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Maximizing the power block efficiency of solar towers plants: dual-pressure level steam generator.

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Abstract

Solar tower plants (STP) are one of the most promising renewable technologies to substitute conventional power plants. To further increase its penetration in electricity markets, it is necessary to maximize its efficiency. This work addresses the challenge of increasing the inlet pressure up to 165 bar at the high pressure turbine (HPT) to improve the power block efficiency. Nowadays, these plants operate Rankine cycle with reheating using steam at 126 bar at the inlet of the HPT. This pressure is limited in current STP by the working temperatures of the molten salt, which range from 285°C and 565°C, the pinch point temperature difference in the evaporator and the current steam generator (SG) layout. A new steam generator design with dual-pressure level evaporation is proposed. The heat exchangers that form the SG are designed thermomechanically and the power cycle performance is analyzed using the first and second laws of thermodynamics. The results show that the novel dual-pressure level SG layout increases the power block efficiency from 44.14% to 44.64%. Assuming a market pricing scenario of two-tier tariff and a power purchase agreement price of 16.3 c€/kWh_e, the new SG layout yields an extra economic benefit of 623 k€ per year due to an increase of energy produced of 5.71 GWh_e/year.

keywords:

solar tower plant; steam generator; dual-pressure level; efficiency; power block

Nomenclature

Abbreviations

COND condenser

D drum

EV evaporator

FWH feedwater heater

35	GA	genetic algorithm
36	HPT	high pressure turbine
37	HRSG	heat recovery steam generator
38	HTF	heat transfer fluid
39	LPT	low pressure turbine
40	PH	preheater
41	RH	reheater
42	SG	steam generator
43	SH	superheater
44	STP	solar tower plant
45	TAC	total annualized cost
46	<i>Symbols</i>	
47	A	heat exchange area per shell (m^2)
48	B_c	baffle cut (%)
49	L_{bc}	heat exchanger cost (k€)
50	D	diameter (mm)
51	h	convective heat transfer coefficient ($\text{W}/\text{m}^2\text{°C}$)
52	LCOE	levelized cost of energy (c€/kWh _e)
53	L_{bc}	baffle spacing (mm)
54	L_t	tube length (m)
55	L_{tp}	tube pitch (mm)
56	l_{ts}	tubesheet thickness (mm)
57	$\dot{m}$	mass flow (kg/s)
58	N_s	total number of shells (-)
59	N_{tp}	tube passes (-)
60	N_{tt}	tube number (-)
61	P	pressure (bar)
62	PPA	power purchase agreement price (c€/kWh)
63	PP_{Ev}	pinch point temperature difference (°C)

64	$\dot{Q}_{in}$	heat absorbed by the molten salt (MW)
65	R	fouling resistance ($^{\circ}\text{Cm}^2/\text{W}$)
66	R_{min}	U-tube minimal radius (mm)
67	T	temperature ($^{\circ}\text{C}$)
68	t_h	head thickness (mm)
69	t_s	shell thickness (mm)
70	U	overall heat transfer coefficient ($\text{W}/\text{m}^2\text{C}$)
71	v	flow velocity (m/s)
72	X	vapor quality (-)
73	$\dot{W}$	power (MW)
74	<i>Greek symbols</i>	
75	ΔP	pressure drop (Pa)
76	η_{PB}	efficiency of the power block (%)
77	θ_{tp}	tube layout ($^{\circ}$)
78	<i>Subscripts</i>	
79	FW	feedwater
80	HP	high pressure
81	HPT	high pressure turbine
82	i	interior, inlet
83	o	outlet
84	LP	low pressure
85	LPT	low pressure turbine
86	net	net power
87	s	shell
88	t	tube

89

90 **1. Introduction**

91 Solar tower plants (STP) have become an attractive alternative to generate clean electricity.
92 A solar field, an energy storage system, a steam generator and a power block form these

plants. Commercial STP, such as Crescent Dunes or Gema Solar [1], employ molten salt to collect the concentrated energy in the receiver. This salt, 60% NaNO_3 and 40% KNO_3 , works as the heat transfer fluid (HTF) in a temperature range of 285°C and 565°C to avoid freezing and corrosion problems, respectively [2].

Many researchers are trying to improve the STP efficiency maximizing the performance of each subsystem. In this way, the heliostat field has been analyzed optimizing its layout [3] or employing different aiming strategies [4, 5]. Other studies have focused on increasing the receiver efficiency proposing new designs [6, 7] or changing the HTF velocity of each panel of the receiver [8]. Similarly, Rodriguez et al. [9] proposed a different approach to reduce the parasitic consumptions recovering the potential energy of the hot HTF that comes from the tower.

STPs use an indirect steam generator (SG) system since it allows the installation of a thermal storage system. Current steam generators (SG) are formed by four heat exchangers arranged in one or two parallel trains: preheater (PH), evaporator (EV), superheater (SH) and reheater (RH). These heat exchangers produce the steam needed in a subcritical Rankine cycle with reheating. Nevertheless, indirect SG systems have the limitation of the pinch point temperature difference in the evaporator (PP_{EV}). On the one hand, the minimum allowed pinch point sets the minimum HTF flow rate through the SG. On the other hand, lower pinch points lead to low temperature differences in the heat exchangers and this increases the required heat transfer areas. Economic and exergetic analyses have shown the relevance of the SG subsystem in the performance of the whole plant [10, 11]. In this sense, a pinch point temperature difference between the HTF and the water in the evaporator of 2.5°C was found as the optimum for the SG design and STP operation.

The majority of the works that have improved the STP efficiency have considered fixed the operating conditions of the steam in the power block [12]. However, as in conventional power plants, the efficiency of the steam cycle can be increased whilst the power block layout is not modified [13]. As the molten salt working temperatures are fixed, a straightforward method to increase the power block efficiency is changing the steam inlet conditions to the high and low pressure turbines (HPT and LPT), i. e. increasing the temperature and/or pressure [13, 14]. On the one hand, the inlet temperature to both HPT and LPT is fixed up to 560°C by the turbine manufacturer [15, 16]. On the other hand, the current layout of the SG, with PH, EV, SH and RH, prevents the increase of the inlet steam pressure to the HPT without violating the pinch point or without increasing the cold temperature of the molten salt. Thus, the inlet pressure to the HPT is limited to 126 bar due to the current SG layout. However, the inlet pressure to the HPT can reach 165 bar and 175 bar for SST-600 and STF15S models, respectively [15, 16]. Therefore, the question that remains is if, considering all STP constrains, there is a SG layout able to increase the steam inlet pressure of HPT in order to increase the power block efficiency.

A feasible way to overcome this drawback has been proposed in the heat recovery steam generators (HRSG) of combined cycles. In these systems, the layout of the heat exchangers presents several pressure levels of evaporation. Bassily [17–19] analyzed several HRSG with dual and triple-pressure levels of evaporation and re-heating. His works optimized dual-

pressure and triple-pressure level HRSG of combined cycles minimizing the irreversibility generated. Srinivas et al. [20] compared between single-pressure, dual-pressure and triple-pressure HRSG. Similarly, other works have studied different HRSGs with several pressure levels in combined cycles [21, 22] or in integrated solar combined cycles [23,24].

In this paper, a dual-pressure level steam generator is proposed to increase the power block efficiency of solar tower plants with molten salts. The new steam generator presents two pressure levels, similarly to HRSG designs of combined cycles [17,20]. A comparison between the current SG of a reference STP, with inlet steam pressure at the high pressure turbine of 126 bar, and the new dual-pressure level SG, with an inlet pressure of 165 bar, is carried out. Heat exchangers are designed according to TEMA standards [25] and ASME pressure Vessel code [26] following the methodology described in [11]. The results show the increase of the power block efficiency while the economic costs are similar for both the current and new SG.

2. Current solar tower plants

The reference STP is a 100 MWe Rankine subcritical plant that comprises solar field, storage system, steam generator and power block, Fig. 1. The energy reflected by the heliostats is concentrated on a central receiver, which is placed on the top of a tower, Fig. 1-a. The hot salt is stored in the hot tank, from where is pumped to the SG to fulfill the energy requirements of the power block. The salt enters to the SG through the SH and RH at $T_{HOT} = 565^{\circ}\text{C}$, and leaves the SG at $T_{COLD} = 285^{\circ}\text{C}$. Currently, the steam generator comprises four heat exchangers that increase the temperature of liquid water from $T_{FW,out} = 245^{\circ}\text{C}$ to steam at $P_{HPT} = 126$ bar and $T_{HPT/LPT,in} = 550^{\circ}\text{C}$, Fig. 1-b. In the power block, the steam follows a regenerative Rankine cycle condensed by air, Fig. 1-c. As the quality at the outlet of the LPT is 0.9163, the reheating pressure is set to $P_{HPT,out} = 45$ bar.

Figure 1. Layout of a solar power tower plant: (a) subsystems division, (b) steam generator, (c) power block.

The thermal behavior of the SG heat exchangers is determined by the pinch point temperature difference between the molten salt and the water at the inlet of the EV. As the optimum pinch point for the plant operation is $PP_{EV} = 2.5^\circ\text{C}$ [10,11], the maximum inlet pressure to the HPT allowable without violating the pinch point is $P_{HPT,in} = 126\text{ bar}$. This fact is illustrated in Figure 2, which shows the temperature profiles of the molten salt and the water for the reference STP layout, Fig. 1, working at $P_{HPT,in} = 126\text{ bar}$ and $P_{HPT,in} = 165\text{ bar}$. The molten salt temperature profile for current conditions, in solid line, shows that the cold temperature of the salt remains at $T_{COLD} = 285^\circ\text{C}$, maximizing the receiver efficiency [27].

On the contrary, if the pressure is increased up to $P_{HPT,in} = 165\text{ bar}$ without violating the pinch, in dashed line, the cold temperature of the molten salt increases up to $T_{COLD} = 293^\circ\text{C}$, Fig. 2. This will cause the reduction of the receiver efficiency due to the modification of the temperature profile in the receiver. Due to that, the heat flux needed on the receiver would change, modifying the aiming strategy [4] [28]. Furthermore, the cold temperature increase may decrease the convective coefficient within the tubes of the receiver, and thus, a higher salt mass flow would be needed, increasing the parasitic consumption of the pumps. Therefore, the cold temperature increase of the molten salt will neutralize the improvement in the power block due to the pressure increase.

Figure 2. Temperature profile for different inlet pressures at the HPT for $PP_{EV} = 2.5\text{ }^{\circ}\text{C}$ of current STP layout.

It is worth to mention the value of the outlet temperature of the feedwater heaters, $T_{FW,out} = 245^{\circ}\text{C}$. This temperature depends on the extraction pressure of the HPT, Fig. 1-c., which is fixed to obtain the maximum power block efficiency [29]. The value of $T_{FW,out} = 245^{\circ}\text{C}$ follows the results shown in [29] while is low enough to avoid salt freezing problems in the PH.

Another option is the change of the HTF to improve its working temperatures. There are many works developing new HTFs [30] [31], for example, Halotechnics SS-700 could work between 257°C and 700°C [31]. However, if Halotechnics SS-700 is used as HTF, the current SG layout (Fig. 1-c) still limits the pressure increase at the HPT. Therefore, future HTF and STP developments will need to modify the SG layout for steam-Rankine power cycles.

3. Dual-pressure steam generator for STP

The challenge of the new SG is to increase the inlet pressure at the HPT without violating the STP thermodynamic restrictions: T_{HOT} , T_{COLD} and PP_{EV} . To simplify the analysis, heliostats, receiver, energy storage system and power block (Fig. 1-a/c) are the same for both SG layouts: the reference case of $P_{HPT,in} = 126\text{ bar}$ (Fig. 1-b) and the new STP at $P_{HPT,in} = 165\text{ bar}$ (Fig. 3-b). Only changes the SG design, which is schematically represented in Fig. 3-a.

In order to obey the $PP_{EV} = 2.5^{\circ}\text{C}$ while increasing the pressure up to $P_{HPT,in} = 165\text{ bar}$, it is necessary to change the SG layout. The new SG design is inspired in the dual-pressure level HRSG of conventional combined cycles [17,20]. The new design shown in Fig. 3-a includes a dual-pressure level SG, in which the evaporation is carried out in two steps, allowing the increase of $P_{HPT,in}$ without violating the pinch point.

A low pressure evaporator receives the water from the FWH at $T_{FW,out} = 245^\circ\text{C}$ at the drum pressure. In the drum (D), part of the water flows through the high pressure line, which includes a pump, the PH, the EV_{HP} and the SH. This mass flow goes through the HPT from $P_{HPT,in} = 165 \text{ bar}$ to $P_{HPT,out} = 45 \text{ bar}$. Prior of reheating, an extraction is made for the last high-pressure FWH, Fig. 3-b. The remaining steam mass flow is mixed with the mass flow that comes from the drum of the low pressure evaporator (EV_{LP}). The percentage of mass flow through high and low pressure lines is calculated from an energy balance in both evaporators.

Figure 3. Proposed STP design for $P_{HPT,in} = 165 \text{ bar}$: (a) dual-pressure level SG, (b) power block.

4. Methodology

Firstly, this section describes the assumptions made to analyze the current STP layout, with $P_{HPT} = 126 \text{ bar}$ (Fig. 1), and the new design, with $P_{HPT} = 165 \text{ bar}$ (Fig. 3). First and second laws of thermodynamics are used to analyze the power block efficiency of the following cases:

- Reference STP: with an inlet pressure to the HPT of $P_{HPT} = 126 \text{ bar}$ and a SG formed by four heat exchangers: PH, EV, SH and RH (Fig. 1-b).
- Dual-pressure level SG for STP: with an inlet pressure to the HPT of $P_{HPT} = 165 \text{ bar}$ and a SG formed by two evaporators: at high pressure (EV_{HP}) and at low pressure (EV_{LP}), a PH, a SH and a RH (Fig. 3-a).

Table 1 summarizes the working conditions of both STP designs. Later, the methodology for the design of the heat exchangers of both the reference and the new SG is explained.

Table 1. Common design parameters of both current and dual-pressure level STP.

Solar field and storage system	
Aperture area (m ²)	1,271,887
Heliostat optical efficiency (%) [32]	51.47
Receiver efficiency (%) [27][28]	75.81
Hot tank temperature (°C)	565
Cold tank temperature (°C)	285
Heat absorbed by HTF ($\dot{Q}_{in}$) (MW)	226.54
Steam generator	
Pinch point in the evaporator (°C) [11]	2.5
Power block	
Inlet HPT & LPT temperature (°C)	550
HPT efficiency (%)	85.5
LPT efficiency (%)	89.5
Isentropic pumps efficiency (%)	80.0
Outlet feedwater temperature (°C)	245
<i>Air condenser</i>	
Condenser inlet quality (-)	0.9163
Condenser pressure (bar)	0.115
Inlet refrigerant temperature (ambient) (°C)	25
Outlet refrigerant temperature (°C)	44
<i>Pressure drop in extraction lines</i>	
Low pressure FWH and deaerator (bar)	$0.05 \cdot P_{ext}$
High pressure FWH (bar)	$0.03 \cdot P_{ext}$
<i>Low pressure closed FWH</i>	
Terminal temperature difference (°C)	1.7
Drain cooling approach (°C)	5.6
<i>High pressure closed FWH</i>	
Terminal temperature difference (°C)	-0.5, -1
Drain cooling approach (°C)	5.6

231

232 4.1. STP modelling

233 The assumptions made for modelling both STP cases are:

- Heliostat field, receiver, energy storage system (Fig. 1-a) and power block (Fig. 1-c and Fig. 3-b) are kept unaltered for both reference and new SG designs following the characteristics shown in Table 1.
- Pinch point temperature difference in the evaporator is $PP_{EV} = 2.5^{\circ}\text{C}$ [10, 11].
- Molten salt minimum and maximum temperatures are $T_{COLD} = 285^{\circ}\text{C}$ and $T_{HOT} = 565^{\circ}\text{C}$, respectively.
- Only the inlet steam pressure to the HPT is modified: $P_{HPT,in} = 126$ bar (Fig. 1) or $P_{HPT,in} = 165$ bar (Fig. 3).
- Water mass flow through the turbines is calculated for an energy captured by the molten salt of $\dot{Q}_{in} = 226.54$ MW.
- Steam turbine SST-600 [15], which is similar to Gemasolar, is chosen for the power block. Table 2 shows the characteristics of this model and turbine STF15S [16], which could be also employed.
- The outlet temperature of the last FWH is set to $T_{FW,out} = 245^{\circ}\text{C}$ [29].
- Extraction points from the LPT are calculated for constant enthalpy drops [13,14]. The 6 extractions from the LPT are connected to 1 high pressure FWH, 1 deaerator and 4 low pressure FWH.
- The quality at the outlet of the LPT is fixed to $X = 0.9163$ in order to minimize the influence of the reheating pressure. Reheating pressure is obtained from this data, being $P_{RH} = 45$ bar.
- Pressure drops due to pipes and joints through the SG are included in pump 2 of the power block with a value of $\Delta P_{pipes} = 22$ bar for the reference case (Fig. 1), and $\Delta P_{pipes} = 25$ bar for the new STP (Fig. 3).
- The efficiency of the power block is calculated using Eqs. 1 and 2:

$$\dot{W}_{net} = \dot{W}_{HPT} + \dot{W}_{LPT} - \sum \dot{W}_{pumps} \quad (1)$$

$$\eta_{PB} = \frac{\dot{W}_{net}}{\dot{Q}_{in}} \quad (2)$$

Water properties are calculated using CoolProp library [33] while molten salt properties are computed from Refs. [2,34].

Table 2. Technical data of the steam turbines available.

	SST-600 [15]	STF15S [16]
Output range (MW)	up to 150	120 – 175
Live-steam pressure (bar)	≤ 165 bar	120 – 175
Live-steam temperature ($^{\circ}\text{C}$)	≤ 565	535 – 565
Reheat steam pressure (bar)	≤ 45	25 – 45
Reheat steam temperature ($^{\circ}\text{C}$)	≤ 565	535 – 565

Condenser pressure (bar)	≤ 1	0.05 – 0.2
Number of uncontrolled extractions	6	6

4.2. Heat exchanger design of the SG

Figure 4 shows the methodology used for the heat exchanger design. In a first step, the preliminary design aspects are selected taking into account the operating conditions. Then, the thermal and hydraulic design calculations are divided in two groups: single phase heat exchanger and two phase heat exchangers. The heat transfer coefficient and pressure drop on the shell-side are calculated according to ESDU [35] and Wills and Johnston [36], respectively. The heat transfer coefficient on the tube-side is calculated using Gnielinski correlation [36]. The heat transfer coefficient and the pressure drop of the two-phase flow is calculated according to the Chen's [37] and Martinelli [36] correlations, respectively. The critical heat flux for the horizontal evaporator is calculated according to the correlation proposed by Wong et al. [38]. The steam drum is sized according to the steam velocity limitation recommended by Ganapathy [39]. The diameters of risers and downcomers are set to 150 and 450 mm, respectively. The number of risers and downcomers is calculated to not overcome the maximum allowable momentum [40]. Then, the pressure drop on the circulation loop of the evaporator and the steam drum is calculated.

Figure 4. Schematic of the heat exchanger design algorithm.

The mechanical design consists on three steps. First, the main wall thicknesses of the heat exchangers (shell, tube, tubesheet, etc) are calculated using ASME Sections VIII and II [26]. In addition, a thermal stress analysis is performed. On the one hand, the minimal U-tube radius is calculated to not overpass the maximum allowable stress on the U-bend due to the thermal tube expansion. On the other hand, the thermal stresses of the tubesheet are checked to not overpass the thermal stress limit according to the O'Donnell method [41]. Lastly, a vibration analysis is performed to assure the integrity of the tubes.

In order to optimize the SG performance, the total annualized cost (TAC) of each heat exchanger is minimized. The capital cost of the heat exchangers are estimated according to Purohit method [42]. The operational pump and start-up costs are also considered in TAC. The heat exchanger optimization is made using a genetic algorithm (GA). Several constrains included to obtain feasible designs are explained in [11] [43] together with the methodology to design the SG.

5. Results and discussion

The design of the proposed dual-pressure level SG is analyzed studying the design of the heat exchangers for the reference and new SG. Then, the cycle performance of both the reference STP, at $P_{\text{HPT,in}} = 126$ bar (Fig. 1), and the new dual-pressure SG for STP, at $P_{\text{HPT,in}} = 165$ bar (Fig. 3), is considered.

5.1. Dual-pressure level SG design

The dual-pressure SG is based on conventional shell and tube heat exchangers. Figure 5 shows the configuration of the dual-pressure SG. The heat exchanger design is made following TEMA [25] and ASME standards [26]. The heat exchanger selection is made according González-Gómez et al. [11]. The SH and RH consist of U-tube/U- shell heat exchangers in order to avoid the high thermal gradients obtained on the single tubesheet heat exchanger designs. EV_{HP} and EV_{LP} are designed in horizontal position with a U-tube/TEMA E heat exchanger. EV_{HP} and EV_{LP} have a forced circulation system to allow the installation of low cost U-tube design. Lastly, the PH consists of a U-tube/TEMA F heat exchanger. In all heat exchangers the salt flows on the shell-side whereas the water/steam flows in the tube-side.

Figure 5. Heat exchanger configuration for the dual-pressure level SG.

The design of the heat exchangers of both the current SG and the dual-pressure SG are presented in Table 3. One of the most interesting results is the increment of the metal wall thicknesses due to the increment of the operating pressure from 126 to 165 bar. As it can be seen, a significant increment of the tubesheet thicknesses is obtained for the PH and the

EV_{HP}. However, in the SH is not obtained a significant increment. This is due to the increment of the pitch ratio (L_{tp}/D_t) from 1.3 to 1.5, which leads to an increment from 0.26 to 0.36 in the ligament efficiency. As a result, a similar tubesheet thickness is obtained for 165 and 126 bar in the SHs. The heat exchanger head is another part susceptible of thickness increment. As it is shown in Table 3, a generalized increment of the head thickness is obtained for all heat exchangers in the new design. Finally, as a result of the metal wall thickness growth on the aforementioned parts and others (head closure, tubes, nozzles, etc), an increment of the heat exchangers metal mass is obtained, especially in the EV_{HP}.

Table 3. Proposed design of the heat exchangers for the dual-pressure SG layout with one train.

Parameter	Superheater 165 (126)	Reheater 165 (126)	Evaporator HP 165 (126)	Preheater 165 (126)	Evaporator LP
Shell diameter, D_s (mm)	915.8 (884)	1010 (1010)	1796 (1796)	1700 (1600)	975
Baffle cut, B_c (%)	29 (28)	27 (22)	24 (23)	26 (23)	37
Baffle spacing, L_{bc} (mm)	546 (612)	394 (317)	756 (569)	650 (658)	926
Tubes ext. diameter, D_t (mm)	12.7 (15.9)	22.2 (25.4)	19.1 (15.9)	15.9 (15.9)	25.4
Tubes int. diameter, D_{ti} (mm)	9 (12.2)	18 (21.2)	14.1 (12.2)	11.7 (12.2)	21.2
Tube pitch, L_{tp} (mm)	19 (20.7)	27.8 (31.8)	23.9 (20.7)	23.9 (23.9)	33.3
Tube layout, θ_{tp} (°)	30 (45)	30 (30)	90 (90)	45 (45)	45
Tube passes, N_{tp} (-)	1	1	2	2	2
Tubes number, N_{tt} (-)	1828 (1219)	1070 (815)	2046-U (2737-U)	1835-U (1615-U)	277-U
Tube length, L_t (m)	18.7 (20.81)	18.6 (22.09)	10.0 (9.43)	10.2 (11.04)	8.4
Shell thickness, t_s (mm)	12.7 (12.7)	12.7 (12.7)	15.9 (15.9)	19.1 (15.9)	9.5
Tubesheet thickness, l_{ts} (mm)	256 (252)	205 (193)	479 (400)	386 (312)	144
Head thickness, t_h (mm)	114.3 (86.9)	34.9 (31.8)	114.3 (85.7)	139.7 (86.9)	19.1
U-tube minimal radius, R_{min} (mm)	384 (395)	606 (713)	44.4 (42)	53.8 (48)	50.8
Mass flow (tube-side), $\dot{m}_t$ (kg/s)	77.5 (86.92)	83.7 (78.7)	349 (567.1)	77.5 (86.92)	203.8
Mass flow (shell-side), $\dot{m}_s$ (kg/s)	345.4 (390.4)	201 (183.3)	546.7 (573.1)	546.7 (573.1)	546.7
Temperature inlet (shell-side), T_{si} (°C)	565 (565)	565 (565)	436 (447.4)	355.4 (332.2)	306.3
Temperature outlet (shell-side), $T_{s,o}$ (°C)	436 (447.4)	436 (447.4)	355.5 (332.2)	306.3 (285.9)	285.3
Temperature inlet (tube-side), T_{ti} (°C)	352 (329.6)	350 (369)	353 (329.6)	265.4 (245)	258.4
Temperature outlet (tube-side), $T_{t,o}$ (°C)	550 (550)	550 (550)	353 (329.6)	352 (327.1)	258.4
Flow velocity (tube-side), v_t (m/s)	10.7 (13.21)	22.6 (23.9)	1.71 (2.53)	0.55 (0.61)	2.59
Flow velocity (shell-side), v_s (m/s)	0.45 (0.65)	0.44 (0.5)	0.56 (0.6)	0.63 (0.70)	0.90
Convective heat transfer coefficient (tube-side), h_t (W/m ² °C)	4806 (3649)	1422 (1227)	22641 (27688)	6224 (6598)	21806
Convective heat transfer coefficient (shell-side), h_s (W/m ² °C)	4143 (5213)	3313 (3656)	3803 (4200)	4042 (4234)	3716

Fouling resistance (tube-side), R_t ($^{\circ}\text{Cm}^2/\text{W}$)	8.825e-5	8.825e-5	2.647e-04	8.825e-5	2.647e-04
Fouling resistance (shell-side), R_s ($^{\circ}\text{Cm}^2/\text{W}$)	8.825e-5	8.825e-5	8.825e-5	8.825e-5	8.825e-5
Overall heat transfer coefficient, U ($\text{W}/\text{m}^2\text{ }^{\circ}\text{C}$)	1246 (1241)	700 (664)	1183 (1295)	1353 (1448)	1265
Heat exchange area (per shell), A (m^2)	1365 (1113)	1392 (1294)	2467 (2595)	1867 (1857)	373
Pressure drop (shell-side), ΔP_s (kPa)	92 (148)	92 (149)	131 (172)	147 (205)	115
Pressure drop (tube-side), ΔP_t (kPa)	128 (105)	81 (70)	46 (70)	10 (13)	89
Shell type	U-shell	U-shell	TEMA E	TEMA F	TEMA E
Baffle type	double-segmental	double-segmental	single-segmental	double-segmental	single-segmental
Heat exchanger weight (only metal) (kg)	2.86e4 (2.47e4)	3.11e4 (3.03e4)	7.01e4 (5.71e4)	5.72e4 (4.22e4)	9.60e3
Total number of shells, N_s (-)	1	1	1	1	1
Heat exchanger cost, C_{Hx} (k€)	1109 (1019)	1207 (1267)	1554 (1437)	750 (568)	266

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330 The results show a pressure drop reduction on the shell-side for the dual-pressure SG
331 design. This is due to the reduction on the minimal allowable shell-side velocity on the
332 optimization algorithm. Hence, a reduction on the operational pump cost on the shell-side is
333 obtained on the SG. This leads to an improvement on the design of the heat exchangers
334 from the cost point of view. For instance, in the case of the SH, the heat transfer coefficient
335 on the shell-side changes from 5213 $\text{W}/\text{m}^2\text{ }^{\circ}\text{C}$ to 4143 $\text{W}/\text{m}^2\text{ }^{\circ}\text{C}$ without a significant change
336 on the overall heat transfer coefficient: 1241 $\text{W}/\text{m}^2\text{ }^{\circ}\text{C}$ and 1246 $\text{W}/\text{m}^2\text{ }^{\circ}\text{C}$ for 126 bar and 165
337 bar SHs, respectively.

338 Another important change in the dual-pressure SG design is the reduction of the
339 water/steam mass flow circulated in the EV_{HP} , where the circulation ratio ($CR =$
340 $\dot{m}_{\text{steam}}/\dot{m}_{\text{total}}$) decreases from 6.5 to 4.5. This means a reduction in the operational pump
341 costs on the tube-side for the EV_{HP} . This improvement is achieved decreasing the minimal
342 allowable velocity in the tube-side for the EV_{HP} . This is because the minimal allowable
343 velocity for two phase flow in horizontal evaporators decreases when the operating pressure
344 increases [44]. However, a relative high circulation ratio (around 20) is obtained for EV_{LP} .
345 This is mainly to fulfill the minimum velocity of a horizontal two-phase flow at 47 bar, which
346 is around 2.59 m/s [44].

347 The start-up energy consumption by the dual-pressure SG is also investigated. The following
348 assumptions are followed to carry out this analysis: (i) isothermal initial conditions at 290 $^{\circ}\text{C}$
349 in the SG and (ii) an energy of 76.5 MWtth to heat-up the steam turbine and the feed-water-
350 heaters [45]. As a result, the start-up energy consumption by the dual-pressure SG is around
351 86.9 MWhtth. This means an increment of 2.25% compared to the energy consumed in the
352 126 bar SG [45].

353 Although the heat exchangers have been designed following the standards considering
354 conservative factors of safety [25] [26], it is worth to mention that the fatigue damage
355 produced by the daily start-up cycles might be higher for the dual-pressure steam generator.
356 This is mainly due to the higher metal wall thicknesses, which leads to a higher thermal

stress if using 150°C/h as for the reference case [45]. Nevertheless, the reliability of the dual-pressure steam generator could be assured working with slower temperature ramp rates. The temperature ramp should be reduced a 45% up to 82.5°C/h for the dual-pressure level SG to obtain similar stress values to the reference case working at design conditions.

5.2. Reference STP compared to dual-pressure level STP

Once both SG are designed, their pressure drop shown in Table 3 is used to calculate the STP behavior. Figure 6 shows the temperature profiles of molten salt and water for the new SG design. It can be seen how the heat duty taken by the EV_{LP} moves the water temperature profile, allowing the pressure increase in the EV_{HP} while keeping $PP_{EV} = 2.5$ °C.

Figure 6. Temperature profile for the proposed dual-pressure level SG at $P_{HPT,in} = 165$ bar.

The drum pressure is set to 47 bar, which corresponds to a saturation temperature of 261°C, in order to avoid freezing problems of the salt in the EV_{LP} [46] [47]. In this sense, there is a low influence of the drum pressure on the power block efficiency. Energy and mass balances of the SG distribute proportionally the mass flow through EV_{LP} and EV_{HP} . For the operating conditions considered in Table 1, this mass flow is distributed in a ratio of 7% through EV_{LP} and 93% through EV_{HP} . Figure 7 compares the T-S diagrams of the reference and the dual-pressure level SG. As the power block layout is the same for both SG designs, the LPT behavior and extractions are kept constant.

Figure 7. Representation of the cycle shown in Fig. 2 in dotted-grey line ($P_{HPT,in} = 126$ bar) and the cycle shown in Fig. 4 in solid-black line ($P_{HPT,in} = 165$ bar).

Because of the dual-pressure level SG, the power block efficiency increases from 44.14% to 44.64% with a 14% increase of the SG costs (Table 3). Table 4 summarizes the performance of the new STP layout. The STP plant efficiency can be estimated as the product of $\eta_{STP} = \eta_{PB} \cdot \eta_{SF} \cdot \eta_{rec}$. As the power block efficiency is decoupled from the solar field and receiver efficiencies since it is only affected by the thermal storage tanks, the STP efficiency improvement comes from the new SG layout. To illustrate this, here we estimate the STP efficiency considering the solar field and receiver efficiencies at solar noon (Table 1), which have an accuracy of 2.5% [28] [32]. Thus, the new STP layout presents an efficiency of $\eta_{STP} = 17.42\%$. The new approach gives an increase of the energy produced of 5.71 GWh_e/year assuming 300 days of operation and 17 hours per day, which considers 11 hours of energy storage.

Table 4. Power block results of reference and new STP layouts.

STP	$\dot{m}_{FW,out}$ (kg/s)	$\dot{W}_{pumps}$ (MW)	$\dot{W}_{net}$ (MW)	Annual electricity (GWh _e /year)	η_{PB} (%)
Reference ($P_{HPT,in} = 126$ bar)	82.22	1.82	100	510	44.14
Dual-pressure level SG ($P_{HPT,in} = 165$ bar)	81.99	2.35	101.12	515.71	44.64

According to [48, 49], the Levelized Cost Of Energy (LCOE) for the reference STP presents a value of 12 c€/kWh for 2018. If the increase of 14% in the capital investment costs of the new SG is included, the LCOE of the dual-pressure STP is reduced up to 11.915 c€/kWh [50]. Such a reduction is caused by the increase of the energy produced.

Furthermore, the increase of 5.71 GWh_e/year produced by the dual-pressure level SG can be translated to an economic profit. This analysis is carried out assuming: (i) a power purchase agreement price (PPA) similar to Noor III PPA = 16.3 c€/kWh [51], (ii) a market pricing scenario of two-tier tariff, which presents a price multiplier over the PPA of 1 during the day and 2.7 during the evening [52] [53]. Therefore, if 5.71 GWh_e/year are produced on the evening for a LCOE = 11.915 c€/kWh, it yields an extra economic revenue of 623 k€/year.

6. Conclusions

A new dual-pressure level steam generator is proposed to increase the inlet pressure of the steam to the HTP for a STP. To carry out this pressure increase without violating the pinch point ($PP_{EV} = 2.5^{\circ}C$) or modifying the minimum and maximum temperatures of the molten salt ($T_{COLD} = 285^{\circ}C$ and $T_{HOT} = 565^{\circ}C$), the evaporation process is carried out in two pressure levels. Thus, the new dual-pressure level SG differs from current SG layout, which is formed by PH, EV, SH and RH, in adding a new evaporator at low pressure.

The heat exchanger designs of the new dual-pressure level SG are compared to the reference SG. The pressure increase produces an increment of the metal wall thicknesses the heat exchangers. As a result, the total heat exchanger mass of the new design increases, the energy needed during the start-up increases in a 2.25% while the temperature ramp rate should be reduced in a 45% during the start-up process. Furthermore, the new dual-pressure SG minimizes the pressure drop on the shell-side, reducing the operational costs of the molten salt pumps, and also, minimizes the pressure drop on the tube-side for the EV_{HP}. From an economic point of view, the overall cost of the new SG increases a 14%, although the LCOE is reduced from 12 c€/kWh_e to 11.915 c€/kWh_e due to the increase of the energy produced by the power block.

The thermodynamic analysis of the STP considers constant the energy captured by the molten salt in the receiver, and thus, keeps the solar field, receiver and power block designs unaltered. The results show an increase of the power block efficiency from 44.14 % to 44.64 % due to the pressure increase of the steam at HPT. The increase of energy produced of 5.71 GWh_e/year yields an extra economic benefit of 623 k€/year assuming a PPA = 16.3 c€/kWh_e and a market pricing scenario of two-tier tariff.

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