Renewable Energy 163 (2021) 530–543

Contents lists available at ScienceDirect

Renewable Energy

journal homepage: www.elsevier.com/locate/renene

Pressure drops, heat transfer coefficient, costs and power block design for direct storage parabolic trough power plants running molten salts

Telma Lopes [a, b], Thomas Fasquelle [c, *], Hugo G. Silva [d]

[a] EMIÁTOMO-Projetos e Manutenção Industrial,Lda, Edifício Zils, Monte Feio, Escritório 515, 7520-064, Sines, Portugal
[b] Renewable Energy Chair, University of Évora, Portugal
[c] Aix Marseille Univ, CNRS, IUSTI, Marseille, France
[d] Department of Physics and Earth Sciences Institute, School of Sciences and Technology, University of Évora, Rua Romão Ramalho, 59, 7000-671, Évora, Portugal

A R T I C L E I N F O

Article history:
Received 13 March 2020
Received in revised form
9 July 2020
Accepted 22 July 2020

Keywords:
Concentrated solar power
Parabolic troughs
Heat transfer fluid
System advisor model
Molten salts

A B S T R A C T

Direct circulation of molten salts in the solar field of parabolic trough solar power plants may be a possible breakthrough to decrease their levelized cost of electricity. While prototypes are being erected around the world, this study addresses the main concerns and changes that are related to the replacement of thermal oils by molten salts, *i.e.* pressure drops, heat transfer coefficient, anti-freezing solutions, cost and power block design. It combines: 1) an analytical comparison of both technologies with respect to pressure drops and heat transfers; 2) simulations of a 50 MW_e/7.5 h-of-storage power plant, using NREL's SAM software, providing details on the dynamics of the outputs and parasitics. It has been observed the following: 1) pressure drops in the solar field are smaller running molten salts instead of thermal oil, thanks to higher operating temperature ranges; 2) HitecXL molten salt leads to lower electricity consumption than Therminol VP-1 oil and Solar Salt (parasitics); 3) a 6.3% reduction of the levelized cost of electricity when running HitecXL, ~14.80 c€/kWh, instead of Therminol VP-1, ~15.80 c€/kWh; 4) simpler power block designs can be considered for the higher operating temperatures of molten salts, resulting in higher efficiencies and/or cheaper power blocks.

1. Introduction

Due to the population increase and the industrial development that the world has undergone over the last decades, energy consumption has increased exponentially, leading to a rarefaction of the resources and an accelerating global warming. With these changes, it becomes increasingly important to look for other solutions that enable energy sustainability, such as renewable energies. Wind energy and solar photovoltaics currently lead the market with a 284 GW increase of installed wind power capacity and a 256 GW increase of installed solar power worldwide between 2010 and 2016 [1].

Concentrated Solar Power (CSP) is still an expensive technology and it remains at a relatively early stage, with most of the running power plants located in Spain and the United States [2]. It presents the possibility of storing thermal energy cheaply and efficiently, allowing the production of energy not only during the period of solar irradiation, but also during times of greater demand. Moreover, Thermal Energy Storage (TES) has the capability to solve the solar intermittency problem (cloud passage, for example) responsible for possible disturbances in the electrical grid.

Most of the commercial CSP systems installed around the world use parabolic trough (PT) collectors as concentration elements and thermal oils (TO) as heat transfer fluid (HTF) [3] and those having TES, use molten salts (MS) as energy storage fluid, yet many do not [4]. The reason why MS are used as heat storage fluid stems from their high thermal capacity, their lower cost and their thermal stability (no risks of ignition).

The possibility of using MS also as HTF has been the subject of various investigations [5–7], since these have several advantages in relation to, such as the possibility of increasing the output temperature of the solar field to 450–565 °C (compared to ~393 °C with TO). Thus, the Rankine cycle's efficiency is expected to be increased from ~37.6% to slightly more than 40% [6]. Moreover, MS have low vapor pressure at high temperatures, high boiling temperature and

* Corresponding author.
E-mail addresses: telma.lopes@emiatomo.eu (T. Lopes), thomas.fasquelle@univ-amu.fr (T. Fasquelle), hgsilva@uevora.pt (H.G. Silva).

https://doi.org/10.1016/j.renene.2020.07.110
0960-1481/

Nomenclature

Acronyms

ASE	Archimede Solar Energy
CAPEX	CAPital EXpenditures
CF	Capacity Factor
CSP	Concentrated Solar Power
DLR	German Aerospace Centre
HTF	Heat Transfer Fluid
LCOE	Levelized Cost of Electricity
MATS	Multipurpose Applications by Thermodynamic Solar
MS	Molten Salt
MSPT	Molten Salt Parabolic Trough
NREL	National Renewable Energy Laboratory
OPEX	OPerational EXpenditures
TMY	Typical Meteorological Year
TO	Thermal Oil
PT	Parabolic Trough
SAM	System Advisor Model
TES	Thermal Energy Storage
T-S	Temperature-Entropy
VHC	Volumetric Heat Capacity

Greek

Δ	Finite difference
ε	Roughness [m]
ρ	Density [kg.m^{-3}]
μ	Dynamic Viscosity [Pa.s]

Variables

beam	Normal direct solar irradiance
C	Specific heat [J.kg^{-1}.K^{-1}]
C_{n_y}	Total cost for year n_y [€ or $]
D_{rt}	Pipe diameter [m]
E_{n_y}	annual electricity production for year n_y [kWh]
F_X	Variable to nullify for friction factor calculation
h	Convection coefficient [W.m^{-2}.K^{-1}]
h_x	Specific enthalpy at point x [J.kg^{-1}]
k	Thermal conductivity [W.m^{-1}.K^{-1}]
$\dot{m}$	Mass flow [kg.s^{-1}]
n	Factor for Dittus-Boelter equation
n_y	Year number
N	Lifespan of a project [year]
Nu	Nusselt number
Pr	Prandtl number
$Pres$	Atmospheric pressure [Pa]
r	Discount rate
Re	Reynolds number
RH	Relative Humidity
$Rough$	Relative roughness
T	Temperature [K or °C]
T_{CS}	Cold Source temperature [K]
T_{dry}	Ambient dry temperature [°C]
T_{HS}	Hot Source temperature [K]
$VP1$	Therminol VP-1
$\bar{v}$	Mean velocity [m.s^{-1}]
w_{spd}	Wind speed [m.s^{-1}]
$\dot{W}$	Electrical work [W]
X	Variable used to calculate friction factor

relatively high thermal conductivity. Their use as HTF may allow a reduction of the capital expenditures (CAPEX) of the plant, by reducing the physical size of the storage system and by removing the need for a heat exchanger between the HTF and the storage medium. On the other hand, their use would require a careful evaluation of certain aspects such as melting point (freezing issues) as well as corrosiveness and installation cost. Presently, various solar tower power plants already use MS as HTF [8,9] because of the reduced length of piping when compared to linear focusing systems. However, there is some investigation about the possibility to use MS in PT power plants as well:

- In 2002, D. Kearney et at [5] began to investigate the feasibility of utilizing MS as both HTF and TES fluid in a PT solar field to improve system performance and to reduce the levelized cost of electricity (LCOE) of the plant (~17%).
- Since July 2013, the first stand-alone Molten Salt Parabolic Trough (MSPT) demo plant, located close to the Archimede Solar Energy (ASE) manufacturing plant in Massa Martana (Perugia, Italy), is in operation. The ASE MSPT demo plant is composed of a single loop with a collecting surface of ~3600 m^2, equipped with high temperature solar receivers and connected to a molten salt storage system constituted of two 25 m^3 tanks [7].
- More recently (2018), the MATS (Multipurpose Applications by Thermodynamic Solar) power plant was inaugurated in Egypt [10]. It has an electrical power of 1 MW and is based on PT collectors with an area of 10,000 m^2. The system uses a mixture of sodium nitrate and potassium nitrate as HTF, heated from ~290 °C to ~550 °C. The power plant also includes a heat storage system with an embedded steam generator.

To tackle the freezing and corrosion problems, as well as achieving higher operating temperatures, a quite important range of different MS was developed within the past few years [11]. Yet, to prove the feasibility of using MS as HTF, an extensive study and a test of various technologies with different MS are still necessary. For instance, Boukelia et al. [12] compared the use of MS (Solar Salt) and TO as HTF in PT plants. Using the System Advisor Model (SAM) and the EBSILON model, they developed an artificial neural network optimizing both MS and TO configurations, and found a 13% LCOE reduction with MS. However, the study does not give any details about pumping consumption, strategies against freezing and does not assess the performance of other MS, such as HitecXL.

Hence, there is still a lack of answers regarding both MS pressure drops and freezing problems, along with their impact on the overall costs of the plant. Pending the completion of experimental power plants and the operational data they will generate, the present study provides reliable indicators to those answers (Section 2 and Section 3), as well as the implied modifications in the power block designs (Section 4).

This work is part of the High Performance Solar 2 (HPS2) project, which is led by the University of Évora and the German Aerospace Centre (DLR) and integrates a consortium of companies and laboratories. Its final objective is the construction of an experimental 3.6 MW$_{th}$ solar plant at the Évora Molten Salt Platform (EMSP) with PT solar concentrators and with MS as heat transfer and storage fluid. HTF will be YaraMost MS that consists of a ternary mixture based on potassium nitrate (KNO_3), sodium nitrate ($NaNO_3$) and as source of calcium nitrate, Yara NitCal-K.

2. Analytical comparison between thermal oils and molten salts as HTF

2.1. Fluid properties

Relevant properties for an analytical comparison between TO and MS are presented in Table 1. Thermal properties of Therminol VP-1, *i.e.* biphenyl and diphenyl oxide, also commercialized under the name Dowtherm A; Solar Salt, 60% $NaNO_3$/40% KNO_3; HitecXL, $Ca(NO_3)_2$–KNO_3–$NaNO_3$; were estimated from correlations provided by D. Kearney et al. [5], K. Vignarooban et al. [8] and A. Bonk et al. [13].

As illustrated in Table 1, the main advantages associated with the use of MS as HTF instead of TO are their higher operating temperatures, their higher Volumetric Heat Capacity (VHC = $\rho.C_p$) and their very low vapor pressure, inducing no risk of vaporization and therefore no need for fluid pressurization. Higher energy density along with lower cost is a significant advantage when considering the high volumes of materials that are needed for TES (27,000 tons of MS in an Andasol-like power plant, *i.e.* 50 MW_e and 7.5 h of TES). On the other hand, MS have less favorable properties such as a higher viscosity and a higher melting point. TO are more interesting as HTF mainly because they do not present risk of freezing at ambient temperature in the very long piping systems (~90 km of receivers and ~1.5 km of interconnecting piping in an Andasol-like power plant), neither they pose any significant corrosion risks.

2.2. Heat transfer coefficient

The heat transfer coefficient drives the thermal efficiency of the receiver tubes. Furthermore, as receiver tubes collect more solar irradiation from the collector facing side, the latter tends to reach higher temperatures. Thus, a low heat transfer coefficient results in a non-uniform thermal expansion of the receiver tubes, leading to an undesirable bending and possibly even breaking the protective glass cover [14,15].

Here, heat transfer coefficient between fluid and pipe are estimated with the Dittus-Boelter equation **(1)**, which is valid for most fluids (0.7 < Pr < 160) in a fully turbulent regime (Re > 10,000) inside a tube [16],

$$Nu = 0.023 \cdot Re^{0.8} \cdot Pr^{n}, \tag{1}$$

with $n = 0.4$ (heated fluid). Since the Nusselt number, *Nu*, involves both the Prandtl number, *Pr*, and the Reynolds number, *Re*, all thermal properties have an impact on the heat transfer rate. Hence, a comparison between the different fluids can be performed by developing this expression, resulting in Equation (2), assuming identical absorbed power and the mass flow depending on the fluid's specific heat:

$$\frac{h_{MS}}{h_{TO}} = \left(\frac{C_{TO}}{C_{MS}}\right)^{0.4} \cdot \left(\frac{\mu_{TO}}{\mu_{MS}}\right)^{0.4} \cdot \left(\frac{\Delta T_{TO}}{\Delta T_{MS}}\right)^{0.8} \cdot \left(\frac{k_{MS}}{k_{TO}}\right)^{0.6}, \tag{2}$$

here, h_{MS} and h_{TO} are convection heat transfer coefficients, C_{MS} and C_{TO} are specific heats, μ_{MS} and μ_{TO} are viscosities, k_{MS} and k_{TO} are thermal conductivities, and ΔT_{MS} and ΔT_{TO} are temperature differences, respectively for MS and TO. Equation (2) shows that the temperature differences that are experienced by the HTF and their conductivities have larger impacts than their dynamic viscosities or their specific heats. Density does not impact the convection heat transfer for a given power.

The comparison between HitecXL and Therminol VP-1, considering a low temperature of 290 °C for both fluids and different high temperatures (~390 °C for the TO, ~500 °C for HitecXL), gives a convection coefficient 30% lower for MS than TO:

$$\frac{h_{HitecXL}}{h_{VP-1}} = \left(\frac{2454}{1400}\right)^{0.4} \cdot \left(\frac{0.000177}{0.00253}\right)^{0.4} \cdot \left(\frac{100}{210}\right)^{0.8} \cdot \left(\frac{0.52}{0.086}\right)^{0.6} = 0.70 \tag{3}$$

To better assess those differences, details of the calculation were given for the case of a 600 m receiver line, such as those found in conventional power plants like Andasol (Table 2). A useful thermal power of ~2.5 MW_{th} was estimated for those calculations.

In Table 2 the symbol $\dot{m}$ stands for fluid mass flow and $\bar{v}$ for the mean fluid velocity. For the same output powers, mass flows with MS are lower than with TO, 68.5 kg s^{-1} *vs.* 10.2 kg s^{-1}, respectively, their lower specific heat being compensated by higher operating ΔT. In addition, higher density of MS results in lower velocities, 0.961.3 m s^{-1} *vs.* 3.9 m s^{-1}, respectively, *i.e.*, lower Reynolds number and therefore lower Nusselt numbers (Prandtl numbers being similar). Nevertheless, the six-times higher thermal conductivity of the MS enables compensating those smaller values, resulting in lower, but relatively close heat transfer coefficient when compared to Therminol VP-1, 2462–2760 W m^2 K^1 *vs.* 3930 W m^2 K^1, respectively. However, for such high values of *h*, this difference should not have a significant impact on the operation of the plant. Another issue would be the possible bending of the absorber tubes. Their manufacturers typically claim that a *Re* > 20,000 is sufficient to homogenize the temperature around them, avoiding problems

Table 2
Heat transfer coefficient (h) of the HTF and respective intermediate calculations.

	Therminol VP-1	Solar Salt	HitecXL
ΔT [K]	100	275	210
$\dot{m}$ [kg. s^{-1}]	10.2	6.0	8.5
$\bar{v}$ [m.s^{-1}]	3.9	0.96	1.30
Re	1,110,346	63,773	64,839
Pr	5.1	5.3	6.8
Nu	3016	312	350
h [W.m^{-2}. K^{-1}]	3930	2462	2760

Table 1
Values of thermal properties at 350 °C for thermal oil and 400 °C for molten salts.

	Therminol VP-1 (*at 350 °C)	Solar Salt (*at 400 °C)	HitecXL (*at 400 °C)
Melting point [°C]	13	220	120
T_{max} [°C]	400	600	500
Density*, ρ [kg.m^{-3}]	761	1839	1913
Dynamic viscosity*, μ [Pa.s]	0.000177	0.00182	0.00253
Specific heat*, C [J.kg^{-1}.K^{-1}]	2454	1511	1400
Thermal conductivity*, k [W.m^{-1}.K^{-1}]	0.086	0.52	0.52
VHC* [MJ.m $^{-3}$.K^{-1}]	1.87	2.78	2.68
Vapor pressure [bar]	7	–	–

due to non-uniform expansion. As the obtained *Re* values are much higher than the required ones (the lowest being 63,773 for Solar Salt), the use of MS as HTF should not present any risk to the receivers or the proper operation of the system.

2.3. Pressure drops

Since TO have lower density and viscosity than MS, they are expected to entail lower pressure drops in the solar field when subjected to the same conditions. With the intention of better assessing those differences, the receiver line power is set to 2.5 MW_{th}, similarly to what was done for calculations of the heat transfer coefficient. Considering the velocity at which each fluid circulates in the receiver tubes to achieve this power with their inlet and outlet temperatures, pressure drops in the solar field are obtained with the following Equation (4):

$$\Delta p_{pipe} = hl_{pm} \cdot \rho \cdot g \cdot L_{pipe}, \tag{4}$$

where ρ represents the fluid density, *g* the gravitational acceleration, L_{pipe} the pipe length and hl_{pm} the head loss per meter given by Equation (5):

$$hl_{pm} = \frac{fr \cdot \bar{v}^2}{2 \cdot D_{rt} \cdot g}. \tag{5}$$

To obtain the value of hl_{pm}, it is necessary to know the friction factor *fr*, the mean velocity of the fluid inside the tube, $\bar{v}$, and the diameter of the receiver tube, D_{rt}. The friction factor is given by Equation (6),

$$fr = \frac{1}{X^2}, \tag{6}$$

where *X* is obtained with Equation (7) and with $F_X = 0$:

$$F_X = X + 2 \cdot \log_{10}\left[\frac{Rough}{3.7} + 2.51\frac{X}{Re}\right], \tag{7}$$

with *Rough* being the relative roughness of the tube ($Rough = \varepsilon/D_{rt}$) [17]. In this case, the value of roughness for commercial new steel receiver tube is used, $\varepsilon = 0.046$ mm, the internal diameter is $D_{rt} = 0.066$ m, and the Reynolds values previously calculated for the same conditions are applied. Considering the different ΔT that are allowed for each fluid, 100 °C for the TO, 210 °C for HitecXL and 275 °C for Solar Salt, the obtained results are gathered in Table 3.

Contrary to what was expected, the pressure drops in the solar field are smaller when using MS as HTF, 264 Pa m^{-1} with Solar Salt and 520 Pa m^{-1} with HitecXL, against 1636 Pa m^{-1} with TO. Although TO are less dense and less viscous, MS have a larger operating ΔT which reduces the flow velocity of the fluid in the receiver tube, resulting in the decrease of pressure drops. This is an important result as it highlights the fact that a significant reduction in pumping electrical power consumption can be achieved by using MS rather than TO. In fact, based on the Δp estimation such reductions would be around 84% when using Solar Salt and 68% when using HitecXL instead of TO. These would eventually compensate the extra energy consumption for MS freezing protection or even reduce the overall consumption. To validate such assumption, numerical simulation results for an Andasol-like power plant are presented in the following section.

Table 3
Pressure drops values per metre [Pa.m^{-1}] obtained for each HTF in their operating conditions.

	Therminol VP-1	Solar Salt	HitecXL
v [m.s^{-1}]	3.9	0.96	1.30
X	7.34	6.95	6.86
fr	0.019	0.021	0.021
h [m/m]	0.219	0.015	0.028
Δp [Pa.m^{-1}]	1636	264	520

3. Techno economic analysis with SAM

The SAM software was used to simulate the operation of a CSP power plant using different fluids as HTF. SAM was developed by the National Renewable Energy Laboratory (NREL) with the aim of modeling a range of renewable energies, including solar thermal PTs. SAM uses an hourly performance model to estimate a power system's total annual output, as well as a cost and financial model [18,19]. In the present analysis, SAM was used to perform an annual simulation of a 50 MW_e power plant with 7.5 h of TES, with local meteorological data from Évora, Portugal, and inputs corresponding to the characteristics of Andasol 3 power plant [20]. A brief description of the parameters used in SAM to simulate the power plant are listed in Appendix 1.

3.1. Meteorological data

The meteorological dataset, namely, normal direct solar irradiance - beam, ambient dry temperature - T_{dry}, relative humidity - RH, atmospheric pressure - Pres and wind speed - w_{spd}, used in this work was obtained from two nearby meteorological stations in Évora, Portugal: one from the Institute of Earth Sciences (38.567686N, 7.91172W) and another from Portuguese Institute for the Sea and the Atmosphere (38.53654N, 7.88795W). Details on the measurement devices can be found on references [21,22]. The meteorological data that were used are the result of a compilation between years 2016, 2017 and 2018. A more accurate analysis would consider typical meteorological years (TMY), instead of individual years, in order to be more representative of the long period of operation, but SAM's TMY for Évora seem not to be appropriate, and not enough years of data were available to calculate TMY.

3.2. Annual production of energy and capacity factor

In the simulation model that was used in SAM, *CSP parabolic trough (physical)*, the annual production of energy represents the amount of electricity that the plant produces during a year and the value is given in MW_eh. From this value, the capacity factor of the plant can be calculated. Capacity Factor, CF, is a comparative measure of the amount of energy produced by a power plant with the maximum energy that could be produced if it was operating at nominal power during the same period, *cf.* **Equation (8)**:

$$CF = \frac{Annual\ \ Energy\ \ Produced}{Maximum\ \ energy\ \ that\ \ could\ \ be\ \ produced}. \tag{8}$$

In the present case, the nominal power of the plant is 50 MWe and the maximum energy that could be produced is calculated by multiplying the nominal power by the number of hours in a year, *i.e.* 8760, which gives 438,000 MWeh. Through the CF it is also possible to determine how many equivalent hours the plant would annually operate at nominal power, for each case (see Table 4).

Interestingly, capacity factors are very similar when comparing the results of the simulation for Therminol VP-1 and HitecXL as HTF, respectively 43.6% and 44.3%, while yields are lower for Solar Salt, 37.6%. These differences will be explained in the following sections.

Table 4
Values of annual production of power energy with each fluid.

	Therminol VP-1	Solar Salt	HitecXL
Annual electricity (produced) [MW_eh]	191,043	164,565	194,169
Capacity factor CF [%]	43.6	37.6	44.3
Number of equivalent hours at nominal power [h]	3819	3294	3881

3.3. Annual thermal power freeze protection

Annual thermal power freeze protection is one of the values provided in the SAM's summary table. It represents the thermal energy required to heat the storage system as well as the solar field, to always ensure the safe temperature range of the HTF. In this way, the amount of thermal energy required to heat the plant with the different HTF is presented in Table 5.

Solar Salt presents the highest needs for freeze protection (~36,468 $MW_{th}h$ in total) because its melting point is very high (220 °C) and thus it requires maintaining the tanks at relatively high temperatures - above 250 °C. HitecXL also needs solar field freeze protection to avoid solidification, but the amount of energy is much lower (~3178 MWh). In the tanks, freeze protection is almost insignificant because tanks have greater thermal inertia. Fig. 1 allows to analyze in more detail the needs for solar field heating according to the HTF used.

With Solar Salt as HTF, up to 8.72 MW_{th} are required during all the nights to avoid freezing, while with HitecXL, a maximum value of 3.88 MW_{th} is required only during moments of low solar availability (typically winter). Therminol VP-1 implies the use of electrical heat tracing in very particular cases and with very low power (<0.01 MW_{th}). Thus, it can be clearly seen that using MS as HTF implies the substantial consumption of energy to maintain the HTF liquid, which is an important drawback of MS when compared to. However, using a low melting-point MS such as HitecXL (instead of Solar Salt) enables to divide the annual freeze protection energy by a factor of 12.

3.4. Pumping consumption

In a conventional PT power plant, circulation of HTF and storage fluid are performed with different pumping system: one pump circulates the HTF in the solar field, one circulates the HTF through the heat exchanger of the TES system during discharge and two pumps circulate the storage fluid from one tank to the other (one pump in each tank, plus back-up pumps). In a similar plant with MS as HTF and storage fluid (direct storage), only two pumps are used: one in the cold tank enables the circulation through the solar field, and one in the hot tank circulates the HTF to the steam generator. In SAM's results, electricity consumption is divided between solar field HTF pump and the others (the three pumps for indirect storage, and the hot tank pump for direct storage). For the present simulation, electricity consumptions of those circulation systems are gathered in Table 6.

It can be verified that TO is the HTF that leads to the highest electricity consumption, with a total value of ~5.2 GW_eh *vs.* ~2.2–2.5 GW_eh for MS. MS have lower pump consumptions thanks to their higher operating temperatures, for the same low operating temperature, ~290 °C, allowing circulation at lower flow rates and thus resulting in lower pressure drops. This consumption varies along the year, as it is illustrated in Fig. 2 and Fig. 3.

Table 5
Quantity of thermal energy needed to heat the storage system and the solar field with the different HTF.

	Therminol VP-1 (TES with HitecXL)	Solar Salt	HitecXL
TES freeze protection [$MW_{th}h$]	3.93	16.23	0.14
Solar field freeze protection [$MW_{th}h$]	0	36,452.05	3178.25
Annual total freeze protection [$MW_{th}h$]	3.93	36,468.28	3178.39

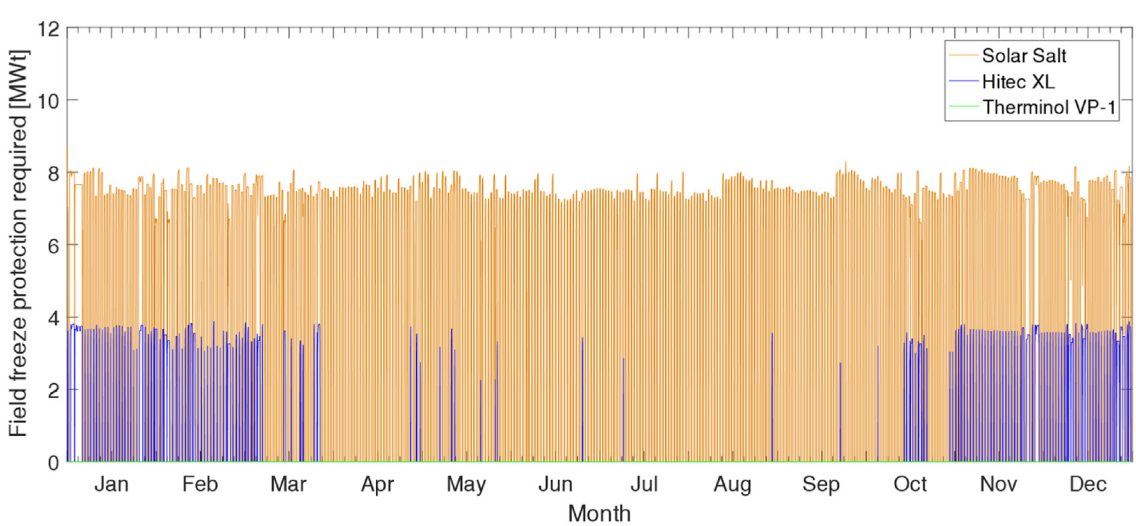

Fig. 1. Annual field freeze protection required in MW_{th}. From highest to lowest values: Solar Salt, HitecXL and Therminol VP-1.

Table 6
Values of pumping consumption.

	Therminol VP-1	Solar Salt	HitecXL
Parasitic power TES and Cycle HTF pump [MW_eh]	2241.41	1765.83	1760.29
Parasitic power solar field HTF pump [MW_eh]	2994.89	411.83	697.06
Total [MW_eh]	5236.30	2177.66	2457.35

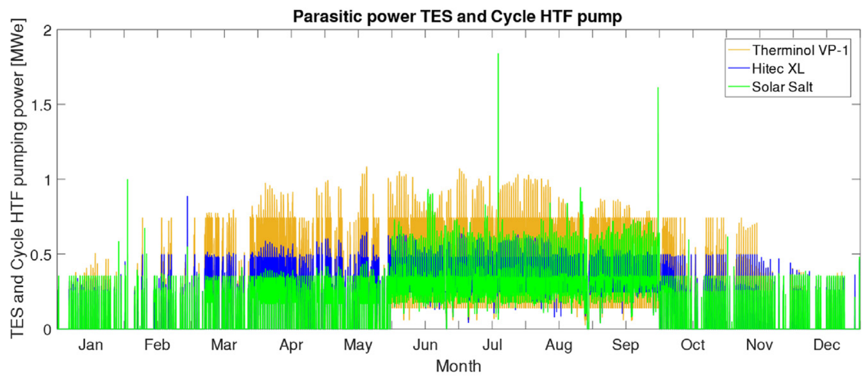

Fig. 2. Parasitic power - TES and Cycle HTF pump.

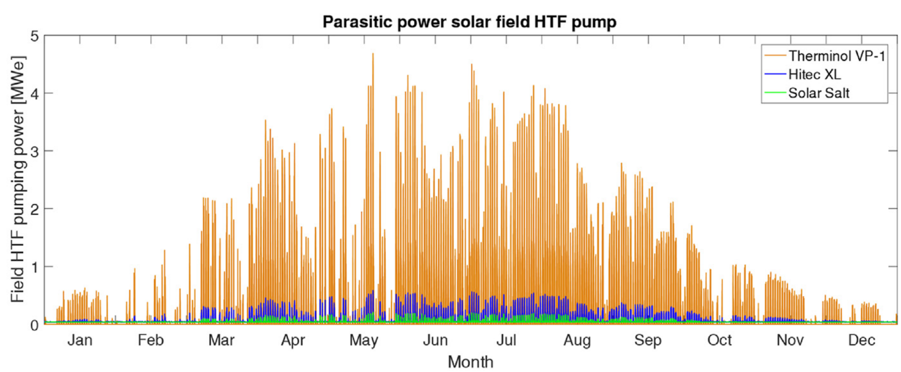

Fig. 3. Parasitic power - solar field pump.

During months with the highest solar energy availability, typically summer, HTF is circulated with the highest mass flows and solar field pumping power for TO reaches up to ~4.68 MW_e for the solar field pump and ~1.01 MW_e for the other pumps. With MS, pumping consumption is considerably reduced, with maximum values around 0.21 MW_e and 0.59 MW_e, respectively. Of the two salts, Solar Salt is the one that needs less pumping energy, mainly because its maximum operating temperature is higher than that of HitecXL and its viscosity is lower.

3.5. *Annual electricity consumption, night operation strategy and cost estimate*

The annual electricity consumption comprises the annual thermal freeze protection (considering an electrical efficiency of 1), the pumping consumption and other consumers, such as auxiliary heater operation, condenser operation (dry cooling), collector drive motors and other parasitic consumptions that are all gathered in one block in Table 7.

The global analysis shows that there is a great discrepancy between electricity consumption with Solar Salt, *i.e.* 49,007 MW_eh, and those of the two other HTF that are quite akin: 16,807 MWh for HitecXL and 16,921 MWh for Therminol VP-1. However, this is the worst-case scenario, in which no specific operation strategy is considered. For example, the circulation of MS from the cold tank through the piping during the night is not scrutinized, and it can significantly reduce the amount of electricity spent on solar field heating.

Indeed, analyzing the meteorological data considered in this work, and using the assumption that a day with insufficient solar resource is a day during which the plant produces less energy than 1 h of production at nominal power (*i.e.* ~380 $MW_{th}h$), 51 days a year with insufficient solar resource are obtained, including:

- 22 periods with only 1 day (2 nights).
- 8 periods with 2 consecutive days (3 nights).
- 3 periods with 3 consecutive days (4 nights).
- 1 period with 4 consecutive days (5 nights).

The thermal inertia capacity of the cold tank is the energy that can be lost by the tank without any risk of salt solidification. For Solar Salt, it is the energy lost by the fluid when cooling down from 290 °C to its minimum safety temperature, 250 °C, resulting in 139.32 $MW_{th}h$. For HitecXL, it is the energy contained by the fluid between 290 °C and 170 °C, *i.e.* 547.34 $MW_{th}h$. Considering total heat losses of 8.72 MW_{th} with Solar Salt (worst case scenario), it

Table 7
Annual electricity consumption counting solar field heating.

	Therminol VP-1	Solar Salt	HitecXL
Thermal freeze protection [MW_eh]	3.93	36,468.28	3178.39
Pumping [MW_eh]	5236.30	2177.66	2457.35
Other parasitics (condenser, collector drive, etc.) [MW_eh]	11,681.26	10,361.42	11,171.01
Total annual electricity consumption [MW_eh]	16,921.49	49,007.36	16,806.75

Table 8
Estimation annual electricity costs with each HTF.

	Therminol VP-1	Solar Salt	HitecXL
Base case annual electricity costs [€]	3,248,926	9,409,413	3,226,896
Cold tank to maintain the system heated during periods with no solar radiation [€]	3,248,926	4,229,076	2,616,645

means that the tank can maintain the system heated at least 16 h since heat losses will decrease with the decreasing temperature. In the case of HitecXL, considering total heat losses of 3.88 MW_{th}, the tank can maintain the system heated during ~141 h. It means that the use of the heat tracing system can be totally avoided, this justifies the interest for this MS to be used as HTF. In fact, considering the annual consumptions and that electricity has an estimated cost of 0.192 €/kWh, a guess of annual electricity costs for the different HTF and the different scenarios is illustrated in Table 8.

Even without considering any further operating strategy to reduce electricity consumption for solar field heating, HitecXL leads to lower electricity consumption than Therminol VP1. As HitecXL has a relatively low melting point compared to Solar Salt, the electricity consumption saved in the pumps is greater than the needed for heating (when compared to). On the other hand, the total electrical consumption when using Solar Salt is approximately 3 times higher than the consumption with the other two fluids which represents a significant increase in operational expenditures (OPEX) of such a plant. Complementary, with the operation strategy that was previously discussed, electricity spending would be avoided for the case of HitecXL and significantly lowered for the case of Solar Salt, however, net production would be lowered in both cases. One may also note that, in terms of exergy, it is much more interesting to use thermal power to heat the piping systems than electricity.

3.6. LCOE calculation

The LCOE (Levelized Cost of Energy) of a direct storage plant with MS is compared with the one of a 50 MW_e conventional plant, based on reference costs (CAPEX and OPEX) from literature [5,23,24], and some calculations that are based on information obtained from HPS2 partners. The costs of HTF correspond to those from Ref. [5]. The energy spent in the pumping of the fluid in the solar field and the value of heat losses are obtained through SAM's results. The power block efficiency is calculated using Equation (9) and assuming an exergy efficiency of 0.7 for both cases, see Equation (10).

$$\eta_{PB} = \eta_{ex} \cdot \eta_{Carnot}, \tag{9}$$

with,

$$\eta_{ex} = 0.7; \eta_{Carnot} = 1 - \frac{T_{CS}}{T_{HS}} \tag{10}$$

Where T_{CS} corresponds to the cold source temperature (ambient) and T_{HS} to the hot source temperature (in K). All the cost and performance details are given in Table 9.

Two different methods were used to calculate LCOE of the two plant types. The first one using SAM, in the *CSP parabolic trough model - LCOE calculator*, and a second method using algebraic calculations. For both cases, an operating period of 25 years was considered [25].

First method: this method uses SAM to calculate the LCOE, inserting CAPEX and OPEX that are reported in Table 9. The obtained results are presented in Table 10.

It can be verified that despite the extra anti-freezing protection requirements for the use of MS as HTF (namely electrical heat tracing, impedance heating and pre-insulated piping supports), the CAPEX value of this type of plant is lower. This is due to the savings in:

- synthetic oil acquisition;
- storage tanks size and quantity of MS, thanks to larger operating temperature as compared with TO;
- number of pumps and heat exchangers, because there is only one working fluid when MS is circulated in both the solar field and TES and, thus, a single system pump (plus a back-up one).

Therefore, as previously shown in Table 9, CAPEX and OPEX are lower for HitecXL than for TO. Besides, thanks to the higher operating temperature and, hence, higher power block efficiency, a greater annual electricity production is found with HitecXL, leading to a reduction of the LCOE by about 6.3% when compared to the case with TO.

Second method: LCOE calculation are made using the values in Table 9, in Euros, considering a discount rate of 2% (value of the Portuguese rate, the electricity production values obtained through the SAM's simulation for Évora and considering a decrease in the plant efficiency along time (0.8%/year), due to ageing components. In these calculations the value of the energy consumed to heat the solar field when using MS is discounted from the total value of produced energy. In addition, the degradation of the TO that occurs in conventional plants is also taken into account [26]. As such, the cost of oil exchanges every 5 years has been included in the OPEX. Considering Equation (11) to calculate the LCOE value:

$$LCOE = \frac{\sum_{n_y=0}^{N} C_{n_y} (1+r)^{-n_y}}{\sum_{n_y=1}^{N} E_{n_y} (1+r)^{-n_y}}, \tag{11}$$

whereby.

- C_{n_y} − total cost, CAPEX and/or OPEX, for year n_y;
- E_{n_y} − annual electricity production (kWh) for year n_y;

Table 9
Costs of conventional power plant and direct storage with molten salts plant.

	CONVENTIONAL POWER PLANT	DIRECT STORAGE WITH HITECXL MOLTEN SALTS
Capital Expenditure (CAPEX)	**366.8 M$|328.7 M€**	**349.5 M$|312.5 M€**
Labour: Site and solar field	62.4 M$|55.9 M€	62.4 M$|55.9 M€
Solar Field	103.9 M$|93.1 M€	103.9 M$|93.1 M€
Synthetic oil	7.8 M$|7.0 M€	-
Molten Salts[1]	26.4 M$|23.7 M€	16.1 M$|14.4 M€
Piping&Insulation	11.4 M$|10.2 M€	11.4 M$|10.2 M€
Pre-insulated piping supports	-	0.20 M$|0.18 M€
Heat tracing	-	13.4 M$|12.0M€
Pumps	Oil: 3.0 M$\|2.7 M€ Salts: 1.6 M$|1.4 M€	Salts: 2.5 M$|2.2 M€
Storage tanks (with insulation and foundation)	9.6 M$|8.6 M€	5.4 M$|4.8 M€
Oil-to-salt heat exchanger	5.1 M$|4.6 M€	-
Electronics, controls, electrical and balance of system	12.6 M$|11.3 M€	12.6 M$|11.3 M€
Power block	20.8 M$|18.6 M€	20.8 M$|18.6 M€
Balance of plant and grid connection	31.2 M$|28.0 M€	31.2 M$|28.0 M€
Others*	71.0 M$|63.6 M€	71.0 M$|63.6 M€
Operational Expenditure (OPEX)	**3.57 M$|3.21 M€**	**3.30 M$|2.97 M€**
Pumping	0.57 M$|0.51 M €	0.30 M$|0.27 M€
Temperature maintain	0 M$|0 M€	0 M$|0 M€**
Others	3.00 M$ |2.70 M€	3.00 M$|2.70 M€
Production		
Power block efficiency	39.1%	43.5%
Heat losses	3.94 MWh_{th}	3,178 MWh_{th}
Heat exchanger efficiency	95.3%	-

[1]In this value are included the cost of salts necessary for the storage in the case of conventional plant and for the storage and HTF in the case of direct storage with molten salts plant.

Table 10
LCOE values obtained through the simulations in SAM.

	CAPEX [M$]/[M€]	OPEX [M$]/[M€]	Annual electricity (produced) [MW_eh]	LCOE [c$/kWh]/[c€/kWh][a]
Conventional CSP power plant	366.80/328.7	3.57/3.21	191,043	17.69/15.80
Direct storage with molten salts (HitecXL)	349.48/312.47	3.30/2.97	194,169	16.57/14.80

[a] Conversion rate: 1 EUR = 1,11844 USD|1 USD = 0,894103 EUR.

- N – lifespan;
- r – discount rate.

The values obtained through the calculation are presented in Table 11.

A reduction of 11.6% in the LCOE value is estimated using HitecXL instead of Therminol VP-1 as HTF. This reduction is lower than the value envisaged by Kearney et al. [5], 17.6%, because of all the extra costs that are considered in the present study (pre-insulated piping supports, heat tracing, etc.) and could be even lower because of more expensive (currently ~25%) high temperature, corrosion resistant receivers.

Table 11
LCOE values obtained through the second approach.

	CAPEX [M$]/[M€]	OPEX [M$]/[M€]	Annual electricity (produced) [MW_eh]	LCOE [c$/kWh]/[c€/kWh]
Conventional power plant	366.80/328.7	3.57/3.21	191,043	14.54/13.00
Direct storage with HitecXL	349.48/312.47	3.30/2.97	190,526	12.85/11.49

With Solar Salt, LCOE estimated with both methods are higher than with TO, 16.02 c€/kWh - 16.98 c€/kWh, because of the substantial needs of electricity to maintain the salt above its minimum safe temperature (see Appendix 6).

One may note that all LCOE values should be lower since a large part of the costs presented in Table 9 was obtained from references already a few years old [5,23,24], from which data is available. In fact, LCOE of current project in Middle East and Morocco are about 7–10 c$/kWh, however details are not available. Nevertheless, it is expected that a comparison between the two fluids with up-dated costs would certainly show the same potential cost reduction running HitecXL as HTF.

4. Power block design with higher T_{HS} and greater ΔT

As MS allow to reach higher T_{HS}, the corresponding power cycles inherently have higher theoretical efficiencies, *cf.* Equation (10). Considering a cold source temperature T_{CS} of 20 °C (ambient), the maximum efficiencies that the cycles can achieve are 62% with HitecXL (T_{HS} = 500 °C) and 65% with Solar Salt (T_{HS} = 565 °C), *vs.* 56% for TO (T_{HS} = 400 °C). However, the actual machines, that follow the Rankine cycle, are hardly close to the ideal ones, reaching ~70% exergy efficiency, *cf.* Equation (9), at the cost of high complexity and CAPEX. In the case of the plants working at 400 °C, this complexity is required to counterbalance the relatively low Carnot efficiency. In the case of MS power plants working at T_{HS} > 500 °C, it might be interesting to create simpler and consequently more cost-effective thermodynamic cycles while attaining similar performances to those working at 400 °C. Concomitantly, this section presents an assessment of power cycle efficiencies that can be expected for various configurations. The results are compared to the efficiency of the cycle that is used in commercial 50 MW_e power plants, like Andasol 3, as described by Ascensión Piquer et al. [27] and using a similar methodology.

4.1. Power cycle of a molten salt plant with direct storage

With the purpose of emphasizing the relation between the complexity of the cycle and its efficiency, the MS plant power block efficiency was assessed for: 1) basic cycle; 2) cycle with reheating; 3) cycle with various bleedings; 4) a complete cycle with reheating and six bleedings.

Several assumptions were made to perform such assessment:

- Water is compressed up to a pressure of 150 bars in the case of HitecXL and 165 bars in the case of Solar Salt.
- In the steam generator, water is vaporized with a 15 bars pressure drop, then superheated up to 490 °C (HitecXL) or 555 °C (Solar Salt), with 10 K pinch between MS and steam.
- Superheated high pressure steam is expanded in a multi-stage turbine with one reheating (pressure drops: 2 bars) and six bleedings. The different bleedings are used to preheat the water through opened and closed feedwater tanks. Turbine is adiabatic and has an isentropic efficiency of $\eta_T = 0.87$. The steam quality at the turbine outlet must be higher than 95% to avoid corrosion and/or cavitation.
- Steam is condensed with a low pressure of 0.06 bars and a temperature of 36 °C, then compressed again.
- The mass flows are determined according to the system conditions and from energy balance conditions.

The values of pressure, temperature, mass flow and enthalpy at each point marked on Temperature-Entropy (T-S) diagram of the cycles, Fig. 4 and Fig. 5, are given in tables available on Appendixes 2 to 5. The cycle efficiencies are calculated from the output work of the turbine stage(s) $\dot{W}_{turbine,1}$ and $\dot{W}_{turbine,2}$, the electrical consumption of the pump(s) $\dot{W}_{pump,1}$ and $\dot{W}_{pump,2}$ and the input heat(s) of the steam generator $\dot{Q}_{heat,1}$ (main heating) and $\dot{Q}_{heat,2}$ (reheating), using Equation (12).

$$\eta_{cycle} = \frac{\dot{W}_{turbine,1} + \dot{W}_{turbine,2} - \dot{W}_{pump,1} - \dot{W}_{pump,2}}{\dot{Q}_{heat,1} + \dot{Q}_{heat,2}} \quad (12)$$

4.2. Cycle efficiency with reheating and one bleeding at 290 °C (Solar Salt) or 200 °C (HitecXL)

The first cycle that is considered is a relatively simple one, with the main objective of preventing salt solidification within the steam generator, thanks to one steam bleeding. In the case of Solar Salt, a steam bleeding at 74.36 bars results on a water saturation temperature of 290 °C which allows to heat liquid water to that temperature before entering the steam generator. In the cycle with a bleeding at 290 °C, represented in Fig. 4 by a green line, a mixer is used at point 7. Its use allows a heat exchange with 100% efficiency, although it implies adding a new pump (from point 7 to point 9).

The bleeding not only solves the problem of high melting point of salts, raising the water temperature from 37.95 °C to 290 °C, but also increases the cycle efficiency from 39.86% to 41.67%.

For the second MS, HitecXL, since the safety temperature is 170 °C, it is possible to use a cycle with a single bleeding at 200 °C (15.54 bars) to prevent solidification. In this case, a power block efficiency of 41.04% is found.

4.3. Complete cycle with HitecXL

As previously mentioned, bleedings are normally used to improve cycle efficiency through increases of water temperature before entering the steam generator. In this way, the complete cycle of a conventional plant (with reheating and six bleedings), adapted to the pressure and temperature conditions of a plant using MS (namely HitecXL) as HTF, is illustrated in Fig. 5.

For this complete cycle, the calculation details of the power block efficiency are given by Equation (12) and Equations (13)–(18).

1) The work carried out by the high-pressure turbine (turbine 1) and the low-pressure turbine (turbine 2).

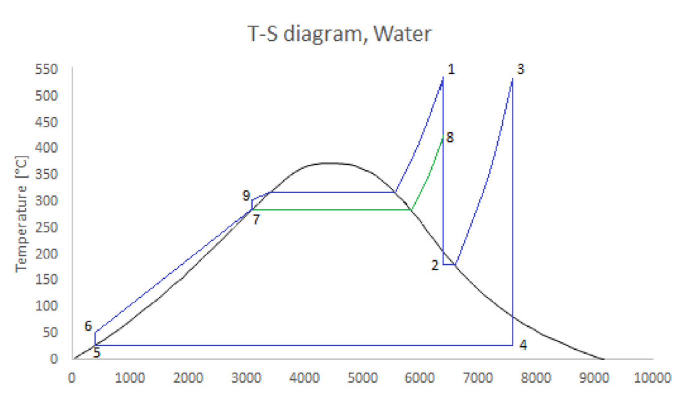

Fig. 4. Rankine cycle with reheating and one bleeding at 290 °C. See Appendix 2 for data and details.

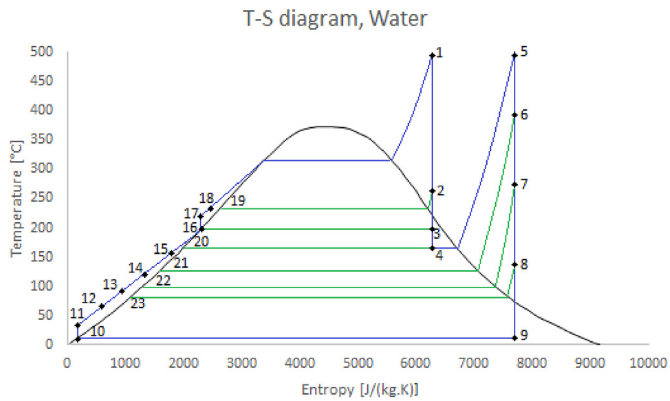

Fig. 5. Complete cycle of direct storage with HitecXL molten salt plant. See Appendix 4 for data and details.

$$\dot{W}_{turbine,1} = \dot{m}_{1\to2}(h_1 - h_2) + \dot{m}_{2\to3}(h_2 - h_3) + \dot{m}_{3\to4}(h_3 - h_4) = 21,669kW, \quad (13)$$

$$\dot{W}_{turbine,2} = \dot{m}_{5\to6}(h_5 - h_6) + \dot{m}_{6\to7}(h_6 - h_7) + \dot{m}_{7\to8}(h_7 - h_8) + \dot{m}_{8\to9}(h_8 - h_9) = 28,331kW. \quad (14)$$

2) The work done by the two pumps (pump 1 and pump 2).

$$\dot{W}_{pump,1} = \dot{m}_{10\to11}(h_{11} - h_{10}) = 53kW, \quad (15)$$

$$\dot{W}_{pump,2} = \dot{m}_{16\to17}(h_{17} - h_{16}) = 479kW. \quad (16)$$

3) And the thermal power that is needed for heating ($\dot{Q}_{heat,1}$) and reheating ($\dot{Q}_{heat,2}$).

$$\dot{Q}_{heat,1} = \dot{m}_{18\to1}(h_1 - h_{18}) = 93,439kW, \quad (17)$$

$$\dot{Q}_{heat,2} = \dot{m}_{4\to5}(h_5 - h_4) = 22,880kW. \quad (18)$$

The obtained efficiency value, 42.53%, is much higher than the one for the complete cycle of a conventional plant, ~39.78% [27]. This increase represents a significant value in the annual production of a plant. Similar procedure is performed for a complete cycle of a Solar Salt plant and all the results are presented in the next section.

4.4. Comparative analysis between the three cycles

In order to visualize with better strictness the differences between the various cycles of the two types of plants with the three different fluids, Table 12 illustrates the efficiencies of all the different power cycles that were considered, calculated with Equation (12). Corresponding exergy efficiencies can be found in Appendix 7.

Results of Table 12 can be analysed in two ways. The first is that the use of MS as HTF allows the achievement of simpler/cheaper power blocks, for example, single bleeding and one reheat, avoiding salt solidification while still having better efficiencies than the complete cycle of a conventional plant with TO as HTF (six bleedings and one reheat): 41.04%–41.67% *vs.* 39.78%. The second way is that with the same complexity, *i.e.*, the same initial investment, a higher efficiency is achieved: 42.53% with HitecXL, and 44.58% with Solar Salt as compared with 39.78% for TO. This choice between one or several bleedings will depend on the investment capacity.

5. Conclusions and future perspectives

In the present work, it has been shown that using MS as HTF instead of TO implies:

- No risk of bending of the absorber tubes thanks to remaining very high Reynolds number (>60,000, knowing that minimum allowed values by manufacturer is 20,000).
- 30% lower heat transfer coefficient within the receivers tubes, although those coefficients are still very high (>3500 W m^{-2} K^{-1}), and heat transfer performance reduction should be negligible.
- 68% (HitecXL) to 84% (Solar Salt) reduction of the pressure drops, thanks to a circulation at relatively low velocity due to the higher temperature difference that is experienced by MS relatively to. This result was not expected since MS are more viscous than TO for similar temperatures.

Results from simulation with SAM software confirmed that pressure drops were lower for MS, and therefore pumping electricity consumption was larger to circulate TO. However, larger needs for electricity to maintain MS above their freezing point were found, especially in the case of Solar Salt. Nevertheless, is has been shown that TES systems generally have sufficient thermal inertia capacity to maintain the whole solar field and interconnecting piping system at a sufficient temperature throughout one to several days with insufficient solar irradiation.

In addition, LCOE estimation showed a potential decrease (>6%) for power plants with HitecXL, especially thanks to smaller TES tanks, no heat exchanger, cheaper HTF and higher power block efficiency.

It has also been shown that instead of high-performance

Table 12
Summary table with all the results obtained. See Appendix 2 to 5 for data and details.

	Conventional plant η_{cycle}	Direct storage HitecXL η_{cycle}	Direct storage Solar Salt η_{cycle}
Basic cycle	13.80%	21.29%	22.89%
Cycle w/reheating	35.85%	38.11%	39.86%
Cycle w/reheating and one bleeding	36.58%	41.04%	41.67%
Cycle w/reheating and two bleedings	37.27%	41.76%	42.89%
Cycle w/reheating and three bleedings	38.01%	42.05%	43.53%
Cycle w/reheating and four bleedings	38.45%	42.14%	43.82%
Cycle w/reheating and five bleedings	39.03%	42.20%	44.07%
Complete cycle	39.78%	42.53%	44.58%

complex power blocks, it is possible to install a simpler and cost-effective equipment with a theoretical efficiency that is still higher than the conventional ones that are adapted to's temperature ranges.

Finally, the use of MS as HTF in a parabolic trough power plant implies extra care regarding freezing protection. Building new linear-focusing systems with MS as HTF will require the design of several components such as electrical heat tracing, impedance heating, pre-insulated piping supports, etc. Only experimental validation of the feasibility of operating such power plants at relatively low cost will confirm the interest of using MS as both HTF and TES fluid.

CRediT authorship contribution statement

Telma Lopes: Methodology, Software, Formal analysis, Investigation, Writing - original draft, Visualization. **Thomas Fasquelle:** Methodology, Validation, Formal analysis, Writing - review & editing. **Hugo G. Silva:** Conceptualization, Methodology, Writing - review & editing, Supervision, Project administration.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Acknowledgements

This work was co-funded by the European Union through the European Regional Development Fund, framed in COMPETE 2020 (Operational Programme Competitiveness and Internationalization), through the ICT project (UID/GEO/04683/2013) with reference POCI-01-0145-FEDER-007690. The authors acknowledge relevant discussions with Michael Wittman and Klaus Hennecke (DLR), Mark and Kay Schmitz (TSK-FLAGSOL), Peter Schmidt (ELTHERM), and Evaristo Esquinas (RIOGLASS). HGS is grateful to Francis Lopes (ICT) for processing the meteorological data and to Samuel Bárias (ICT) and Jorge Neto (IPMA) for providing it.

Appendix 1. Inputs of SAM

Table 13
Manual inputs of the SAM.

System Design		Field aperture [m^2]	510120
		Hours of storage at design point [h]	7.5
		HTF	Therminol VP-1/HitecXL/Solar Salt
		Loop inlet HTF temperature [°C]	290
		Loop outlet HTF temperature [°C]	400/500/565
Solar Field	Solar Field Parameters	Row spacing [m]	18.9
		Piping thermal loss coefficient [$W.m^{-1}.K^{-1}$]	0.85/0.68/0.65
		Wind stow speed [$m.s^{-1}$]	14
	Heat Transfer Fluid	Freeze protection temperature [°C]	43/170/250
		Min single loop flow rate [$kg.s^{-1}$]	0.3/4/4
		Max single loop flow rate [$kg.s^{-1}$]	12.4/12.7/9
		Header design min flow velocity [$m.s^{-1}$]	0.06/0.72/0.36
		Header design max flow velocity [$m.s^{-1}$]	3.47/1.33/0.96
Storage System		Tank height [m]	14
		Tank fluid minimum height [m]	0.7
		Parallel tank pairs	1
		Wetted loss coefficient [$W.m^{-2}.K^{-1}$]	0.163
		Initial hot HTF percent	0
		Cold tank heater temperature set point [°C]	200/200/250
		Cold tank heater capacity [MW_e]	0.02
		Hot tank heater temperature set point [°C]	300
		Hot tank heater capacity [MW_e]	0.02

Appendix 2. Values used for efficiency calculation of cycle with reheating and 1 bleeding at 290 C (Solar Salt)

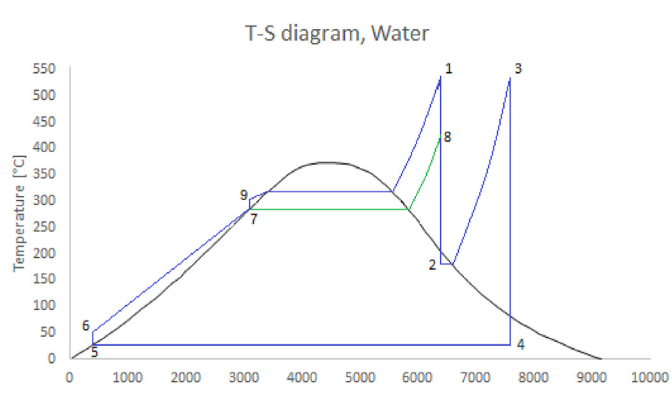

Fig. 6. Rankine cycle with reheating and 1 bleeding at 290 °C.

Table 14
Values of pressure, temperature, mass flow and enthalpy for each point.

Point	Pressure [Bars]	Temperature [°C]	Mass flow [kg.s^{-1}]	Enthalpy [kJ.kg^{-1}]
1	150	555	44.79	3461.71
2	10	180	28.33	2833.19
3	8	555	28.33	3599.81
4	0.06	36.17	28.33	2594.38
5	0.06	36.17	28.33	151.27
6	74.36	37.95	28.33	158.7
7	74.36	290	44.79	1289.64
8	74.36	432.95	16.46	3236.5
9	150	291.81	44.79	1297.20

$$\eta_{cycle} = \frac{m_1(h_1 - h_8) + m_2(h_8 - h_2) + m_3(h_3 - h_4) - m_5(h_6 - h_5) - m_7(h_9 - h_7)}{m_1(h_1 - h_9) + m_3(h_3 - h_2)} = 0.3972$$

Appendix 3. Values used for efficiency calculation of cycle with reheating and 1 bleeding at 200 °C (HitecXL)

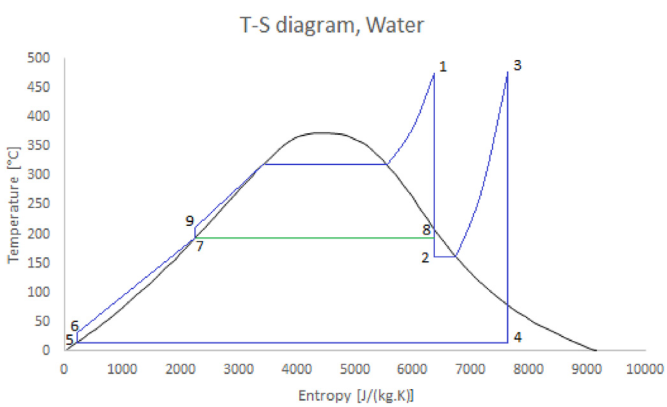

Fig. 7. Rankine cycle with reheating and 1 bleeding at 200 °C.

Appendix 4. Values used for efficiency calculation of complete cycle with HitecXL

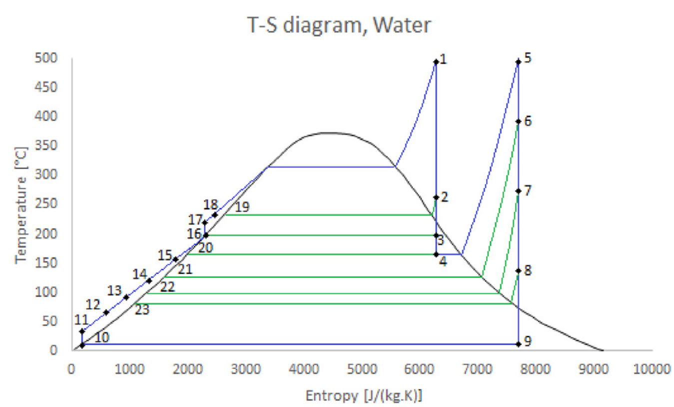

Fig. 8. Complete cycle of direct storage with molten salts plant.

Table 15
Values of pressure, temperature, mass flow and enthalpy for each point.

Point	Pressure [bars]	Temperature [°C]	Mass flow [kg.s^{-1}]	Enthalpy [kJ.kg^{-1}]
1	135	490	40.80	3300
2	10	180	29.88	2750.17
3	8	490	29.88	3458.71
4	0.06	36.17	29.88	2530.09
5	0.06	36.17	29.88	151.27
6	15.54	36.54	29.88	152.82
7	15.54	200	40.80	852.74
8	15.54	200	10.93	2766.60
9	135	202.86	40.80	864.69

$$\eta_{cycle} = \frac{m_1(h_1 - h_8) + m_2(h_8 - h_2) + m_3(h_3 - h_4) - m_5(h_6 - h_5) - m_7(h_9 - h_7)}{m_1(h_1 - h_9) + m_3(h_3 - h_2)} = 0.4104$$

Table 16
Values of pressure, temperature, mass flow and enthalpy for each point.

Point	Pressure [bars]	Temperature [°C]	Mass flow [kg.s^{-1}]	Enthalpy [kJ.kg^{-1}]
1	135	490	40.05	3300
2	27.95	257.32	2.16	2886.80
3	15.54	200	3.42	2766.60
4	10	180	2.18	2750.17
5	8	490	32.29	3458.71
6	2.7	324.73	1.51	3120.09
7	1.01	202.46	1.69	2879.74
8	0.31	84	1.40	2655.15
9	0.06	36.17	27.70	2530.09
10	0.06	36.17	34.47	151.27
11	15.54	36.54	34.05	152.82
12	15.54	66.65	34.05	268.04
13	15.54	96.67	34.05	402.34
14	15.54	126.67	34.05	528.62
15	15.54	156.67	34.05	654.10
16	15.54	200	40.05	852.74
17	135	202.86	40.05	864.69
18	135	227.29	40.05	966.81
19	27.95	230	2.16	990.77
20	10	180	2.18	763.18
21	2.7	130	3.68	546.28
22	1.01	100	5.38	419.71
23	0.31	70	6.77	293.89

Appendix 5. Values used for efficiency calculation of complete cycle with Solar Salt

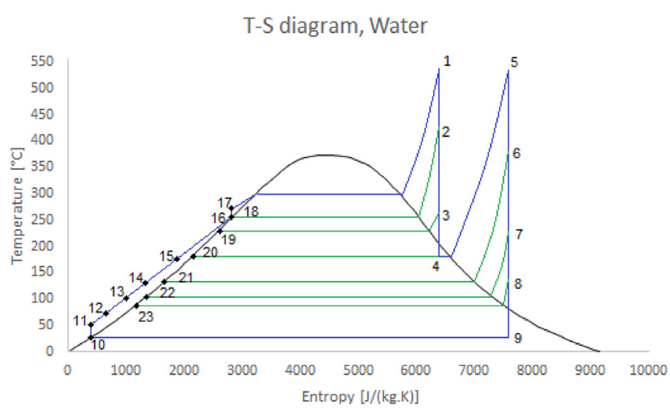

Fig. 9. Complete cycle of direct storage with Solar Salt.

Table 17
Values of pressure, temperature, mass flow and enthalpy for each point.

Point	Pressure [bars]	Temperature [°C]	Mass flow [kg.s^{-1}]	Enthalpy [kJ.kg^{-1}]
1	150	555	39.62	3461.71
2	74.36	432.95	4.37	3236.5
3	27.95	290.31	3.51	2975.07
4	10	180	3.08	2833.19
5	8	555	28.66	3599.81
6	2.7	380.36	1.23	3234.39
7	1.01	245.7	1.35	2965.46
8	0.31	119.43	1.34	2722.08
9	0.06	36.17	24.74	2594.38
10	0.06	36.17	31.74	151.27
11	27.95	36.84	31.74	154.06
12	27.95	66.68	31.74	278.82
13	27.95	96.67	31.74	404.15
14	27.95	126.67	31.74	529.55
15	27.95	174.67	31.74	730.19
16	27.95	230	39.62	990.77
17	150	232.92	39.62	1002.98
18	150	284.29	39.62	1217.71
19	74.36	290	4.37	1289.64
20	10	180	3.08	763.18
21	2.7	130	4.31	546.28
22	1.01	100	5.66	419.71
23	0.31	70	7	293.89

Appendix 6. LCOE calculation with Solar Salt

Table 18
LCOE values obtained for a power plant running Solar Salt.

	CAPEX [M$]/[M€]	OPEX [M$]/[M€]	Annual electricity (produced) [MWh]]	LCOE [c$/kWh]/[c€/kWh]
First method (with SAM)	339.5/303.55	3.25/2.92	164,565	18.99/16.98
Second method (manual calculation)	339.5/303.55	3.25/2.92	133,577	17.92/16.02

Appendix 7. Values of exergy efficiency

Table 19
Summary table with values of exergy efficiency of each cycle.

	Conventional plant η_{cycle}	Direct storage HitecXL η_{cycle}	Direct storage Solar Salt η_{cycle}
Basic cycle	24.64%	34.34%	35.22%
Cycle w/reheating	64.02%	61.47%	61.32%
Cycle w/reheating and one bleeding	65.32%	66.19%	64.11%
Cycle w/reheating and two bleedings	66.55%	67.35%	65.98%
Cycle w/reheating and three bleedings	67.88%	67.82%	66.97%
Cycle w/reheating and four bleedings	68.66%	67.97%	67.42%
Cycle w/reheating and five bleedings	69.70%	68.06%	67.80%
Complete cycle	71.04%	68.60%	68.58%

References

[1] Iea, Installed Power Generation Capacity by Source in the Stated Policies Scenario, IEA, 2000-2040 [Online]. Available, https://www.iea.org/data-and-statistics/charts/installed-power-generation-capacity-by-source-in-the-stated-policies-scenario-2000-2040.

[2] Nrel, National renewable energy laboratory (NREL) home page | NREL [Online]. Available, https://www.nrel.gov/.

[3] D. Baharoon, H. Rahman, W. Omar, S. Fadhl, Historical development of concentrating solar power technologies to generate clean electricity efficiently, Renew. Sustain. Energy Rev. 41 (2015) 996–1027.

[4] P. Gauché, J. Rudman, M. Mabaso, W. Landman, T. von Backstrom, A. Brent, System value and progress of CSP, Sol. Energy 152 (2017) 106–139.

[5] D. Kearney, U. Herrmann, P. Nava, B. Kelly, R. Mahoney, J. Pacheco, R. Cable, N. Potrovitza, D. Blake, H. Price, Assessment of a molten salt heat transfer fluid in a parabolic trough solar field, J. Sol. Energy Eng. 125 (2003) 170–176.

[6] M. Wittmann, M. Schmitz, H.G. Silva, P. Schmidt, G. Doppelbauer, R. Ernst, P. Santamaria, T. Miltkau, D. Golovca, L. Pacheco, D. Hogemann, M. Meyer-Grunefeldt, B. Seubert, HPS2 - Demonstration of a Molten-Salt in Parabolic Trough Plants - Design of Plant, *SolarPaces*, 2018.

[7] A. Maccari, D. Bissi, G. Casubolo, F. Guerrini, L. Lucatello, G. Luna, A. Rivaben, E. Savoldi, S. Tamano, M. Zuanella, Archimede solar energy molten salt parabolic trough demo plant: a step ahead towards the new frontiers of CSP, Energy Procedia 69 (2015) 1643–1651.

[8] K. Vignarooban, X. Xu, A. Arvay, K. Hsu, A.M. Kannan, "Heat transfer fluids for concentrating solar power systems – a review, Appl. Energy 146 (2015) 383–396.

[9] F. Casella, E. Casati, P. Colonna, Optimal operation of solar tower plants with thermal storage for system design, IFAC Proceedings Volumes 47 (2014) 4972–4978.

[10] Enea, MATS Project - Multipurpose Applications by Thermodynamic Solar, 2019 [Online]. Available, http://www.mats.enea.it/.

[11] A. Gil, M. Medrano, I. Martorell, A. Lázaro, P. Dolado, B. Zalba, State of the Art on High Temperature Thermal Energy Storage for Power Generation. Part 1 - Concepts, Materials and Modellization, *Renewable and Sustainable Energy Reviews*, 2010, pp. 31–55.

[12] T. Boukelia, O. Arslan, M. Mecibah, Potential assessment of a parabolic trough solar thermal power plant considering hourly analysis: ANN-based approach, Renew. Energy 105 (2017) 324–333.

[13] A. Bonk, S. Sau, N. Uranga, M. Hernaiz, T. Bauer, Advanced heat transfer fluids for direct molten salt line-focusing CSP plants, Prog. Energy Combust. Sci. (2018) 69–87.

[14] S.M. Akbarimoosavi, M. Yaghoubi, 3D Thermal-structural analysis of an absorber tube of a parabolic trough collector and the effect of tube deflection on optical efficiency, Energy Procedia 49 (2014) 2433–2443.

[15] S. Khanna, V. Sharma, S.B. Kedare, S. Singh, Experimental investigation of the bending of absorber tube of solar parabolic trough concentrator and comparison with analytical results, Sol. Energy 125 (2016) 1–11.

[16] I. Tosun, Modeling in Transport Phenomena - A Conceptual Approach, second ed., Elsevier Science, Turkey, 2007.

[17] Nrel, Technical manual for the SAM physical trough model [Online]. Available, https://www.nrel.gov/docs/fy11osti/51825.pdf.

[18] N. Blair, A. Dobos, J. Freeman, T. Neises, M. Wagner, T. Ferguson, System Advisor Model - General Description, National Renewable Energy Laboratory, Golden, CO, 2014.

[19] Nrel, System Advisor moldel (SAM) [Online]. Available, https://sam.nrel.gov/download, 2019.

[20] Nrel, Concentrating solar power projects - andasol-3 [Online]. Available, https://solarpaces.nrel.gov/andasol-3.

[21] F.M. Lopes, R. Conceição, T. Fasquelle, H.G. Silva, R. Salgado, P. Canhoto, M. Collares-Pereira, Predicted direct solar radiation (ECMWF) for optimized operational strategies of linear focus parabolic-trough systems, Renew. Energy 151 (2020) 378–391.

[22] F.M. Lopes, R. Conceição, H.G. Silva, T. Fasquelle, R. Salgado, P. Canhoto, M. Collares-Pereira, Short-term forecasts of DNI from an integrated forecasting system (ECMWF) for optimized operational strategies of a central receiver system, energies 12 (7) (2019) 1368–1386.

[23] Irena, Renewable Energy Technologies: cost analysis series [Online]. Available, https://www.irena.org/documentdownloads/publications/re_technologies_cost_analysis-csp.pdf, 2012.

[24] C. Turchi, M. Mehos, C.K. Ho, G.J. Kolb, Current and Future Costs for Parabolic Trough and Power Tower Systems in the US Market, U.S.A, 2010.

[25] Dlr, Concentrating Solar Power Now - Clean Energy for Sustainable Development, 2002.

[26] H. Grirate, N. Zari, A. Elmchaouri, S. Molina, R. Couturier, Life time analysis of thermal oil used as heat transfer fluid in CSP power plant, AIP Conference Proceedings 1734 (2016), 040005.

[27] A.A. Piquer, Determinación del ciclo de vapor de una central solar termoeléctrica CCP de 50MW, Universidad Carlos III de Madrid, Madrid, 2012.