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Optimization of a Decoupled Combined Cycle Gas Turbine Integrated in a Particle Receiver Solar Power Plant

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Abstract. The Next-CSP project aims at improving the performances of central receiver Solar Thermal Electric (STE) plants through the development and integration of a new technology based on high temperature (800°C) particles used as both heat transfer fluid and storage medium.

The objective of the present paper is to define the best combined cycle configuration by exploring various design options where Turbine Exhaust Temperature (TET) and compressor outlet temperature vary. Regarding the Dense Particle Suspension Heat eXchanger (DPS-HX) train that provides the heat input to the gas turbine's working air, a particular attention is paid to the particle-side layout.

A decent net cycle efficiency can be achieved by a two-reheat topping cycle with a TET of 600°C, with equal expansion ratios and a 3 pressure-reheat Rankine cycle at 160-20-3 bars / 585 / 575°C. Further increasing the low pressure (LP) turbine expansion ratio would result in a less bulky and cheaper LP DPS-HX, at the expense of cycle efficiency. Adjusting the HP and IP pressure ratios can simplify the particle-side layout of the DPS-HX train, again at the expense of efficiency (at least 0.5% pt).

With a two-reheat topping cycle, the particles cannot be expected to enter the receiver at a temperature far below 600°C. The impact of that high receiver inlet temperature and of the resulting moderate storage density on the plant's efficiency and economics should be discussed further. Similarly, a detailed techno-economic optimization of the DPS-HX train should be performed in order to define the optimal pressure drops and temperature differences.

INTRODUCTION

In state-of-the-art molten salts tower STE plants, the heliostat field, the solar receiver and the storage system account for roughly half of the Levelized Cost Of Electricity (LCOE) [1]. In order to reduce these costs, an option is to increase the power cycle's working temperature in order to increase its efficiency and hopefully the storage density. However, molten salts in their current formulation are already working at their maximum allowable temperature (565°C [2]). A change in heat transfer fluid (HTF) and storage medium is therefore necessary to increase both cycle efficiency and storage affordability.

The Next-CSP project aims at improving the performances of STE plants through the development and integration of a new technology based on high temperature (800°C) particles used both as heat transfer fluid and storage medium.

In the concept developed in Next-CSP, an upwards bubbling fluidized bed of particles (preferably olivine [3]) is heated up to around 800°C in a tube receiver. Hot particles can then be extracted from the storage tank and heat a working fluid (air, CO_2, steam…). Within the Next-CSP project, various validation-scale prototypes of the key components as well as a complete 1.5 MWe unit including a solar receiver and a gas turbine will be developed and tested at the THEMIS solar tower in Targasonne (France).

Amongst the options envisioned for a power block working with such a hot source, subcritical steam is too limited in efficiency [4], supercritical steam is expensive and available for large plants (>250MWe) only [5]. As for supercritical CO_2 cycles, they are not yet an industrial solution and suffer from an increased dependence on their cold source, which is usually mediocre in desert areas that are best suited to Solar Thermal Electric Power [6]. Therefore,

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a low-TIT (Turbine Inlet Temperature) CCGT (Combined Cycle Gas Turbine) appears to offer the best trade-off between performances and technological readiness.

Several studies about gas turbines and combined cycles for solar applications are available in the literature. In particular, Fraidenraich et al. [7] put forward the existence of an optimal TIT for combined cycles coupled with pressurized air receivers. Al-attab et al. [8] provided an extensive review of the currently operating externally heated Brayton cycles and of the associated high temperature heat exchangers. Andreades et al. [9], [10] proposed a design based on commercially available gas turbines (reheat-air Brayton combined cycle), aimed at solar or nuclear applications, with optional hybridization for peaking. Finally, Puppe et al. [11] compared several cycles to be coupled with a pressurized air receiver, including a low-TIT combined cycle. All the aforementioned studies either focus on systems without storage, or with regenerative storage that render reheating on the Brayton cycle impractical, and usually involve hybridization at least for peak production.

The objective of this paper is to assess various design options for a utility-scale (150 MWe) combined cycle gas turbine integrated in a STE plant using a storable hot source at ~800°C without hybridization.

OVERALL LAYOUT OF THE COMBINED CYCLE

A previous study [12] indicates that the most efficient topping cycle for such a low-TIT CCGT is a two-reheat gas turbine without intercooling, as shown in Figure 1. The bottoming cycle is typically a three-pressure, single reheat steam cycle (3P-RH). The remarkable feature of this base case is that all expansions have the same pressure ratio, close to the optimum value found by Siros and Fernández Campos [12] for a TIT of 800°C. Since the polytropic efficiencies are the same, the exhaust temperature is the same for every turbine (600°C) and the resulting compressor outlet temperature is 407°C. The assumptions made for the base case are summarized in Table 1.

The lower the pressure of the air going successively through the 3 stages (HP/IP/LP) of the DPS-HX train, the higher its volumetric flow. Keeping a constant relative air pressure drop ($\Delta P/P$) for the 3 stages would therefore require to dramatically increase the bulk of the LP (and, to a lesser extent, IP) exchanger. Therefore, a techno-economic trade-off leads to increasing the $\Delta P/P$ when the air pressure decreases, as shown in Table 1.

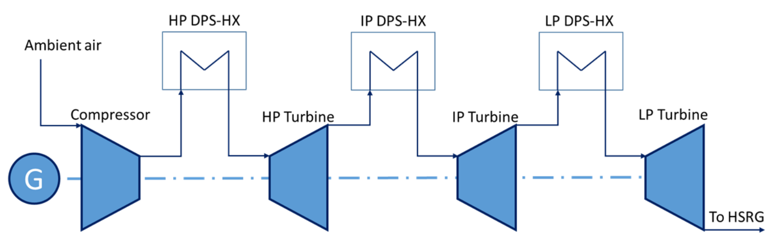

FIGURE 1. Layout of the hot air Brayton cycle envisioned

The additional power output provided by the bottoming cycle was defined as a function of the sole TET (Turbine Exhaust Temperature) to be added to the gas turbine's output (see paragraph 0), thus allowing to focus the study on the gas turbine only. Additionally, since the pressure losses in the DPS-HX (Dense Particle Suspension Heat eXchanger) are fixed and the HP and IP turbines have the same polytropic efficiency, both turbines have the same impact on the cycle's performances.

TABLE 1. Main assumptions and cycle parameters in the base case

Parameter	Value	Remarks
Turbine Inlet Temperature (TIT)	780°C	Considering particles exiting the receiver at 800°C
Ambient air temperature	25°C	
Polytropic efficiency of turbomachines	90%	Typical value
Absolute pressure drop at compressor inlet	10 mbar	Typical value for an industrial gas turbine (inlet air duct + filtration system)
Relative pressure drop at DPS-HX (% of the outlet pressure)	3% HP 5% IP 7% LP	See above why the $\Delta P/P$ increases when the air pressure decreases
Gas turbine exhaust pressure drop	30 mbar	Typical value, mainly Heat Recovery Steam Generator pressure drop
Gas turbine auxiliary consumption (% of the gross electric output)	1%	Accounts for transformer losses and miscellaneous auxiliaries
Generator nameplate efficiency	98.5%	

Starting from this base case layout, the most efficient combined cycle configuration will be determined by exploring various design options where TET and compressor outlet temperature vary.

CASE STUDY OF THE CYCLE'S MAIN PARAMETERS

Influence of the TET

As mentioned above, for a fixed air flow through the hot air turbine, the electric power generated by the bottoming cycle depends only on the TET of the topping Brayton cycle. Various bottoming steam cycles were designed using Thermoflow's GT-PRO software for temperatures ranging from 550 to 650°C, with an air flow fixed at 400 kg/s. The hypotheses are summarized in Table 2 and the T-Q diagram of the Heat Recovery Steam Generator (HRSG) designed for TET = 600°C (chosen as an example) is shown in Figure 2.

TABLE 2. Main hypotheses for bottoming cycle design in GT-PRO
(RH = Reheater, HPS/IPS/LPS = HP/IP/LP Superheater)

Architecture	3 pressure levels with reheat (3P-RH)
Ambient air temperature	25°C
Condenser temperature	45°C
Turbomachines polytropic efficiencies	Estimated by GT-PRO
Pinch at evaporators	5 K
Min superheaters approach temperature difference (HP+RH / IP / LP)	12 K / 17 K / 17 K
Relative pressure drops (HPS / RH / IPS / LPS)	3.5% / 7.0% / 2.0% / 5.0%

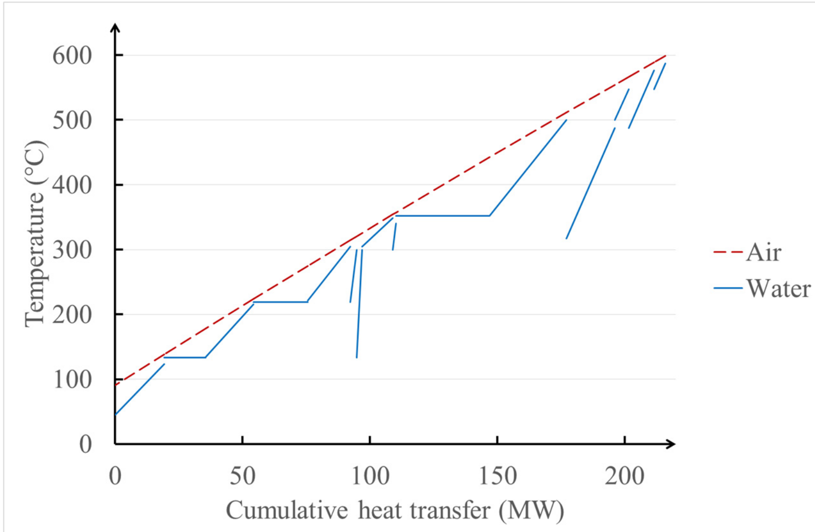

FIGURE 2. T-Q diagram of the HRSG designed for a TET of 600°C

The performances of the resulting bottoming cycles are plotted in Figure 3. In the temperature range considered, the net power output of the bottoming cycle appears to be almost linearly dependent on the TET, thus allowing the use of the following equation in order to evaluate the power generated by the bottoming cycle:

$$\text{Rankine cycle net power output (kW)} = 240.21 \times \text{TET(°C)} - 68645 \tag{1}$$

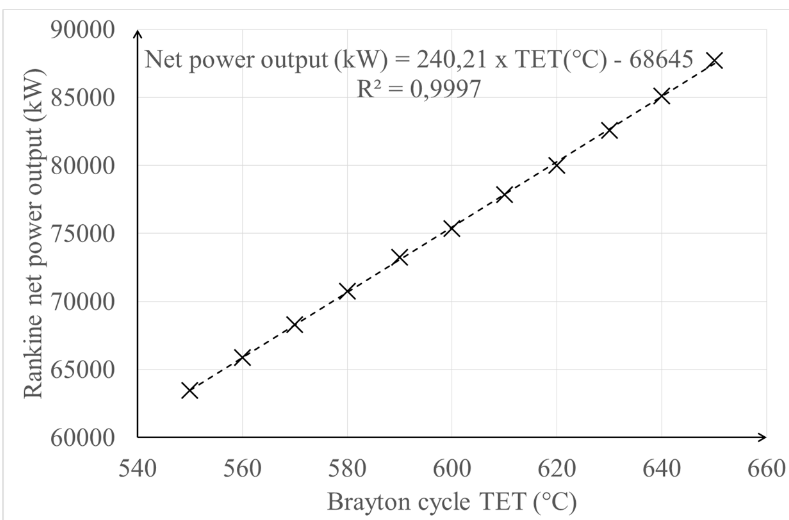

FIGURE 3. Net power output of the designed bottoming cycles for an air flow fixed at 400 kg/s

A sensitivity study on the pressure ratios (especially the LP one that defines the TET) led to the definition of a first alternate case that allows the maximum combined cycle efficiency for the considered TIT. Its parameters are described in Table 3 and its performances are discussed in paragraph 0.

Influence of the Compressor Outlet Temperature

The design of the cycle should also take into account the particle side of the exchangers. In the base case described above, the compressor outlet temperature is different from the HP and IP turbine exhaust temperatures. Therefore, the HP DPS-HX does not cover the same temperature range as do the IP and LP DPS-HX, and therefore cannot be strictly parallel to them. In that case, the particles are mixed at the same temperature before entering a complementary low-temperature HP DPS-HX with a high mean temperature difference, see Figure 4.

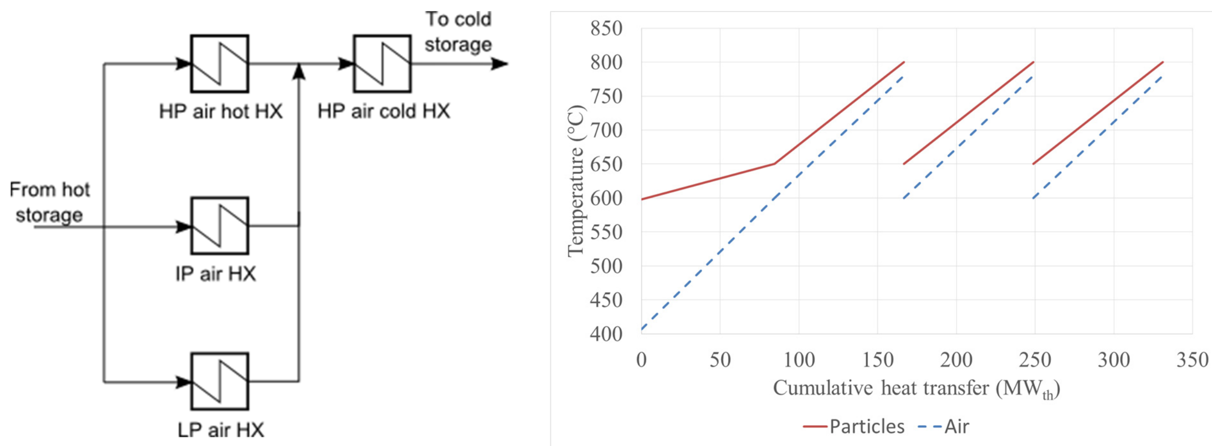

FIGURE 4. Particle path in the case of a complementary low-temperature DPS-HX, and corresponding T-Q diagram in the base case described in part 0, based on an assumed temperature difference of 50 K at every DPS-HX inlet.

This temperature mismatch between cold HP air and cold IP/HP air requires a complex layout (such as that shown on Fig. 4) and results in exergy losses due to high temperature differences. Another solution would be to mix the particles at different temperatures after the DPS-HX network. For a fixed total particle flow, this would result in the same particle exit temperature and exergy losses, but with a more expensive HP DPS-HX, since the logarithmic mean temperature difference would be lower.

Alternate case #1 aims at maximizing the combined cycle efficiency thanks to a higher TET that corresponds to a lower LP turbine expansion ratio. The HP and IP expansion ratios remain equal and become higher than the LP expansion ratio so that the compressor pressure ratio and consequently the compressor outlet temperature remain roughly identical to those of the base case.

The considerations mentioned in paragraph 3.2 led to the definition of three other case studies described in Table 3 as alternate case #2, #3 and #4. Alternate case #2 has the same TET as the base case, but the HP and IP expansion ratios are adjusted so that all DPS-HX have the same working temperatures. Alternate case #3 is the only layout for which all turbines have the same expansion ratio and all DPS-HX work at the same temperatures, given the assumptions made on the polytropic efficiencies and pressure drops in the cycle. Finally, alternate case #4 has the same TET as alternate case #1, which is supposed to maximize the combined cycle efficiency, but the HP and IP expansion ratios are again adjusted so that all DPS-HXs work with the same temperature range.

TABLE 3. Main parameters of the cases studied

	Base case	Alt. case #1	Alt. case #2	Alt. case #3	Alt. case #4
HP/IP turbine expansion ratio	2.3	2.6	3.1	2.85	3.35
LP turbine expansion ratio	2.3	1.8	2.3	2.85	1.8
HP/IP turbine outlet temperature (°C)	600	574	540	555	524
TET (= LP turbine outlet temperature) (°C)	600	649	600	555	649
Compressor outlet temperature (°C)	407	411	540	555	524

Performances of the Cases Studied

The cases presented in Table 3 were simulated with Thermoflex, a software developed by Thermoflow Inc., to perform steady-state simulations of power cycles. The resulting performances are given in Table 4.

TABLE 4. Performance of the cycles studied

	Base case	Alt. case #1	Alt. case #2	Alt. case #3	Alt. case #4
Brayton net power output (MWe)	85.6	84.7	80.8	80.2	80.0
Rankine net power output (MWe)	75.5	87.3	75.5	64.7	87.2
Total cycle power output (MWe)	161.0	172.0	156.3	144.9	167.2
Heat input (MWth)	331.3	352.6	326.7	306.3	348.0
Particles temperature at DPS-HX inlet (°C)	605	580	590	605	575
Air pressure at LP DPS-HX inlet (bars)	2.4	1.9	2.4	3.0	1.9
Cycle net efficiency	48.6%	48.8%	47.8%	47.3%	48.1%

In any case, adjusting the pressure ratios for a simpler DPS-HX layout (cases #2 to #4) implies a significant loss in cycle efficiency, namely at least a 0.5% pt loss compared to the base case. Alternate case #1 brings a small improvement compared to the base case in terms of efficiency. Yet it should be noted that the LP DPS-HX operates at a lower pressure, which would significantly increase the capital costs needed to keep the same relative pressure drop.

The outlet temperature of the particles, i.e. the temperature of the particles that enter the receiver, only varies in a 30 K range, meaning a ~15% change in storage density. That may also impact the receiver design, though it is only a second order parameter compared to the particles temperature at the receiver outlet.

Finally, the bottoming cycle produces about half of the plant's net power, whereas in common gas-fired combined cycles that share is usually close to 1/3. It may therefore be profitable to use a more expensive HRSG (i.e. with lower pinches and possibly lower pressure drops on the steam side).

CONCLUSION

Several options were examined for the design of a combined cycle using a two-reheat gas turbine externally heated by fluidized particles used as both heat transfer fluid and storage medium. Beyond electrical efficiency, a focus was made on the particle-side of the DPS-HX network and the working pressure of the LP DPS-HX.

The best efficiency of 48.8% is obtained for a high TET, and thus a low pressure at the LP DPS-HX. This translates into a more expensive LP DPS-HX, a heavier pressure drop across it, or both, as well as a more expensive HRSG since the thermal power to be recovered is higher.

However, a decent net cycle efficiency can be achieved with a TET of 600°C, with equal expansion ratios and a 3P-RH Rankine cycle at 160-20-3 bars / 585/575°C. It allows higher LP DPS-HX working pressures, allowing for cheaper HRSG and LP DPS-HX. It incurs a higher particle temperature at the outlet of the DPS-HX network, thereby lowering the storage density by around 12%. For lack of data on the industrial scale receiver design, the impact on the receiver's performance cannot be assessed, but it is expected to be minor.

Further increasing the LP turbine expansion ratio would result in a less bulky and cheaper LP DPS-HX, at the expense of cycle efficiency. Adjusting the HP and IP pressure ratios can simplify the DPS-HX particle-side layout, again at the expense of cycle efficiency (at least 0.5% pt).

With a two-reheat topping cycle, the particles cannot be expected to enter the receiver at a temperature far below 600°C if the topping cycle is not intercooled. However, intercooling would significantly penalize the cycle efficiency. The impact of that high receiver inlet temperature and of the resulting moderate storage density on the plant's efficiency and economics should be discussed further. Similarly, a detailed techno-economic optimization of the DPS-HX design should be performed in order to define the optimal pressure drops and temperature differences.

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REFERENCES

1. IRENA, "Concentrating Solar Power", in *Renewable Energy Technologies: Cost Analysis Series*, 2012
2. K. Lovegrove and W. Stein, *Concentrating Solar Power Technology: Principles, Developments and Applications* (Woodhead, Oxford, 2012)
3. J. Baeyens and H. Zhang, "Next-CSP Deliverable 1.1 – Report on particle selection for solar heat capture and storage", April 2017
4. R. Gicquel, *Systèmes Energétiques* (Les presses de l'Ecole des Mines, Paris, 2009)
5. J. H. Peterseim and A. Veeraragavan, "Solar Towers with Supercritical Steam Parameters - is the Efficiency Gain worth the Effort?", Energy Procedia 69 (2009), pp. 1123-1132
6. J. Dyreby, S. Klein, G. Nellis, and D. Reindl, "Design Considerations for Supercritical Carbon Dioxide Brayton Cycles With Recompression", Journal of Engineering for Gas Turbines and Power 136 (2014)
7. N. Fraidenraich, J. M. Gordon, and C. Tiba, "Optimization of gas-turbine combined cycles for solar energy and alternative-fuel power generation", Solar Energy 48 (1992), pp. 301-307
8. K. A. Al-attab and Z. A. Zainal, "Externally fired gas turbine technology: A review", Applied Energy 138 (2015), pp. 474-487
9. C. Andreades, R. O. Scarlat, L. Dempsey, and P. Peterson, "Reheat-Air Brayton Combined Cycle Power Conversion Design and Performance Under Nominal Ambient Conditions", Journal of Engineering for Gas Turbines and Power 136 (2014)
10. C. Andreades, L. Dempsey, and P. F. Peterson, "Reheat Air-Brayton Combined Cycle Power Conversion Off-Nominal and Transient Performance", Journal of Engineering for Gas Turbines and Power 136 (2014)
11. M. Puppe *et al.*, "Hybrid High Solar Share Gas Turbine Systems with Innovative Gas Turbine Cycles", Energy Procedia 69 (2015), pp. 1393-1403
12. F. Siros and G. Fernández Campos, "Optimisation of a Low-TIT Combined Cycle Gas Turbine With Application to New Generation Solar Thermal Power Plants", ASME Turbo Expo Proceedings (ASME, Charlotte, North Carolina, 2017)