Supercritical ORC Turbogenerators Coupled with Linear Solar Collectors

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1. Introduction

The modern development of concentrated solar power (CSP) plants worldwide started with the first oil crisis in the early 70's, and peaked with the nine commercial SEGS plants built in the Mojave Desert in California, with an average yearly direct normal insolation (DNI) up to 2727 kWh/m^2 year, between 1984 and 1990 (Fernández-García et al., 2010). It is mainly because of these installations that plants based on parabolic trough collectors (PTC) are the most mature and cost-effective CSP technology (Price et al., 2002). Several other solar power plants of this kind are currently being built. Another option for line-focus concentrators to which consistent R&D efforts are being devoted involves linear Fresnel collectors. This technology has the potential of significantly reducing the cost of the solar field, which constitutes the largest share of the total initial investment and the cost with the highest margin for reduction (Pitz-Paal et al, 2007). Linear Fresnel collectors are now in the pre-commercial stage (Mills, 2004).

In conventional solar thermal power plants a heat transfer fluid (HTF, typically synthetic oil) is used in a closed loop to carry the thermal energy over to the working fluid of the steam Rankine cycle power system. Techno-economic optimization leads to a typical power capacity of the order of 50 MW_E due to the complexity and cost of a steam power plant. The corresponding footprint is approximately 25 hectares. The value of the turbine inlet temperature, which is directly related to the efficiency of the thermodynamic cycle, is limited by the HTF thermal stability. Available oils are compatible with operating temperatures up to approximately 390 °C, thus limiting the live steam temperature to a lower value. Conversely, the thermal efficiency of the solar field decreases with increasing temperature of the absorber, since higher temperatures increase the heat losses to the environment. Major research and development efforts are underway to improve the performance of absorbers at higher operating temperatures (Benz, 2007). In order to increase the steam temperature at the turbine inlet, other HTF's such as molten salts are being investigated (Eck and Hennecke, 2007; Kearney, 2007; ENEA, 2011). An all-together different concept is to directly generate vapor in the collector, thereby making the intermediate HTF-loop redundant. Direct steam generation (DSG) would allow a substantial reduction of the cost of the solar electricity produced mainly by (Müller, 1991; Cohen and Kearney, 1994; Pitz-Paal et al., 2007)

- increasing the power plant efficiency. Live steam up to 500 °C and 100 bar (at the solar field outlet) has been proposed (Benz, 2007; Birnbaum et al., 2010).
- decreasing both the cost and the complexity of the plant as the HTF loop and the expensive heat exchanger between the HTF and the working medium of the Rankine cycle is avoided.

The main drawbacks of DSG technology can be summarized as follows:

- the absorbers and the piping in the whole solar field must endure both elevated pressures and temperatures, which poses technical challenges;
- operational difficulties might arise because the two-phase flow in the absorber can cause unacceptable circumferential temperature gradients under certain operating conditions in terms of heat-flux and mass flow (Eck and Steinmann, 2004; Martínez and Almanza, 2007). In addition, a Ledinegg-type hydrodynamic instability might also occur due to the evaporating flow, leading to potentially risky pressure's oscillations and thus requiring additional equipment and complications in term of control of the loop (Taitel, 1990; Odeh et al., 2000). Extended research activities on these phenomena have been carried on in the recent past, with the aim of identifying feasible operational strategies (Valenzuela et al., 2005; Valenzuela et al., 2006; Eck and Hirsch, 2007).
- Thermal storage concepts suitable for DSG are inherently more complicated and less efficient, if compared to systems for conventional solar thermal power plants. This is due to the need for a

multi-stage system with a combination of sensible- and latent-heat storage sections (Tamme et al., 2004; Steinmann and Tamme, 2008; Pitz-Paal et al., 2007).

Nonetheless, the feasibility of the DSG configuration has been proven under real solar conditions at the PSA DISS facility in Almería, Spain (Zarza et al., 2004). Further work is ongoing with the aim of demonstrating the technology at a pre-commercial scale (Zarza eta al., 2006).

Organic Rankine cycles (ORC) turbogenerators are technically and economically attractive energy conversion systems for so-called external thermal energy sources in the small-to-medium power range (from few kW_E up to few MW_E); the advantages of an organic working fluid compared to water for such plants' sizes have been demonstrated (Angelino et al., 1984). ORC power systems are successfully employed for the conversion of medium and low temperature energy sources such as industrial waste-heat, biomass combustion, geothermal reservoirs, and their market is growing at a fast pace. In this power range and for moderate and low temperature (up to 400 °C), ORC based conversion units are arguably more efficient and cost-effective than steam power plants (Quoilin et al., 2009).

The use of ORC turbogenerators for the conversion of high-grade energy source has been studied (Obernberger et al., 2004; Invernizzi et al., 2007) and first ORC-based CSP solar plant prototypes were put into operation several years ago (Verneau, 1978; Angelino et al., 1984); new commercial plants are planned and currently under construction worldwide (Turboden, 2011).

Recently, a new paradigm for the successful development of thermal solar plants has arisen (Price and Hassani, 2002): economy of production can be achieved by means of high-volume manufacturing of small-capacity standard and modular systems, suitable for distributed energy conversion, instead of larger centralized power plants (Prabhu, 2006). On this same development path is also the technology of Stirling engines powered by a solar-dish coupled: the Maricopa (AZ) pilot plant, made of 60 dish-Stirling systems (each producing a nominal gross power of 25 kW_E), started operation on January 2010 (Solar Paces, 2011). As a matter of fact, after the success of the SEGS plants (ranging from 14 up to 80 MW_E), no more commercial plants have been built in the US between 1990 and 2004 and the reason is the high risk connected with the large initial investment and footprint associated with multi-Megawatt solar plant. Even recently, several projects for the realization of large solar power plants have not gone beyond the initial phase in many countries (Fernández-García et al., 2010).

The advantages of the new small-scale approach to the development of solar thermal power plants can be synthesized as follows:

- Cost-effectiveness can be reached by large-series production of standard modular units (both PTC and power conversion unit), thereby achieving economy of production. Standard units can be assembled in the factory and delivered skid-mounted for fast and easy installation, thus reducing associated costs.
- Market penetration in the initial phase is eased by the low absolute value of the initial investment (at least one order of magnitude). Even if the specific cost is comparatively high, but not unreasonable, the risk can be managed. First technically successful installations can promote rapid growth.
- A small-modular plant can provide local value and meet local customer's needs, rather than generating power for the wholesale market.
- Small power plants can be fully automated and remotely controlled, significantly reducing O&M costs, which are a non-negligible component of the operating costs of large installations.

The first research and development project which investigated the feasibility of this concept, named STORES (Prabhu, 2006), resulted in the identification of ORC turbogenerators as the optimal conversion technology for distributed thermal solar power, mainly because of its reliability and sufficient performance. The outcome of the study has been the construction of the first solar plant of this kind in the Saguaro Desert, AZ (average yearly DNI over 2500 kWh/m^2 year). The plant has a nominal power of 1 MW_E and an average annual efficiency of 12%, it features a small-capacity thermal storage unit and no need for onsite staff (Canada et al., 2006; Kolb and Hassani, 2006). This was the first new PTC solar-thermal power plant to come on-line in 15 years.

The study documented in this paper is related to the concept of Direct Supercritical Fluid Generation (DSFG) in the solar absorber of a parabolic trough, in combination with a high-efficiency small-capacity ORC turbogenerator (figure 1). The idea stems from the observation that siloxanes, a suitable working fluid for ORC turbogenerators, are also employed, as mixtures, in the very same thermal oil loops of large solar power plants. The underlying objective is the evaluation of the possibility of reaching comparatively higher energy conversion efficiency and lower initial investment, thanks to the simplicity of the overall plant configuration.

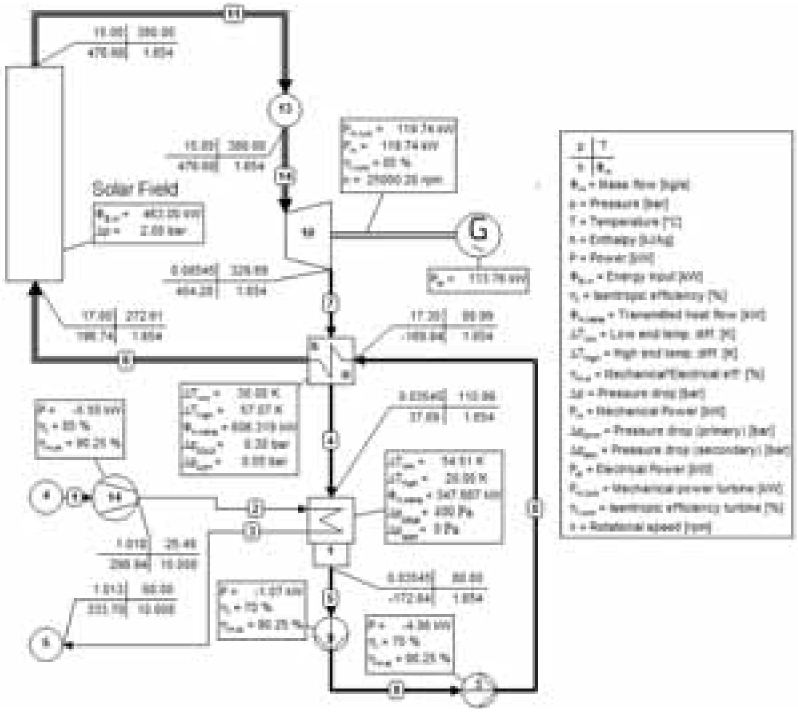

Fig. 1: Conceptual plant layout of the DSFG system (from the Cycle Tempo graphical user interface) dimensioned to be installed in Tucson, AZ. The nominal power output is 100 kW_{EL}, and the corresponding estimated net efficiency is η_{NET}=22%. Solar field constituted of a single ET-150 SCA, A_{SF} = 817.5 m^2, $Availability_{SF}$ = 0.96, DNI_{DES} = 950 W/m^2, η_{OPT_DES} = 75 %.

2. Plant configuration and working fluid

With reference to figure 1, the organic fluid in the supercritical state (pressure greater than the critical pressure of the fluid) is circulated directly in the solar absorber, where it is heated and reaches the maximum cycle temperature (also supercritical), before expanding in the turbine. Due to the thermodynamic characteristic of the fluid, namely the large heat capacity, the vapor at the outlet of the turbine is at high temperature in a dry superheated state, therefore a heat exchanger (the regenerator) transfers its thermal energy to the liquid at the outlet of the main pump, substantially increasing the thermal conversion efficiency. An air-cooled condenser further cools the organic vapor, condenses it, and brings it to a sub-cooled temperature at the inlet of the pre-feed pump.

The working fluid considered in this study belongs to the family of siloxanes (Angelino and Invernizzi, 1993; Colonna, 1996; Angelino and Colonna, 1998), and more specifically it is D_4 (octamethylcyclotetrasiloxane, $[(CH_3)_2\text{-}Si\text{-}O]_n$, with n=4, a cyclic molecule), see Tab. 1. Linear siloxanes are already employed in commercial ORC turbogenerators and they are non-toxic, environmentally friendly (ODP and GWP are both zero), low-flammable, bulk-produced and highly thermally stable (Colonna 1991, Angelino 1993, Colonna, 1996, Angelino 1998). The selected cyclic molecules are slightly more complex than currently adopted linear siloxanes, thus arguably more suited for the power capacity considered in this preliminary study (100 kW_E) (Angelino et al., 1984). Specific models for the calculations of thermodynamic properties for these fluids have recently been developed (Colonna et al., 2006; Colonna et al., 2008).

One of the key points is that, due to the low critical-pressure values of suitable working fluids (around 10 bar), it is technically and economically feasible to pressurize the fluid in such a way that the pressure in the collectors is kept above its critical value. In addition, though some technical challenges are connected to heat transfer and fluid dynamic features, supercritical flows of siloxanes could work as heat transfer media equally well to currently adopted pressurized liquid mixtures of siloxanes.

The supercritical ORC configuration has already been studied for space power applications (Boretz, 1986; Colonna, 1991; Angelino and Invernizzi, 1993), for geothermal applications (Bliem and Mines, 1989), and more recently also for heat recovery applications (Schuster et al., 2010; Astolfi et al., 2011). A number of concepts have also been proposed for high temperature systems (Fernández et al., 2011).

The envisaged advantages of the proposed DSFG solar ORC turbogenerator are:

- High conversion efficiency: the cycle configuration reduces thermodynamic losses, as thermal energy is directly transferred to the working fluid at comparatively higher temperature;
- plant simplification: as in DSG steam power plants, the primary heat exchanger, the phase separator, and the HTF are not needed; furthermore the pressure levels are much lower.
- ease of operation: all the problems related with the evaporative section of the solar absorber can be avoided;
- high thermal capacity of the working fluid: the working fluid might be used also for direct thermal storage (Beckam and Gilli, 1984).

Tab. 1: Main properties of D4 (Colonna et al., 2006)

	T_{cr} [°C]	P_{cr} [bar]	MW [g/mol]	$T_{BOILING}$ [°C]	ρ_{cr} [kg/m^3]
D_4	313.34	13.32	296.62	175.35	305.79

3. Method

In order to obtain preliminary results about the performance of the plant, several models have been developed and implemented in computer programs, which have been used to perform simulations. Initially a 1D model of the solar absorber employing supercritical organic fluid has been developed in order to correctly estimate the thermal energy made available to the power conversion unit. The validated model has been reduced in order to make it compatible with a steady-state system model of the entire power plant (solar field and power conversion unit). The reduced model of the absorber and of the power conversion unit have been implemented into a commercial software (see fig. 1) capable of on- and off-design simulations (Van der Stelt et al., 1980-2011). The steady state simulation program has been already validated with field data and used successfully in numerous studies on ORC power plants (Angelino and Colonna, 1998; Angelino and Colonna, 2000a; Angelino and Colonna, 2000b). A library for the estimation of thermophysical properties of fluids (Colonna and Van der Stelt, 2004) is used for thermodynamic and transport properties calculations in all cases. Finally, the simulation of one year of operation utilizing data related to solar radiation over a typical location has been carried out. A steady state simulation is run for each hour of the year, using the appropriate DNI and weather data from a database. The results are post-processed in order to obtain the most important performance parameters.

4. Models

4.1. Absorber

The solar energy collecting equipment (SECE) for the proposed plant does not differ from a conventional PTC, but for the working fluid, which is in the supercritical state. Main components of a SECE are the solar collector assembly (SCA), including mirrors, the structures, the tracking system, and the absorbers tubes. SCAs are connected in a series forming a loop. The solar field (SF) is made of many such loops. A one-dimensional steady-state physical model of the absorber (SOLAB) has been implemented in *Fortran90*, as no suitable program for the detailed simulation of an absorber containing a supercritical fluid has been found elsewhere. A similar finite-volumes absorber model has been recently proposed and extensively validated against experimental data for the case oil as HTF within the absorber (Forristall, 2003). The absorber is discretized in heat collecting elements (HCE). Thermal losses can be calculated along the length of the adopted solar loop by solving energy balance equations accounting for the heat transfer fluid, the receiver, the glass envelope and the environment. Pressure losses are estimated with correlations, as well as the main field and optical losses.

Almost no scientific literature is available regarding the experimental assessment of heat transfer or pressure drop correlations applicable to the flow of complex supercritical organic fluids in pipes. Experiments and theory show that in water and carbon dioxide, an enhancement of the heat transfer coefficient (HTC) in the near critical region can occur for a certain range of the heat fluxes, while the phenomenon of heat transfer deterioration (HTD) can take place under conditions of high q / G (being q the heat flux and G the mass flux) (Pioro et al., 2004a). Figure 2 shows the variations of some properties of interest in heat transfer and pressure drop calculations in the critical region, according to the models applied for D_4.

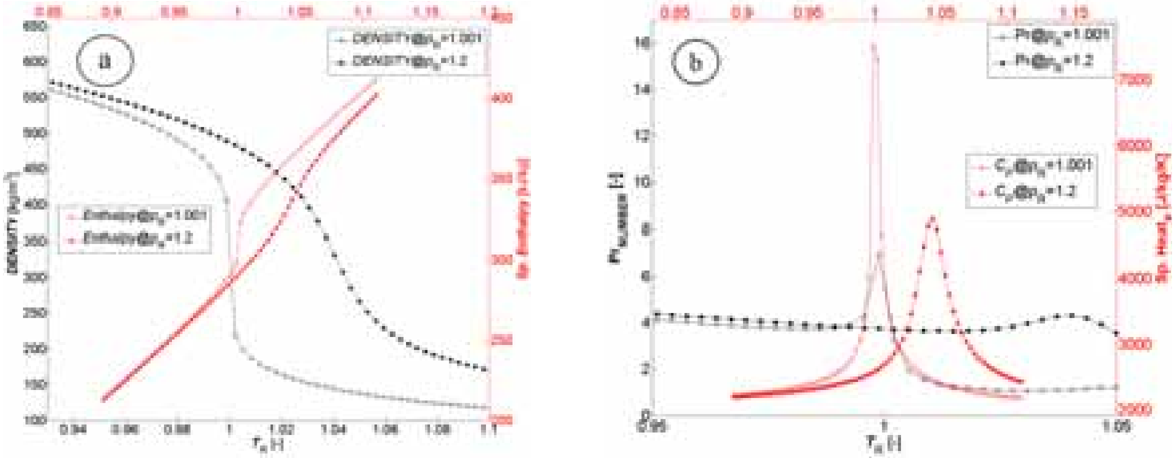

Fig. 2: Variation of thermo-physical properties of interest for the estimation of the heat transfer coefficient in the critical region. The fluid is D_4. a) Density and specific enthalpy, b) Prandtl number and isobaric specific heat. Properties vs. reduced temperature are shown along two isobars (reduced pressure).

In this work a recent correlation (Bae and Kim, 2009) for the estimation of the heat transfer coefficient has been selected. This correlation is capable of predicting with sufficient accuracy also the deteriorated heat transfer regime and it has been validated against experimental data also for HCFC-22 in the near critical and supercritical region. According to the authors, if the non-dimensional group $Bu = Gr_b/(Re_b^{2.7}Pr_b^{0.5})$ (accounting for HTD) is small enough ($Bu<10^{-8}$), heat transfer can be predicted with classical Dittus-Boelter-like equations. Given the foreseeable values of the process parameters involved (mainly in terms of heat flux and mass flux), this limit is not likely to be reached in any operating condition. These conclusions are also coherent with the limits suggested by other authors (Hitch et al., 1997; Grabezhnaya and Kirillov, 2006). Thus the following equation has been used in the present work (Bae and Kim, 2009):

$$\mathrm{Nu} = 0.021 \cdot \mathrm{Re}_b^{0.82} \cdot \mathrm{Pr}_b^{0.5} \cdot \left(\frac{\rho_w}{\rho_b}\right)^{0.3} \cdot \left(\frac{\overline{c_p}}{c_{pb}}\right)^{n} \tag{1.1}$$

Where n is a function of T_{WALL}, T_{BULK} and $T_{PSEUDO\text{-}CRITICAL}$. The estimation of the pressure drops in the absorber tube can be performed with usual methods provided that some kind of correction is applied, mainly to take into account the accelerations along the tube consequence of sharp density changes (Pioro et al., 2004b). The proposed formula has the form

$$\Delta P_{\mathrm{TOT}} = \Delta P_{\mathrm{FRICTION}} + \Delta P_{\mathrm{ACCELERATION}} = \xi_{\mathrm{FR}} \cdot \left(\frac{L}{D} \cdot \frac{G^2}{2\rho}\right) + G^2 \cdot \left(\frac{1}{\rho_{\mathrm{OUT}}} - \frac{1}{\rho_{\mathrm{IN}}}\right) \tag{1.2}$$

Where the friction factor ξ_{FR} is evaluated as follows (Filonenko, 1954)

$$\xi_{FR} = \left(\frac{1}{\left(1.82 \log_{10} \mathrm{Re}_b - 1.64\right)^2} \right) \tag{1.3}$$

Figure 3a shows the comparison between data calculated with SOLAB and the reference data (Forristal, 2003): calculations are performed with the same hypothesis in terms of SECE, working fluid – Therminol-VP1 (Solutia Inc., 2009) – and operating conditions. The curves display the global efficiency (thermal efficiency at peak optical efficiency) of the collector. One of the calculated iso-DNI curve (DNI=300 W / m^2) shown in Figure 3a shows also values for supercritical D_4 instead of Therminol-VP1. Being the heat transfer properties and specific heat of the two fluids quite similar, there are no appreciable differences in the efficiency of the collector (mainly function of absorber's outer wall temperature) for the two cases. Figure 3b displays the pressure and the convective HTC between the fluid and the inner wall in the HCE as a function of the position along the absorber. An increase in the HTC is predicted and this is due to values of the fluid temperature close to the critical one.

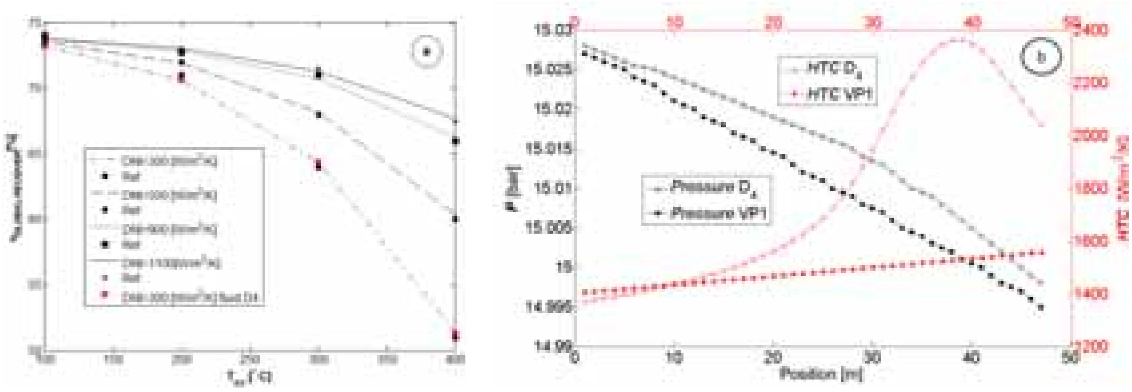

Fig. 3: a) Validation of the 1-D absorber model by comparison with reference data (Forristal, 2003): collector global efficiency vs. average HTF temperature, the incidence is zero (peak optical efficiency), the fluid is Therminol VP1, the mass flow is 8 kg/s. b) Convective heat transfer coefficient and pressure in the absorber for fluids VP1 and D_4 as a function of the position along the absorber. T_{INL} = 300 °C, DNI = 850 W/m^2, T_{OUT} = 325 °C. LS-2 Collector and Solel UVAC Cermet selective coating.

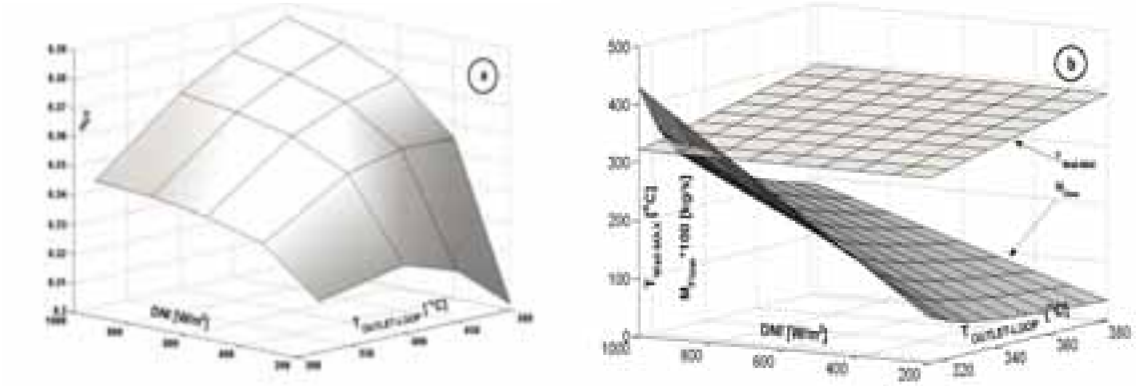

Fig 4: a) Exergy efficiency of an ET-150 collector (supercritical D_4) under varying T_{OUT} and irradiation. T_{IN} is 250 °C, and the incidence is zero (peak optical efficiency); exergy of solar radiation calculated according to Candau (Candau, 2003). b) Maximum wall temperature and working fluid mass flow as a function of DNI and T_{OUT} (T_{IN} is 250 °C).

Figure 4a shows the map of the computed exergy efficiency for a SCA ET-150 with the Schott PTR-70 heat collecting element. The working with fluid is D_4 entering at 250 °C, with varying outlet temperatures and irradiation levels. As expected (Bejan, 1981; Fiaschi and Manfrida, 2010), unlike thermal efficiency (see fig. 3a), the exergy efficiency does not show a monotonic behavior with respect to fluid temperature, but the optimal operating temperature is lower than the maximum achievable under certain irradiation conditions. Figure 4b shows, for the same conditions of the results of figure 4a, the map of the maximum wall temperature and of the mass flows as a function of the loop outlet temperature and DNI. It can be seen that, due to the efficient heat transfer process, conditions at which thermal decomposition of the fluid can be expected, i.e. approximately $T_{WALL\text{-}MAX}$ = 400 °C (Colonna P., 1991; Angelino G., Invernizzi C., 1993) are not reached for any of the foreseen operating conditions.

4.2 Solar Field

The solar field simulated for this study is composed of one single ET-150 SCA with Schott PTR-70 absorbers; the selection of such a relatively large SCA could be plausible wherever a footprint to approx 6 m x 150 m is available. This could be the case of a shopping center in a sunny country.

In order to obtain the model of a solar field which is suitable for the year-round simulation of the performance of the entire power plant in terms of complexity/computational time, a lumped parameters model has been derived by reduction of the 1-D model. Heat-losses (HL) have been modeled as a function of T_{FLUID}, T_{ENV}, DNI, θ_{INCID}, and V_{WIND} through the functional form (Burkholder and Kutscher, 2009)

$$\begin{aligned}\text{HL} = {} & A_0 + A_1\left(T_{\text{HTF}} - T_{\text{env}}\right) + A_2 T_{\text{HTF}}^2 + A_3 T_{\text{HTF}}^3 + A_4 \cdot \text{DNI} \cdot IAM \cdot \cos\vartheta_{\text{INC}} \cdot T_{\text{HTF}}^2 + \\ & + \sqrt{V_{\text{wind}}}\left[A_5 + A_6\left(T_{\text{HTF}} - T_{\text{Amb}}\right)\right]\end{aligned} \tag{1.4}$$

Where IAM is the incident angle modifier for the given SCA (Forristal, 2003). The coefficients have been determined using the Levenberg-Marquardt algorithm for least squares curve fitting (Press et al., 1992), starting from the results obtained with SOLAB. As shown in the previous section (see figure 3a), the use of supercritical fluid has negligible influence on the thermal efficiency of the collector, if compared to a HTF, thus the values of the calculated coefficients are similar to those taken as the reference (Burkholder and Kutscher, 2009). Given the almost linear behavior of thermal losses with respect to fluid's temperature, the power lost across the whole loop can be determined by integrating eq. 1.4. An additional loss of 10 W / m^2 has been considered in order to account for piping losses (Forristal, 2003). Optical losses for off-design conditions have been modeled with the usual approach, i.e., by considering the IAM and the end-losses from the collector (shadowing losses have not been accounted for, as in this case there is only one SCA). Finally, the instantaneous, useful thermal energy (per unit length of collector) is calculated as

$$Q_{\text{input}} = \eta_{\text{OPT}} \cdot \text{DNI} \cdot W_{\text{AP}} \cdot \cos\theta \cdot \text{IAM} \cdot \text{End Loss} - Q_{\text{Thermal loss}} \tag{1.5}$$

Where η_{OPT} is the peak optical efficiency, W_{AP} [m] is the collector aperture, θ is the incidence angle (mainly dependent on the plant's location, time of the year and tracking axis orientation) (Duffie and Beckam, 2006). Given the values expected for the pressure drops across the SCA (less than 1 bar in design conditions), this quantity has been considered fixed and independent from the mass flow. A value of 2 bar is set for the pressure drop such that it includes all the other losses in the cold header and in the valves.

4.3 Components of the power conversion unit

Given the considered power capacity and the need for modularity, compact plate heat exchangers have been modeled for both the recuperator and the air-cooled condenser; the dimensioning procedure (determining exchangers layout and surface areas satisfying the given process constraints) has been carried out by means of a commercial package for the design of compact heat exchangers (Aspentech, 2007). All the main parameters related to on-design operation of the ORC power unit are reported in figure 1. For off-design conditions the overall heat transfer coefficient $U = \left(\frac{1}{h_{\text{IN}}} + \frac{1}{h_{\text{OUT}}}\right)^{-1}$ has been varied proportionally to the mass flow G (neglecting thermal resistance on the process side) as (Incropera and De Witt, 2002),

$$h_{\text{Off-Design}} = h_{\text{Design}}\left(\frac{G_{\text{Off-Design}}}{G_{\text{Design}}}\right)^n \tag{1.6}$$

The exponent n has been determined by fitting off-design simulation results performed with the commercial package. The fluid is considered to enter and exit the condenser in saturated conditions, and the condensation pressure is considered fixed by the control of the coolant mass-flow. The pressure drop across the heat exchangers is also considered constant in off-design conditions for simplicity.

The modeled expander is a single-stage supersonic axial turbine. The turbine is assumed chocked in all operating conditions, and the result of the design calculation is the critical nozzle area (where sonic conditions occur) (Dixon, 1998). For off-design simulations there exists a relation between the mass flow and inlet pressure. The isentropic efficiency of this type of turbomachinery strongly depends from the expansion ratio β: the efficiency is a parabolic function of u/C_0 for different values of β, being u the rotational speed and C_0 the ideal discharge velocity ($C_0 = \sqrt{2\Delta h_{IS}}$) (Verneau, 1987). The locus of maximum efficiency can in turn be described by a logarithmic function of the expansion ratio. The rotational speed is considered to be controlled in order to operate the turbine at maximum efficiency for a given expansion ratio. For simplicity, the efficiency of the pumps and of the fans are assumed constant and equal to the design values for all off-design conditions, as well as the efficiency of the high-speed generator.

For this first study, the temperature at the outlet of the solar field is assumed controlled and kept at a constant value (see fig. 4a), even though it is known that such a control strategy is a sub-optimal; the mass-flow therefore can fluctuate as a response following the available irradiation. The control strategy must also guarantee that the pressure of the working fluid in the solar field is supercritical for all operating conditions. In order to fulfill these requirements, a valve is placed at the outlet of the solar field (see fig. 1), such that the pressure at the inlet of the chocked turbine can be reduced in case of low irradiation. Conversely the the main pump is appropriately regulated if the mass-flow must be higher than the nominal value.

5. Simulation of one year of operation

The hourly simulation of one year of operation has been carried out in order to evaluate the performance of the proposed system. For this exemplary case, the data for solar radiation and environmental conditions are those of the Saguaro power station in Tucson, AZ (Canada et al., 2006; Kolb and Hassani, 2006). Data on air temperature, wind speed and DNI have been obtained from the TMY3 weather-data bank (NREL, 2008).
An availability factor of 0.96 has been assumed for the solar field, coherent with an unmanned installation. A reducing factor of 50% has been considered for hours where irradiation was enough to run the plant, but no sun had been available for the previous five hours. In this way a rough estimation of the energy contribution required for start-up operation is introduced (Cabello et al., 2011); the value is chosen considering the high flexibility of ORC turbogererators.
The performance of the 100 kW$_E$ DSFG ORC system is finally compared to the performance of the 1 MW$_E$ Saguaro plant (Canada et al., 2004); being the SECE and thus the optical performance almost the same for the two cases (Fernández-García et al., 2010), the main difference is the performance of the power conversion units. Tab. 2 shows the main performance parameters of the simulated and the actual installations.

Tab. 2: Performance comparison between 100 kW$_{EL}$ DSFG system and 1 MW$_{EL}$ Saguaro power station (Canada et al., 2004)

Plant Location	Saguaro Desert, Tucson, AZ	
DNI [kWh/m^2 year]	2636	
Plant	Saguaro APS	DSFG-scORC
Plant Size [kW$_E$]	1000	100
SECE collectors/absorbers	Solargenix DS-1/Schott PTR70	Eurotrough ET-150 /Schott PTR70
Solar Field Size [m^2]	10340	817
Solar Field availability	-	0.96
Solar Field HTF	Xceltherm 600	D_4 (supercritical)
ORC working fluid	n-Pentane	D_4
Nominal Data		
Condensation Temperature [°C]	15	80
ORC Turbine gross output [kW$_E$]	1160	120
$T_{INLET-SF}$ [°C]	120	270
$T_{OUTLET-SF}$ [°C]	300	380
$T_{INLET-ORC-TURBINE}$ [°C]	204	380
$p_{INLET-ORC-TURBINE}$ [bar]	22.3	15
η_{ORC}	20.7 %	22.3 %
$\eta_{SOLAR-TO-ELECTRIC}$	12.1 %	13.5 %
Results		
Annual Capacity Factor	23 % (Gross)	25% (Net)
$\eta_{SOLAR-TO-ELECTRIC}$ Annual	7.5 % (Gross)	13.5% (Net)

6. Conclusions and future work

The study documented in this paper is focused on the feasibility of a new concept for a small-scale CSP plant (100 kW_E). The power conversion unit is a high-efficiency, therefore high-temperature, air-cooled ORC turbogenerator powered by Direct Supercritical Fluid Generator: the working fluid (a siloxane) is pressurized at supercritical level and the supercritical fluid receives thermal energy directly in the absorber. A supercritical cycle with the selected working fluid features a low maximum pressure and no need for an intermediate heat transfer loop. The system configuration is therefore simple and suitable for standardization and high-volume production. The simulations show that in condition of optimal insolation the maximum net conversion efficiency is 22.3 % while the net electrical efficiency over one year of simulated operation is 13.5%. These projected values are higher than those available for the best solar ORC plant in operation, which is water cooled, and close to those of proposed high-efficiency air-cooled large solar power stations (Mittelman and Epstein, 2010).
A high value of the turbine inlet temperature (380 °C), together with a relatively high value of the condensation temperature (80 °C) of the thermodynamic cycle have been selected. The thermal stability of the fluid over long period of operation should be better investigated, though preliminary results are promising provided that the fluid is properly treated and the plant made completely air-tight.
Future work, made possible by the availability of the tools developed for this study, will consider the optimization of the operating parameters, the possible advantage of using mixtures as working fluids, the possibility of using the working fluid also for thermal storage. Several improvements to the models are under scrutiny, namely more complex and accurate models for the heat exchangers, of the turbine and of the heat transfer characteristic of the supercritical fluid. A dynamic model of the system is under development for control studies. The construction of a proof-of-concept prototype power conversion unit with commercial participation is also presently discussed. Tests on an absorber in which supercritical siloxanes is flown should also be undertaken.

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