The Potential of Scheffler-type Solar Receivers in Direct Steam Generation Power Systems

Paolo Iodice, Francesco Saverio Marra, Giuseppe Langella, Amedeo Amoresano

***Abstract*—This paper investigates the potential of Scheffler-type solar receivers used as heat sources for direct steam generation power systems for power up to 500 kW. Scheffler-type receivers provide good energy performance and acceptable efficiency in comparison with other technologies to exploit solar power because the focal receiver can decrease thermal losses even at high degrees of vaporization temperature. In this research, comprehensive evaluation and energy performance optimization of the proposed solar receiver are thoroughly obtained in a broad range of working states and under variable solar irradiations. For this aim, numerical optimization of the main thermodynamic parameters is conducted via a specific thermodynamic model to compute each energy loss which affects the heat transmission process in the cavity receiver, so calculating the energetic efficiency of the considered solar receivers at part-load working states. Power plants based on Scheffler solar systems can become a hopeful technology for civil applications since they can be adopted to supply the energy needs of new urban settlements while ensuring, at the same time, easy construction, reduced expenses and acceptable energy conversion efficiency.**

***Index Terms*—energetic performance optimization, renewable energy resources, Scheffler solar receiver, solar thermal power generation**

I. Introduction

IN the last years, due to energetic crisis alertness and serious environmental alarms (such as global warming and pollutant emissions), many studies have conducting on advanced technological solutions to exploit renewable energy sources that are alluring rising attention worldwide [1],[2]. Presently, solar power is the most ample and handy amongst other types of renewable energy resources [3]. In this regard, photovoltaic and solar thermal power generation are two well-known technologies that could replace conventional power plants based on fossil fuel for power lower than 1 MW. However, nowadays the development of photovoltaic power generation is penalized by low energy conversion efficiency in exploitation of solar power and high production costs [4].

Manuscript received April 27, 2024; revised December 20, 2024.

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Nevertheless, as is known, nowadays the development of photovoltaic power generation is penalized by both low energy conversion efficiency in exploitation of solar power and high production costs [4].On the other side, significant technological improvements have been attained in the field of renewable energy power plants based on utilization of solar energy which presently represent a well-known technology [5],[6].

In this regard, currently, concentrated solar power (CSP) plants are often based on direct steam generation (DSG). In such CSP plants, solar power is typically collected in linear Fresnel mirrors or in parabolic trough collectors (PTCs), which are well-known systems to use solar energy; actually, they deliver nearly 85% of the global capacity of solar thermal power systems [7]-[9]. Nevertheless, in PTC solar systems, decrements in solar power collector efficiency are significant at high vaporization temperatures owing to high convective and radiative thermal losses [10]-[12]. Furthermore, in these solar receivers, there is not negligible variability in solar power collection efficiency with variable sun radiation; therefore, under specific weather situations, the reduction in radiation intensity can significantly penalize and reduce the mechanical power of solar power plants based on PTCs [13].

For all these reasons, this study proposes Scheffler (SC) solar receivers as heat sources for innovative DSG solar systems, with water used as both heat transfer and working fluid in a steam Rankine cycle (SRC). These innovative renewable energy power systems could be planned to satisfy the energy demand of small residential areas, so supplying electric power between 10 and 500 kW with good energy conversion efficiency, reduced size and adequate investments and costs.

Given the above, the SC solar receivers are based on an excellence reflector which includes a humble tracking device and a sole fixed focus solar concentration system so that these solar receivers can be simply built and operated with satisfactory expenses all over the word [14]. Really, SC solar systems can be designed to become low-cost lightweight devices; in fact, production and installation expenses of a single SC solar system are around €10000 [15].

Moreover, the energy performance of SC solar systems is better than that of conventional PTCs because of improved compactness and higher efficiency of the focal receiver which can lessen convective and radiative heat losses, even for extreme evaporation temperatures. Besides, unlike traditional PTCs, the decline in solar collection efficiency of SC receivers with decreasing solar radiation intensity is acceptable [16].

In this analysis, to perform the energy assessment of the solar receivers under examination, a specific numeric model for part-load behavior of the SC receiver is examined, so evaluating all the possible energy losses concerning the heat transfer phase in the cavity receiver [17]. Consequently, by implementing this numerical model in specific MATLAB subroutines, thermodynamic optimization of the main parameters is performed, so computing the energetic efficiency of the Scheffler-type receiver at part-load operating states. As key results of the numerical simulations conducted on the planned solar receivers, variations in solar power collection efficiency will be calculated with sun radiance intensity and vaporization temperature.

Clearly, comprehension of the impact of these thermodynamic factors on the energy evaluation of the SC receiver appears essential for the further phases of commercialization of these advanced solar systems applied in DSG power systems. As will be carefully described in the next sections, the findings reached in the present research establish that Scheffler-type receiver could become a hopeful technology for optimal harnessing of solar power in CSP systems based on direct steam generation.

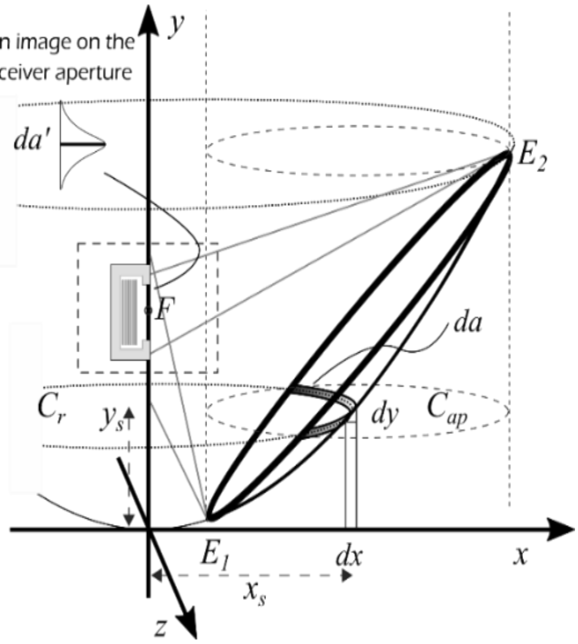

Fig. 1. Scheme of the Scheffler receiver and reflector

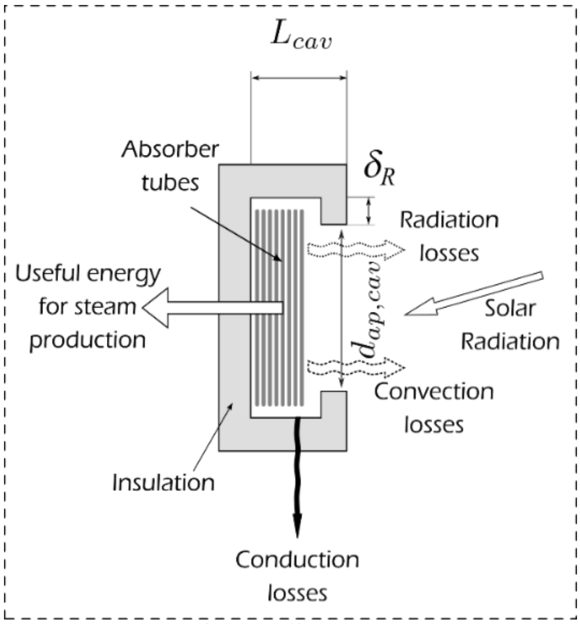

Fig. 2. Expanded design of the Scheffler receiver

II. Modelling

The energy conversion efficiency of all solar power systems, as well known, always depend on the energy performance of its core elements, hence basic criteria and numerical model for the Scheffler receivers part-load behavior have to be established in this study. Besides, to show performance analysis of Scheffler-type receivers compared to traditional PTC solar systems, thermodynamic models and basic equations of these two types of solar receiver are investigated and exposed in this section.

A. *The Scheffler Numerical Model*

To calculate each energy loss that can possibly impact on the thermal transmission process in the cavity of the Scheffler receiver, a thermal model on this solar system is proposed and evaluated [18].

By applying the receiver thermal model examined by Fraser [19]-[21], first the receiver absorbs solar energy from the reflector (Fig. 1) and then it transfers thermal power to the working fluid in a cavity receiver (Fig. 2). Such cavity receiver uses solar energy from the reflector through a bundle of cylindrical pipes and, after all the energetic losses affecting the heat transfer phase, it collects the remaining thermal energy in the working fluid [16],[22], as described below.

The energy balance in the receiver is described in Eq. (1), where the net heat available to transfer at the working fluid, Q_{av}, is evaluated as the difference between the heat gathered in the cavity, Q_{rec}, and all heat losses Q_{loss} including convection, conduction, and radiation losses [14],[18],[21]-[23].

$$Q_{av} = Q_{rec} - Q_{loss} = Q_{rec} - Q_{rad,ref} - Q_{rad,emi} - Q_{conv,nat} - Q_{conv,for} - Q_{conv,cond} \quad (1)$$

To evaluate the level of thermal energy stored in the cavity, Q_{rec}, it can be used Eq. (2), where I_d is the solar irradiation (W/m^2), $A_{ap,ref}$ (m^2) represents the aperture area of the reflector, ρ (-) is the surface reflection coefficient and φ (-) denotes the interception factor of the SC collector [17]. All factors reported in Eq. (2) are fixed, excepting the interception parameter that is computed as the fraction between the power intercepted by the receiver $P_{intercept,tot}$ and the total power reflected by the collector $P_{reflect,tot}$ (see Eq. (3)) [24],[25].

$$Q_{rec} = I_d\, A_{ap,ref}\, \rho\, \varphi \quad (2)$$

$$\varphi = \frac{P_{intercept,tot}}{P_{reflect,tot}} = \frac{\int_A^B I(a)\, da}{\int_{-\infty}^{\infty} I(a)\, da} \quad (3)$$

To evaluate the integral in the numerator of Eq. (3), the contribution of an element $dA_s = ds\, x_s\, d\theta$ of the mirror surface is considered. dA_s is captured by the element dx along the x axis (see Fig. 1) and the element of arc $x_s\, d\theta$ of the circumference C_r revolving the paraboloid at the same position and having limits on the elliptical boundary of the mirror (with coordinate $\pm z_b$ along the z axis), hence depicting the shaded area da indicated in Fig. 1. Since $ds^2 = dx^2 + dy^2$, Eq. (4) is obtained by replacement of the equation of the parabola:

$$ds = \sqrt{1 + \left(\frac{x}{2f}\right)^2}\, dx \tag{4}$$

Coordinates $\pm z_b$ as a function of x_s are calculated by overlapping the equation of the circumference C_r with radius x_s and the circumference C_{ap} of the sun interception area lying on the plane $y = y_s$ and having the center at point (x_c, y_s) and radius r:

$$\begin{cases} (x - x_c)^2 + z^2 = r^2 \\ \quad x^2 + z^2 = x_s^2 \end{cases}$$

whose solutions are:

$$x_b = \frac{x_c^2 + x_s^2 - r^2}{2\, x_c}, \quad z_b = \pm\sqrt{x_s^2 - x_b^2}$$

Let $\theta_b = \arcsin(z_b/x_s)$. The integral in the numerator of Eq. (3) can be obtained as in Eq. (5), and the integral reported in denominator of Eq. (3) is calculated as $P_{reflect,tot} = I_d \pi r^2$.

$$\int_{E_1}^{E_2} I(a)\, da = \int_{x_1}^{x_2} \int_{-z_b}^{+z_b} I(a)\, dA_s = \int_{x_1}^{x_2} \int_{-\theta_b}^{+\theta_b} I_d \cos(\psi/2) \sqrt{1 + \left(\frac{x_s}{2f}\right)^2}\, x_s\, d\theta\, dx \tag{5}$$

The maximum error of the polynomial approximation $I(a)$ is presumed as a continuous function in the SC concentrators. The normal distribution of the focal point radiation is reported in Eq. (6), with $t = [1 + a\, \nu(x_s)/2]^{-1}$. The value of $\nu(x_s)$ depends on the diameter of the aperture of the receiver cavity. The elements appearing in Eq. (6) are expressed in Table 1; this equation is resolved when the rest *res* is presumed very small.

$$I(x) = I_d \left(1 - \frac{2}{\sqrt{2\pi}} e^{-\frac{\nu(x)^2}{8}} (k_1 t + k_2 t^2 + k_3 t^3 + k_4 t^4 + k_5 t^5) + 2\, res\right) \tag{6}$$

Successively, the interception factor φ of Scheffler collector can be determined by replacing the geometric, statistical and optical values (depending on the dimensions and characteristics of the Scheffler concentrator) in Eq. (3) and Eq. (5), and solving the integral of Eq. (5), so obtaining Eq. (7).

TABLE I
CONSTRAINTS OF THE NORMAL DISTRIBUTION.

a	*k1*	*k2*	*k3*	*k4*	*k5*
0.23164	0.31938	-0.3565	1.78147	-1.82125	1.33027

$$\varphi = \frac{\sum_{x_1}^{x_2} \sum_{-\theta_b}^{\theta_b} I(x_s) \cos(\psi/2) \sqrt{1 + \left(\frac{x}{2f}\right)^2}\, x_s \Delta x \Delta\theta}{I_d(\pi r^2)} \tag{7}$$

Once the energy stored in the cavity is established, all the energy losses Q_{loss} can be estimated by determining each contribution illustrated in Eq. (1). The heat losses due to the reflected radiation, $Q_{rad,ref}$, are analysed using Eq. (8). In this equation, $(1 - \alpha_{eff})$ symbolizes the reflectance of the cavity calculated in Eq. (9), with $A_{ap,cav}$ representing the aperture area of the cavity and α_{cav} that corresponds to the absorbance factor.

$$Q_{rad,ref} = (1 - \alpha_{eff}) Q_{rec} \tag{8}$$

$$\alpha_{eff} = \left(\frac{\alpha_{cav}}{\alpha_{cav} + (1 - \alpha_{cav})\left(\frac{A_{ap,cav}}{A_{cav}}\right)}\right) \tag{9}$$

Radiation loss $Q_{rad,emi}$ owing to emitted radiation is determined by applying Eq. (10), where T_{cav} is the temperature of the cavity and T_{amb} is the ambient temperature.

$$Q_{rad,emi} = \varepsilon_{eff} \sigma^* A_{int,cav} (T_{cav}^4 - T_{amb}^4) \tag{10}$$

In the expression of Eq. (10), $A_{int,cav}$ indicates the internal surface of the cavity and the effective emission factor, ε_{eff}, is estimated as in Eq. (11) [24], in which L_{cav} represents the cavity length (see Fig. 1) and by accepting $\varepsilon_{cav} = \alpha_{cav}$.

$$\varepsilon_{eff} = \left(\frac{(1 - \varepsilon_{cav})}{\varepsilon_{cav}\left(1 + \frac{4 L_{cav}}{2 R_{cav}}\right)} + 1\right)^{-1} \tag{11}$$

The impact of convection heat losses on the energy performance of a cavity receiver is very important: they refer to the heat flow that comes out of the cavity opening due to the heating of the air inside it which rises by buoyancy or by pressure exerted by the wind. Following this model, two different convection heat losses are assessed: forced and natural convection losses. The natural convection heat loss in a cavity receiver, $Q_{conv,nat}$, is calculated in Eq. (12), where $A_{int,cav}$ denotes the internal surface of the cavity and $h_{int,nat}$ represents the natural convection coefficient that is assessed by using the Nusselt number.

$$Q_{conv,nat} = h_{int,nat}\, A_{int,cav} (T_{cav} - T_{amb}) \tag{12}$$

Eq. (13) and Eq. (14) can be used to calculate the Nusselt's number that depends on the geometry and temperature of the cavity, where $d_{ap,cav}$ indicates the aperture diameter of the cavity, θ denotes the inclination angle of the cavity and Gr represents the Grashof number for natural convection inside the cavity (as reported in Eq. (15). In this last equation, g correspond to the gravitational acceleration, while ν_{cav} and β_{cav} denote, respectively, the kinematic viscosity and the isobaric compressibility coefficient of the gas in the cavity receiver.

$$Nu_{int,nat} = 0.088\ \mathrm{Gr}^{\frac{1}{3}} \left(\frac{T_{cav}}{T_{amb}}\right)^{0.18} \cos(\theta)^{2.47} \left(\frac{d_{ap,cav}}{d_{cav}}\right)^{m} \quad (13)$$

$$m = -0.982 \left(\frac{d_{ap,cav}}{d_{cav}}\right) + 1.12 \quad (14)$$

$$Gr = \frac{g\beta_{cav}\,(T_{cav}-T_{amb})d_{cav}^{3}}{\nu_{cav}^{2}} \quad (15)$$

The forced convection loss is determined as the sum of two different parts, caused by lateral wind and front wind. The heat transfer coefficient due to lateral wind V_s is assessed in Eq. (16). The heat transfer coefficient caused by frontal winds V_f is computed in Eq. (17), where $f(\theta)$ is calculated in Eq. (18). Thus, the forced convection loss, $Q_{conv,for}$, can be evaluated as in Eq. (19):

$$h_{forced,side-on} = 0.1967\, V_s^{1.849} \quad (16)$$

$$h_{forced,head-on} = f(\theta)\, V_f^{1.401} \quad (17)$$

$$f(\theta) = 0.163 + 0.749\sin(\theta) - 0.502\sin(2\theta) + 0.327\sin(3\theta) \quad (18)$$

$$Q_{conv,for} = \left(h_{forced,side-on} + h_{forced,head-on}\right) A_{int,cav}(T_{cav} - T_{amb}) \quad (19)$$

Finally, conduction heat loss is transported through the inner walls of the cavity in the direction of the outward ambient through convection process (Eq. (20) [16]. The convection heat transfer coefficient, $h_{ext,cav}$, comprises both forced and natural convection coefficients, as indicated in Eq. (21). To assess the natural and forced convection coefficients ($h_{ext,nat}$ and $h_{ext,for}$) on the outside surface of the cavity ($A_{ext,cav}$) when orientated vertically, the Nusselt number is calculated as in Eq. (22) and Eq. (23), where *Pr* and *Re* are the Prandtl and Reynolds numbers, respectively.

$$Q_{conv,cond} = \frac{(T_{cav}-T_{amb})}{\frac{L}{k_{int}A_{int,cav}} + \frac{1}{h_{ext,cav}A_{ext,cav}}} \quad (20)$$

$$h_{ext,cav} = (h_{ext,nat}^{3} + h_{ext,for}^{3})^{\frac{1}{3}} \quad (21)$$

$$Nu_{ext,nat} = 0.27 Re^{1/4} \quad (22)$$

$$Nu_{ext,for} = 0.664 Re^{1/2} Pr^{1/3} \quad (23)$$

Conclusively, the overall solar power collector efficiency η_{SOL} of SC receivers is assessed as in Eq. (24), calculating the fraction between the net thermal power Q_{av} effectively transferred to the working fluid and the total sun irradiation $I_d{\cdot}A_{ap,ref}$. Thus, this efficiency includes all the potential optical losses and thermal dispersions of Scheffler receivers.

$$\eta_{SOL} = \frac{Q_{av}}{I_d{\cdot}A_{ap,ref}} \quad (24)$$

B. *The PTC Numerical Model*

Solar power systems based on PTCs are considered a well-known technical solution in the field of solar thermal power generation. Really, such CSP plants account for nearly 85% of the total capacity of actual solar power systems. Farther, solar power systems based on PTCs represent a favorable technology to exploit solar energy also for cost savings [18].

The solar collector efficiency, η_{PTC}, of a sole PTC can be evaluated as in Eq. (25), in which T_a (K) denotes the ambient temperature, T (K) represents the temperature at the PTC inlet and I_d (W/m^2) is the solar irradiation [6].

$$\eta_{PTC}(T) = 0.762 - 0.2125 \cdot \frac{T-T_a}{I_d} - 0.00167 \cdot \frac{(T-T_a)^2}{I_d} \quad (25)$$

In CSP systems using PTCs, many elementary collectors are positioned thus, to appraise the whole solar collector efficiency, the average operating temperature of adjacent basic modules is presumed to change equally from one basic collector to another.

Nonetheless, in CSP systems based on PTCs, the solar field involves vapor-liquid blend. Really, water, that heats up flowing in the PTCs, is both in binary phase and liquid phase region; consequently, at PTC outlet, it converts in dry saturated steam [6].

In liquid phase region, the required collector area A_l to achieve a specific outlet temperature T_{out}, starting from a fixed inlet temperature T_{in}, is assessed as in Eq. (26), where $C_p(T)$ is the thermal capacity of water in the liquid state and is assessed in Eq. (27) by first order approximation. In Eq. (27), T represents the temperature of water in the liquid state ranging between T_{in} and T_{out}, $C_{p,0}$ indicates the thermal capacity of water at the reference temperature T_0 (which is the PTC inlet temperature T_{in} in this case) and α is the slope of the first order approximation [6].

$$A_l = \int_{T_{in}}^{T_{out}} \frac{\dot{m}\cdot C_p(T)}{\eta_{PTC}(T)\cdot G_b}\, dT \quad (26)$$

$$C_p(T) = C_{p,0} + \alpha(T - T_0) \quad (27)$$

Once the analytic solution to the integral of Eq. (26) is obtained, solar energy collector efficiency in liquid state is calculated in Eq. (28), where $\dot{m}$ represents the mass flow rate of water flowing in the PTCs and Δh_l is the enthalpy increase in the liquid phase region of water.

$$\eta_{PTC,l} = \frac{\dot{m}\cdot\Delta h_l}{A_l\cdot I_d} \quad (28)$$

Evidently, the solar collection efficiency in binary phase region, $\eta_{PTC,b}$, can be evaluated as in Eq. (25) because water temperature is always constant in this region. Therefore, the necessary collector area, A_b, in binary phase region is assessed in Eq. (29), in which Δh_b indicates the enthalpy rise in the binary phase region of water.

$$A_b = \frac{\dot{m}\cdot\Delta h_b}{\eta_{PTC,b}\cdot G_b} \quad (29)$$

Then, for solar power systems based on PTCs, the entire solar collection efficiency is determined in Eq. (30), since the solar field involves liquid-steam mix. Obviously, in such an equation, $\dot{Q}$ signifies the heat thermal power transferred to the water (both in the binary phase region and liquid phase region).

$$\eta_{PTC} = \frac{\dot{Q}}{I_d \cdot (A_l + A_b)} = \frac{\Delta h_l + \Delta h_b}{\frac{\Delta h_l}{\eta_{PTC,l}} + \frac{\Delta h_b}{\eta_{PTC,b}}} \tag{30}$$

Clearly, the PTC solar collection efficiency can also be estimated as in Eq. (31), in which A signifies the global area (specifically the sum of single parabolic collectors) and Δh_{tot} is the total enthalpy rise of water from PTC inlet to PTC outlet [6].

$$\eta_{PTC}(T) = \frac{\dot{Q}}{I_d \cdot A} = \frac{\dot{m} \cdot \Delta h_{tot}}{I_d \cdot A} \tag{31}$$

III. Numerical Results

In this paragraph, to perform a comparison amongst PTC and Scheffler solar systems, a deep evaluation of the relevant energy performance is conducted by adopting the thermodynamic models presented in the preceding sections.

The energy appraisal of the Scheffler-type receiver is steered under numerous operating states by applying the receiver thermal model previously presented. In this way, all potential energetic losses impacting on the thermal transfer process in the cavity receiver can be estimated. Thus, in the present research, a numerical code based on the dedicated thermodynamic model is used to perform the parametric optimization of the chief thermodynamic factors and to calculate the solar collector efficiency of the planned solar system for part-load working situations.

In detail, vaporization temperature of water at the Scheffler outlet is supposed to rise constantly from 180 °C to 300 °C at intervals of 10 °C, while sun radiance intensity I_d ranges from 400 W/m^2 to 1000 W/m^2. Afterwards, if thermodynamic parameters for numerical simulations are set, the SC solar power collection efficiency can be calculated against evaporation temperature by applying the proposed mathematical model for all levels of solar power.

Then, to evaluate all optical and thermal losses which impact on the reflector and receiver of a Scheffler system, SC solar collector efficiency is estimated versus vaporization temperature of water for many levels of solar radiation intensity (Fig. 3). Clearly, such solar collection efficiencies, η_{SOL}, indicate the portion of the whole sun radiation arriving on the reflector ($I_d \cdot A_{ap,ref.}$) that can be efficiently transferred to the operating fluid as net heat power Q_{av} (see Eq. (24)). By observing the results reported Fig. 3, clearly, the solar collection efficiency always declines with rising vaporization temperature under each value of sun irradiance intensity [16].

Heat collection efficiency of PTC receivers is estimated in Fig. 4 against vaporization temperature and solar beam intensity (analogously to Fig. 3). Such results have been attained by applying the thermodynamic model proposed for PTC solar systems in the preceding paragraph.

By assessing and comparing the numerical results of Fig 3 and Fig. 4, it clearly appears that SC solar systems turn out better performing than standard PTC because of adequate efficiency and excellent compactness of the Scheffler focal receivers that are able to cut the heat exchange area with the external environment [14],[15]. Really, the solar collector efficiency of SC solar systems persists adequately high even at high values of vaporization temperature (Fig. 3).

At the same time, for high values of evaporation temperature, the reduction in solar collection efficiency is considerable in the PTC solar systems (Fig. 4), caused by considerable radiative and convective energy losses that increase with increasing evaporation temperatures [6]. In effect, at the highest level of temperature (set at 300 °C in the mathematical elaborations), the PTC solar collection efficiency ranges between 27.0 % and 55.7 % under growing sun beam intensity from 400 W/m^2 to 1000 W/m^2, while the SC thermal collection efficiency ranges between 38.1 % and 62.5 % in the same solar radiation range.

Besides, by assessing a comparison between the numerical results of Fig. 3 and Fig. 4, the reduction in the solar collection efficiency with reducing solar irradiance intensity appears less significant for the Scheffler-type systems than for the PTCs. In fact, under the lowest level of sun radiance intensity I_d (set to 400 W/m^2), the SC thermal collection efficiency increases from 38.1 % to 63.6 % with declining vaporization temperature from 300 °C to 180 °C; the PTC solar collection efficiencies increase from 27.0 % to 54.9 % in the same evaporation temperature range.

Therefore, such an energy benefit can even alleviate the negative effect of low solar intensity on the mechanical energy produced by DSG solar power plants equipped with SC receivers as thermal source compared to comparable solar systems based on PTC receivers [16].

The heat power collection efficiencies assessed for the Scheffler and PTC solar systems are compared, under the same solar radiation intensities, in Fig. 5, Fig. 6 and Fig. 7 at vaporization temperature fixed to 180 °C, 240 °C and 300 °C, respectively. Hence, in these figures, under each set level of vaporization temperature, the efficiencies reached by using the Scheffler-type solar systems for medium solar beam intensities (also lower than 600 W/m^2) can even equalize the highest thermal power collection efficiencies obtained in the PTCs under the maximum solar irradiation (fixed to 1000 W/m^2 in our numerical simulations).

Consequently, all the numerical results shown in this investigation are profitable and helpful to appraise, in real applications, energy benefits of DSG power systems based on SC receivers for several levels of thermal power delivered by these particular solar systems. Indeed, the findings achieved in this research demonstrate that the Scheffler-type receiver represents a hopeful technical solution for solar thermal power generation.

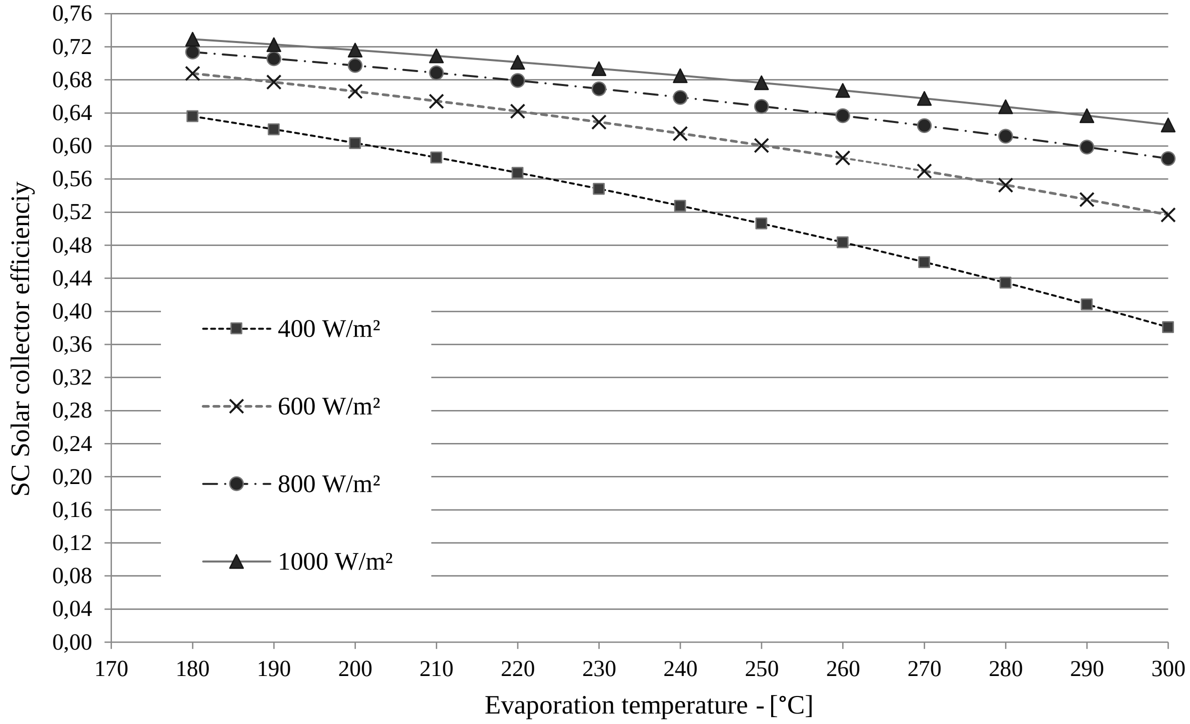

Fig. 3. SC Solar power collection efficiency computed versus evaporation temperature and solar radiance intensity.

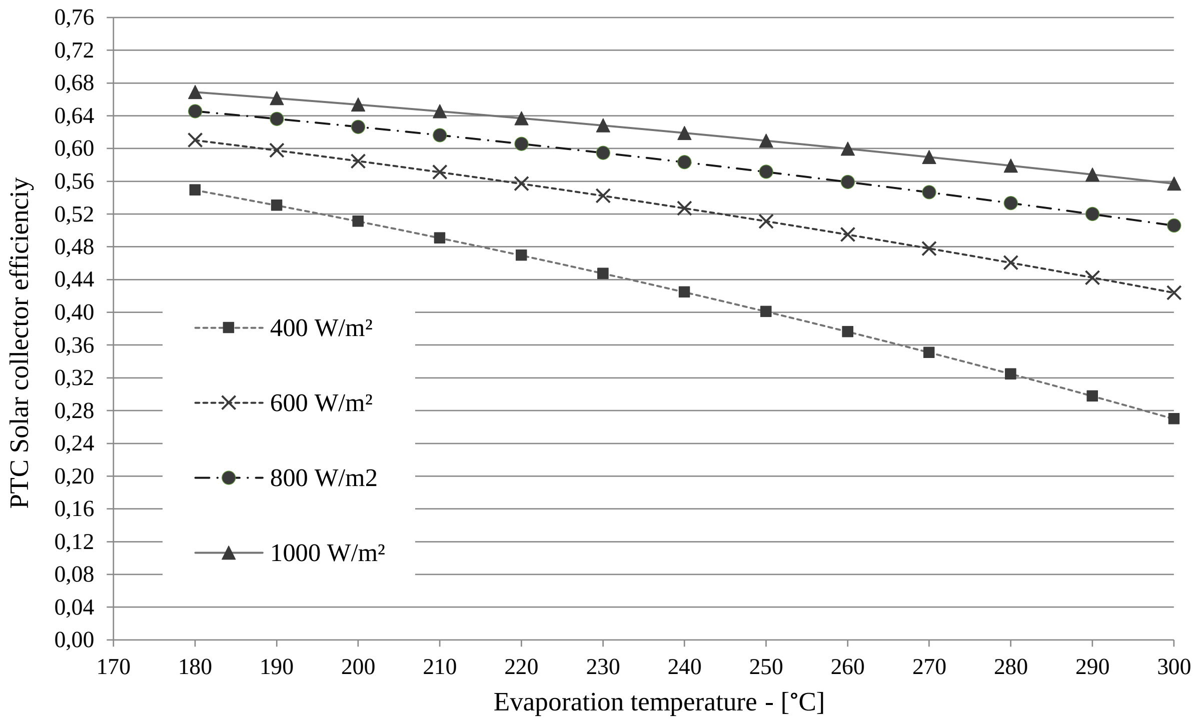

Fig. 4. PTC Solar power collection efficiency computed versus evaporation temperature and solar radiance intensity.

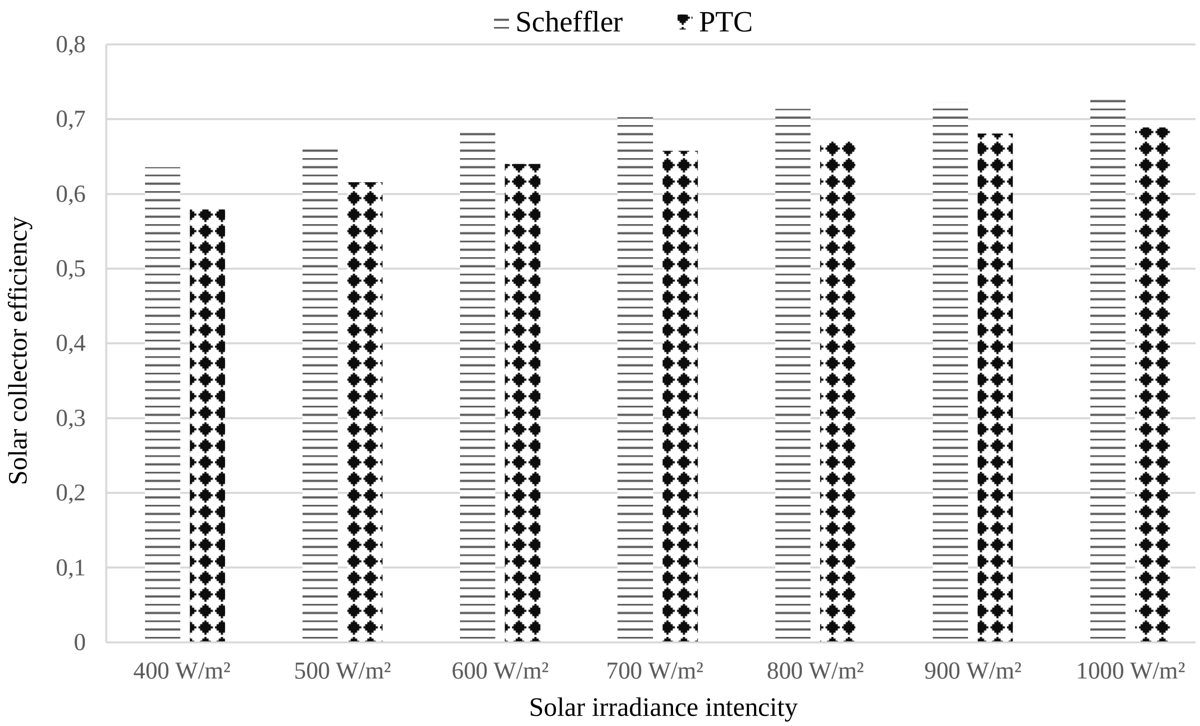

Fig. 5. Solar power collection efficiency evaluated against solar irradiance intensity: relationship between Scheffler and PTC solar systems at vaporization temperature fixed to 180 °C.

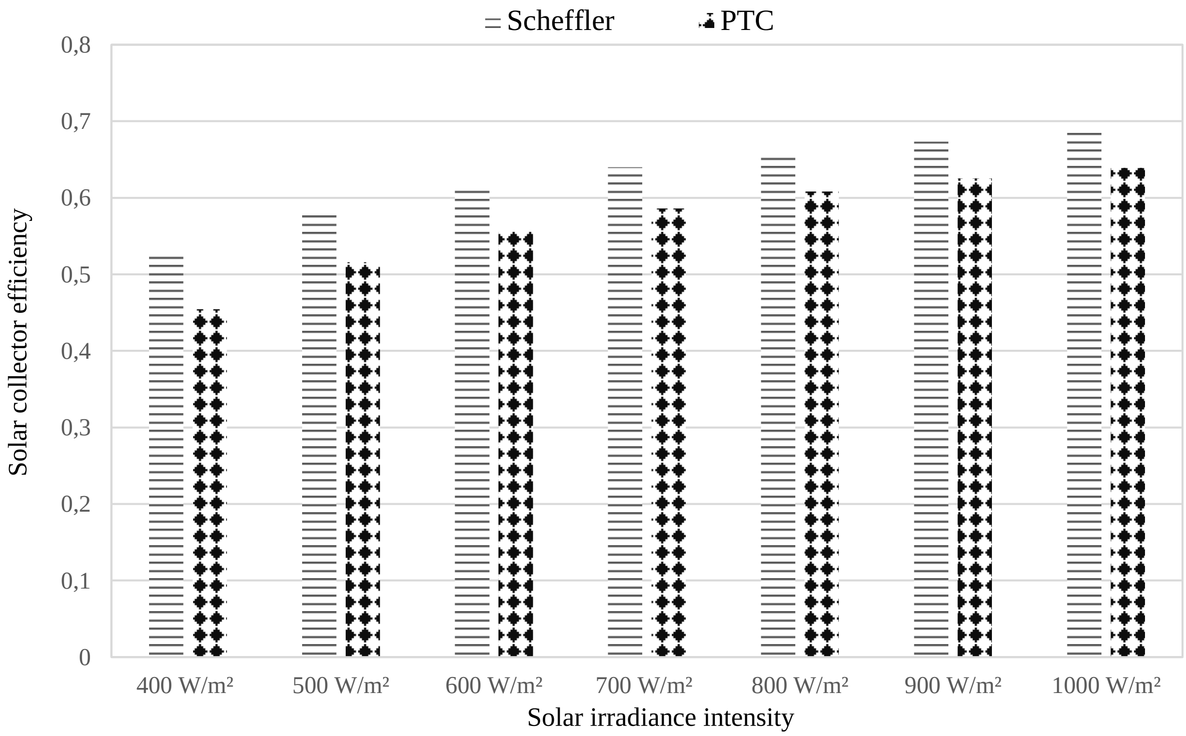

Fig. 6. Solar power collection efficiency evaluated against solar irradiance intensity: relationship between Scheffler and PTC solar systems at vaporization temperature fixed to 240 °C.

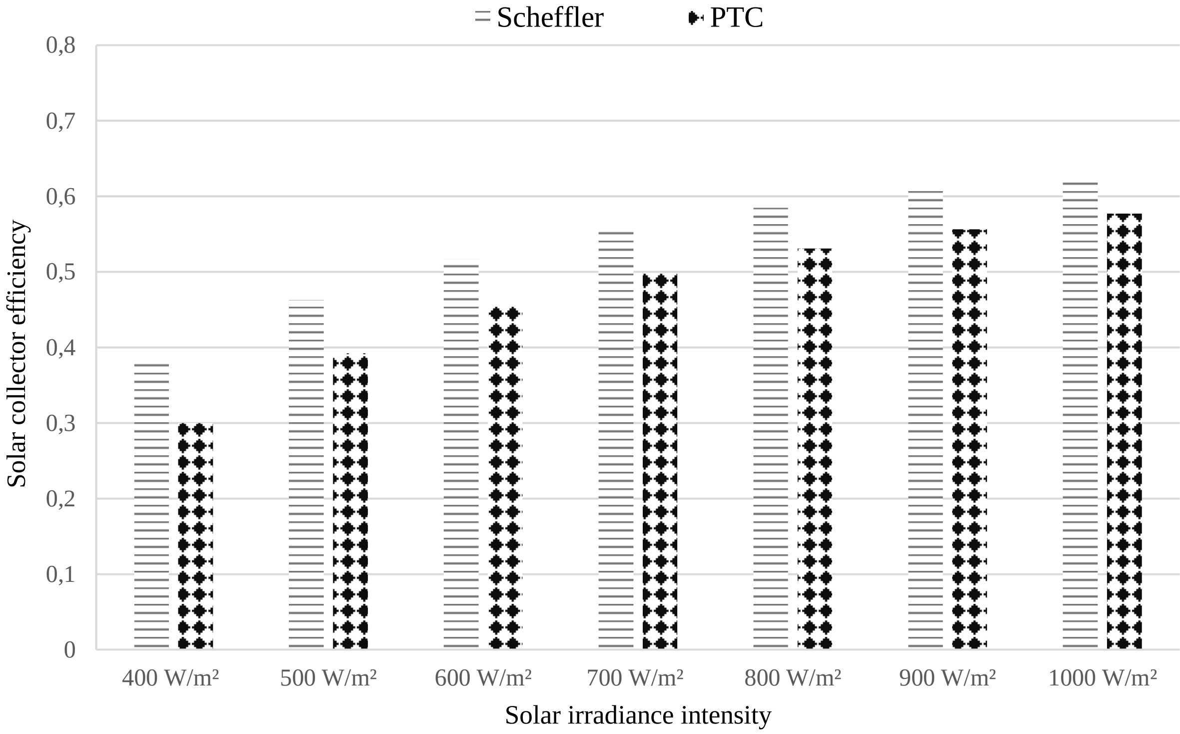

Fig. 7. Solar power collection efficiency evaluated against solar irradiance intensity: relationship between Scheffler and PTC solar systems at vaporization temperature fixed to 300 °C.

IV. Conclusions

The purpose of this investigation was to introduce a dedicated thermodynamic model to assess the energetic performance of Scheffler solar receivers integrated in renewable energy power systems. For this aim, the energy efficiency of this solar receiver was estimated by calculating all energy losses influencing the thermal transfer phase in the cavity receiver. As chief results of numerical simulations based on this model, variations in solar power collector efficiency were computed with sun irradiation intensities and vaporization temperatures.

The main findings obtained in this analysis demonstrate that, for each leval of solar power, the energy performance of Scheffler receivers remains better than that of conventional PTC systems. Besides, the Scheffler solar collection efficiency proves to be adequately high even for high levels of vaporization temperature owing to low thermal losses and good compactness of the focal receiver.

Additionally, Scheffler-type solar receivers persist less sensitive to variations in sun beam intensity when compared with usual PTCs. Thus, for unfavorable weather situations, solar power plants adopting SC receivers as heat sources, in comparison with standard solar power systems which use PTCs, can operate without considerable declines in net power and in energy conversion efficiency.

For all these reasons, this study demonstrates that DSG solar power plants based on Scheffler-type systems can attract increasing consideration as gainful and sustainable solar power systems. In effect, the proposed renewable energy power system can be adopted in the future to supply net power for civil applications from 10 to 500 kW, so providing the energy supplies of slight urban areas with easy construction, reduced investments and satisfactory energy conversion efficiency.

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