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ScienceDirect

Energy Procedia 45 (2014) 101 – 110

Energy Procedia

68th Conference of the Italian Thermal Machines Engineering Association, ATI2013

Comparison of medium-size concentrating solar power plants based on parabolic trough and linear Fresnel collectors

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Abstract

This paper compares the performance of medium-size Concentrating Solar Power (CSP) plants based on an Organic Rankine Cycle (ORC) power generation unit integrated with parabolic trough and linear Fresnel collectors. The CSP plants studied herein use thermal oil as heat transfer fluid and as storage medium in a two-tank direct thermal storage system. The performance of the CSP plants were evaluated on the basis of a 1 MW ORC unit with a conversion efficiency of about 24% and by considering different values of solar multiple and thermal storage capacity. The comparative performance analysis of the two CSP solutions was carried out with reference to the direct solar energy availability of Cagliari, Italy (1720 kWh/m^{2}y) on a yearly basis by means of specifically developed simulation models.

The results of the performance assessment demonstrate that CSP plants based on linear Fresnel collectors lead to higher values of electrical energy production per unit area of occupied land. The highest specific energy production of CSP plants based on linear Fresnel collectors is about 55-60 kWh/y per m^2 of occupied land and it is achieved with solar multiples in the 1.74-2.5 range and storage capacities in the range of 4-12 hours. The highest specific production of the solutions based on parabolic trough collectors is about 45-50 kWh/y per m^2 of occupied land and is achieved with lower solar multiples (around 1.5-2.3). Owing to their better optical efficiency, the use of parabolic troughs gives better values of energy production per unit area of solar collector (about 180-190 kWh/m^2 vs. 130-140 kWh/m^2).

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Keywords: CSP plants; Parabolic trough collectors; Linear Fresnel Collector

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doi:10.1016/j.egypro.2014.01.012

Nomenclature

A	Area	γ	Azimuth angle
F	Focal length	η	Efficiency
L	Collector length	CSP	Concentrating Solar Power
m	Mass flow	DNI	Direct Normal Irradiation
R	Lines distance	IAM	Incidence Angle Modifier
Q	Thermal power	LFC	Linear Fresnel Collector
T	Temperature	ORC	Organic Rankine Cycle
W	Collector width	PTC	Parabolic Trough Collector
α	Elevation angle	SC	Storage Capacity
θ	Incidence angle	SM	Solar Multiple

Introduction

Power generation from solar energy is one of the most interesting options in reducing fossil fuel consumption and related CO_2 emissions. Nowadays, one of the most effective solutions for power generation from solar energy is represented by Concentrating Solar Power (CSP) systems. The current CSP world generating capacity is around 3000 MW, more than 2000 MW of additional capacity is currently under construction and an installed CSP capacity of about 10 GW is expected before 2015. Spain is the country with the highest CSP production, thanks to the operation of more than 40 power plants with an installed capacity of more than 2 GW [1-2].

Different options are available for solar fields (parabolic trough, linear Fresnel, solar tower and solar dish systems), power block (steam Rankine and organic Rankine cycles, Stirling engines, etc.), heat transfer fluid (thermal oil, molten salts, steam, etc.) and storage medium (thermal oil, molten salts, solid rocks, etc.). Today, the construction of large-size CSP plants (over 20-50 MWe) based on Parabolic Trough Collectors (PTC) using thermal oil as heat transfer fluid, molten salts as storage medium and integrated with steam Rankine cycles is the preferred choice. The standard configuration of a commercial 50 MWe CSP plant includes a solar field with 500000-700000 m^2 of aperture area (600-800 lines with a length of about 150 m), which requires the availability of about 150-250 hectares of land. Thermal oil with a maximum bulk temperature of 390-400 °C is the most used heat transfer fluid while molten salts are often used in the thermal energy storage section. The power generation section is based on a steam Rankine cycle with superheated steam produced at about 370-380 °C and 80-100 bar, high-pressure and low-pressure steam turbines with reheating and 4-6 steam extractions for feed-water heating [3-6].

Large-size CSP plants are today the best tradeoff between capital costs and overall conversion efficiency. However, in many countries (and in Italy, in particular) the wide extension areas required by large-size CSP plants are very hard to find. In this case, medium- and small-size CSP plants (around 1 MWe) may be a more suitable option due to the lower land requirement. Moreover, medium- and small-size power plants are also a very interesting solution for the diffusion of distributed power generation.

The technology options available for small- and medium-size CSP plants significantly differ from those of large-size plants. In fact, steam turbines are generally not suitable for power outputs in the range of few MWe and therefore the power generation section of such CSP plants cannot be based on steam Rankine cycles. The most interesting alternative is represented by Organic Rankine Cycles (ORC), where steam is substituted by organic fluids with high molar weight. ORC systems require thermal energy inputs with temperature levels in the range of 250-350 °C and show conversion efficiencies of about 20-25% [7-9]. Moreover, Linear Fresnel Collectors (LFC) may be a viable alternative to PTC especially if land requirement is an important feature. In comparison to PTC, LFC has a simpler design, uses less expensive mirrors and tracking systems, shows lower land requirements and lower capital costs. However, the optical efficiency of LFC is lower than that of PTC [10-13].

This paper reports a comparative performance analysis of CSP plants using parabolic trough and linear Fresnel collectors, thermal oil as heat transfer fluid and an Organic Rankine Cycle power generation unit. The CSP plants studied herein also include a two-tank direct thermal storage system based on the use of the same thermal oil as storage medium. The comparative study was carried out on the basis of a 1 MW ORC plant by considering different values for both the solar multiple and thermal storage capacity.

1. Configuration of the CSP power plants

Figure 1 shows a simplified diagram of the CSP plant analysed in this paper. The CSP plant includes three main sections: the solar field, the power block and the thermal energy storage (TES) system.

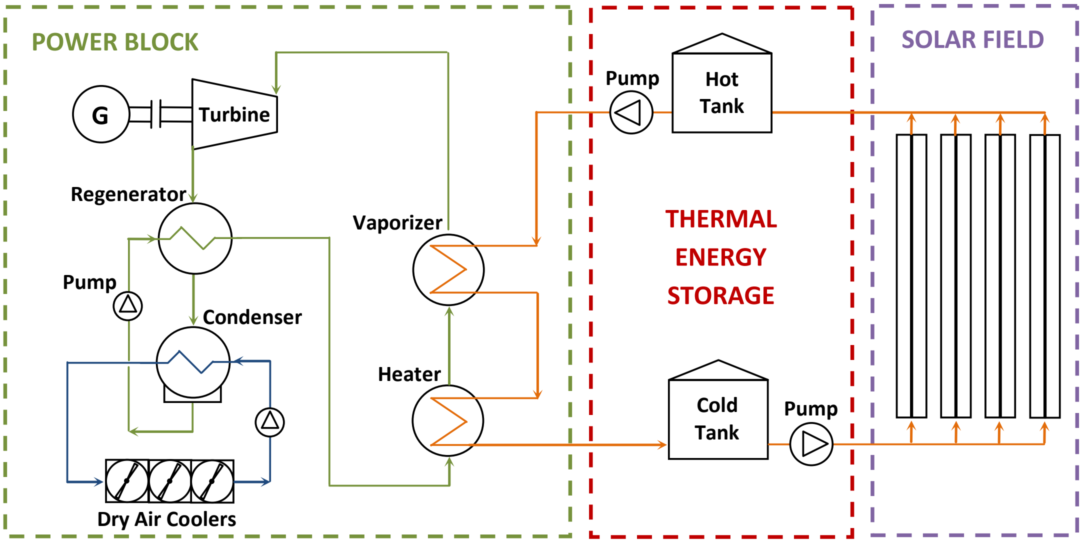

Fig. 1. Process flow diagram of the CSP power plant.

The solar field is based on several lines of PTC or LFC connected in parallel to achieve the required thermal oil mass flow and therefore the required thermal power output. Parabolic trough and linear Fresnel collectors are designed for the same thermal power output and the same input and outlet thermal oil temperatures. However, due to the different geometrical structure and the different collector efficiency, the collecting area and the number of lines show a significant variation.

For both PTC and LFC, each collector line includes several collector modules connected in series. In the case of PTC, each solar collector module is composed of parabolic mirrors mounted on a steel structure supported by pylons. The evacuated receiver tube is placed in the focal line of the parabolic surface and is connected to the cold and hot header pipes with flexible joints to allow the rotation of the overall structure and the thermal expansion of the receiver tube with increasing temperatures. The solar collector lines are aligned along the North-South direction and are equipped with a single-axis tracking system that allows the PTC to follow the sun path from East to West. The main geometrical features of PTC are the aperture of the parabola (about 5-6 m), the total length (about 100-150 m) and the focal distance (1.5-2.5 m).

In the case of LFC the solar collector module is composed of several rows of flat mirrors whose slope continuously changes following the sun's position. The mirrors are mounted on a fixed steel structure placed near the ground (about 0.5-1.0 m above the ground) and concentrate solar radiation onto a fixed receiver installed several meters above the mirror plane. Moreover, the receiver includes a secondary reflector that redirects the incoming solar rays towards the evacuated receiver tube. Similarly to PTC, the solar collector lines are aligned along the North-South direction and are equipped with a single-axis tracking system to follow the sun's path from East to West. However, in this case the overall steel structure is lighter than that of PTC because only the mirror rows rotate to follow the sun's path. The main geometrical features of LFC are the number and width of the mirror rows (7-15 mirror rows 0.5-0.6 m in width), the total length (about 150-200 m) and the receiver height (4.0-8.0 m). Overall, LFCs are less expensive than PTCs, using flat mirrors (or mirrors with only a very small curvature), a lighter supporting structure and a simpler tracking system. However, the optical efficiency of LFCs is lower than that of PTCs. The thermal energy produced by the solar field can be directly used in the power block or stored in the TES system for deferred use. The power block is based on an ORC unit, where thermal energy is converted to electrical energy by using an organic fluid (a siliconic oil in this case) that follows a regenerated Rankine cycle. As shown in

Fig. 1, the thermal energy produced by the solar field is used in the ORC unit to heat and vaporize the organic fluid. The produced organic vapour expands in the turbine, is cooled in the regenerator and condensed. After the condenser, the organic fluid is compressed by the feeding pump and then preheated in the regenerator. The condensing heat is removed by dry water coolers.

The excess thermal energy produced by the solar field during periods of high solar availability is stored in the TES section and used by the ORC unit to produce electrical energy during periods of low or zero solar availability. As shown in Fig. 1, the CSP plant studied here is based on a two-tank direct system using the same thermal oil as the storage medium. In particular, the thermal oil from the low-temperature storage tank (the cold tank) flows through the solar field, where it is heated and then sent to the high-temperature storage tank (the hot tank). From the hot storage tank the required mass flow of thermal oil is pumped to the ORC unit and then sent back to the cold storage tank.

The piping system of the CSP plant includes the two circulating thermal oil pumps, the cold and hot header pipe (one for distributing the cold oil throughout the collector loops and the other for collecting the hot oil), the main pipes, valves, fittings and pressure, temperature and flow meters.

2. System modelling and assumptions

The performance of the CSP plants were evaluated on a yearly basis by means of a specifically developed simulation model. In particular, the simulation model evaluates the performance of PTC and LFC as a function of solar radiation and solar position, for given values of the main geometrical and technical characteristics of solar collectors, as well as for assigned thermodynamic properties of the heat transfer fluid.

As is known, the solar energy available for a CSP plant is conventionally defined by the product of Direct Normal Irradiation (DNI) and collecting area A_C. For PTC the collecting area is represented by the aperture area of the collector, while for LFC it is given by the overall area of the mirror rows. Evaluation of the annual energy production of CSP plants requires the availability of detailed data (at least on an hourly basis) about solar radiation and solar position.

The present comparative study was carried out by using a data set for a typical meteorological year obtained from the Meteonorm software [14] for the site of Cagliari (39°13’25’’N, 9°07’20’’E), in the south of Sardinia (Italy). In particular, the meteorological data set includes DNI, solar azimuth and elevation, ambient temperature, relative humidity and wind velocity. Table 1 summarizes the most important meteorological data of the site and the design conditions assumed for the CSP plant.

Table 1. Meteorological data for Cagliari.

Available DNI	1720 kWh/m^{2}y
Average ambient temperature	17.2 °C
Average wind velocity	3.96 m/s
Design DNI	800 W/m^2
Design elevation/azimuth angles	74.2 °/0.0 °
Design ambient temperature	32.0 °C

As is known, in the case of linear collectors, the adoption of a single-axis tracking system reduces the usable DNI since the collector surface can be kept normal to the sun’s rays only with double-axis tracking systems. For linear collectors, the incidence angle θ between solar rays and the normal direction to the collector surface is usually defined. For a given collector geometry, the incidence angle depends on the solar position, which is completely defined by the azimuth angle γ (that is, the angle between the projection of the solar rays on the horizontal plane and the south direction) and the elevation angle α (that is, the angle between solar rays and their projection on the horizontal plane). Moreover, two different components of the incidence angle θ are calculated: the longitudinal component θ_L (the angle between the normal direction to the horizontal surface and the projection of the solar rays on the longitudinal plane of the collector) and the transversal component θ_T (the angle between solar rays and their projection on the transversal plane of the collector). These two components can be expressed in function of γ and α by means of the following equations:

$$\tan(\theta_T)=\frac{\sin(\gamma)}{\tan(\alpha)} \quad (1)$$

$$\sin(\theta_L)=\cos(\gamma)\cdot\cos(\alpha) \quad (2)$$

It should be observed that in the case of PTC, the tracking system allows rotation of the overall collector along its longitudinal axis such that the normal vector to the collector aperture area is always contained in the same plane of the solar rays. Therefore, the component θ_T directly gives the rotation angle of the PTC during the daily sun path.

The simulation model used in this comparative study evaluates the thermal power output and the conversion efficiency of the overall solar field as a function of solar radiation and solar position. The simulation model is mainly based on the energy balance of the solar collector, which includes two main components: the solar concentrator and the receiver tube. The energy balance of the solar concentrator gives:

$$Q_{SOL} = A_C \cdot DNI = Q_{RCV} + Q_{OPT} \quad (3)$$

In eq. (3) Q_{SOL} denotes the solar power input (that is, the available direct solar radiation), Q_{RCV} is the thermal power concentrated onto the receiver tube and Q_{OPT} are the optical losses of the solar concentrator. In particular, thermal power Q_{RCV} has been evaluated here by means of the following equation:

$$Q_{RCV} = A_C \cdot DNI \cdot \eta_{OPT,R} \cdot IAM(\theta) \cdot \eta_{END} \cdot \eta_{SHD} \cdot \eta_{CLN} \quad (4)$$

As shown by eq. 4, the thermal power available for the receiver tube is lower than the solar power input owing to the presence of different types of optical losses. Reference optical efficiency $\eta_{OPT,R}$ depends on mirror reflectivity, glass tube transmissivity, absorptivity of the selective coating of the receiver tube, imperfections in the collector mirrors, tracking errors, shading of receiver supports on mirrors, etc. Reference optical efficiency is commonly evaluated for θ=90° because the optical properties on the different materials (mirrors, glass tube, selective coating) depend on the incidence angle of the incoming solar rays. For this reason, reference optical efficiency is multiplied by the Incidence Angle Modifier (IAM) to give the effective optical efficiency. The Incidence Angle Modifier depends on the incidence angle θ and is often divided into the longitudinal and transversal IAM components:

$$IAM(\theta) = IAM\ (\theta_L) \cdot IAM\ (\theta_T) \quad (5)$$

Figure 2 shows the two IAM components considered in this study for linear Fresnel and parabolic trough collectors in function of the longitudinal and transversal components θ_L and θ_T. It should be noted that for PTC the IAM transversal component can always be assumed equal to 1 because the normal vector to the collector aperture is always contained in the same plane as the solar rays. With longitudinal components θ_L lower than 90°, the useful mirror area of linear collectors is reduced by the geometrical end-losses, which depend on collector length L and focal length F. Overall, these losses are taken into account by means of the end-loss optical efficiency:

$$\eta_{END} = 1 - \frac{F}{L} \cdot \tan\theta_L \quad (6)$$

Moreover, in eq. 4, the term η_{CLN} is mirror and glass tube surface cleanliness efficiency and η_{SHD} is the shadow efficiency, which gives the energy losses due to the shadow from the different lines of collectors. Shadow efficiency applies only to solar fields with multiple lines of PTC and is given by the following equation:

$$\eta_{SHD} = 1 - \frac{R}{W} \cdot \tan\theta_T \quad (7)$$

where R is the distance between the PTC lines and W the collector aperture width.

Thermal power Q_{FLD} transferred to the thermal oil is given by the difference of receiver available power Q_{RCV} and thermal losses of solar field Q_{THR}:

$$Q_{FLD} = m_O \cdot Cp_O \cdot \left(T_{O,out} - T_{O,in}\right) = Q_{RCV} - Q_{THR} \tag{8}$$

Thermal losses were evaluated by the sum of receiver thermal losses and piping thermal losses:

$$Q_{THR} = \left[q_{tube} + q_{pipe}\right] \cdot A_C = \left[(a_1 \cdot \Delta T + a_2 \cdot \Delta T^2) + q_{pipe}\right] \cdot A_C \tag{9}$$

where ΔT is the difference between average oil temperature in the receiver tube and ambient temperature.

Table 2 shows the main geometrical and performance parameters of PTC and LFC used in the following comparative study. The main power consumption of the solar field is due to the collector tracking system and the oil circulating pumps. The first term was assumed equal to 0.5 W/m^2 of collecting area while the second was evaluated by imposing a fluid velocity of about 1 m/s and a pump efficiency equal to 75%.

Table 2. Main geometrical and performance parameters of PTC and LFC.

	PTC	LFC
Collector length L	150 m	180 m
Collector width W	5.77 m	16.56 m
Collector area A_C	865.5 m^2	2054.4 m^2
Focal length F	1.71 m	7.40 m
Lines distance R	17.31 m	4.50 m
Reference optical efficiency $\eta_{OPT,R}$	0.75	0.67
Cleanliness efficiency η_{CLN}	0.98	0.98
Inlet/outlet oil temperature	204/305 °C	204/305 °C
a_1 coefficient	-	0.056 W/m^{2}K
a_2 coefficient	0.00047 W/m^{2}K^2	0.000213 W/m^{2}K^2
Specific piping losses q_{pipe}	5 W/m^2	5 W/m^2

The power block considered in this comparative performance analysis is an ORC unit with a power output of 1 MWe integrated with a dry water cooler. Design and off-design performance of the ORC module were evaluated with reference to data available for commercial units [15]. In particular, Table 3 reports the most important operating parameters of the ORC unit while Fig. 3 shows its off-design efficiency as a function of thermal load (cooling water at 25 °C) and air temperature (thermal power input of 4043 kW and 7 °C of temperature difference between cooling water and air).

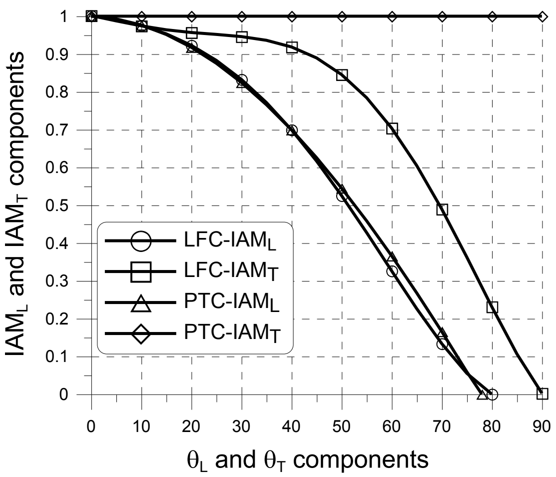

Fig. 2. Longitudinal and transversal IAM components.

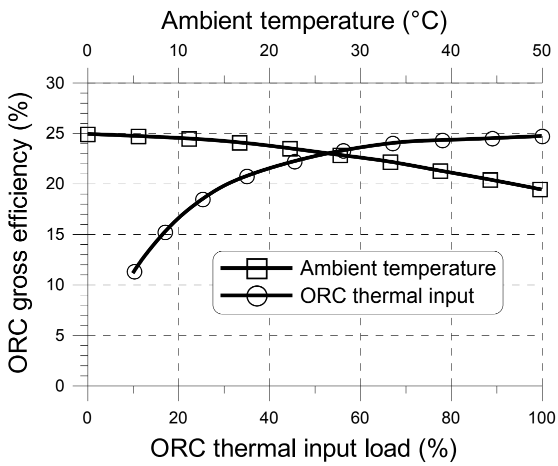

Fig. 3. Off-design efficiency of the ORC unit.

Table 3. Operating parameters of the ORC unit.

Gross power output	1000 kW
Thermal power input	4043 kW
Oil inlet/outlet temperature	305/204 °C
Condenser power output	3040 kW
Water inlet/outlet temperature	25/35 °C
Gross electrical efficiency	24.7%
ORC internal consumption	36 kW
Dry-coolers electrical consumption	1.5% of cond. power

As already mentioned, the TES system considered here is based on two identical thermal oil tanks. The mass and volume of thermal oil to be stored was evaluated in function of the required storage capacity (here expressed in terms of equivalent hours of ORC thermal supply) and thermodynamic properties of thermal oil (for the design of the two storage tanks, an average specific heat of 2.5 kJ/kgK and a density of 750 kg/m^3 at 305 °C were assumed here). Moreover, the volume of each storage tank was increased by 10% compared to the thermal oil volume. Finally, a thermal energy loss of 2% of stored energy was considered.

3. Performance of CSP power plants

For a given power output of the ORC unit, the collecting area of the solar field and storage tank volume mainly depend on two important design parameters: the Solar Multiple (SM) and Storage Capacity (SC) expressed in terms of equivalent hours of thermal supply of the power block. The solar multiple is the ratio between the thermal power produced by the solar field at design conditions and the thermal power required by the power block at nominal conditions and therefore the size of the solar field linearly depends on SM. Table 4 compares the main design specifications of the two CSP plants based on PTC and LFC for a solar multiple equal to 1. In particular, Table 4 shows the net efficiency of the CSP plant, here defined by the ratio of net power output and solar power input.

Table 4. Main performance parameters of CSP plants based on PTC and LFC.

	PTC	LFC
Solar field collecting area	7515.1 m^2	8501.3 m^2
Solar field thermal power output	4043 kW	4043 kW
Thermal oil mass flow	16.01 kg/s	16.01 kg/s
Solar field conversion efficiency	67.25 %	59.45 %
ORC thermal power input	4043 kW	4043 kW
ORC gross power output	931.3 kW	931.3 kW
ORC gross efficiency	23.03 %	23.03 %
CSP overall power consumption	102.2 kW	91.4 kW
CSP net power output	840.1 kW	839.9 kW
Required land area/collecting area	3.0	1.30
CSP net conversion efficiency	13.97 %	12.35 %

For a given thermal power output, solar fields based on LFC require a larger collecting area (by about 12-13%) than PTC due to lower conversion efficiency (59.5% vs. 67.3%). On the contrary, the CSP-LFC solution requires about 56% less land area than the CSP-PTC plant. It should be noted that gross power output and efficiency of the ORC unit are lower than the reference values of Table 3 due to the higher air temperature (the ORC unit integrated with the dry water cooler requires about 18 °C of ambient temperature to give the reference gross efficiency of Table 3). Overall, at design conditions the net efficiency of the CSP-PTC plant is about 1.6 percentage points higher than that of the CSP-LFC solution. Obviously, it should be observed that solar field needs a finite number of collector lines and therefore the actual solar multiple of both PTC and LFC configurations must be higher than 1 to give the power output required by the ORC unit (the CSP-PTC solution requires at least 9 collector rows while the CSP-LFC solution requires at least 5 collector rows).

For a given ORC unit, increasing the solar multiple increases the thermal power output of the solar field and therefore the required collecting and land area. Land area depends on number, width and distance of the collector lines, the free space around the collectors (10 m in this case) and the area required by the power block and the TES section (here assumed equal to 2500 m^2). Figure 4 shows the land requirement for CSP plants based on PTC and

LFC in function of SM (plot labels also give the corresponding number of collector rows). As shown in Fig. 4, the PTC land requirement is higher than that of LFC, especially for large SM values. Moreover, the volume of each storage tank and the mass of thermal oil contained therein depends only on SC because the product of ORC thermal power input and the equivalent hours of storage capacity directly gives the energy to be stored by the TES section. Figure 5 shows the oil mass and tank volume in function of SC.

Figure 6 shows the net electrical production on a yearly basis for the two CSP configurations in function of both solar multiple and equivalent hours of thermal energy storage. As expected, annual energy production increases with SM and SC. However, the increase in storage capacity improves net energy production only for increasing values of SM. Moreover, for given values of both SM and SC, the CSP-PTC solution gives higher energy yields in comparison to the CSP-LFC one (by about 15-25% with SM in the range of 2.0-2.5 and by about 10-15% for SM in the range of 4.0-4.5) owing to its better solar field efficiency. The importance of a proper trade-off between solar multiple and storage capacity is pointed out in Figure 7, which shows the effectiveness of the TES section. In fact, the increase in SM increases the thermal energy produced by the solar field and therefore raises the surplus of energy that cannot be directly used by the ORC unit. The effective utilization of this surplus energy for power generation requires a suitable TES capacity. In particular, the TES effectiveness in Figure 7 is defined here as the ratio of thermal energy actually stored in the TES system and surplus energy produced by the solar field. As shown in Figure 7, the achievement of high TES effectiveness (in the range of 0.9-1.0) requires energy storage capacity to increase with the solar multiple. For example, in the case of the CSP-LFC configuration the achievement of a TES effectiveness higher than 0.9 requires a SM of about 1.5-1.8 for SC=4 and about 2.4-2.8 for SC=12. The same TES effectiveness for the CSP-PTC solution requires a SM of about 1.2-1.5 for SC=4 and about 2.0-2.5 for SC=12.

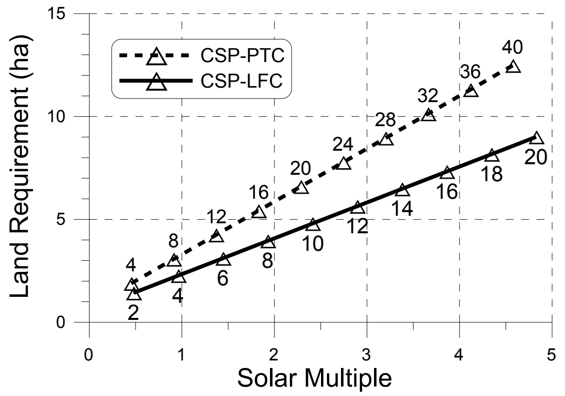

Fig. 4. Land requirement for CSP plants in function of SM.

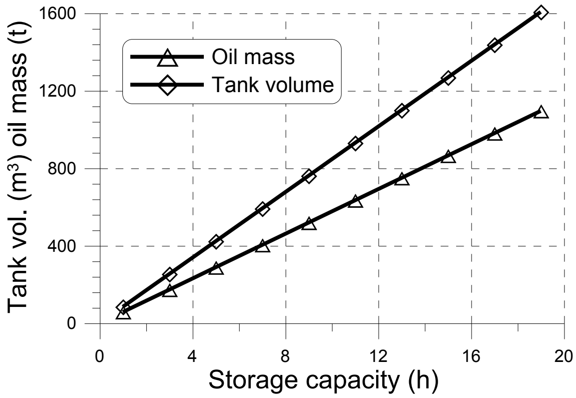

Fig. 5. Tank volume and storage oil mass in function of SC.

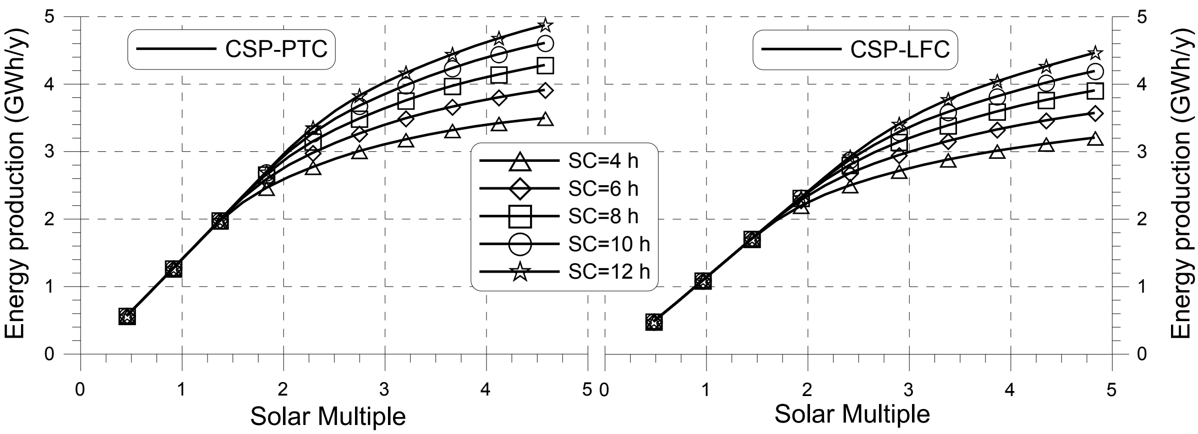

Fig. 6. Annual energy production of the two CSP power plants in function of solar multiple and energy storage capacity.

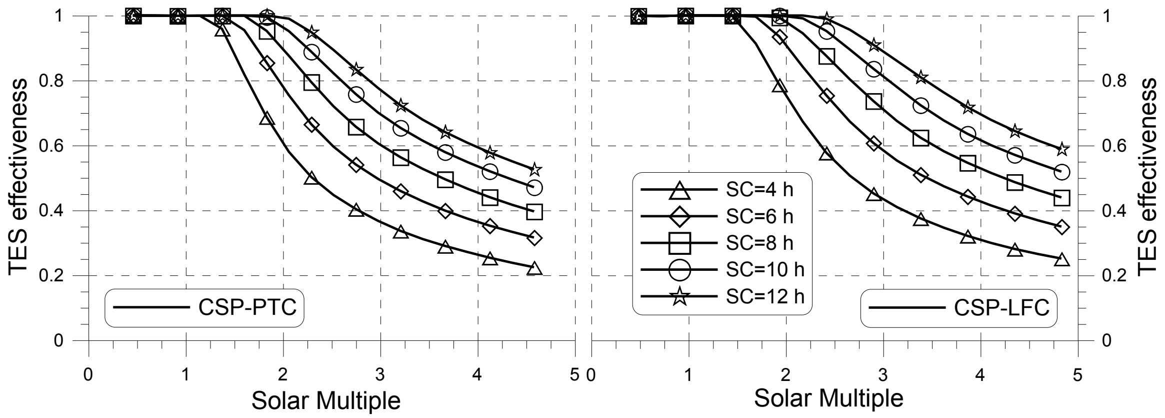

Fig. 7. Thermal energy storage effectiveness of the CSP power plants in function of solar multiple and energy storage capacity.

As mentioned, in many countries (and in Italy in particular) the finding of the large areas required by CSP plants can be difficult and therefore an important performance parameter of such power plants is specific energy production per unit area of solar field. For this reason, Figure 8 reports the specific annual energy production of the CSP configurations studied here in function of both SM and SC. In particular, the left side of Figure 8 refers to energy production per unit area of collector aperture (E_{PC}) while the right side refers to energy production per unit area of required land (E_{PL}). Figure 8 demonstrates that if land availability is the most limiting factor, CSP-LFC solutions are very interesting owing to their highest energy production per m^2 of occupied land.

The highest specific production of CSP-LFC plants is about 55-60 kWh/y per m^2 of occupied land. In the field of SC considered here (4-12 hours) the latter specific production is achieved with SM values around 1.75-2.5. Figure 8 shows that the highest specific production of CSP-PTC plants is about 45-50 kWh/y per m^2 of occupied land and that it is achieved with lower SM values (around 1.5-2.3). On the other hand, the left side of Figure 8 demonstrates that the CSP solutions based on PTC give higher values of specific energy production per unit area of collector aperture (about 180-190 kWh/m^2 vs. 130-140 kWh/m^2), owing to their better optical efficiency in comparison to the LFC solutions. Obviously, as annual solar energy available per m^2 of collector area is the same for all CSP solutions (1720 kWh/m^{2}y for Cagliari), energy production per unit area of collector aperture is proportional to the average conversion efficiency of the CSP plant. For example, the aforementioned specific energy production of CSP-PTC plants (about 180-190 kWh/m^2) leads to an average conversion efficiency of about 10.5-11.0% while that of the CSP-LFC plant (130-140 kWh/m^2) gives an average efficiency of about 7.6-8.1%. Finally, it should be observed that the completion of the comparative study between PTC and LFC requires an economic analysis to identify the solution with the lowest energy production cost.

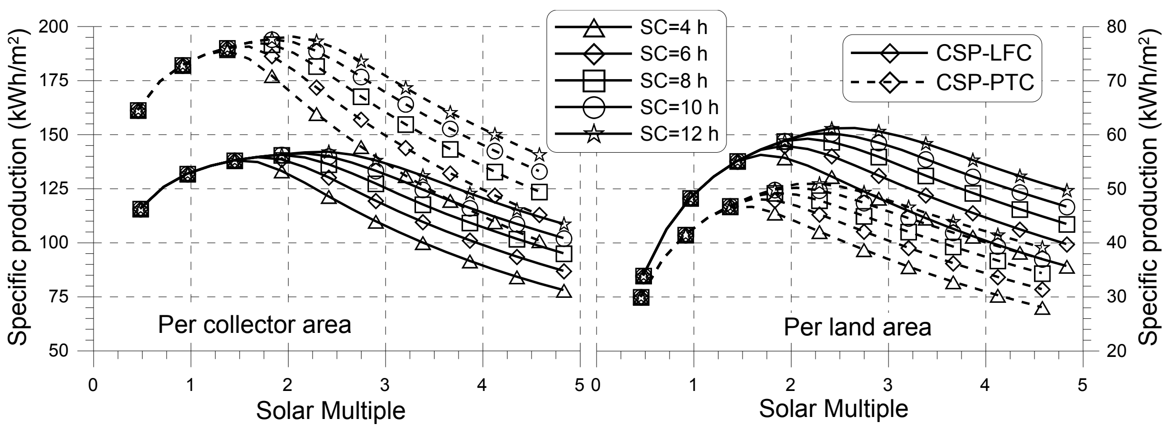

Fig. 8. Specific energy production of the two CSP power plants in function of solar multiple and energy storage capacity.

4. Conclusions

In this paper a comparative performance analysis of medium-size (1 MWe) Concentrating Solar Power (CSP) plants based on an Organic Rankine Cycle (ORC) power generation unit integrated with parabolic trough and linear Fresnel collectors is presented. Owing to the lower land requirement, this kind of CSP plant can be a very interesting option in countries (like Italy), where the areas required by large size CSP plants are very hard to find.

Annual energy production of the two CSP configurations increases with solar multiple and storage capacity. However, the increase in storage capacity significantly improves annual energy production only for increasing values of the solar multiple. In the field of storage capacities considered here (4-12 hours), the best performance is achieved for solar multiples of about 1.75-2.5 for CSP plants based on LFC and 1.5-2.3 for PTC.

The results of the performance assessment demonstrate that if land availability is the most limiting factor, CSP solutions based on linear Fresnel collectors are the preferred option owing to their higher energy production per m^2 of occupied land (about 55-60 kWh/y per m^2 of occupied land vs. 45-50 kWh/y produced by solutions based on PTC). However, owing to their better optical efficiency, the use of parabolic troughs gives better values of energy production per unit area of solar collector (about 180-190 kWh/m^2 vs. 130-140 kWh/m^2) and therefore better conversion efficiencies (about 10.5-11.0% vs. 7.6-8.1%). Finally, it should be observed that the completion of the comparative study between PTC and LFC requires an economic analysis to identify the solution with the lowest energy production cost.

Acknowledgements

This work was carried out as part of a collaboration agreement with the Consortium “Sardegna Ricerche” for the management, scientific coordination and development of research activities of the Laboratory "Concentrated Solar Technologies and Hydrogen from RES".

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