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CSP-PT gas plant using air as Heat Transfer Fluid with a packed-bed storage section

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Abstract

Concentrated Solar Power technologies represent an important alternative able to replace in a medium/long term fossil fuel sources. Current technology has several drawbacks which prevent a large diffusion: the principal one is the choice of the Heat Transfer Fluid which involves a certain complexity, including the heat storage section. Conventional plants in operation, consider diathermic oil and, more recently, molten salts. The potential of gases as working fluid has been underestimated till now and its use has not still fully exploited. Using gas would determinate a simpler conversion section increasing reliability. The gas, as proposed by the authors, can expand directly in a series of inter-reheated turbines after a series of intercooled compressions, reaching an acceptable overall global efficiency of the conversion section. The paper describes the optimum choice for the thermodynamic cycle which approaches an Ericsson cycle, integrating it with a comprehensive mathematical model for the heating section of the gas inside the solar receiver. A Thermal Energy Storage section based on the use of a packed bed of rocks has been considered, merged at the plant to insure production continuity. The overall software platform for the plant can be used as design tool in order to set up most important alternatives related to the plant characteristics and specific parameters.

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Keywords: *Solar Energy, Concentrated Solar Power, Parabolic-Trough, Heat Transfer Fluid, Thermal Energy Storage.*

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1. Introduction

The most important challenge of the nowadays "Sustainable Development" is to find a meeting point between energy production, energy consumptions and CO_2 reduction. However, in spite of the commitments, energy consumption continues to raise (in 2016, +1.3% than the 2015 level, +17.8 % from 2006 to 2016) and the fossil fuel quota, moreover, increases (+10.9% of oil, +24.5% of gas, +13.3 % of coal from 2006 to 2016), [1]. Main concern of this evolution is the atmospheric CO_2 level. The scientific community puts on the stage various scenarios based on the relation "CO_2 ppm-temperature grow-environmental effects-mankind effects". Based upon current understanding, the stabilization of CO_2 at 450 ppm will likely result in a global equilibrium warming from 1.4 to 3.1 °C, with a best guess of about 2.1°C: this would require a reduction of current annual greenhouse gas emissions by 70-80% by 2100. Actual CO_2 concentration ranked at an unfortunate level of 409 ppm in 2018, so a very urgent intervention should be in the agenda of policy makers, [2]. Excluding nuclear and conventional hydro, renewables accounted for 3.2% of global consumptions, [1], despite +350% respect 2006. This discouraging result calls for new conversion technologies with greater potential and a strong industrial feasibility. Concentrated Solar Power (CSP) technologies seem to have these issues and could represent, if duly supported, a real renewable energy breakthrough. Making reference to the Parabolic Trough (PT), technology almost commercially proven, [3], a Heat Transfer Fluid (HTF) at high temperature (290-390 °C) which moves inside solar receivers acts as high temperature source, suitable to feed almost conventional thermoelectric power plants. An intermediate thermal storage system is needed in order to give continuity to the energy production. The downstream conversion section is quite conventional being done via a thermoelectric power station. So, CSP-PT technology matches downstream conventional generation technologies having upstream a section in which solar energy is captured and concentrated by means of collectors which produce a high temperature heat source. Literature production in recent years testifies a huge interest toward CSP, [4][5], particularly focusing on PT technology. As evidenced in [6], the technical potential of CSP-PT is much greater than the real electricity consumption of the World. Furthermore, the possible combination between electrical needs and thermal needs, together with the chance of meeting territorial needs (i.e. fresh water), encourage the interest toward this technology. Main attention in literature was focused on the HTF and related technologies to concentrate solar energy and on the thermal storage. Present technology considers Diathermic Oil (DO) whose properties have been studied and the majority of the plants under construction makes use of DO in spite of some important drawbacks: oil is pollutant, toxic, explosive, difficult to be managed, with a maximum working temperature of 420-450 °C. Piping components are not conventional and still deserve improvement in the specific sector. A step ahead will be done by Molten Salts (MS) [7]. Main advantage is the high thermal capacity and the possibility to increase maximum component temperature till to 580 °C. The reduced cost represents also an interesting feature; the main concern is the solidification temperature below 240 °C which requires a continuous monitoring and an energy assistance during night-time or low irradiation periods. The problem of chemical compatibility with the conventional alloys, limited to the thermo-mechanical stresses and strains induced (from high operational temperature and mechanical load), make components technology even more complex than the same for oil. These critical issues limited the industrial interest and the expected development which was predicted a decade ago and is still far from being achieved. Actually, the high thermal capacity of MS is used for thermal storage as for DO: this section also is characterized by a high costs continuous severe control and operational conditions. The use of MS also matches as conversion section a conventional steam power plant whose efficiencies are in the range of 20-30%, [7]. The presence of a condensing section of the steam introduces some limitations in desert areas which are the best candidate for CSP, [8] and this happens also when using DO. This paper will investigate the capabilities of using gas in the energy conversion section simplifying the plant layout and increasing its reliability. To insure a certain night-time continuity of the power generation, a thermal energy storage section based on the use of a packed bed rocks will be also examined.

2. DEC layout and sensitivity analysis

2.1. Gases as Heat Transfer Fluid and plant layout

A real step change for PT-CSP plants could be represented by the use of gases as HTF, [8][10][11]. All the plant components in the solar field will take benefit of this choice in terms of cost, reliability maintenance, safety, etc. Plant size could be really reduced and several operational aspects would be simplified, being more conventional with respect to the DO and MS use. The use of gases calls for new conversion sections with respect to those presently considered, [12][13]. Considering this option, typical gas cycles can be considered as reference, [14]. In order to compare cycle efficiency, considering that maximum temperature must be limited with respect to those typical of the gas turbine plants, a candidate cycle layout would be based on:

- a sequence of inter-cooled compression tills to a given maximum pressure;
- a sequence of inter-re-heated expansions. The principal heating is represented by solar fields;
- a regeneration phase which increases the air temperature at the compressor exit, before the heating inside a solar field. The heating fluid is represented by the air leaving the last turbine at ambient pressure;
- external heaters, fed by biomass, used to compensate temperature variability and define its maximum value.

Hence, the thermodynamic transformations which happen inside the plant approaches an Ericsson cycle, in which the compressions and the expansions tend to approach an isothermal transformation: this happens through a sequence of adiabatic and intercooled isobaric transformations (DEC, Discrete Ericson Cycle), [15]. Authors already presented a mathematical model which receives as input solar radiation and gives as output the temperature of the HTF. The model privileges a global approach and the possibility of quickly setting up all the different and characteristic parameters of a CSP plant [16][17].

A more detailed analysis is presented in this paper discussing an optimization which defines the best number of compressions and turbines which characterize the compression and expansion phases. These aspects have been discussed and integrated with a thermal storage system based on packed bed technology which has been fully integrated into the CSP plants. Thanks to a mathematical modeling of the charging and discharging phases, continuity in the electrical energy production has been insured. For a given mass flow rate of the working fluid (air), the thermal storage has been designed. In order to insure steady conditions during the charging phase and also to restore design conditions at the turbine inlet despite the radiation fluctuation, an external heater fed by biomass has been considered. This stage is also useful to assist the plant during transients (plant startup) or during day-night time transition. The mathematical model of the plant, mainly referred to solar receivers and thermal storage sections, allowed to produce a model-based design of the overall plant allowing to include some operating plant conditions of particular interest from an engineering point of view. In fact, main concern when gases are used as working fluid is represented by the variation of its speed during the heating process. This causes that, when a maximum speed value is reached (25 m/s), the flow rate must be split into branches which operate in parallel. Consequently, the performances of the solar receivers must be re-calculated, considering different heat transfer coefficients, pressure losses, etc. Maximum metal temperature of 580 °C has been considered as limiting factor too, increased by the low heat transfer coefficient which are the consequence of flow rates splits and lower operating pressures, [18].

2.2. Sensitivity analysis

Having fixed a number of expansions in 2 stages, [16], the choice of an optimized plant layout that consider number of compressions, maximum pressure, specific power and net plant efficiency is evaluated through a sensitivity analysis, *Figure 1a* and *1b*, in which the solar irradiation is kept constant (900 W/m^2) and the length of the solar collectors is not consider a technological constraint. In fact, actual technologies consider modular length of the solar plants with step of 12 meters each. *Figure 1a* highlights the effect of the number of compression stages and maximum pressure on plant performances, in terms of net efficiency and specific power. Increasing the numbers of compression stages, the performance gain decreases and becomes less significant above a maximum pressure of 30 bar. *Figure 1b* confirms these observations, also highlighting that maximum pressures lower than 30 bar reduces the contribution of solar power below 75% of the total renewable power introduced (solar + biomass). Hence, the optimized layout chosen has 2 stages of expansion, 4 stages of compression and a maximum pressure of 30 bar. This choice finds a further confirmation if the ratio between this power and the receiver length is considered: in fact, representing the receiver length an important contribution of the plant cost, an optimization is suitable in order to maximize the plant power per unit of the receiver length ratio [17]. According to the previous analysis, an optimum plant layout has been chosen, with 4 compression stages and 2 expansion stages for a maximum pressure of 30 bar. *Figure 2* shows the transformations in the T-s plane, having fixed maximum temperature and pressure and suitable values for the component efficiencies and pinch points at the heat exchangers. Initial temperature is 40 °C and base pressure is 1 bar.

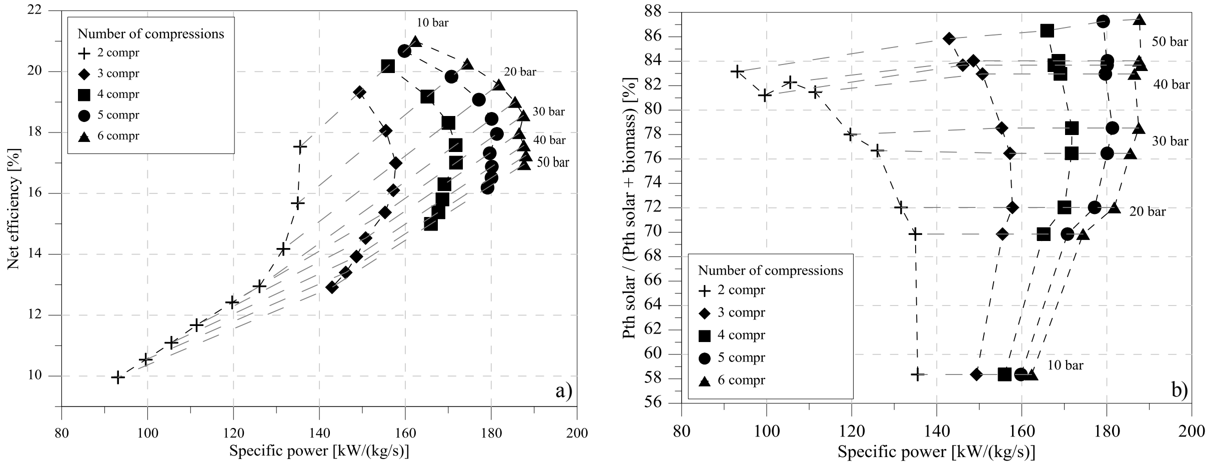

Figure 1: Net efficiency (a) and solar contribution (b) per specific power for DEC layout sensitivity analysis.

The simulation considers 1 kg/s of air inside the receivers. *Table 1* reports main data concerning the plant, including the heat absorbed, the power produced and the plant efficiency: DEC-based plant demonstrates high electrical efficiency (about 17%) which is really interesting if compared with actual technology for the conventional CSP plants that using oils and salts has HTF, in order to 15-18% [7].

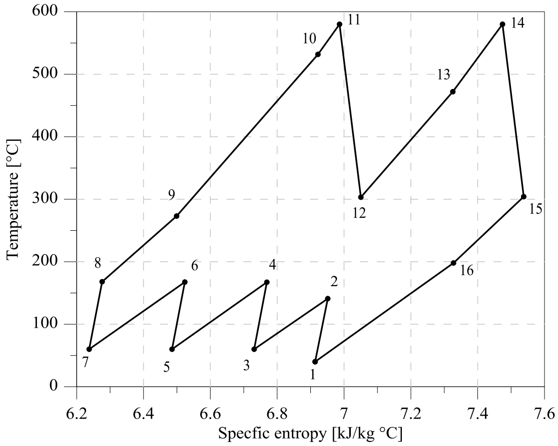

Figure 2: T-s cycle for the optimized DEC plant.

Table 1: DEC-based plant parameters.

Parameter	Value	Quantity
Temperature levels	40-580	°C
Pressure levels	1-30	bar
Mass flow rate	1	kg/s
Plant Power	170	kW
Collector length	12	m
Number of collector modules	12	-
SF total length	144	m
Number of compressions	4	-
Number of expansions	2	-
SF thermal efficiency (average)	61,4	%
Cycle efficiency	26.9	%
DEC efficiency	17.1	%
Solar share	75.2	%
Biomass boiler efficiency	70	%
P/L ratio	1,175	kW/m

The behaviour of the optimized plant layout, shown in *Figure 3*, has been then considered in two typical days, representative of summer (June the 21th) and winter (December the 21th) fluctuation of solar irradiation. *Figure 4* reports the air temperature at the solar collectors exit and solar/biomass power introduced in the DEC-based plant, that produces a constant mechanical power of about 170 kW from 6 am to 6 pm each day. As shown, the solar field contribution is relatively low in the winter day, accounting for about 15.0% of energy introducing daily in the DEC-based plant (54.5% in the summer day). Moreover, significant margins of heat recovery at low-medium temperatures (up to 60-180 °C) exist in the inter-refrigeration stages (A, B, C, D; about 15-20 kW) and at medium-high temperatures (350-580 °C) from flue gases of biomass combustion downstream the air heaters (E, F; about 7-10 kW), considered to assure the maximum peak temperature (580 °C) at the turbines inlet. These heat recovery margins could be used at medium temperature to feed an ORC plant (for a R245 organic fluid, in specific temperature range and mass flow rate, it is possible to hypothesize a power recover in the order of 10%, [21]) and at low temperature to feed desalination plants for fresh water production [19][20]. Figure 4 shows, according to the two days considered, the temperature of the air exiting the solar field - Figure 4a -from the LP and HP fields: the remaining temperature increase till to 580 °C is given by the external heater biomass-fuelled. In Figure 4b the thermal power exchanged inside the solar fields and the remaining thermal power to be given by the external heater: this plant choice was made in order to keep constant the conditions of the air at the entrance of the turbines. A limited off design could be also accepted if a reduction of the contribution given by the heaters is the goal. Data reported in Figure 4 are the result of the comprehensive mathematical model of the solar receiver presented by the Authors in [16][17], in which efficiencies are evaluated using Equations 1-3.

$$\eta_{therm} = \frac{Q_{in\_coll}}{Q_{sol}}; \quad \eta_{cycle} = \frac{L_{exp} - L_{compr}}{Q_{in\_coll} + Q_{in\_boil}}; \quad \eta_{DEC} = \frac{L_{exp} - L_{compr}}{Q_{sol} + \frac{Q_{in\_boil}}{\eta_{boil}}} \qquad \text{Eq (1-3)}$$

Assuming the operation of the plant only when the solar source provides a sensible contribution (from 6.00 am to 18.00 pm), the average of solar contribution is quantifiable of the order to 50% without considering the eventual recovery produced by ORC-based power units.

2.3. The importance of a storage section in a CSP plant

The plant layout in *Figure 3* highlights, also, a storage section which replaces collector thermal production during night-time and the management of transients for solar radiation irregularities: this is a crucial point in a CSP plant for the energy production.

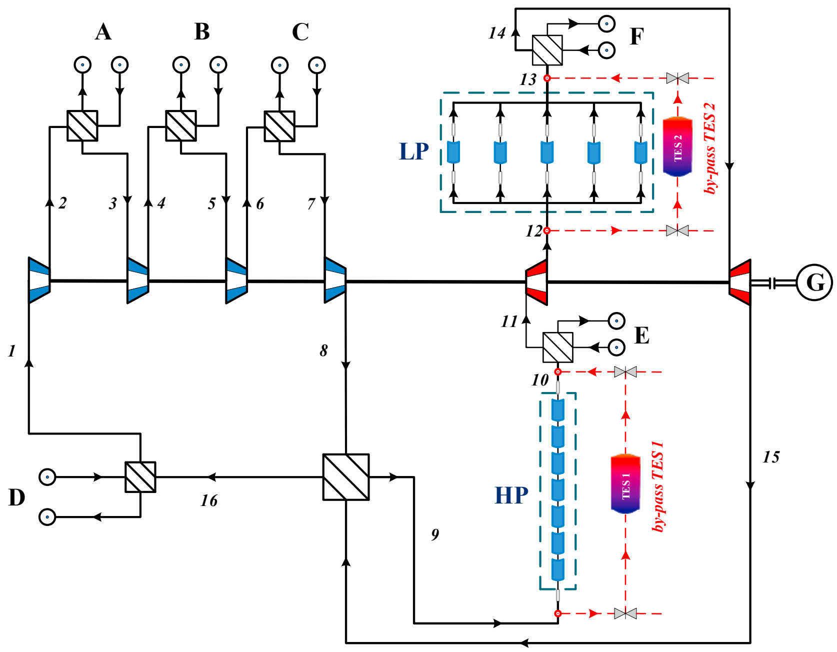

Figure 3: Optimized DEC plant layout

In fact, the energy storage in CSP plants has been considered since the first plants put in operation: several papers [21][23][24][25] presented modeling approaches of TES (Thermal Energy Storage), using various methods for storing heat, especially in form of sensible heat. The use of molten salts seems to represent the most efficient method, being it also candidate as HTF. In case of gaseous HTF, however, packed bed systems appear to be the right choice. As recently presented in [23][25] using a packed bed system with some precautions, performances comparable with the storage with MSs or other systems (oil) seem to be obtainable. In particular, in [25][26] the problem of hysteresis is presented, but when charging and discharging phases are done at the maximum temperatures, this aspect is limited and is in agreement with others papers, [23][24], so a packed bed technology has been considered in this paper in order to propose a preliminary integration hypothesis with the DEC plant.

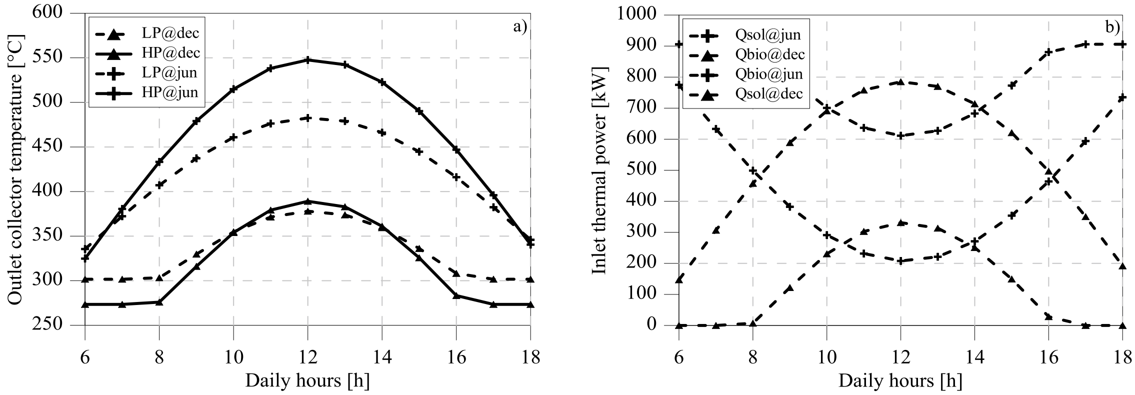

Figure 4: Air temperature at the exit solar field exit (a); thermal power exchanged in the solar field and added by the external heater (b).

3. Thermal energy storage section in DEC plant

The thermal storage technology presented is based on the storage of heat by a porous packed bed of rock heated by the gaseous HTF coming from a dedicated solar field and cycle, opportunely designed. The porous rock bed is contained inside an insulating tank. It is “thermally” charged during the day and is discharged during the night, so insuring a partial night-time electrical energy production depending of the tank size and specific parameters (mass flow rate, pressure and temperature).

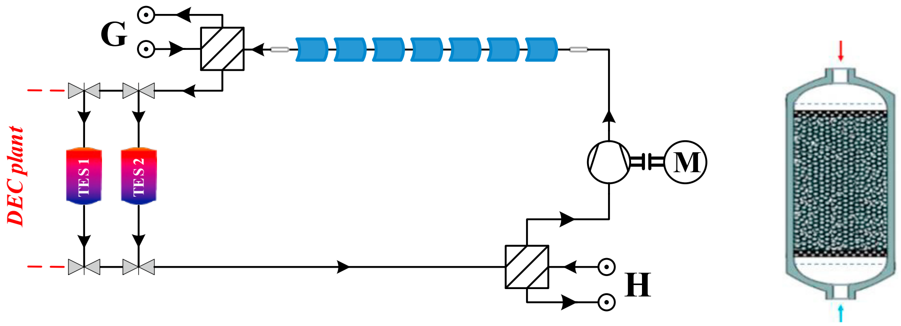

Figure 5: Thermal energy storage dedicated plant layout (a) and tank scheme (b).

During the charging phase, these rocks are heated by a flow of air that crosses them from top to bottom, obtained through an ad hoc CSP solar system, sized to supply air at 550 °C towards the tanks. While during the discharge phase the air, at a low temperature, passes through this hot bed in the opposite direction, heating up, *Figure 5*. Once the first tank is loaded, the air flow is switched to the other. A compressor is also available to compensate for pressure losses along the cycle: plant characteristics are shown in the *Table 2*. The advantages are: a) the possibility of varying the maximum and minimum temperature of the rocks without problems; b) the use a low-cost solid material (Alumina) with a high specific heat; c) the possibility of directly inserting the tank in the DEC system avoiding heat exchangers. In addition, only one tank per solar field is needed because it represents, at the same time, the hot and the cold heat source. In fact, by inserting hot air on one side, the different density that the gas takes along the tank establishes a transition region of natural temperature, called thermocline, hence the name of the technology. Therefore, the upper part of the tank represents the hot source and the lower part the cold one. Another advantage of the porous packed bed is to prevent mixing, which is harmful to thermal stratification. The main disadvantages are the low heat exchange coefficient of a gas compared to a liquid HTF and the need to pressurize the gas with the consequent increase in compression work. The mathematical modelling of the tank is based on the equations formulated by Schumann [27] and has been carefully studied and validated with experimental data, [25][26]. This model allows to determine at any time the temperature of the air, of the rocks and the accumulated energy, both in charge and in discharge. Furthermore, after this validation, a sensitivity analysis was carried out, in order to size the main geometrical parameters of the tank (particle diameter, aspect ratio, mass flow) and obtain maximum efficiency, [26]. For the storage system sizing, the following considerations are made: a) the quantity of thermal energy to be stored comes from the study of the DEC cycle, in particular from its total efficiency and output useful power; b) to replace the two solar fields and provide thermal energy to the air, *Figure 5*, at least two tanks are needed, respectively for the 9-10 and 12-13 cycle sections. Furthermore, the air inlet temperature in the tanks is higher than the ambient temperature (around 270° C). This means that the porous packed bed must be kept at a minimum temperature equal to that of the air inlet, otherwise the fluid will give off heat instead of absorbing it. Ultimately, this fact involves the permanent accumulation of a part of energy in the tanks that must remain confined to keep the porous packed bed in temperature. In *Figure 6* the continuous curve represents the temperature of the packed bed when the tank is unloaded, while the dotted curve when the tank is loaded. The area between the two curves is proportional to the accumulated energy and can be released in the discharge phase. While the area below the continuous curve is the energy stored permanently in the tank. This sizing allows to store thermal energy for 3 hours of DEC operation (where the 2 charged tank operates in parallel, each for every expansion section), with a period of charging of 3 hours for each tank for a total of 6 hours charging phase.

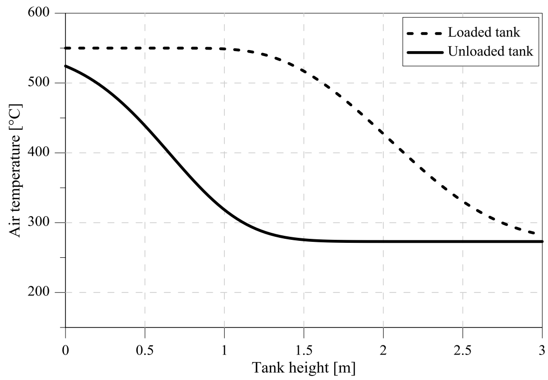

Figure 6: Charging and discharging trends for the storage section.

Table 2: Storage section parameters.

Mass flow rate (m)	1 kg/s
Tank inlet temperature (T_{max})	550 °C
Tank outlet temperature (T_{min})	270 °C
Pressure (p)	30 bar
Solar field length (L)	84 m
Diameter tank (D)	2 m
Tank height (H)	3 m
Diameter of Alumina rocks (d)	4 mm
Void Fraction (ε)	0.39
Charging time for each tank (T_{ch})	3 h
Discharging time for each tank (T_{dis})	3 h
Energy accumulated (E_{store})	894 kWh_t
Energy returned in discharge (E_{out})	815 kWh_t

The charge of the two tanks is carried out at the same pressure, while the discharge at different pressures. However, this does not affect the discharge time, which is the same for both tanks. As in *E* and *F* heat exchangers, *G* one is used in order to reach maximum peak inlet temperature in the tank that can't be reaches using solar collectors in every times of the considered day: considering the best irradiation hours, a contribution of solar field is in order of 72.4%. *H* heat exchanger provides a regulation in order to guarantee a constant inlet temperature in the solar field.
In the discharging phase, lasting 3 hours, the maximum temperature is near to the maximum one, with a biomass contribution negligible. Another aspect to underline is that each tank, slowly loading up, supplies air at outlet temperature above the minimum design temperature (270 °C). If excessive, this may result in a variation in the length of the solar field, in order to have the output always at 550 °C. However, in our case, as shown in *Figure 6*, the variation of the minimum T is in the order of a few degrees and this problem is therefore negligible too. Finally, to reduce charging and discharging phases, mass flow rate can be increased, so requiring a greater solar collector length. This is mainly due to the need of splitting the flow rate into parallel-operating solar receiver when a maximum allowable speed of the air is reached inside the solar receiver.

4. Conclusions

The paper goes deep in the use of gases (compressed air) as working fluid inside CSP-PT. This choice invites to make reference to gas cycles in the conversion section, simplifying it with respect to actual plants in which a thermoelectric plant is considered. In the conversion section, the compressed air is directly expanded across gas turbine stages and, among these stages, a reheating process is done by means of solar fields. Being the compression phase done by a a sequence of inter-refrigerated transformations, a regeneration stage can be done: this allows to increase air temperature at the solar field inlet and to save solar collectors lenght. So, the reference cycle is represented by an Ericcson Cycle approximated by a sequence of intercooled compressions and inter-reheated expansion, (DEC: Discrete Ericson Cycle). The paper considers a thermal storage section according to the packed bed technology. Biomass-fuelled external heaters are considered in order to keep the plant at constant temperature at the turbines inlet, during day and night times (3 hours).

Considering a compressed at 30 bar and 1 kg/s as flow rate with a maximum temperature equal to 580 °C, DEC-based plant (4 compressions and 2 expansions) has about 170 kW of mechanical power. The global efficiency is close to 20 % which could be further increased recovering the low-high grade thermal energy recoverable from the compressed air cooling and from the biomass-fueled external heaters.

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