiations higher than 500 W/m^2, it is evident that high temperatures T_3 lead to moderately

lower overall efficiencies η_G. By contrast, for low temperatures T_3 the global efficiency η_G is damaged by extreme under-expansion conditions which cause poor SE performance (blowdown effect) [12].

Subsequently, in order to detect optimum operating conditions under the fixed condensation pressures, the first derivative of each of these functions indicating variations in global efficiency η_G with solar radiation G_b and evaporation temperature T_3 were calculated and fixed to zero [1]. Using this analytical procedure, variations in the optimum evaporation temperature $T_{3,op}$ and the resultant maximum solar thermal power efficiency $\eta_{G,op}$ are calculated and shown in Fig. 8 and Fig. 9, respectively.

In effect, the optimum global efficiencies are attained when the condensation pressure is assumed equal to 0.1 bar, because a rise in the obtainable enthalpy of expansion with a declining condensation pressure p_4 overcomes a reduction in overall efficiency of screw expanders in over-expansion working conditions. The results reported in Fig. 8 indicate that, when condensation pressure is assumed equal to 0.1 bar, the optimum evaporation temperature rises from around 161 °C to 221 °C with growing solar radiation. On analyzing Fig. 9, under this specific operating condition ($p_4 = 0.1$ bar), the maximum overall efficiency $\eta_{G,op}$ increases from around 9.9% to 13.2% when solar radiation rises from 300 W/m^2 to 900 W/m^2. Hence, this SE-based DSG solar plant, in the established range of the optimum working conditions, can operate at lower evaporation temperatures than similar DSG power plants adopting steam turbines, without considerable reduction in overall efficiency [1].
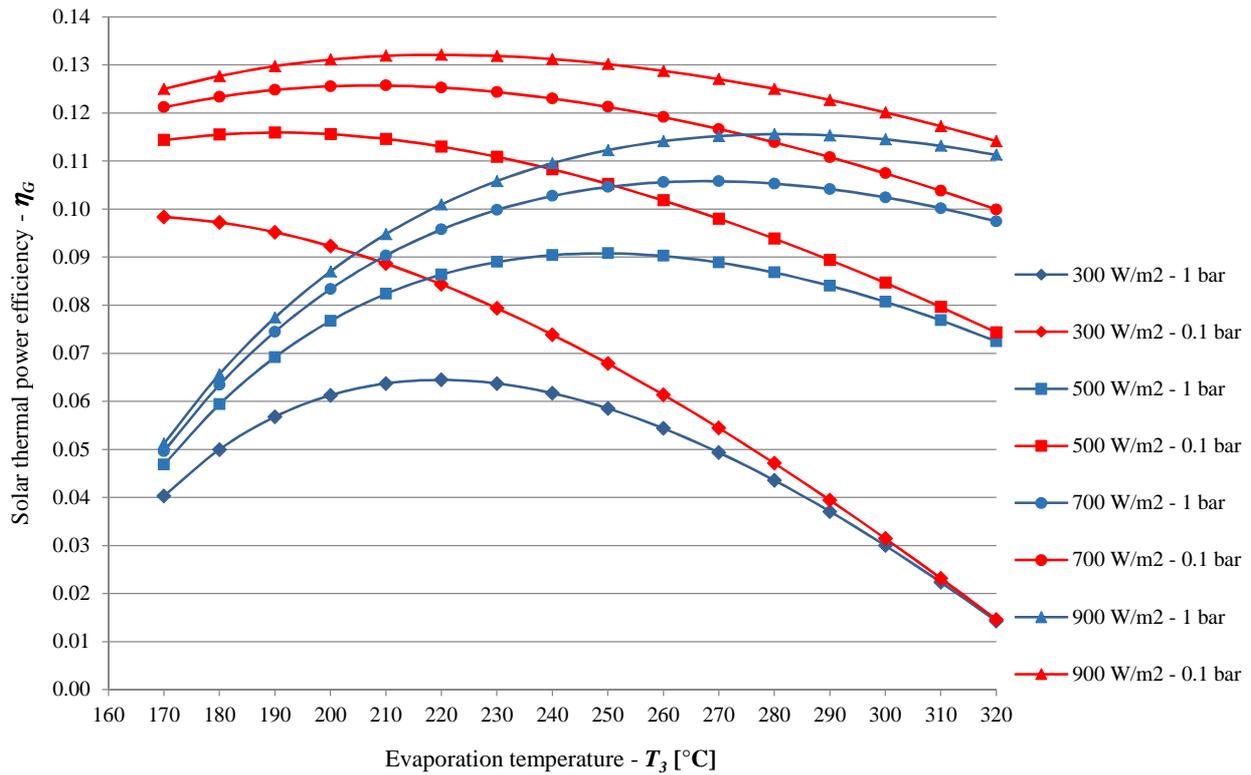
Fig. 7. Variations in global efficiency with evaporation temperature when condensation pressure

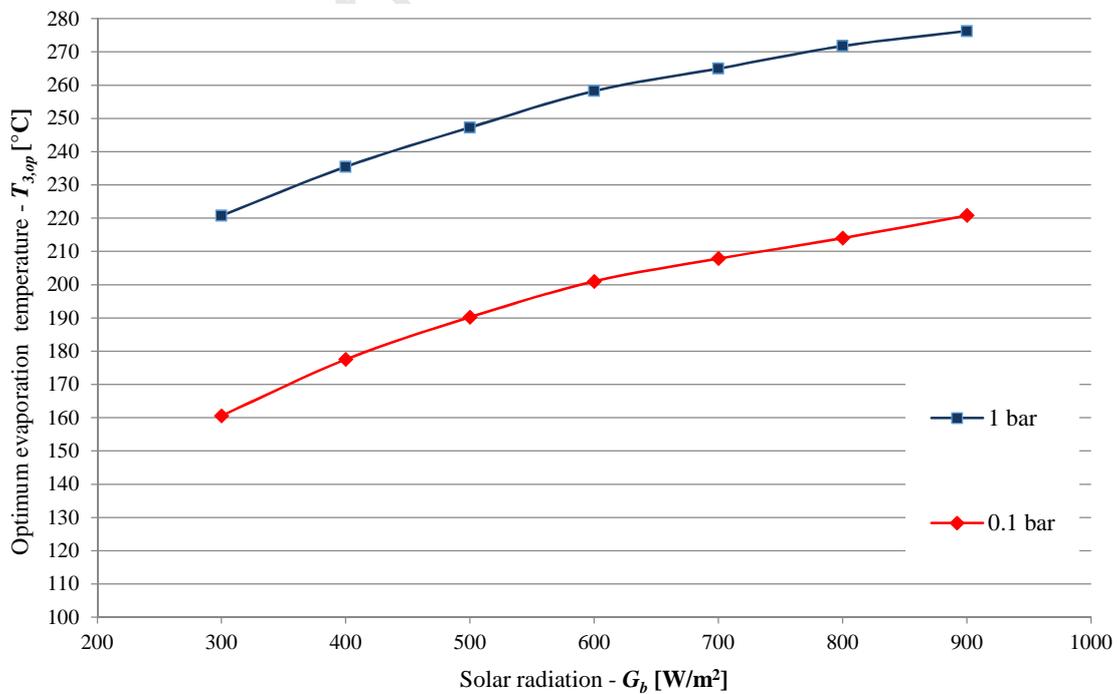is assumed equal to 1 bar and 0.1 bar under each solar radiation.

Fig. 8. Variations in optimum evaporation temperature with solar radiation when condensation

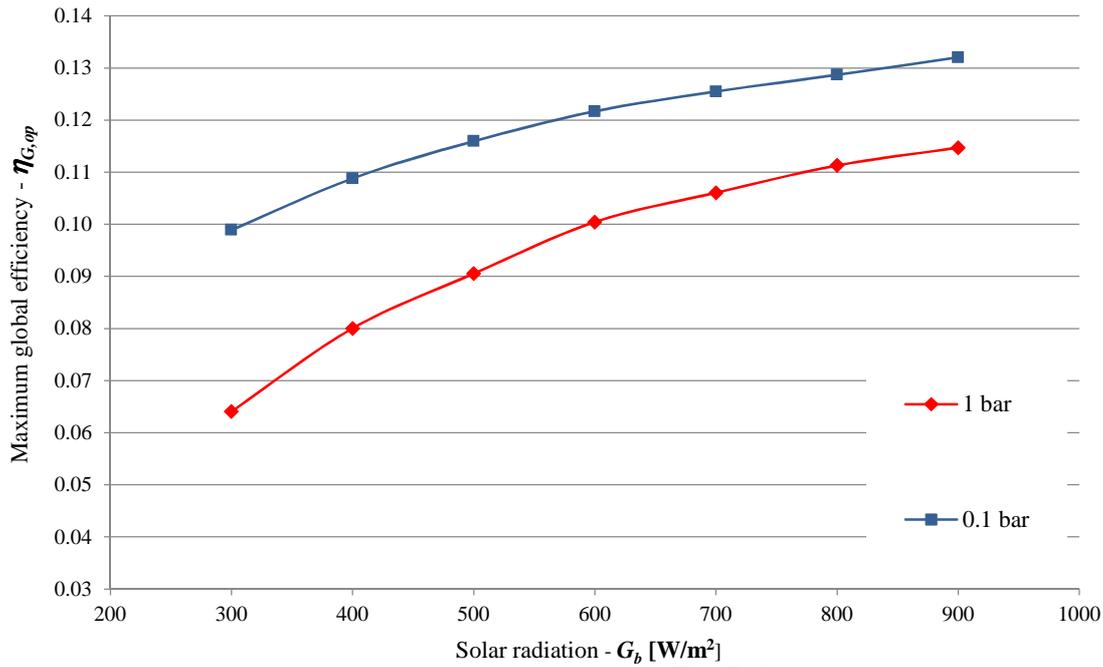pressure is assumed equal to 1 bar and 0.1 bar

Fig. 9. Variations in optimum overall efficiency with solar radiation when condensation pressure is assumed equal to 1 bar and 0.1 bar

Conclusions

The PTC-SRC power plants based on steam SEs offer several advantages regarding the low temperature and pressure in the solar field, reduced size and low technical requirements. The solar electricity generation plant analyzed in this study adopts screw expanders and parabolic trough collectors, and is based on the steam Rankine cycle. To assess best operating conditions and maximize efficiency of the whole DSG power plant at part-load working conditions, parametric optimization was then improved in a wide range of variable evaporation temperatures under fixed condensation pressures.

The chief conclusions on the energy performance and parametric optimization established on this kind of SE-based DSG solar plant can be summarized as follows:

- Under condensation pressure assumed equal to 0.1 bar, low evaporation temperatures produce a little lower or nearly constant global efficiencies in comparison with its

maximum levels. Instead, for high evaporation temperatures, the global efficiency decreases owing to energy losses from the blowback effect.

- Under condensation pressure assumed equal to 1 bar and when beam solar radiation is higher than 500 W/m^2, high evaporation temperatures lead to moderately lower global efficiencies. By contrast, for low evaporation temperatures, the overall efficiency is damaged by extreme under-expansion conditions which cause reduced SE performance.

- The best global efficiencies are reached when the condensation pressure is supposed equal to 0.1 bar because a rise in the available enthalpy of expansion with a decreasing exhaust pressure overcomes a reduction in overall efficiency of screw expanders in over-expansion operating conditions. For this specific working condition, the optimum evaporation temperature is around 220 °C under higher solar radiation, with the resultant maximum global efficiency that is around 13.2%.

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Direct steam generation solar systems with screw expanders and parabolic trough collectors: energetic assessment at part-load operating conditions

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The authors state that there are not any financial and personal relationships with other people or organizations that have inappropriately influenced this paper