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# Accepted Manuscript

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# CSP plants with thermocline thermal energy storage and integrated steam generator – Techno-economic modeling and design optimization

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#### 9 Abstract

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Although CSP has reached technological maturity, high capital investment and specific electricity cost remain the major development barriers. To reduce them, highly efficient, integrated, and cheaper CSP components are urgently needed. In this paper, we investigate a novel CSP plant configuration with a single-tank Thermal Energy Storage (TES) fully integrated with the steam generator.

The objective of this research is twofold: i) provide a reliable model of single-tank thermal storages with integrated steam generator; ii) identify two optimized CSP plant designs to achieve best energetic and economic performances. To achieve these aims we developed a numerical model of the main system components and validated it against experimental data. This model was then integrated in a full simulation and heuristic design optimization of the plant.

The results revealed that the system proposed can generate electricity in middle-Italy (Rome) at a cost of 230.25 \$/MWh with a 15 % reduction compared to the double tank option. Furthermore, if cogeneration is used to recover the waste heat, this system is an interesting option for users such as small districts, university campuses and hospitals. In the latter case, the optimized system pays off in 6 years and covers 80 % of the heating and cooling requirements.

#### 25 Highlights

- A novel CSP plant with thermocline TES and integrated steam generator is modeled in details
- The solar field and the integrated TES models are validated with experimental data
- The single tank configuration lowers the LEC of 42 \$/MWh
- Cogeneration lowers the LEC of 28 %
- 30

- 31 Keywords: Concentrated Solar Power; Integrated Steam Generator; Molten Salts; Techno-economic
- 32 Optimization; Thermocline Energy Storage;

## 33 Nomenclature

Latin let	ters
A	Area $[m^2]$
С	Cost [\$]
$c_p$	Specific Heat $\left[\frac{J}{kgK}\right]$
$C_y$	Yearly cost $\left[\frac{\$}{\text{year}}\right]$
d	Diameter [m]
E	Yearly Electrical energy $\left[\frac{MWh}{year}\right]$
Ė	Electrical power [W]
е	Specific kinetic energy $\left[\frac{J}{kg}\right]$
Eu	Euler number [-]
FIT	Feed-In Tariff $\left[\frac{\$}{MWh}\right]$
$f_{labor}$	Labor cost index ratio[ – ]
f <sub>M&amp;S</sub>	Marshall & Swift cost index ratio [-]
h	Specific enthalpy $\left[\frac{J}{kg}\right]$
k	Thermal conductivity $\left[\frac{W}{m K}\right]$
$k_{I}$	Geometric factor for helicoidal heat exchangers [-]
L	Length [m]
LEC	Levelized Electricity Cost $\left[\frac{\$}{MWh}\right]$
т	Mass flow rate $\left[\frac{kg}{s}\right]$
п	Scale factor [ – ]
Nu	Nusselt number [-]
р	Pressure [Pa]
Pr	Prandtl number [-]
Q	Yearly Thermal Energy $\left[\frac{GJ}{year}\right]$
Ż	Thermal power [W]

R	Revenues [\$]
Re	Reynolds number [-]
$R_y$	Yearly revenues $\left[\frac{\$}{\text{year}}\right]$
S	Characteristic size [-]
SPBT	Simple Payback time [years]
Т	Temperature [K]
t	Time [s]
TCLF	Thermal Load Capacity Factor [ – ]
U	Global heat transfer coefficient $\left[\frac{W}{m^2 K}\right]$
th	Thickness [ <i>m</i> ]

# 35 Greek letters

- $\alpha$  Heat transfer coefficient  $\left[\frac{W}{m^2 K}\right]$
- $\eta$  Efficiency [ ]
- $\rho$  Density  $\left[\frac{\text{kg}}{\text{m}^3}\right]$

# 36 Subscripts

abs	Absorbed
b	Buoyancy
bc	Boundary condition
bl	Boling
cont	Contingencies
dec	Decommissioning
dir	Direct
ec	Economizer
el	Electrical
ev	Evaporator
f	Friction
Fo	Fouling
FW	Feed-water
h	Hydraulic
HTF	Heat Transfer Fluid
i	Internal

in	Incoming
inv	Investment
lam	Laminar
ls	Liquid-Solid phase transition
MS	Molten Salts
nom	Nominal
0	External
O&M	Operation and Maintenance
out	Outgoing
ref	Reference
SF	Solar Field
sh	Superheater
sol	Solar
t	Tube
th	Thermal
turb	Turbulent
У	Yearly

#### 38 1 Introduction

Despite having been under investigation for several decades, Concentrated Solar Power (CSP) is still hardly competitive with conventional fossil-based power plants and the expected market development in the Mediterranean region remains an unfulfilled promise. The high upfront investment cost and the difficult siting [1] are the two major barriers to a rising share of CSP in the future energy mix. It is thus clear that the primary focus of future research should be the reduction of both the investment cost and the specific cost of electricity, which will extend the CSP market also to mid-size plants located at intermediate latitudes.

The first step in this direction is the simplification of the power plant loop. In this regard, ENEA (Italian National Agency for New Technologies, Energy and Sustainable Economic Development) has promoted [2] the use of a thermocline (i.e. single-tank) Thermal Energy Storage (TES) with an integrated Steam Generator (SG) submerged in the heat storage medium. The plant can be further simplified through the use of the molten salts mixture, which was commonly found as heat storage medium, also as the Heat Transfer Fluid [3] with consistent benefits to the efficiency of the power cycle.

53 The submerged steam generator technology is well-known within the nuclear community and many 54 models have been developed in the past. For instance Ref. [4] proposed a lumped parameter approach 55 considering three regions (i.e. the subcooled, the boiling and the superheater region) with movable 56 boundaries, while in Ref [5] the Authors refined the discretization to get a 1D finite volume approach 57 [6]. However, only a few studies [7], considered the natural circulation on the coolant side, with a 58 design close to the submerged steam generator proposed by ENEA. Furthermore, literature is rich in 59 thermocline TES model. For instance, Yang and Garimella [8] investigated the performance of a 60 molten salts thermocline tank filled with quartzite rock through a 2D axial-symmetric finite volume 61 model; they show that the discharge efficiency raises for tanks with a high aspect ratio and operated 62 at small Reynolds number. Strasser et al. [9] adopted a similar approach to show that the cycle 63 efficiency can be further enhanced with a structured concrete network instead of conventional packed bed material. The use of latent heat storage in CSP has also been studied in great details: 64 65 Nithvanandam et al. [10] studied the performance of a packed bed TES with encapsulated PCM during partial charging and discharging cycles while Fornarelli et al. [11] developed a detailed 66 67 numerical model of a shell-and-tube TES with Phase Change Material showing that natural 68 convection can be conveniently exploited to reduce the melting time. Despite this great availability 69 of literature on the topic, only Ref. [12] studied the integrated storage-steam generator system. The 70 great level of detail of their finite volume model makes it ideal for technology development but 71 impractical for system analysis and plant optimization, which demand for more compact modeling 72 approaches.

73 For what it concerns the reduction of the specific cost of electricity, a possible field of competitiveness 74 improvement for small CSP is represented by polygeneration. The option of CSP-driven desalination 75 has been widely investigated [13 14], since regions with high water scarcity generally have a large 76 solar resource. Another interesting cogeneration option is the CSP-driven biomass gasification, which 77 has lately received considerable attention in the scientific community [15]. On the other hand, it 78 should be noted that only a few researchers [16, 17] have investigated the cogeneration of power, 79 heating and cooling in a single CSP plant, which could be an ideal opportunity to enlarge the market 80 of CSP to users like small districts, university campuses and hospitals.

We believe that the innovative match of these two concepts, i.e. the ENEA compact system and the cogeneration option, has the potential to open the doors of CSP to small-scale facilities in regions with moderate solar resources. In order to quantify this potential, in this paper we utilize the tools of energy and economic analysis, which have been proficiently applied in the past to solar tower combined cycle [18, 19], parabolic through plants for process heat generation [20] and to CSP desalination plants [21].

87 This paper stems from the need of filling the literature gaps we highlighted in this introduction. 88 Firstly, we aim at providing a reliable (i.e. validated with experimental data) and computationally 89 cheap (i.e. suited for system-level annual simulations) modeling framework of the storage tank with 90 integrated steam generator. Secondly, this paper has the objective of proposing two optimized designs 91 of the small CSP cogeneration system as well as to analyze their performances. The first design is 92 thought for an ideal thermal user and has the aim of establishing the potential of the technology. The 93 second one is targeted to a specific user, i.e. a hospital, and has the aim to analyze the performances 94 of the system when coupled with a real user in a real energy market.

#### 95 2 The CSP cogeneration plant with thermocline TES and integrated Steam Generator

#### 96 2.1 Power plant description

97 Figure 1 presents the system proposed by ENEA. The molten salts pump (MSP) circulates the "solar 98 salt" (i.e. an eutectic mixture with 60 wt % NaNO<sub>3</sub> and 40 wt % KNO<sub>3</sub>) from the storage tank into 99 the receiver tubes of the Parabolic Trough Solar Collectors (PTSC) (streams 1, 2 and 3). Once the 100 fluid reaches the desired temperature (stream 4), it is circulated back to the storage tank (stream 6). 101 If the system conditions do not allow to reach the desired temperature the salts can be circulated back 102 to the solar collectors with a by-pass valve (stream 5). The storage tank contains a steam generator, 103 which is immersed in the molten salts; this sub-system is called Storage Tank with Integrated Steam 104 Generator (STISG). The steam produced (stream 7) flows to the steam turbine and it is eventually condensed in the condenser (WCD) (stream 8). In cogeneration mode, the thermal power collected 105 106 by the steam condenser (stream 11) is used to satisfy the thermal requirements of a heat consumer or can be fed to an Absorption Chiller Unit (ACU) to satisfy a cooling load. Finally, the Rankine cycle 107 108 is closed with the use of a water pump (WP1).

In the following sections, a summary of the modelling approach of the three main subsystems of the plant is given, namely the solar field, the STISG and the power block. The modeling approach utilized in this paper results from a trade-off between fidelity of the system representation, complexity of input data needed to model a real installation and computational effort required for the design optimization of such a complex system. The good agreement with experimental results suggests the validity of the modeling assumptions and the correctness of the code implementation.

#### 115 **2.2 Main assumptions**

116 The following assumptions have been made to model the plant:

• The maximum design temperature in the receiver is set to 550 °C. This value is suggested based on the experience maturated at the 5 MW Archimede plant in Priolo Gargallo [22]. At

- higher temperatures, alkaline hydroxides and carbonates are produced at higher rate. These
  species present a limited solubility in molten nitrates and precipitate rapidly yielding to pipes
  and valves occlusion.
- The high pressure level of the steam cycle has been fixed to 40 bar.
- The condenser minimum driving temperature difference, i.e. at the pinch point, has been set to 10 °C.
- The temperature required by the waste heat recovery unit (stream 12) has been set equal to 90
   °C.
- The efficiency of the heat distribution system is set to 90 % and the Coefficient of Performance
   of the absorption chiller to 60 %, as suggested as a reasonable value for single effect Water LiBr absorption machines (e.g. in [23]). This means that, in winter, 90 % of the recovered
   heat is available as heating power, while, in summer, 60 % of the recovered is available as
   cooling power. Distribution losses are neglected.
- 132 **3** Mathematical modelling of the power plant components

#### 133 **3.1 Solar field**

#### 134 3.1.1 Components description and Model

We have modeled a concentrator similar to the one already in operation at the Archimede plant [22]. 135 136 The parabolic through reflector is a 12.5 m long parabolic mirror with 5.76 m of aperture and a focal height of 2.01 m. It sustains a 4.06 m long receiver tube consisting of an absorber inside a glass 137 138 envelope with bellows at either end. The absorber is a stainless steel tube (70 mm in diameter) which 139 is treated with selective coating to obtain a high absorptance in the solar energy spectrum, and low emittance in the infrared (i.e. 95% and 7.3% respectively from manufacturer specifications). The 140 141 glass envelope (125 mm in diameter) is made of Pyrex and guarantees a transmittance higher than 96 142 % in the full range of operating temperatures. The annulus space between the absorber and the glass envelope is under vacuum (1 x  $10^{-4}$  mbar) to reduce thermal losses. 143

In the present work, the analytical equations of Ref. [24] are used for the solar position and the optical model of the receiver while a more detailed approach is followed for the thermal model of the receiver tube. A quasi 1D model is implemented: the receiver is discretized along the axial direction and, for each of the finite volume, a thermal balance is written considering non-advective heat transfer (i.e. conduction and radiation) only in the radial direction. This approach is widely used for the simulation of thermal systems of this type [24, 25]

150	Specifically, the formulation presented in Ref. [25] is considered: assuming steady-state and for a
151	negligible change in potential energy we can write:
152	$\dot{Q}_{net} = \dot{m}_{HTF} \left( h_{in} + e_{in} - h_{out} - e_{out} \right) \tag{1}$
153	$\dot{Q}_{net}$ is the radiative power effectively transferred to the heat transfer fluid and can be calculated as:
154	$\dot{Q}_{net} = \dot{Q}_{abs} - \dot{Q}_{losses} \tag{2}$
155	In steady-state conditions, the concentrated radiation absorbed on the surface of the absorber tube can
156	be either transmitted to the heat transfer fluid or rejected towards the environment. In the first case,
157	we have a series of the following thermal resistances (Fig. 2)
158	• Conduction from the outer surface of the absorber tube to the inner surface of the absorber
159	tube
160	• Convection from the inner surface of the absorber tube the heat transfer fluid
161	In the second case, the thermal power path is the following (Fig. 2):
162	• Radiation/convection heat transfer from the outer surface of the absorber tube to the inner
163	surface of the glass envelope
164	Conduction heat transfer across the glass envelope

- Radiation/convection heat transfer from the external surface of the glass envelope towards the environment
- 167 The thermal properties of the materials and the correlations proposed in [25] were used for the 168 calculation of the heat transfer coefficients. The irradiance data were obtained from the HelioClim3 169 database [26] and the wind speed and ambient temperature data from the EnergyPlus database [27], 170 both providing data with a 15 minutes sampling.

171 *3.1.2 Experimental validation* 

165

166

172 The test bench consists of a 50 meters parabolic though solar field, similar to the one described in the 173 modeling section of this paper. The experimental string is composed by 4 reflectors in series. Four 174 thermocouples are soldered on the external surface of the receiver tube at each joint between consecutive 175 reflectors. Two submerged thermocouples are placed at the inlet and at the outlet of the experimental facility, 176 i.e. at x = 0 m and at x = 50 m respectively. The soldered thermocouples provide a highly varying measurement 177 along the angular coordinate which cannot be accounted for in our quasi 1D model. Hence, only the 178 measurements provided by the submerged thermocouples is used. The measurement at x = 0 m provides the 179 inlet boundary condition while the measurement at x = 50 m is used to validate the model. Figure 3 compares 180 temperature measured at x = 50 m with values predicted by our numerical model. The average mass 181 flow rate during the test is 6.39 kg/s with a standard deviation of 0.23 kg/s. 182 The largest difference between experimental and numerical results arises when the inlet temperature

- The fargest unreferee between experimental and numerical results arises when the mild temperature
- is varied over the duration of the test because the model does not account for transient effect, while a

184 good approximation is visible during steady-state conditions. Nevertheless, a steady-state model 185 remains suitable to reproduce the normal operating conditions of a commercial CSP plant, where the 186 mass flow rate is varied by the control system to maintain a constant temperature levels across the

receiver tubes. The average first law efficiency is calculated as 0.54 with a standard deviation of 0.05.

#### 188 **3.2** Storage tank with integrated steam generator

#### 189 *3.2.1 Component description and model*

The steam generator is a once-through counterflow shell-and-tube heat exchanger with a helicoidal tube bundle: on the shell-side, in an annulus-shaped channel, the molten salts flow downward and, on the tube side, water flows upward becoming superheated steam. This heat exchanger operates in natural circulation mode on the molten salts side thanks to the strong fluid; in fact, within the range of temperatures considered, the density of the fluid experiences nearly a 10 % variation which is exploited as motion driving force.

196 Figure 4 schematically illustrates the STISG system for the small CSP cogeneration plant described197 in Section 2.

- The temperature of fluids along the axial dimension of the steam generator are calculated with a onedimensional finite volume numerical model [28] using a double iteration loop to solve the natural circulation problem (Figure 5):
- 201 1. The molten salts mass flow is guessed

202 2. The outlet temperature of the molten salts (bottom side of the steam generator) is guessed

- 3. The thermal problem is solved following a fist-order upwind approximation on the water side
   until the temperatures of the two fluids in the upper side of the steam generator are obtained,
   i.e. the molten salts inlet temperature and the steam outlet temperature
- 4. The calculated inlet temperature of the molten salts is compared with the boundary condition.
  If the convergence criterion is not met, a new outlet temperature is calculated and the code
  returns to step 3. Otherwise, the algorithm is allowed to proceed to step 5.
- 5. The pressure drop on the molten salts side is calculated and it is compared to the buoyancy
   pressure difference. If the convergence criterion is not met, the algorithm calculates a new
   mass flow and returns to step 1. Otherwise the algorithm returns the solution.
- 212 Convergence criteria are written as absolute differences where the tolerances, i.e.  $\varepsilon_{th}$  and  $\varepsilon_{fd}$ , are set

213 to  $10^{-3}$  °C and  $10^{-2}$  Pa for the thermal and fluid-dynamic model respectively. The heat transfer and

214 pressure drop correlations presented in [28] were used for the calculations and are briefly summarized

215 below.

216 The water internal heat transfer coefficient in the economizer and in the superheater are calculated

217 with the Dittus-Boelter [29] and the Heinemen correlation [30] respectively. In formulas:

218 
$$Nu_{ec,w} = 0.023 \,\mathrm{Re}^{0.8} \,\mathrm{Pr}^{0.4}$$
 (3)

219 
$$Nu_{sh,w} = 0.133 \,\mathrm{Re}^{0.84} \,\mathrm{Pr}^{0.333}$$

In the evaporating section, the Chen correlation [31] was used to calculate the heat transfercoefficient:

(4)

222 
$$\alpha_{ev,w} = \alpha_{bl}S + \alpha_{ls}F \tag{5}$$

where the suppression factor S accounts for the reduction of the boiling heat transfer coefficient when
convective boiling becomes dominant; F is the Chen phase multiplicator. For further details the reader
is referred to the original work of Chen [31].

On the molten salts side, the steam generator can be modeled as a bank of helicoidal tubes in crossflow. The Nusselt number was calculated combining a turbulent and a laminar term in the following
way [30]:

229 
$$Nu_{MS} = 0.3 + \sqrt{Nu_{lam}^2 + Nu_{turb}^2}$$
 (6)

where:

231 • 
$$Nu_{lam} = 0.664\sqrt{\text{Re}} \text{Pr}^{\frac{1}{3}}$$
 (7)

232 
$$\circ Nu_{turb} = \frac{0.037 \,\mathrm{Re}^{0.8} \,\mathrm{Pr}}{1 + 2.433 \,\mathrm{Re}^{-0.1} \left(\mathrm{Pr}^{\frac{2}{3}} - 1\right)}$$
(8)

Once the internal and external heat transfer coefficients, i.e.  $\alpha_i$  and  $\alpha_o$ , are known the global heat transfer coefficient of the j-th volume U<sub>i</sub> is obtained as:

235 
$$U_{j} = \left(\frac{1}{\alpha_{i}} + \frac{r_{i}}{r_{o} \alpha_{o}} + \frac{r_{i} \log\left(\frac{r_{o}}{r_{i}}\right)}{k_{t}}\right)^{-1}$$
(9)

236 The heat transfer rate exchanged in the j-th volume  $\dot{Q}_i$  is hence calculated as follows:

237 
$$\dot{Q}_j = U_j S_j (T_{FW_j} - T_{MS_j})$$
 (10)

The temperature profiles on the water and molten salts side are then calculated according to an upwindscheme. For the economizer and superheater sections we write:

240 
$$T_{FW_{j+1}} = T_{FW_{j+1}} - \frac{\dot{Q}_j}{m_{FW}c_{p,FW}}$$
 (11)

In the evaporating section, the water temperature is always equal to the saturation temperature and we monitor the evolution of the vapor fraction  $x_{FW}$  as following:

243 
$$x_{FW_{j+1}} = x_{FW_j} + \frac{Q_j}{m_{FW}h_{fg}}$$
(12)

244 On the molten salts side we have:

245 
$$T_{MS_{j+1}} = T_{MS_{j+1}} - \frac{\dot{q}_j}{m_{MS}c_{p,MS}}$$
 (13)

The correlations for a bank of helicoidal tubes in cross flow proposed in [30] are used for fluiddynamic calculations. After the preliminary calculation of the geometrical factor  $k_{1,}$  the Euler number Eu is obtained as follows:

249 
$$\frac{Eu}{k_1} = 0.263 + \frac{0.867 \cdot 10^{-2}}{\text{Re}} - \frac{2.02}{\text{Re}^2}$$
 for  $\text{Re} < 2 \cdot 10^3$  (14)  
250  $\frac{Eu}{k_1} = 0.235 + \frac{0.198 \cdot 10^{-4}}{\text{Re}} - \frac{0.124 \cdot 10^8}{\text{Re}^2} + \frac{0.312 \cdot 10^{11}}{\text{Re}^3} - \frac{0.274 \cdot 10^{14}}{\text{Re}^4}$  for  $2 \cdot 10^3 \le \text{Re} < 2 \cdot 10^6$   
251 (15)

As far as the modeling of the stratification in the TES is concerned, we consider the Reynoldsaveraged version of the turbulent Navier-Stokes equations. Mathematically:

$$254 \qquad \frac{\partial \rho}{\partial t} + \frac{\partial \rho u_j}{\partial x_j} = 0 \tag{16}$$

255 
$$\frac{\partial \rho u_i}{\partial t} + \frac{\partial}{\partial x_j} (\rho \ u_j u_i) = -\frac{\partial P}{\partial x_i} + \frac{\partial \sigma_{ij}}{\partial x_j} + F_i$$
(17)

256 
$$\frac{\partial \rho E}{\partial t} + \frac{\partial}{\partial x_j} (\rho u_j H) = \frac{\partial}{\partial x_j} (u_i \sigma_{ij}) - \frac{\partial}{\partial x_j} \left( \left( \frac{\mu}{PR} + \frac{\mu_T}{PR_t} \right) \left( \frac{\partial T}{\partial x_j} \right) \right)$$
(18)

257 where  $\sigma_{ij}$  is the tensor of viscous stresses,  $S_{ij}$  is the tensor of shear stresses and H is the total

enthalpy. Closure of the turbulent equations is provided through a  $k - \omega$  model.

The governing equations are converted to algebraic equations using the finite-elements method in 259 260 COMSOL Multiphysics [32] with 2nd order Lagrange finite elements for the velocity field and linear elements for the pressure and temperature fields. Time integration lies on a fully-implicit variable-261 262 order variable-time step BDF (Backward Differentiation Formula) scheme with maximum and minimum order set to 5 and 2 respectively. The time step is accepted if the  $L_2$  norm of a predictor-263 based relative error estimates is below 1e-3. The set of nonlinear equations arising from the spatial 264 265 and temporal discretization are solved via the under-relaxed Newton method. Preliminary numerical experiments have shown that setting the under-relaxation factor to 0.85 is a good trade-off between 266 reliable convergence and computational cost. Convergence is considered satisfactory when the  $L_2$ 267 268 norm of the residuals drops below 1e-6. At each Newton iteration, the system of linearized equations 269 is solved via the MUltifrontal Massively Parallel Sparse direct Solver (MUMPS) [33]. Verification 270 has been performed with a-posteriori error estimates based on the use of the Richardson extrapolation 271 [34]. A free-triangular mesh with 4.1e4 elements has been chosen as the one that guarantees a Grid Convergence Index (GCI) [35] below 1 %. The element size in the axial direction is 0.5 cm while in 272 273 the radial direction is 1.5 cm

The CFD approach is too demanding for system-level simulations. Hence, in this paper we use a logistic distribution function to represent the non-dimensional molten salts temperature profile of a vertical fluid column inside the tank. The function was parametrized statistically, using 18 Computational Fluid-Dynamic (CFD) simulations. This approach proved to be extremely convenient for the adaptation of CFD results to annual system-level simulation and optimization. For an extensive discussion on the reduction methodology, the reader is referred to [36]

#### 280 3.2.2 Experimental validation

Figure 6(a) shows the results for the steam generator operated with 85 % of the nominal mass flow 281 rate (0.11 kg/s) of water and for a molten salts inlet temperature of 520 °C. The molten salts side 282 283 results show very good agreement with experimental data. The skin temperature (i.e. metal 284 temperature of the external surface of the receiver tube) calculation is quite accurate in the 285 evaporating section while it shows a non-negligible deviation in the superheating section. However, 286 the trend is well reproduced and the large error is mainly due to the sharp increase of the water 287 temperature in the superheating section. A small difference of the water mass flow, in fact, can result 288 in large relative errors.

Figure 6.(b) and Figure 6.(c) show the results obtained with a water mass flow 10 % and 20 % higher than the nominal value. Differently from the previous case, the water side is quite well approximated except for the top section of the steam generator

In all the tests conducted we obtained an average absolute error in the molten salts temperature of 3.16 °C with a standard deviation of 3.22 °C. Furthermore, an excellent agreement between predicted and experimental inlet/outlet temperatures of water and molten salts is achieved, thus the model can be confidently used for system-level simulations.

- The validation of the CFD model is an essential step of the methodology proposed in the present paper, allowing to proceed with multiple simulations in different conditions and to characterize the reduced model (i.e. the logistic function) by statistical means. Validation has been performed using experimental data taken from 14 thermocouples equally spaced every 10 cm on a long rod that is immersed vertically in the tank at r = 0.5 m.
- The validation of the discharging process is shown in Figure 7(a). Solid lines are the results obtained by the CFD simulation, while starred indicators are the experimental data. The average absolute error is 1.18 °C with a standard deviation of 2.53 °C. As far as the standby process is concerned, the results are compared for a total period of approximately 27.8 hours. Referring to Figure 7(b), starred red markers indicate the experimental results, while blue solid lines are obtained by the CFD simulation A very good agreement is reached in the upper part of the tank where the rate of temperature drop in

- 307 time is perfectly predicted by the CFD model. Also in this case, the comparison with experimental 308 data is satisfactory with an average absolute error if 1.91 °C and a standard deviation of 3.14 °C.
- 309 The reduced model was then tested against the CFD simulations to verify its accuracy. The prediction
- 310 achieved through the two modeling approaches are compared in Figure 8 for both a charging and a
- 311 discharging process. The results obtained with the reduced model show a nearly perfect agreement
- 312 with the CFD ones. The interested reader is advised to examine [36], for the full details and the
- 313 potential applications of this model reduction approach.

#### 314 **3.3** Power block

- 315 The power block sub-system includes three main components: the steam turbine, the steam condenser
- and the feedwater pump.
- 317 The thermodynamic performance of the steam turbine is modelled according to Medina Flores et al.
- 318 [37]. The Authors proposed to write the isoentropic efficiency of the turbine as a function of the steam
- 319 pressure at the inlet and at the outlet section of the turbine.
- 320 In summary, the electrical power output can be written as:

321 
$$\dot{E}_{el} = \frac{1}{\beta} (\dot{m} (h_1 - h_{2,iso}) - \alpha)$$
 (19)

- 322 where  $\alpha$  and  $\beta$  are two pressure-dependent fitting parameters calculated as proposed in the original 323 reference.
- According to Ref. [37], the power output of the turbine during the startup can be obtained through the use of a startup factor  $F_{startup}$  in the following way:

$$326 \qquad \dot{E}_{el} = F_{startup}(t)\dot{E}_{nom} \tag{20}$$

327 The correction factor ranges from 0 to 1, at the beginning and at the end of the startup process328 respectively, and increases quadratically in time. It can be calculated with:

329 
$$F_{startup}(t) = \left(\frac{t_{sin ceStartup}(t)}{t_{startup}}\right)^2$$
(21)

- 330 In the framework of this paper,  $t_{startup}$  is set to one hour.
- The condenser considered in the present work is a shell-and-tube heat exchanger as the one describedin Ref. [38]. Its axial coordinate is discretized and in each of the volume considered, the thermal
- 333 power  $\dot{Q}$  is calculated by means of an energy balance.
- The global heat transfer coefficient is determined as follows [38]:

335 
$$U = \left( R_{fo} + \left( \frac{1}{\alpha_i} + R_{fi} \right) \frac{d_o}{d_i} + \frac{th_t}{k_t} \frac{d_o}{D_m} + \frac{1}{h_o} \right)^{-1}$$
(22)

Here,  $R_{fi}$  is the fouling factor, d is the diameter,  $th_t$  is the tube thickness,  $k_t$  is the tube conductivity 336

337 and  $\alpha$  is the heat transfer coefficient. Subscripts *i* and *o* apply for internal and external side of the tube respectively.  $D_m$  is the mean diameter calculated as follows: 338

$$D_m = \frac{d_o - d_i}{\ln\left(\frac{d_o}{d_i}\right)}$$
(23)

Moving to the feedwater pump, the approach followed is the one of Pelster [39], where the power 340 consumption of the device  $\dot{E}_{pump}$  is calculated with: 341

342 
$$\dot{E}_{pump} = \frac{1}{\eta_h} \dot{m}_{FW} \left(\frac{\Delta p}{\rho}\right)$$
 (24)

Following the approach proposed by the same author, the pump outlet temperature  $T_{out}$  is computed 343 344 as [38]:

345 
$$T_{out} = T_{in} + \frac{(1 - \eta_h)\dot{E}_{pump}}{\dot{m}\bar{c}_p}$$
(25)

Due to the high degree of maturity of these conventional components, the power block model is 346 347 considered reliable enough, and in this case, no experimental validation is performed.

#### 348 **Cases considered** 4

349

350 In this paper, two different optimization cases are considered:

351

CASE 1: THE MODULAR DESIGN: which has the aim of showing the potential of the 352 technology and the advantages of cogeneration in a deliberately general setup

353 CASE 2: THE TAILORED DESIGN, which has the aim of demonstrating the 354 competitiveness of the technology in a specific market with real thermal users.

355 The modular design is a single-objective optimization of the system configuration in order to minimize the Levelized Cost of Electricity. In this case, we fix the size of the plant to 1 MW<sub>e</sub>, which 356 357 should fit many mid-size industrial users and we consider the system to be located in Rome. As far 358 as cogeneration is concerned, since the objective of this case is to quantify the maximum economic 359 advantages that cogeneration can bring, we consider an ideal thermal load where waste heat is always 360 fully utilized to satisfy heating and cooling needs.

361 On the other hand, the tailored design is a double objective optimization built to minimize the payback 362 time and to maximize the fraction of user's heating and cooling load satisfied by the solar system, i.e. the Thermal Load Capacity Factor (TLCF). Hence, in this case, we consider both a real thermal load 363 364 and a real power market. The user in question is a 500 beds hospital, located in middle-Italy. The 365 name of the hospital cannot be revealed due to non-disclosure agreements. The heating load is

366 completely satisfied by a simple natural gas boiler while cooling is obtained through a hybrid system 367 where vapor compression chillers are used for base load and gas absorption chillers are used for peak 368 shaving and security of supply. The absorption machines are single-effect water-LiBr chillers. Their 369 operation is modeled with a constant COP of 60 %, as suggested in [23] for this type of machines. 370 Since the cooling system is already in place and its installation is rather recent (dated 2013), we do 371 not account for it in our economic analysis.

372 The fraction of the building thermal load satisfied through natural gas is monitored through hourly readings of the meter. Figure 9(a) and 9(b) show the thermal load on a typical winter day and on a 373 374 typical summer day respectively. These graphs are obtained by averaging the measurements over the season considered. In winter, two load peaks are visible, one in the morning around 7 am and one at 375 376 night around 8 pm. The load is much steadier in summer where only small fluctuations are visible 377 between 1 pm and 8 pm. The cumulative power distribution in the year considered is given in Figure 378 9(c). A base load is well identified to be slightly more than 500 kW and the peak demand is roughly 379 2500 kW.

- 380 Compared to Case 1, the tailored design should include two additional design variables:
- The size of the power plant, which should fit the specific needs of the user
- The size of a hot water storage tank, which is needed to handle successfully possible
   mismatches between power block operations and the heating/cooling load
- 384 The building is located in a slightly populated area, with large ground availability for the installation 385 of the solar field. The vicinity of the solar field makes the distribution thermal losses negligible.

#### **386 5 Optimization setup**

Evolutionary algorithms are acknowledged to be the most suitable choice for the optimization of
complex energy systems, which often result in Mixed Integer highly Non-Linear Problems (MINLPs)
with several non-feasible holes in the design space [40].

- 390 In this paper, we use the parallel implementation of the GA of the MATLAB Optimization Toolbox
- 391 with a total of 10 decision variables (summarized in Table 1) and a population size of 50 individuals.
- 392 The initial population is randomly generated in the feasible region.
- 393 Convergence is considered reached when the average L<sub>2</sub> norm step in the normalized objective(s)
- space drops below 1e-2. This happened after a total of 52 and 62 generations for the 1<sup>st</sup> and 2<sup>nd</sup> case
   respectively.
- The design variables are selected to enhance freedom in the design of the most relevant power plant components, i.e. the TES, the power block, the solar field, the steam generator and the waste heat
- 398 utilization system.

399 Starting from the TES, the number of storage hours NH is an intuitive representation of the storage 400 tank size. This value is the number of hours of continuous nominal operation that could be guaranteed 401 to the power block during an ideal discharge process, i.e. starting from the tank fully charged at the 402 maximum temperature and assuming no mixing or diffusion during the discharge. The aspect ratio of 403 the tank is defined as the ratio of the tank diameter D to the tank height H and its choice is the trade-404 off between two competing phenomena: a large tank aspect ratio brings a small average Reynolds 405 number of the molten salts during the charging and discharging phase which reduces thermocline 406 degradation due to turbulence effects. On the other hand, a small tank aspect ratio, although reducing 407 the area of contact with the cold and the hot fluid, brings more turbulent degradation.

We chose the design turbine power  $P_e$  as the representative variable for the power block. Once  $P_e$  is set, the remaining power block components are sized according to the design thermodynamic cycle obtainable through the assumptions outlined in Section 2.2.

411 Moving to the solar field, the solar multiple (SM) is defined as the ratio of the total mirror area to the 412 "exact mirror area". This last quantity is the solar field aperture area required to deliver to the power 413 cycle the thermal power needed to operate the turbine in nominal conditions. Besides the total area, 414 the optimal number of collectors per string n<sub>coll</sub> should also be carefully identified: this design variable 415 affects the average heat transfer fluid velocity, whose value is a tradeoff between heat transfer 416 efficiency and pressure losses. Moreover, the orientation of the solar field is expected to play a major 417 role on the annual performance of the system. The optimizer can vary this variable between 1 and 2, 418 being the former the N-S orientation and the latter the E-W orientation. Finally, the solar field design 419 has one more degree of freedom, the solar field spacing d<sub>spacing</sub> between adjacent strings of solar 420 collectors. A too compact solar field design can yield a high self-shadowing effect between solar 421 collectors and a consequent drop in the optical efficiency. On the other hand, a too far placement 422 implicates a higher land cost.

Two design variables were identified for the steam generator: the number of tubes n<sub>tubes</sub> and the height
H. The bounds have been set according to some preliminary design performed by ENEA in the
framework of the OPTS European project [41].

426 In Case 2 we decided to evaluate the installation of a hot water storage tank placed right after the 427 condenser. Hence, a new design variable was created that is the water storage capacity quantified in 428 terms of full load hours  $NH_{water}$  of the heating/cooling system. This quantity is defined as the number

429 of hours of continuous operation guaranteed to the heating and cooling systems at maximum load.

430 As far as the objective functions are concerned, we consider:

• The Levelized Electricity Cost (LEC) for Case 1

• The Simple Pay-Back Time (SPBT) for Case 2

• The Thermal Load Capacity Factor (TLCF) for Case 2

The LEC was preferred over other economic indicators, e.g. the Net Present Value (NPV), for its great adoption in the field of CSP, hence making comparison with other studies straightforward. Also please note that the plant operator and the thermal user are considered two different entities in this study, hence any purchase for the electrical grid or consumption of back-up natural gas by the latter is disregarded.

The first two objectives, i.e. the ones accounting for the economic performance of the plant, aredefined as:

441 
$$LEC = \frac{CRF \cdot C_{inv} + C_{y,O\&M} + C_{y,dec} + C_{y,cont} - R_{y,heat\&cold}}{E_y}$$
(26)

442 
$$SPBT = \frac{C_{inv}}{R_{y,electricity} + R_{y,heat\&cold} - C_{y,O\&M} - C_{y,dec} - C_{y,cont}}$$
(27)

443 where:

• CRF [-] is the annualization factor that can be computed as:

445 
$$CRF = \frac{i(1+i)^n}{(1+i)^n - 1} + k_{ins}$$
(28)

446 In the previous equation the interest rate *i* and insurance rate  $k_{ins}$  are set to 7 % and 2 % 447 respectively as suggested in Ref. [39].

•  $C_{inv}$  [\$] is the investment cost of the plant obtained summing the investment costs of all the 449 components, i.e.  $C_{inv} = \sum C_i$ . The investment cost of the i<sup>th</sup> component  $C_i$  is calculated through 450 the use of cost functions, which stem from a best-fit on a wide range of market data and relate 451 the cost of component to a specific size parameter S<sub>i</sub>. Mathematically [39]:

452 
$$C_i = c_{ref} \left(\frac{S_i}{S_{ref}}\right)^n f_{M\&S}$$
(29)

The adopted  $f_{M\&S}$  index for the present study is the one of 2011 obtained from Ref. [42] and set to 1546.5. The full reference data of  $c_{ref}$ ,  $S_{ref}$ , and n for the power block is obtained by [39], while the ones related to the solar field and the molten salts TES are gathered from [43]. The characteristic dimensions, their reference value and the specific reference costs of the components considered are listed in Table 2. The characteristic dimensions are obtained directly from the definition of the design variables. For more details on the cost function approach used to calculate the components investment costs, the reader is referred to [40, 48].

- $C_{y,O\&M}\left[\frac{\$}{year}\right]$  represents the Operation & Maintenance costs. We consider service contracts for ground keeping, mirrors washing and water treatment, material maintenance for the equipment and operation cost due to personnel. All the data obtained through [43] are normalized on the plant electrical capacity to obtain a specific O&M cost.
- $C_{y,cont} \left[\frac{\$}{year}\right]$  and  $C_{y,dec} \left[\frac{\$}{year}\right]$  refer to contingencies costs and decommissioning costs. In the 467 present paper we follow the approach presented in Ref. [39] and set them to 10 % and 5 % 468 respectively of the total project cost.
- 469

465

•  $R_{y,heat\&cold}\left[\frac{\$}{year}\right]$  accounts for the revenues from the heat market considered as savings brought by the CSP cogeneration installation with respect to a conventional natural gas boiler and a H<sub>2</sub>0-LiBr absorption chiller. In mathematical terms:

473 
$$R_{y,heat\&cold} = p_{NG} \left( \frac{Q_{heat} + \frac{Q_{cold}}{COP_{ACU}}}{\eta_{boiler}} \right) = p_{NG} \frac{Q_{wasteHeat}}{\eta_{boiler}}$$

474

(30)

475 where the second equality sign holds only in case of complete sale of the plant waste heat on 476 the market, i.e. case 1. We set the thermal efficiency of the typical natural gas boiler  $\eta_{boiler}$  to 477 90%, and the price of natural gas  $p_{NG}$  equal to 10.087 €/GJ, that is the market price for 478 industrial users as set by the Italian Ministry of Development and Economic Resources [44].

•  $R_{y,electricity}\left[\frac{\$}{year}\right]$  accounts for the revenues from the electricity market and it is calculated as:

- 481 Where  $p_e$  is the price at which electricity is sold in the italian power market, which is 482 determined by summing the fixed incentive of the Feed-In-Tariff (FIT) scheme and the 483 liberalized price with which electricity producers are remunerated on the day-ahead market. 484 Those last data are obtained on the GME (Italian Electricity Market manager) website [45] 485 for the year 2014 while the incentive tariff is set according to the Italian Ministerial Decree 486 of 6 Jul. 2012 to 320 €/MWh [46].
- Table 2 summarizes the most relevant data implemented in the economic model. For a moreexhaustive breakdown at the component level, the reader is advised to consult [43].
- The last objective function (i.e. TLCF) quantifies the performances of the CSP plant when used incogeneration mode. It is calculated as:

491 
$$TLCF = 1 - \frac{E_{backup}}{Q_{heat} + Q_{cold}}$$

492 that is the solar fraction of the annual heating and cooling demand.

#### 493 6 Results and discussion

#### 494 6.1 Case 1

Table 3 presents the optimized design specifications for Case 1. The optimal design presents a high value of solar multiple and storage tank size in order to increase the capacity factor of the steam turbine. However, the optimal storage tank size and solar multiple are far from the upper bound set for the optimization routine. This means that an optimum is present in the range considered and that

- the marginal cost of adding storage capacity and more mirrors to the solar field does not pay off.
- 500 On the other hand, the number of collectors per string is maximum which means that increasing the 501 length of the single string results in a higher annual yield of the solar field.
- The height of the steam generator and the number of tubes selected are in close agreement with the preliminary design proposed by the manufacturer for the European Project OPTS. Finally, the optimal tracking axis orientation found is N-S which brings an 11% increase in the annual electricity yield of the unit square meter compared with E-W orientation.
- 506 The main annual energy flows and first-law efficiencies are summarized in Table 4. The proposed 507 system in the optimized configuration generates 3864 MWh of electricity per year, which results in a 508 capacity factor of the power block of 38.6 %.
- 509 The second-law analysis of the system in the optimized configuration is conducted to identify the 510 most critical components. A summary is presented in Table 5 while a representation of the exergy
- 511 streams in the CSP plant is given in Figure 10. The exergy efficiency for each component is calculated
- 512 as following [44]:

513 
$$\eta = \frac{E_p}{E_f} = 1 - \frac{E_d + E_l}{E_f}$$
 (33)

- 514 where  $E_p$  is the exergetic product of the component,  $E_f$  are the exergetic resources used to drive it, 515  $E_d$  and  $E_l$  represent the exergy destruction and the exergy losses of the component.
- 516 Most of the solar exergy hitting the reflectors, i.e. 51.4 %, is lost before reaching the receiver tube 517 due to imperfect concentration. Another big portion is lost or destroyed in the receiver tube such that 518 only 23.3 % of the solar exergy reaches the storage unit. Hence, it is clear that the most critical 519 components are the solar-to-thermal converters. An effective strategy to increase the second-law 520 efficiency of the system is to adopt reflectors with higher optical efficiency and/or multiple axis 521 tracking. This would certainly modify the optimal design of the plant: the increased system products

522 yield per surface ratio would make a larger solar field convenient. On the other hand, most of the 523 exergy optimization studies consider the unit exergetic cost [49] of the functional products as the 524 performance measure of the plant. To this aim, the storage unit, which presents a higher exergetic 525 efficiency, can be improved by reducing the exergy destruction in the thermocline, as shown in [28].

526 Little room for improvement is left in a mature component like the power block.

527 Anyhow, the exergetic performance of the system is not considered in the optimization problem 528 formulation and further investigations on this matter are left to future extensions of this work.

529 The stratified storage has an acceptable exergetic efficiency, i.e. roughly 83 %. This figure of merit 530 allows for a performance comparison between single tank and double tank storage systems. The 531 exergy product of a storage unit can be written in the following form:

(34)

(35)

532 
$$E_p = \rho_E V \eta$$

where  $\rho_E$  is the exergy density of the unit (MWh/m<sup>3</sup>) and V is the total volume. If the double-tank installation (denoted by the subscript DT) is designed to deliver the same exergy of the single-tank installation (denoted by the subscript ST) we can estimate the required volume ratio as:

$$536 \qquad \frac{V_{DT}}{V_{ST}} = \frac{\rho_{ST}\eta_{ST}}{\rho_{DT}\eta_{DT}}$$

where  $\rho_{ST}/\rho_{DT} = 2$ . If we consider an ideal (i.e. with 2<sup>nd</sup> law efficiency equal to unity) double-tank storage system we obtain  $V_{DT}/V_{ST} = 1.67$ . The additional investment cost of the double-tank alternative results in a LEC of 272.59 \$/MWh, which is 18 % higher than the one obtained with a single-tank system.

Moving to the analysis of economic performances, the system requires a total capital investment of 541 542 14.56 million of US\$ and can generate electrical power at the levelized cost of 230.25 \$/MWh. From 543 a comparison with studies on Parabolic Through solar plants ([50-52]), where the estimated LEC 544 ranges from a minimum of 200 \$/MWh to a maximum of 360 \$/MWh for plants sizes in the range of 545 50 MW<sub>e</sub> to 100 MW<sub>e</sub>, it is clear that the solution proposed has competitive economic performances. The CAPital Expenditures (CAPEX) and LEC breakdown are represented in the pie charts of Figure 546 547 11. The cost of the solar field is still the major contributor to the total power plant investment cost 548 accounting for 54 % of the total. The second largest item in the plant's owner expenditures list is the 549 power block, which accounts for 15% of the total cost. Finally, the storage tank represents only 10% 550 of the total cost in the optimized configuration. The other pie chart represents the Levelized Electricity 551 Cost breakdown where also the revenues generated from the heat sold on the market are included. In 552 this way, it is possible to notice that cogeneration has the potential to decrease the specific cost of electricity of 28 % and this option is thus crucial for the economic viability of small CSP systems. 553

#### 554 6.2 Case 2

Figure 12 presents the Pareto front obtained from the multi-objective optimization of Case 2. It can 555 556 be noticed that the minimum possible payback-time found is slightly higher than 6 years and the 557 maximum fraction of thermal load covered by the solar resource that can be reached is very close to 558 87 %. A complete thermal load coverage is extremely non-economical. In order to satisfy completely 559 the winter request, where the thermal load is maximum and the solar yield minimum, the system 560 would be oversized for most of the year and a large fraction of the thermal energy would not be utilized nor remunerated. The hybridization of the heating and cooling system looks from the curve 561 562 the most interesting option.

563 Two extreme points are selected from the Pareto front and their annual performance is analyzed. Point 564 1 is the most economically viable solution, while the second design, i.e. Point 2, is the one that 565 guarantees the highest solar coverage of the heating and cooling load.

The design specifications and the techno-economic performance of the system in the two selected points are summarized in Table 6. The first issue to notice is that in both points the tracking axis is selected to be East-West oriented. It is well known that this orientation choice guarantees a steadier output throughout the year compared to the N-S counterpart at the price of a lower yearly energy yield. However, we found that the N-S orientation results in a high amount of thermal energy wasted during summer months due to a solar generation that largely exceeds the demand.

572 The optimal combination of solar multiple, molten salts storage tank size and nominal power of the 573 steam turbine is very interesting. It is found, in fact, that is more convenient to buy a large steam 574 turbine coupled with a small tank at the cost of a low capacity factor rather than investing in a big 575 storage tank. On the other hand, there are no appreciable differences in the steam generator design 576 between the two Pareto points which confirms the observations of the previous optimization run. The design of point 2 gives a total efficiency decrease of 2 %. The electrical capacity factor of the power 577 578 block is very similar in the two points and differences in the total electricity generation are mainly 579 due to a slight difference in the nominal steam turbine power selected for the optimal design.

The total investment cost of the design in Point 2 is roughly 3.5 M\$ higher than the one in point 1. The difference comes mainly from the solar field cost, from the molten salts storage tank cost and from the water tank cost. The total cost breakdown in the two points is depicted in Figure 13. The relative investment in storage technologies, i.e. water tank and Molten Salts storage tank is nearly 10 % higher for Point 2 than for point 1. A larger portion than expected is attributed to the purchase of the hot water storage.

586 In order to investigate more in details the trends behind the solution found, the system is simulated 587 with different combinations of steam turbine and storage tank sizes. The turbine size is allowed to

vary between 1000 kW and 3000 kW with steps of 400 kW, while the tank storage size is allowed to vary between 2 and 18 hours with steps of 4 hours for a total of 20 design points analyzed. The size of the solar field, in terms of mirror area is fixed to 28000  $m^2$ , as obtained for the design of Point 1. The water tank size is set to a very small value, i.e. 2 hours, in order to exclude the influence of this parameter on the system performance. All the other parameters are set equal to the design specifications of Point 1.

594 In all the possible combinations obtained through this procedure, we analyze the normalized 595 breakdown of the power plant revenues, i.e. we consider:

- The incentive-related revenues per unit of investment, in the form of Feed-In-Tariff (FIT) due
   to the amount of electricity generated
- The heat-related revenues per unit of investment, due to the heat sold to the user.
- The power market-related revenues per unit of investment, due to the selling of power to the
   electrical grid.

The trend of the FIT-related specific annual revenues per unit of investment  $\left[\frac{\$}{year \$_{inv}}\right]$  for different 601 602 combinations of the two decision variables selected is depicted in Figure 14.(a). If this was the only earning source of the plant, most convenient designs would be obtained for small steam turbine sizes 603 604 with big storage tanks. However, by looking at Figure 14.(b), it is clear that specific revenues 605 connected to real market trends are greater for big turbines and for small storage tank size. The reason 606 for such a behavior is that those plants deliver a higher amount of energy right in the middle of the 607 day and thus the average price at which the power is sold is higher. Increasing the operating hours of the steam turbine only adds cost to the system and lower the average price of electricity. As far as the 608 heat-related revenues are concerned (Figure 14(c)), turbine sizes in the range between 1600 kW and 609 610 2300 kW are recommended for the hospital considered because higher Thermal Load Capacity Factors can be achieved. 611

612 This last analysis shows that the optimal system configuration may vary considerably depending on 613 the incentive policy framework in which the plant is operated.

#### 614 7 Conclusions

In this paper, we firstly presented an efficient and flexible modeling framework that can accurately predict the performance of the single storage tank with integrated steam generator. In particular, the 1D finite volume model of the steam generator predicts the molten salts temperature with a mean absolute error of 3.16 °C, while the analytic approach used to reduce the CFD model of the tank can reproduce the vertical temperature profile with a mean absolute error of 1.18 °C.

620 We used this validated model to optimize the system designs in two different cases. In the first case, we optimized the design of a 1MW<sub>e</sub> plant located in Rome with an ideal thermal load in order to 621 assess the potential of the technology for mid-size users. We revealed that this type of system could 622 generate power at a price of 230.25 \$/MWh, if it is operated for 38 % of the year. In particular, the 623 624 possibility of utilizing locally the waste-heat, being responsible of a 28 % reduction of the Levelized 625 Cost of Electricity, is crucial for the economic viability of this kind of plants. Furthermore, we found 626 that the single tank configuration with integrated steam generator allows to decrease the specific electricity of another 42 \$/MWh compared to double tank option. 627

In the second case, we conducted a case-study with a 500 beds Italian hospital with the aim of investigating the performances of the system with a real user in a real market framework. We found

630 that, if the system is properly designed, the investment costs can be recouped in a period between 6

and 7 years and a range between 80 % and 87 % of the heating and cooling demand can be satisfied

632 with the solar system.

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- 758





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764<br/>765Figure 2. Electrical analogy used to model the heat losses in the receiver tube [22]. (1) Heat Transfer Fluid, (2) absorber inner<br/>surface, (3) absorber outer surface, (4) glass envelope inner surface, (5) glass envelope outer surface, (6) air, (7) sky.





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Figure 3. Validation of the solar field model at ENEA Research Center La Casaccia.





Figure 5. Computational model flow chart of the naturally-circulated steam generator



- 777<br/>778Figure 6. Experimental validation of the steam generator model in three different conditions. (a): 520 °C, 45 bar, mass flow<br/>85% of nominal. (b): 520 °C, 40 bar, mass flow 110% of nominal. (c): 480 °C, 46 bar, mass flow 120 % of nominal
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Figure 9. (a): Thermal load on the typical winter day. (b): Thermal load on the typical summer day. (c): Cumulative power
 distribution in the year considered.



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Figure 10. Annual exergy streams of the CSP plant



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799<br/>800Figure 12. Pareto front: set of solutions of the multi-objective optimizations. Red indicators are obtained by the optimization,<br/>the blue solid line is a polynomial regression function used to highlight the trend





#### Figure 13 Cost breakdown comparison between the two Pareto points selected

Figure



Figure 14. (a):. Revenues from the power market per unit of investment [-]. (b): Revenues from selling heat per unit of investment [-]. (c): Revenues from incentive per unit of investment [-].

Desision variable	Lower	Upper	Type	Utilizeed in	Utilized in	
Decision variable	bound	bound	Type	Case 1	Case 2	
Number of hours of molten	1	24	Continuous	Ves	Ves	
storage NH [hours]	1	27	Continuous	1 03	105	
Tank aspect ratio D/H [-]	0.2	5	Continuous	Yes	Yes	
Design Turbine Power Pe	500	2500	Continuous	Na	Vac	
[kW]	300	2500	Continuous	NO	res	
Solar Multiple SM [-]	1	8	Continuous	Yes	Yes	
Spacing between collectors	5	25	Continuous	Vec	Vec	
d <sub>spacing</sub> [m]	5	23	Continuous	1 C3	1 00	
Steam generator height H [m]	1	4	Continuous	Yes	Yes	
n. of tubes of the steam	3	20	Integers	Yes	Yes	
generator n <sub>tubes</sub> [-]						
n. of collectors in a string $n_{coll}$ [-	2	8	Integer	Yes	Yes	
]	-	0	megor	1.00	1.00	
Tracking system axis [-]	1 = N-S	2: E-W	Integer	Yes	Yes	
Number of hours of water	2	24		N	V	
storage NH <sub>water</sub> [hours]	Z	24	Continuous	INO	res	

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Table 1. Decision variables overview

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DIRECT COSTS	Characteristic	Cost	Unit	n	S <sub>ref</sub>	Sources
	dimension					
Solar field trough	Mirror surface	357	\$/m <sup>2</sup>	1	-	[43] &
						ENEA
Tank Envelope	External surface	2364	\$/m <sup>2</sup>	0.8	190	[43]
					9	
Fluid, Foundations	Total volume	1131	\$/m <sup>3</sup>	0.82	106	[43] &
and Handling system					0	ENEA
(Tank)						
Steam Generator	Number of tubes	11904	$/(n_{tubes})$	0.78	84	ENEA
Steam turbine	Design electric	473	\$/MW	0.67	25	[39]
	power					
Condenser	Heat transfer	585	\$/m <sup>2</sup>	1	25	[39]
	surface					
Pump, BOP,	Design electric	376	\$/MW	0.8	110	[39]
buildings, Safety	power					
systems(Power						
block)		6				
Water tank	Total volume	660	\$/m <sup>3</sup>	1		[47] &
						ENEA
INDIRECT COSTS						
Engineering,	<b>O</b>	11.8	% of direct	-	-	[43]
Procurement,			capital cost			
Construction &	$\sim$					
Project costs						
C						
SERVICES and						
O&M						
Grounds/house	Ground surface	0.04	\$/m <sup>2</sup>	-	-	Elaborated
keeping						from [43]
Mirror washing	Mirror surface	0.41	\$/m <sup>2</sup>	-	-	[43]
Water Treatment	Design electric	1318	\$/MW	-	-	[43]
	power					

Materials	-	3.2	% of capital	-	-	[43]
Maintenance			cost			
TES and Power	Design electric	5564	\$/(MW y)	-	-	Elaborated
block Labor	power					from [43]
Solar field Labor	Mirror surface	2.07	$/(m^2 y)$	-	-	Elaborated
						from [43]
					$\mathcal{O}$	
OTHER COSTS					$\sim$	
Contingencies	-	10	% of total	-	-	[42]
			project cost			
Decommissioning	-	5	% of total		-	[42]
			project cost			
Interest rate	-	7	%	-	-	[39]
Insurance rate	-	2	%	-	-	[39]

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Table 2. Summary of economic model data

1.16
4.12
15.23
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2.58
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N-S
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Table 3. Optimal design in the basic configuration

Total power generation [MWh]	3864
Total heat sold [MWh]	10206
Total heat wasted [MWh]	0
Total auxiliaries [MWh]	117
Capacity Factor [%]	38.64
Power block efficiency [%]	25.41
Optical efficiency solar field [%]	48.55
Thermal efficiency solar field [%]	80.68
System gross electrical efficiency [%]	9.72
System net electrical efficiency [%]	9.45
System total efficiency [%]	38.44

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#### Table 4. Energy flows and efficiencies

Exergetic Efficiency of concentrating device[%]	48.56
Exergetic Efficiency Receiver [%]	47.87
Exergetic Efficiency Storage [%]	83.48
Exergetic Efficiency Turbine [%]	87.50
Exergetic Efficiency Condenser [%]	84.63
Exergetic efficiency System [%]	16.08

Table 5 Calculated second-law efficiencies of the main components

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	Point 1	Point 2
MS storage size [hours]	1.9	3.2
Design Turbine Power [kW]	2294	2366
Solar multiple [-]	1.87	2.1
Height Steam Generator [m]	2.69	2.48
Number of tubes steam generator [-]	12	12
Water storage size [hours]	8.89	18.91
Tracking	E-W	E-W
Direct investment costs [M\$]	16.39	19,95
LEC [\$/MWh]	296	344
Revenues from heat [k\$]	740	794
Revenues from incentive [k\$]	2006	2200
Revenues from market [k\$]	622	738
Simple Pay-Back Time [years]	6.0	6.8
Capacity Factor power block [%]	24.54	24.94
Thermal Load Capacity Factor [%]	79.22	86.46
Total Power [GWh]	4.71	5.16
Electrical net efficiency [%]	9.32	8.96
Total efficiency [%]	39.62	37.52

Table 6. Pareto-point analysis