12CNIT-2022 - FULL PAPER

Hybrid Brayton supercritical CO_2 power cycle using a gas turbine and a tower solar receptor with an ORC as a waste heat recovery system

José Ignacio Linares[1], Eva Arenas[1,2], Alexis Cantizano[1,2], Maria José Montes[3], Antonio José Rovira[3], José Porras[1]

[1]Rafael Mariño Chair in New Energy Technologies, Comillas Pontifical University, Alberto Aguilera 25 – 28015 Madrid, Spain, e-mail: linares@comillas.edu

[2]Institute for Research in Technology, Comillas Pontifical University, Santa Cruz de Marcenado 26 – 28015 Madrid,

[3] Department of Energy Engineering, Universidad Nacional de Educación a Distancia (UNED), Juan del Rosal 12, 28040 Madrid, Spain

Keywords: *Regenerative gas turbine; CSP; ORC; combined cycle; supercritical CO_2 power cycle*

TOPIC: RENEWABLE ENERGIES, ENVIRONMENTAL IMPACT AND CIRCULARITY

1. Introduction

Spain has a large installed power capacity of concentrating solar power (CSP) plants and a relevant scientific production in that field. The current share in the electricity mix is low (around 2%, with a total installed power capacity of 2300 MWe) [1]. A significant growth in that installed power is expected in the following decade, supported by the dispatchability of CSP plants based on thermal energy storage systems (TES). As an effort to reduce the high current levelized costs of electricity of CSP, integrated solar combined cycles (ISCC) have been proposed. Their designs are based on the solar integration, in parallel to the boiler, by means of parabolic trough collectors (PTC) [2]. Economic feasibility analysis of these solutions concluded that incentives were necessary [3]. Other solar integration systems have been analyzed, including a central tower receiver (CTR) in parallel to the exhaust gas of a gas turbine or preheating the air coming from the compressor. All of these choices are less developed than the conventional ISCC with PTC, not being usual to include any thermal storage as the solar energy is used as a secondary contribution [4].

The current state of the art in ISCC lies in non-conventional advanced power cycles for heat recovery. Although the Rankine cycle is the most widespread and commercially available, other power cycles with lower complexity have been proposed, such as the Organic Rankine Cycle (ORC), suitable for intermediate to low temperatures. Chacartegui et al. presented one of the first works in this field [5], postulating the advantages of using a recuperative gas turbine, after further developments [6]. Supercritical CO_2 Brayton power cycles (S-CO2) have also been proposed for CSP, taking advantage of its suitability for intermediate-high temperatures. So, the National Renewable Energy Laboratory of the United States, in the Concentrating Solar Power Gen3 Demonstration Roadmap [7], stated the key role of the Brayton supercritical CO_2 power cycle as an efficient energy conversion system.

The project AdInCCSol (Advanced integration of combined cycles in solar thermal power plants) aims to study and develop four different solar technologies with four corresponding advanced thermodynamic cycles [8]. This paper shows the first steps in defining one of these technologies, being the Brayton supercritical CO_2 power cycle (S-CO2) coupled to a gas turbine cycle, the key of this hybrid configuration.

2. Materials and method

2.1. Layout

Figure 1 shows a conceptual diagram of the proposed power plant. A recuperative gas turbine (upper part in Figure 1) is used as the topping cycle of a combined one, whereas an S-CO2 is used as the bottoming cycle (central part in Figure 1). The S-CO2 power cycle receives heat from two thermal sources: a high-temperature and an intermediate-temperature source. The high-temperature thermal source consists of both the turbine outlet in the topping cycle (High Temperature Heat Recovery, HTHR) and the solar receiver (Central Tower Receiver, CTR). The intermediate-temperature thermal source (Intermediate Temperature Heat Recovery, ITHR) consists of the flue gases. These are obtained after mixing the gases coming from the gas turbine recuperator and those coming from the heat recovery boiler (gas turbine outlet). The waste heat available downstream the intermediate-temperature thermal input (ITHR) is collected in the low-temperature heat recovery (LTHR) and converted into electricity by means of an organic Rankine cycle (ORC). Heat rejection is carried out by a cooling tower through the precooler (PC) in the S-CO2 power cycle and through the condenser (COND) in the ORC.

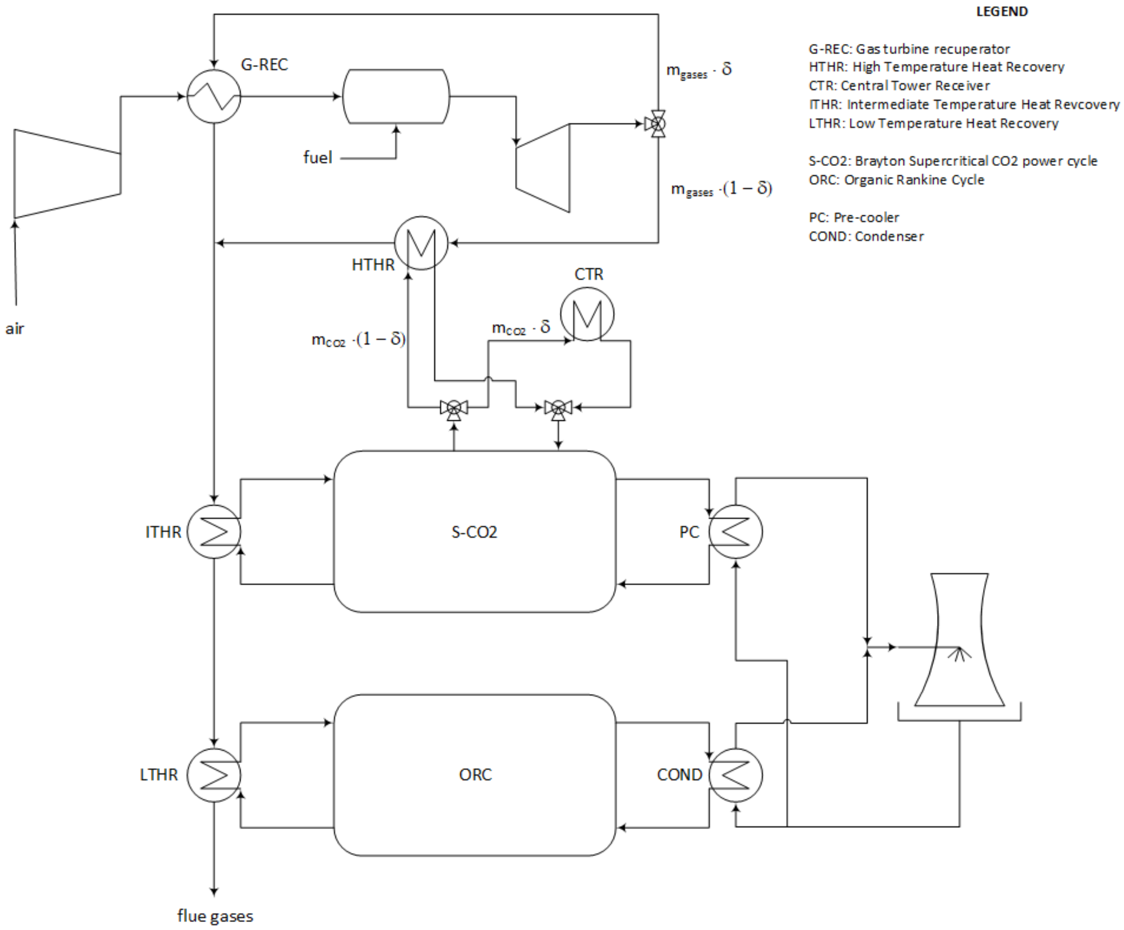


Figure 1. Conceptual layout.

As pointed out in Figure 1, the dispatchability of the plant is achieved by combining the solar radiation and the gas turbine flue gases as the high-temperature thermal source of the S-CO2 cycle. The flue gases leaving the gas turbine are split into two streams, so a mass flow rate fraction δ is sent to the recuperator and 1-δ to the HTHR. Such fraction is determined by the solar radiation: the higher radiation, the higher value of δ. Thus, high radiation leads to a low mass flow rate of flue gases through the HTHR and a high one through the gas turbine recuperator, reducing the natural gas consumption. This concept has been designated as partial recuperation [9].

The use of a flue gases stream as a thermal source entails the use of a cross-flow heat exchanger (both streams unmixed), flowing the gases outside the tubes and CO_2 (HTHR or ITHR) or the organic fluid (LTHR) inside of them. In order to maximize the S-CO2 efficiency, 300 bar is established as the highest pressure, which would require a large tube thickness. One way to avoid that is locating the thermal input of the S-CO2 power cycle in the low-pressure side [10]. Figure 2 explains this concept, where the recuperator HTR transfers the thermal energy taken downstream the turbine by the HTHR/CTR to the turbine inlet. As the low pressure in S-CO2 is 85 bar, no special features in those heat exchangers are required; moreover, that pressure allows using a direct solar receiver, that is, CO_2 can be used as the working fluid in the power cycle and also as the cooling medium in the receiver. The solar receiver has been designed for this application, based on both the microchannel [11] and the star receiver [12] concepts. Therefore, the receiver consists of a radial structure of four absorber panels converging on the central axis of the tower. Each absorber panel is a compact structure of microchannels, which receives solar radiation from both exposed surfaces, thus reducing the thermal gradient along the panel thickness. The microchannel concept is especially suitable for pressurised gaseous fluids and the star receiver is a light-trapping geometry that reduces radiation losses, which are especially important when working at such high temperatures. The supercricital CO_2 enters the receiver at 510ºC and leaves at 700ºC, with a thermal efficiency of 88.5%.

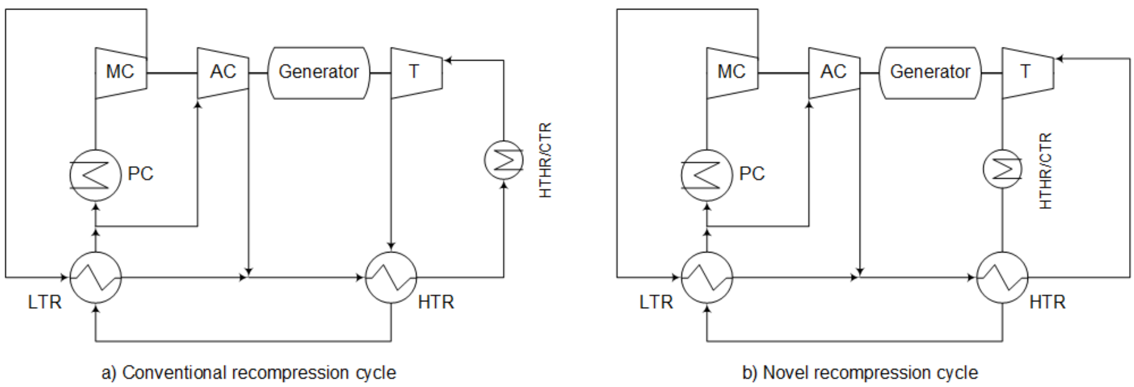


Figure 2. Conventional recompression layout (a) versus modified one (b) employed in the proposed power plant to supply high-temperature thermal energy to the S-CO2.

Both layouts given in Figure 2 show the recompression concept, i.e. using an auxiliary compressor (AC) before the precooler (PC) to reach the required high pressure and a temperature comparable with that in the LTR high-pressure outlet. This arrangement makes it possible to use a lower mass flow rate in the high-pressure stream of the LTR, so compensating its higher specific heat and achieving a balanced temperature profile in this heat exchanger. The implementation of the S-CO2 in the proposed power plant replaces the heating effect of the auxiliary compressor by the heat transfer from the ITHR. The heat is now supplied through an intermediate loop which allows the use of a suitable pressure (85 bar) inside the tubes of the ITHR. This arrangement can be seen in Figure 3, which includes the entire subsystems (gas turbine, S-CO2 and ORC) layout. The split fraction (α) at the compressor outlet is determined to obtain the same temperature approach at both extremes of the LTR. The dual high-temperature thermal source (flue gases and solar receiver) entails the use of two HTR recuperators. The management of the flue gases imposed by the solar radiation allows maintaining nearly constant conditions at the S-CO2 turbine and compressor.

Regarding the ORC, a regenerative transcritical cycle is proposed. The highest pressure is 50 bar, so the working fluid directly flows inside the tubes of the LTHR.

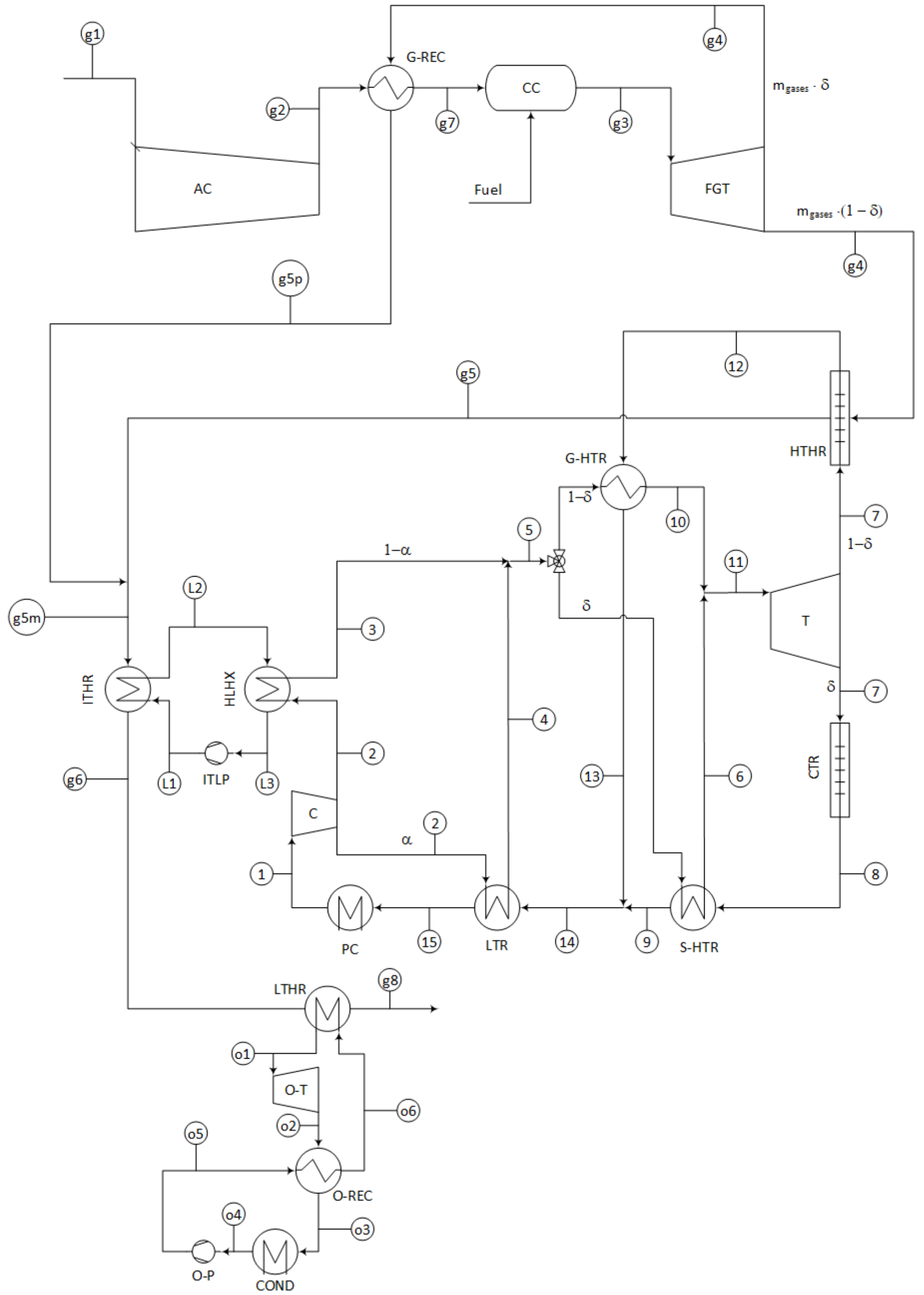


Figure 3. Layout of the proposed power plant detailing all the systems components.

2.2. Main hypothesis

This paper deals with the design performance. That is, cooling water and air compressor inlet conditions are set to the design values, whereas the radiation contribution (measured by means of the flue gases split fraction, δ) varies. Cooling water comes from the cooling tower at 25 ºC, and its mass flow rate is changed in order to maintain 35 ºC as the CO_2 compressor inlet temperature in the S-CO2 cycle and the condensation temperature in the ORC. Ambient conditions (inlet conditions of the gas turbine compressor) are set to 15 ºC and 1 bar.

The compressor and turbine isentropic efficiencies in the gas turbine cycle are set to 85% and 90%, respectively. Fuel is assumed to be pure methane, coming into the combustion chamber at 25 ºC and at a convenient pressure. Complete combustion is assumed. The turbine inlet temperature is set to 1,500 ºC. At the design point, the exhaust gases temperature (g4) is set to 950 ºC and the net power to 100 MWe. The air mass flow rate and the pressure ratio derived from these conditions are maintained constant. Pressure drops are neglected [13].

In the S-CO2 cycle, the inlet temperature at both HTR low-pressure streams (points 8 and 12 in Figure 3) is set to 700 ºC. The design temperatures approach are 10 K at HTRs, 5 K at LTR and 30 K at HTHR. Compressor inlet conditions are 85 bar and 35 ºC, whereas compressor outlet pressure is 300 bar. A pressure drop of 40 kPa is assumed in the CO_2 streams of the heat exchangers. Compressor isentropic efficiency is fixed to 87% and 92% for the turbine. Intermediate-temperature loop pump (ITLP) isentropic efficiency is 75% [10].

In the ORC, the chosen working fluid is R600 (n-buthane) due to its properties, mainly because it is a natural refrigerant with null ODP and low GWP. Saturation pressure at 35 ºC is 3.3 bar, which avoids the air entrance in the condenser. In contrast, critical pressure is 39 bar, which makes it possible to establish a transcritical cycle at an intermediate value of the turbine inlet pressure. Turbine isentropic efficiency is assumed to be 80%, and the efficiency of the pump 75%. At the design point, turbine inlet conditions are set to 50 bar and 200 ºC, the temperature approach at the recuperator to 10 K and the LTHR flue gases outlet temperature to 115 ºC. Pressure drops are neglected [14].

2.3. Performance model

Ambient conditions and cooling temperature are maintained constant, whereas solar radiation changes (measuring this variation through the gases split fraction, δ). In the S-CO2 power cycle, this radiation variation entails significanty variable conditions at the two HTRs (with higher duty in the S-HTR than in the G-HTR for a large solar contribution) and at the high-temperature thermal sources (HTHR and CTR). On the other hand, LTR, PC and HLHX heat exchangers conditions are nearly constant, as well as at the CO_2 compressor and the turbine. In fact, there will be some slight variations due to the adjustment of the mass flow rate in order to maintain 700 ºC at points 8 or 12 (Figure 3).

An energy balance and Second Law restrictions are imposed at the main components. Compressors and pumps are assumed adiabatic and governed by equation (1), where w stands for specific work, h for enthalpy and η for isentropic efficiency. Subscripts i stands for "inlet", o for "outlet", *c/p* for "compressor/pump", and s represents the state with the same entropy as the inlet and the same pressure as the outlet. Turbines are also assumed adiabatic and governed by equation (2), where subscript T stands for "turbine", and the rest of notation is the same as in equation (1). An energy balance in the heat exchangers is described by equation (3), where $\dot{m}$ stands for mass flow rate, $\dot{Q}$ for heat power exchanged, and the subscripts h and c stands for "hot" and "cold" respectively; the rest of the notation is the same as in equation (1).

$$w_{c/p} = \frac{h_{o,s}-h_i}{\eta_{c/p}} \quad (1)$$

$$w_T = \eta_T \cdot \left(h_i - h_{o,s}\right) \quad (2)$$

$$\dot{Q} = \dot{m}_h \cdot (h_{hi} - h_{ho}) = \dot{m}_c \cdot (h_{co} - h_{ci}) \quad (3)$$

Regarding the flow splitters, α is determined by balancing the LTR, whereas δ is imposed by the solar contribution.

For the operation of the heat exchangers, that is, their performance when mass flow rates vary, the model of Patnode has been followed, along with the ε-NTU method [15]. Their expressions have been taken from EES libraries [16], the simulation environment where the entire model has been implemented. Equation (4) gives the relationship between the thermal conductance (*UA*) and the mass flow rates at design (subscript *N*) and off-design conditions, where the rest of the notation has been defined in equations (1) and (3). This model has been applied to the heat exchangers affected by the solar contribution variations: HTHR, G-HTR and S- HTR in the S-CO2 power cycle, G-REC in the gas turbine and LTHR and O-REC in the ORC. Mass flow rates are nearly constant in the LTR, HLHX, and ITHR. Cooling water mass flow rates in the PC and COND vary to maintain CO_2 and R600 temperatures constant, having a low variation in their duty.

$$\frac{UA}{UA_N} = \frac{\frac{1}{\dot{m}_{c,N}^{0,8}} + \frac{1}{\dot{m}_{h,N}^{0,8}}}{\frac{1}{\dot{m}_{c}^{0,8}} + \frac{1}{\dot{m}_{h}^{0,8}}} \tag{4}$$

Equations (5) to (7) calculate the net power of different systems: gas turbine (GT), Brayton supercritical CO_2 (S-CO2) and ORC. Subscripts are defined in Figure 3. Equation (8) gives the combined cycle efficiency, taking into account the overall net power and the heat power input from both the combustion chamber (CC) and the central tower receiver (CTR).

$$\dot{W}_{GT} = \dot{W}_{FGT} - \dot{W}_{AC} \tag{5}$$

$$\dot{W}_{S-CO2} = \dot{W}_{T} - \dot{W}_{C} - \dot{W}_{ITLP} \tag{6}$$

$$\dot{W}_{ORC} = \dot{W}_{O-T} - \dot{W}_{O-P} \tag{7}$$

$$\eta_{CC} = \frac{\dot{W}_{GT} + \dot{W}_{S-CO2} + \dot{W}_{ORC}}{\dot{Q}_{CC} + \dot{Q}_{CTR}} \tag{8}$$

3. Results

In the gas turbine, the air mass flow rate obtained in the compressor is 195.1 kg/s with a pressure ratio of 6.75, both determined without considering recuperation, that is, without solar contribution (δ = 0). Figure 4 shows the net power breakdown of each subsystem as a function of the solar contribution. Each of them exhibits a flat behavior, reaching an overall production of 180 MWe (56% from the gas turbine, 39% from the S-CO2 and 5% from the ORC). Although the combined cycle efficiency presents a maximum with a solar contribution of 50%, the absolute value ranges from 56.8% to 57.5%, that is, a nearly constant value with an average of 57.2%. The solar contribution allows reducing the CO_2 emissions from 346 g CO_2/kWhe (without solar contribution) to 236 g CO_2/kWhe (with maximum solar contribution). A maximum reduction of 32% can be achieved.

Figure 4 confirms the validity of the assumption of not considering the variation in the thermal conductance in both the precooler (S-CO2 heat release) and the condenser (ORC heat release), as all the power cycles maintain a flat behavior. The same assumption has been made in other heat exchangers of S-CO2 (LTR, HLHX and ITHR), based on a nearly constant CO_2 mass flow rate. Such a hypothesis can be verified in Figure 5, where the maximum variation of this mass flow rate with solar contribution is lower than 2.7%.

Figure 6 shows the effect of the solar contribution in the T-Q profile of both HTHR and ITHR. That effect can be appreciated in the heat transferred in the HTHR, which entails a higher slope in the plot at the right. However, extreme temperatures are maintained. A similar effect can be observed at the gas turbine recuperator (Figure 7), where a high temperature approach can be seen. Such high temperature approach is due to the fact that the thermal conductance of this recuperator is determined at the maximum solar contribution point to obtain the same temperature at point (g5p) than at point (g5) with the minimum solar contribution. This sizing allows to reduce the irreversibility in the mix of streams (g5) and (g5p) at intermediate solar contributions.

Figure 8 shows T-Q profiles at the G-HTR in two different cases of solar contribution (S-HTR would be the reciprocal plots). Although this heat exchanger cannot be balanced (both streams have the same mass flow rate of CO_2), the high operation temperature produces similar specific heats at both streams, and the temperature approach reached at the hot stream outlet is nearly maintained through the heat exchanger.

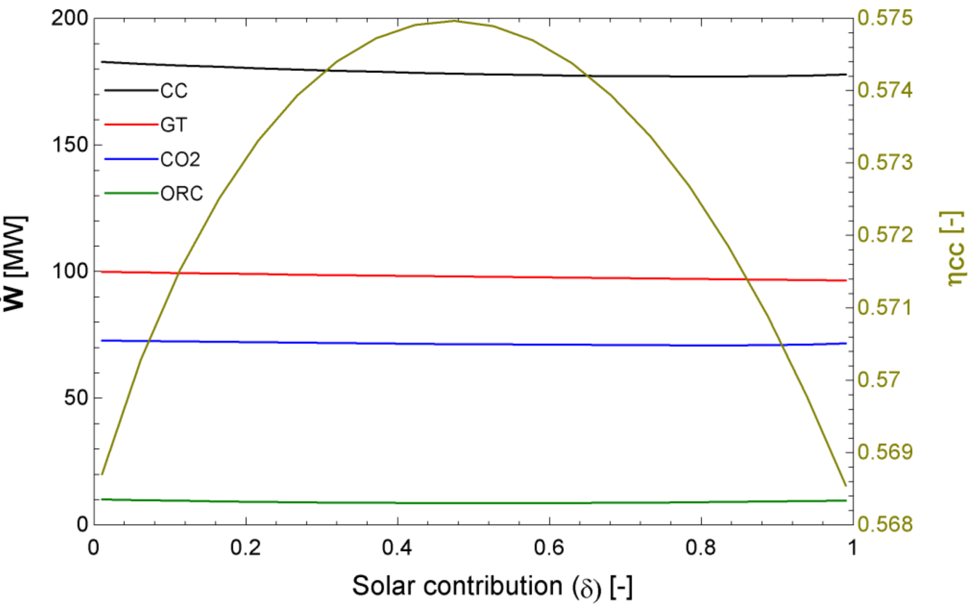


Figure 4. Power of different subsystems and global combined-system efficiency

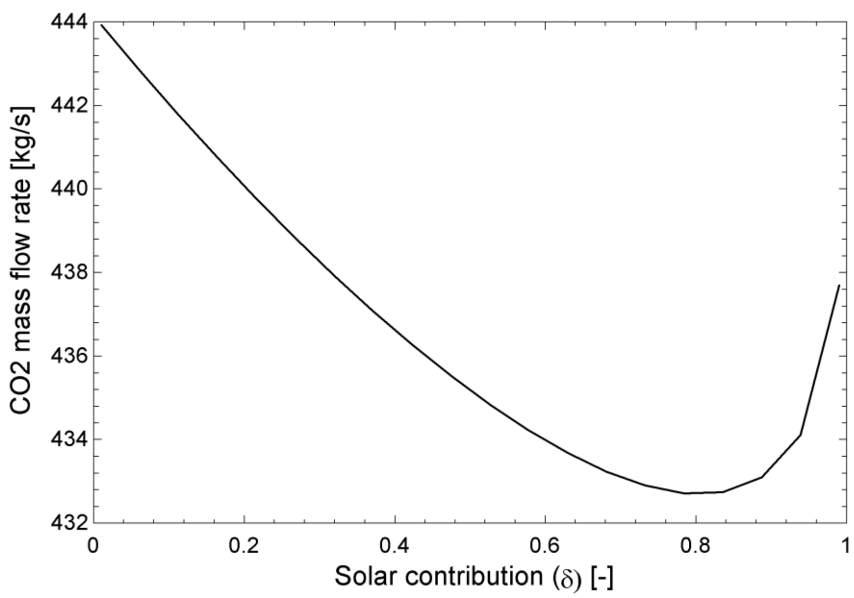
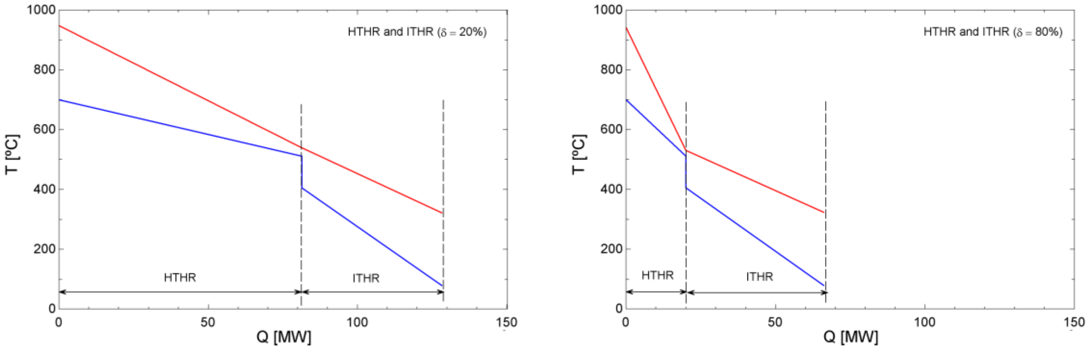
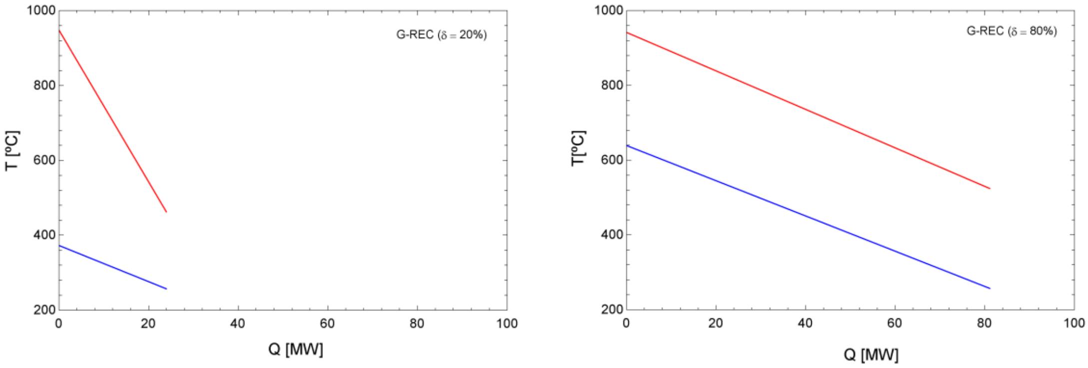


Figure 5. Mass flow rate variation in working fluid of S-CO2.



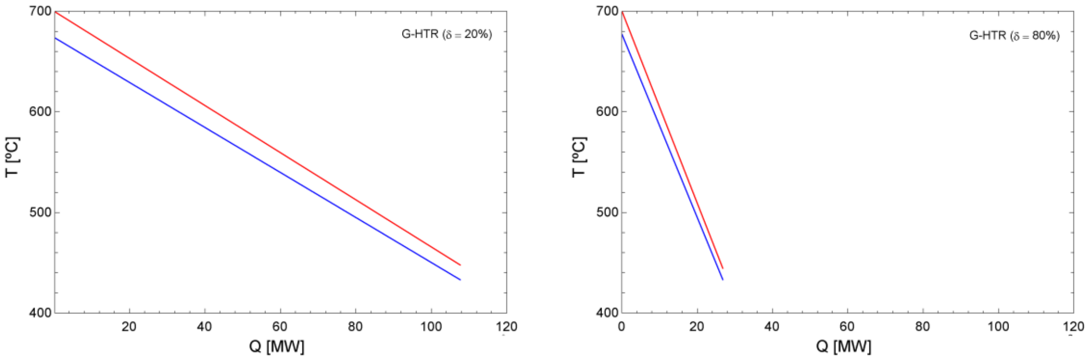
(a) (b)

Figure 6. T-Q profiles at high and intermediate heat recovery heat exchangers (a: 20% of solar contribution; b: 80% of solar contribution).



(a) (b)

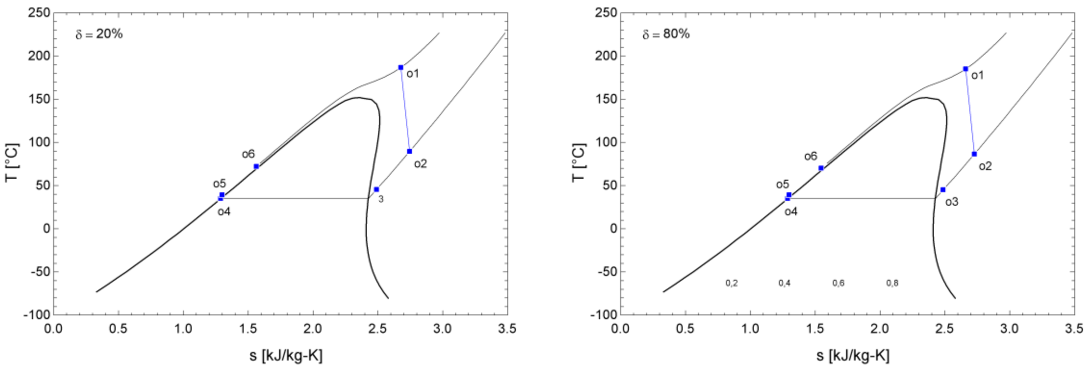
Figure 7. T-Q profiles at gas turbine recuperator (a: 20% of solar contribution; b: 80% of solar contribution).



(a) (b)

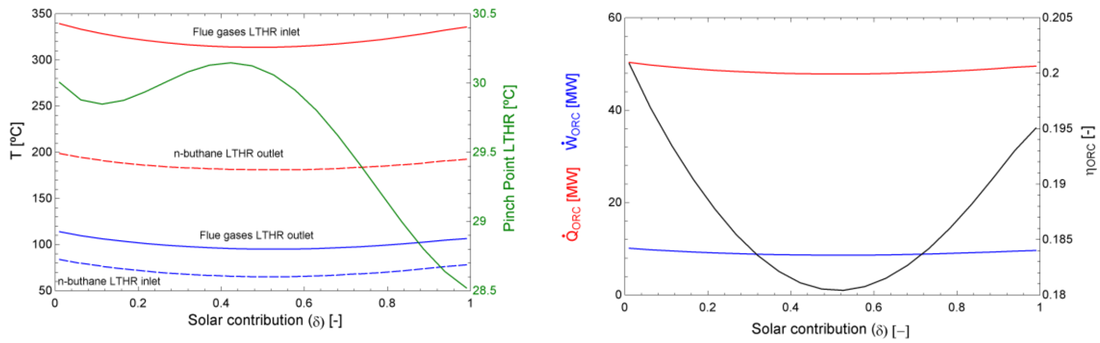
Figure 8. T-Q profiles at G-HTR (a: 20% of solar contribution; b: 80% of solar contribution).

Regarding the ORC, Figure 9 shows the T-s diagram at two different solar contributions, showing a transcritical cycle. Mass flow rate is maintained in the working fluid. The condensation pressure is also maintained, adjusting the cooling water mass flow rate, that taking into account Stodola equation, entails maintaining the high pressure value [15]. However, the turbine inlet temperature varies, although in a slightly way (from 200 ºC at $\delta = 0$ to 192.6 ºC at $\delta = 1$). Figure 10 shows the inlet/outlet temperatures of both streams at LTHR (left plot) and the performance of ORC (right plot). It can be seen how the LTHR thermal conductance has been sized to produce a minimum flue gases outlet temperature of 95 ºC (at 50% solar contribution). ORC efficiency ranges from 18% to 20%, working with nearly constant heat recovery (50 MWth) and net power (9 MWe). Due to the low contribution of the ORC to the power output (5%), the intermediate temperature of flue gases at the LTHR (≈320 ºC to ≈100 ºC) and the heat available (50 MWth), the heat recovery might be used instead as an useful thermal energy for industrial processes, depending on the allocation of the power plant. This would reduce the combined cycle efficiency to 54%, but would simplify the plant, gaining flexibility (it would be a combined heat and power plant). Another possibility would be storing this thermal energy in molten salts to meet peak demands.



(a) (b)

Figure 9. T-s diagram of ORC (a: 20% of solar contribution; b: 80% of solar contribution).



(a) (b)

Figure 10. Inlet/outlet temperatures at LTHR (a) and performance of ORC (b).

4. Conclusions

A hybrid combined cycle GT/S-CO2 supported by a solar central receiver system, including an ORC to recover the low-temperature flue gases energy, has been presented. The performance of the system has been analyzed at its design point, that is, at constant ambient and cooling conditions, with a variable solar contribution. The gas turbine includes a recuperator, which is fed at a variable mass flow rate of the flue gases, in accordance with the solar contribution. This modulation allows to reduce CO_2 emissions up to 1/3, maintaining a nearly constant power output (180 MWe) and global efficiency (57.2%). The gas turbine cycle contributes to the power output in 56%, the S-CO2 in 39% and the ORC in 5%.

The use of partial recuperation with the gas turbine to compensate the variable solar contribution allows to maintain the operational conditions at the turbomachines and at several heat exchangers. This results in a nearly constant (variation lower than 3%) mass flow rate of CO_2 inside the S-CO2 cycle. Taking into account the low contribution of the ORC to the power output, its heat input might be derived to process heat. Another option would be to maintain its use to power production, but shifting to peak demands by including thermal energy storage with molten salts.

Acknowledgements

This work has been supported by the Spanish Ministry of Economy and Competitiveness through the PID2019-110283RB-C33 project.

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