International Conference on Renewable Energies and Power Quality (ICREPQ'19)
Tenerife (Spain), 10th to 12th April, 2019
Renewable Energy and Power Quality Journal (RE&PQJ)
ISSN 2172-038 X, Volume No.17, July 2019

Thermal and Exergy Efficiency Analysis of a Solar-driven Closed Brayton Power Plant with Helium & s-CO_2 as Working Fluids

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Abstract. Solar Thermal Energy power plants operating with traditional steam Rankine cycles have a low thermal and exergy efficiency. An attractive pathway to increase the competitiveness of this technology is to investigate Closed Brayton cycles working with different fluids with desirable properties that show potential for improving their efficiency
In this work a solar driven regenerative Brayton cycle is studied employing two different working fluids: Helium and supercritical CO_2. The cycle efficiencies are determined for different turbine inlet temperatures and for the optimal compressor pressure ratios. Additionally, an exergy analysis breakdown of the different plant components is shown for each case, while the solar field sizes and working fluid flows are calculated for a fixed gas turbine output.

Key words

Helium, Closed Brayton, solar thermal, exergy, supercritical CO_2

1. Introduction

Solar thermal plants usually operate with subcritical steam Rankine cycles at low temperature. The efficiency of these cycles is low. The study of alternative thermodynamic cycles with higher efficiency is fundamental to the development of solar thermal energy [1].

This work evaluates the thermal and exergy efficiency of a solar driven regenerative closed Brayton cycle with Helium (He) and supercritical CO_2 (s-CO_2) as working fluids. The influence of the compression pressure ratio (Pr) and Turbine Inlet Temperature (TIT) on plant performance is analyzed. The closed Brayton cycles have already been employed in nuclear energy [1]. This work aims to address the advantages of one working fluid over another from an efficiency perspective.

2. System Description

Figure 1 Simple scheme of the power plant modeled in Unisim Design R451 Figure 1 shows a simple scheme of the power plant. It consists of a regenerative closed Brayton cycle with two compression stages, an intercooler, a recuperator, an expansion stage and a precooler. When the working fluid is Helium, an intermediate cycle and an exchanger is required to transfer the solar energy to the Helium cycle. This configuration has been recommended in references [2] and [3]. The solar technology is thus a solar tower with pressurized air at 10 bar as heat transfer fluid. The outlet air temperature from the solar tower is such to achieve a Minimum Temperature Approach (MITA) of 30ºC with the Helium stream. The air flow rate is fixed to achieve an exchanger effectiveness of 80%. Because of its cheaper price compared to a costly gas such as Helium, the potential leaks at the solar tower are assumable when using s-CO_2 as working fluid and therefore it can be directly routed to the solar receiver, avoiding the pressurized air cycle and exchanger.

The main operational parameters of the cycle are included on table I. The same value of pressure losses ΔP is assumed for all heat exchangers, while for the solar receiver, a value of 2% was adopted. It was assumed that irrespective of the working fluid employed or the TIT, the solar receiver efficiency remained constant, in order to address exclusively the relative differences of the Brayton cycles with each working fluid.

Table I. Main model parameters of the cycle.

Parameter	Symbol	Value	Reference
Turbine polytropic efficiency	η_t [%]	89	[3]
Compresor polytropic efficiency	η_c [%]	86	[3]
Recuperator effectiveness	[%]	90	[3]
Pressure losses	ΔP [%]	1	[4]

LP Compressor inlet pressure He/s-CO_2	P_1 [bar]	1.5/74	[3]
LP Compressor inlet temperature	T_1 [K]	303.15	[3]
Heliostat field efficiency	η_{hel} [%]	64.28	[5]
Solar Receiver efficiency	η_{rec} [%]	75	[5]

Since the specific heat capacity of CO_2 at pressures close to the critical pressure undergoes significant variations with temperature. Therefore, in the simulations employing s-CO_2 as working fluid, a reasonable MITA of 15ºC was specified. The resulting (feasible) recuperator effectiveness was always lower than that specified for the recuperator in the case of He as working fluid.

The gas turbine output was set to 50 MW (at turbine terminals), which is the gross power limit stipulated by Spanish legislation for solar thermal power plants as specified in [6] and [7]. For each working fluid and operational conditions, the solar field size (in MW) is determined. Note that due to the auxiliary consumption of the blower, which is not negligible, the net plant output is somewhat larger when s-CO_2 is chosen over He, since the intermediate air cycle is avoided. The Intercooler and Precooler were modelled assuming a cooling water inlet temperature of 25ºC and a 10ºC temperature rise. The model was built in Unisim Design R451 [8] using Peng- Robinson equation of state for thermodynamic property estimations.

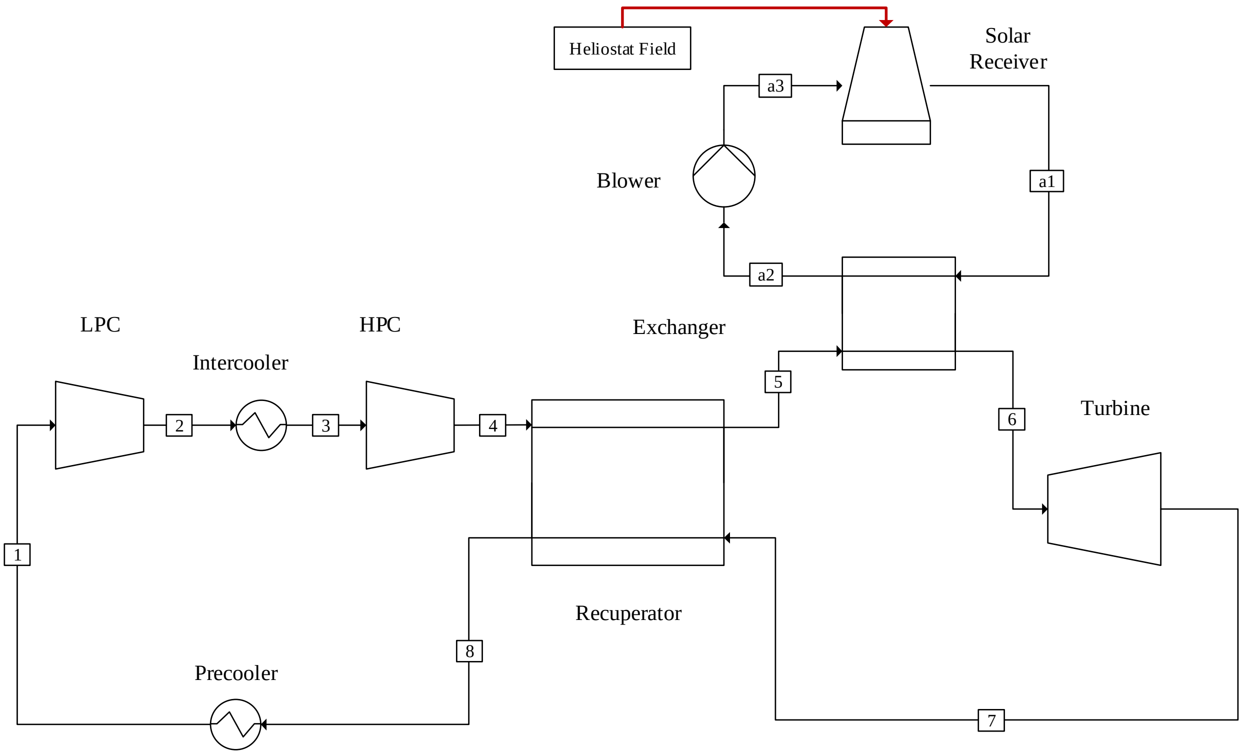

Figure 1 Simple scheme of the power plant modeled in Unisim Design R451

3. Methodology

The plant is evaluated from a thermal and an exergy point of view. The following operating parameters are introduced in the model:

- Compressor pressure ratio P_r (varied from 1.5-5) It is defined by the equation 1. P_2 was determined in such a way that the pressure ratios of LPC and HPC are kept the same.

$$P_r = \frac{P_4}{P_1} \tag{1}$$

- Turbine Inlet Temperature (TIT) i.e $\boldsymbol{T_6}$ (varied from 750ºC to 950ºC).

The model calculates the following output parameters given the specifications above:

- Working fluid & air flow rates: $\dot{\boldsymbol{m}}_{\boldsymbol{He}}$, $\dot{\boldsymbol{m}}_{\boldsymbol{sCO2}}$ & $\dot{\boldsymbol{m}}_{\boldsymbol{air}}$.
- Net power output $\dot{\boldsymbol{W}}$: It is defined as sum of the expansion power and compression power, minus the auxiliary consumption of the blower, if present.

$$\dot{W} = \dot{W}_T + \dot{W}_{LPC} + \dot{W}_{HPC} + \dot{W}_{blower} \tag{2}$$

- Solar power input $\dot{\boldsymbol{Q}}_{\boldsymbol{SF}}$: It is calculated as the ratio between the thermal power input $\dot{\boldsymbol{Q}}_{\boldsymbol{S}}$ on the cycle and the product of heliostat field efficiency $\boldsymbol{\eta_{hel}}$ and receiver efficiency $\boldsymbol{\eta_{rec}}$.

$$\dot{Q}_{SF} = \frac{\dot{Q}_S}{\eta_{hel} \cdot \eta_{rec}} \tag{3}$$

- Thermal efficiency ϵ: It is the ratio between the net power output $\dot{W}$ and the solar power input $\dot{Q}_{SF}$.

$$\epsilon = \frac{\dot{W}}{\dot{Q}_{SF}} \tag{4}$$

- Exergy efficiency ψ: It is calculated by equation 5.

$$\psi = \frac{\dot{W}}{\dot{E}_{SF}} = 1 - \frac{\sum \dot{I}_k + \dot{I}_{loss}}{\dot{E}_{SF}} \tag{5}$$

Where:

- $\dot{I}_k$ Exergy destruction of section k. It is calculated by adding of the exergy destruction of each power plant component j of section k. These exergy destruction values have been grouped in three sections: solar block, turbomachinery (Compressors, Turbine and Blower) and heat exchangers (Precooler, Intercooler, Recuperator & Exchanger). The exergy destruction for component j is determined by application of the exergy balance (6) under stationary conditions to an open, rigid system. The definition of flow exergy of a stream i as an inlet/outlet of component j is given in equation (7):

$$\frac{\partial B_j}{\partial t} = \dot{Q}_j\left(1 - \frac{T_0}{T}\right) - \left(\dot{W}_j - P_0\frac{\partial V_j}{\partial t}\right) - \dot{I}_j + \sum_i e_{i,j}\dot{m}_{i,j} \tag{6}$$

$$e_i = (h_i - h_0) - T_0(s_i - s_0) = h_i - T_0 s_i - g_0 \tag{7}$$

-$\dot{I}_{loss}$: It is the exergy transferred by the working fluid to the cooling water in Precooler and Intercooler. This is a relatively small amount (close to 0.5% of the total exergy input) compared to the exergy destruction terms.

- $\dot{E}_{SF}$: Solar exergy input: It is calculated by the equation 9 [6].

$$\dot{E}_{SF} = \dot{Q}_{SF} \cdot \left(1 - \frac{4}{3}\cdot\left(\frac{T_0}{T_s}\right) + \frac{1}{3}\cdot\left(\frac{T_0}{T_s}\right)^4\right) \tag{9}$$

Where $\dot{Q}_{SF}$, is the solar power input, T_s is the sun temperature (5778 K) and T_0 is the ambient temperature.

These parameters are calculated from the thermodynamic properties (enthalpy, entropy and exergy) of each stream of the cycle modelled with Unisim Design R451 in a spreadsheet. The ambient temperature and pressure are taken as 298.15 K and 1.01325 bar respectively.

4. Simulation and Results

A. Thermal and Exergy Efficiency Performance

Figure 2 and Figure 3 represent influence of the compressor pressure ratio on the thermal & exergy efficiency for the two working fluids. For s-CO2, the maximum efficiency is reached at unrealistically high pressure ratios. Henceforward a HPC discharge pressure of 200 bar is taken, which corresponds to a P_r of 2.7, when evaluating the effect of TIT. On the other hand, the optimal pressure ratio is specified when Helium is used as working fluid.

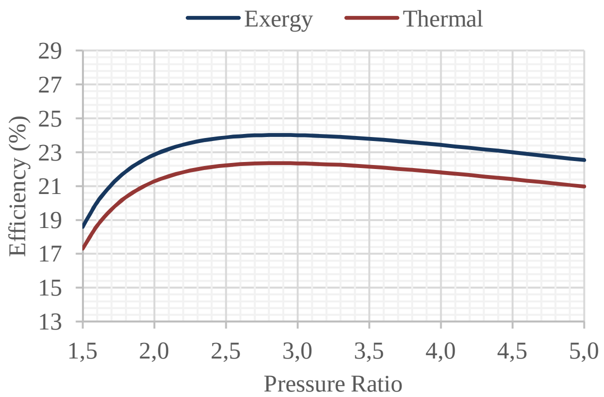

Figure 2 Thermal & exergy efficiency with He as working fluid for a TIT = 950ºC vs. Pressure ratio

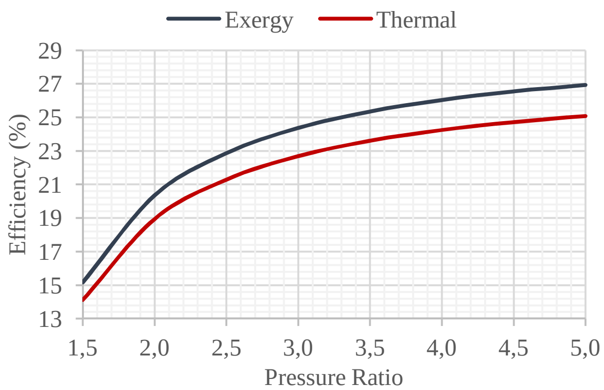

Figure 3 Thermal & exergy efficiency with s-CO_2 as working fluid for a TIT = 950ºC vs. Pressure Ratio

These figures illustrate that the compression ratio has a significant effect on thermal and exergy efficiency. The same comparison is given in Figure 4 & Figure 5 for the range of TITs considered showing a linear increase of both efficiencies for higher TITs.

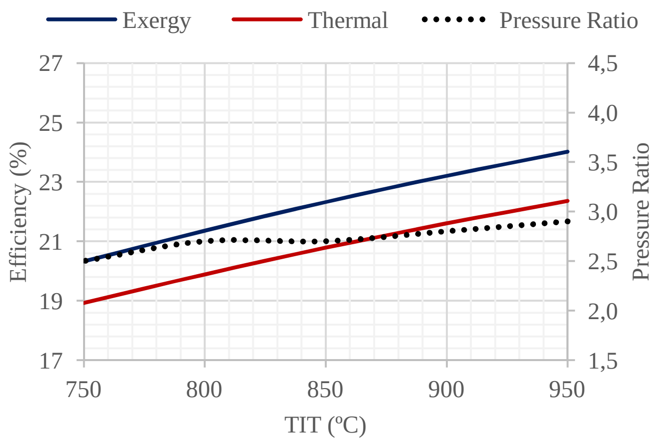

Figure 4 Thermal and exergy efficiency at Optimal Pressure Ratios with He as working fluid vs. TIT

Figure 4 reveals that the optimal pressure ratio for Helium varies slightly from 2.9 at a TIT of 950 ºC to 2.6 at a TIT of 750ºC.

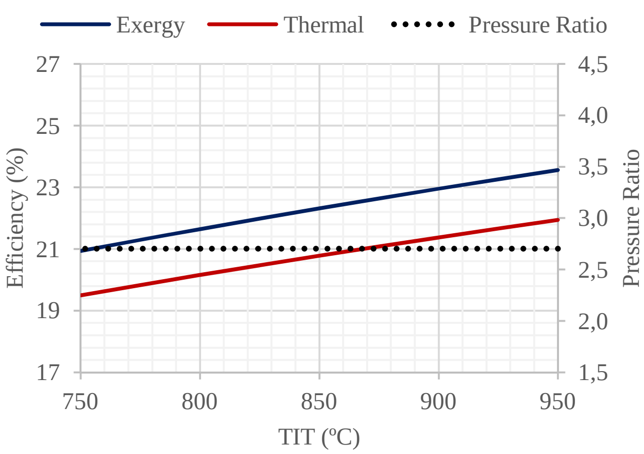

Figure 5 Thermal & exergy efficiency at a pressure ratio of 2.7 with s-CO_2 as working fluid vs. TIT

B. Plant Analysis

In this section the operational variables, namely solar field size and working fluid flow rates, are shown for varying pressure ratios and TITs.

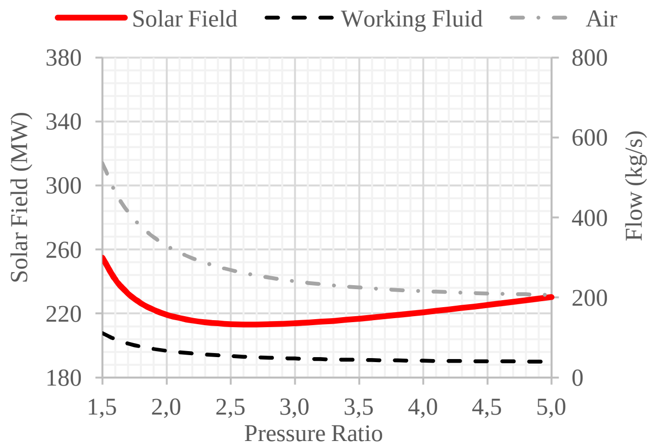

Figure 6 Solar field size and He flow for a TIT = 950ºC vs. Pressure ratio

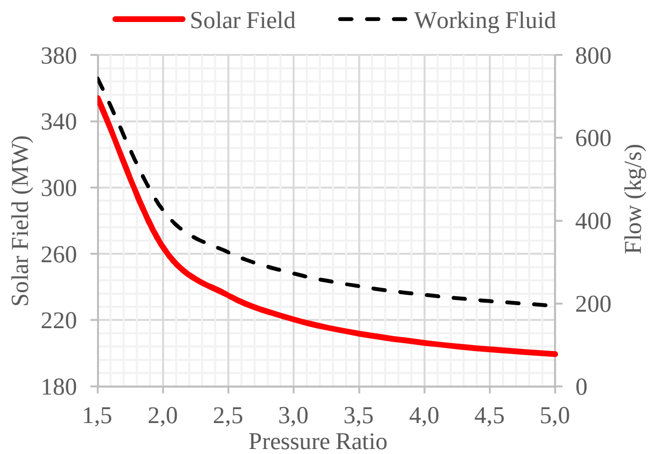

Figure 7 Solar field size and s-CO2 flow for a TIT = 950ºC vs. Pressure ratio

Figure 6 and Figure 7 show the influence of pressure ratio on these operational variables for Helium and s-CO_2 as working fluids respectively. For the Helium case, the pressure ratio of the smallest solar field (2.6) closely corresponds with that of highest thermal and exergy efficiency (2.9), with a rather small solar field size difference of 0.38 MW. It is noteworthy to mention the sharp increase in solar field size for s-CO_2 as working fluid for low pressure ratios, and the substantial higher flow rates required compared to that of Helium.
The same evaluation is done for varying TIT, as show in Figure 8 and Figure 9.

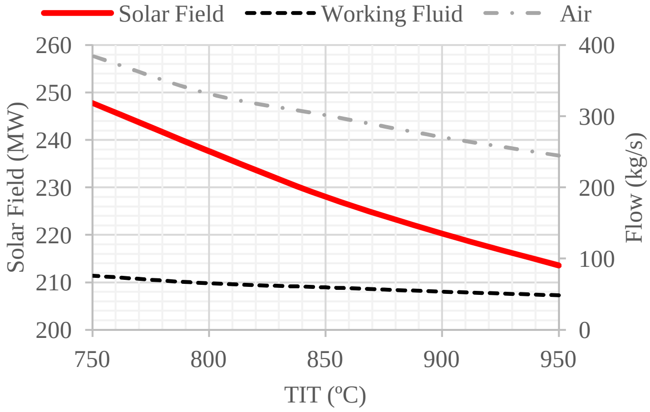

Figure 8 Solar field size & He flow at optimal pressure ratio vs. TIT

The most efficient cases (TIT =950ºC) correspond to the smallest solar field.

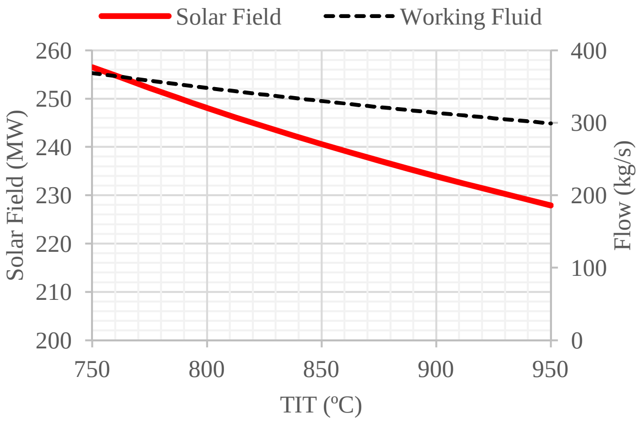

Figure 9 Solar field size and s-CO_2 flow at pressure ratio = 2.7 vs. TIT

Finally, a comparison between the recuperator temperature profile for Helium and s-CO_2 is given in Figure 10, which shows the attractiveness of Helium from a thermal exchange perspective compared to s-CO_2.

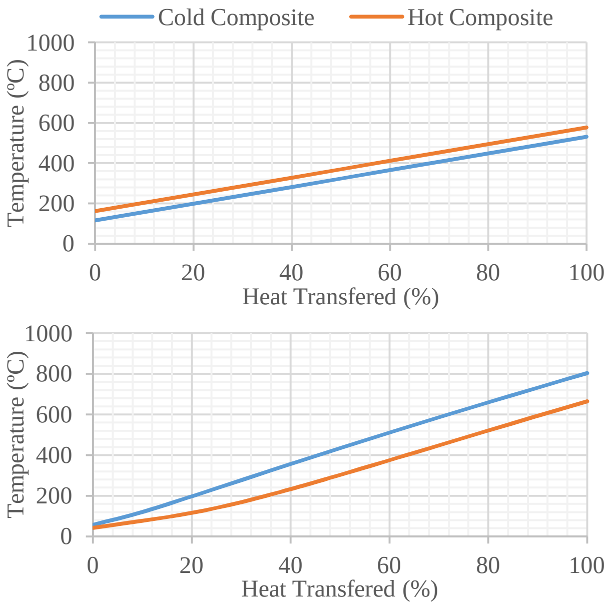

Figure 10 Recuperator temperature profiles for He (above) [Pr =2.9, TIT =950ºC] and s-CO2 (below) [Pr =2.7, TIT=950ºC]

C. Exergy Breakdown Analysis

By applying equation (6) to each element of the cycle and grouping terms as described in section 3, Figure 11 and Figure 12 show exergy profiles obtained just for different TITs for He and s-CO_2 as working fluids respectively.

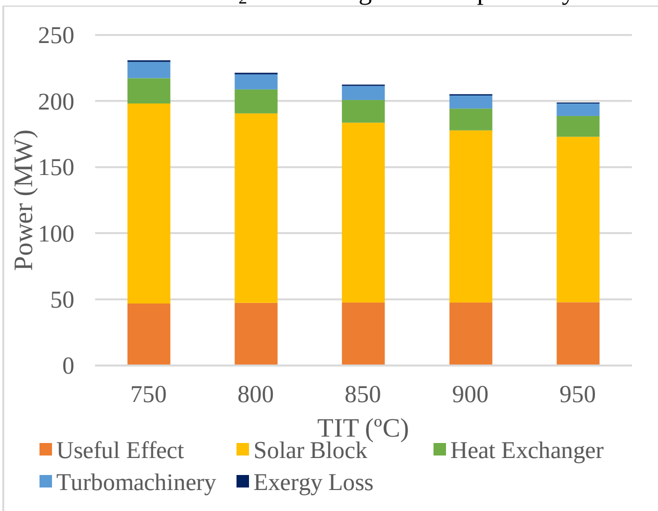

Figure 11 Total exergy breakdown for He as working fluid at optimal pressure ratio vs. TIT

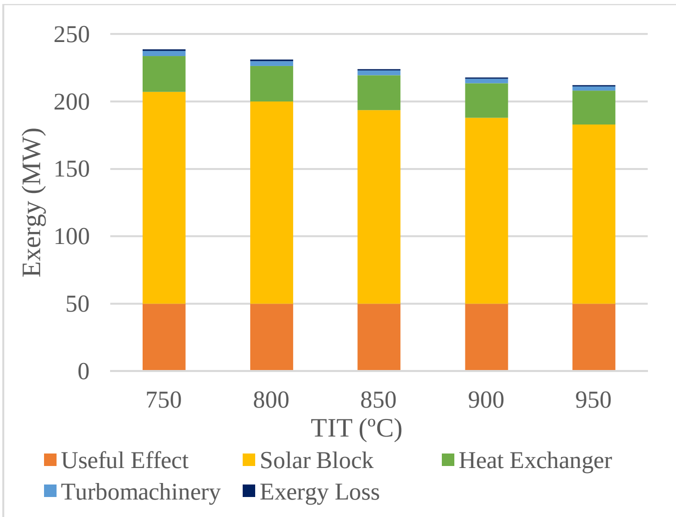

Figure 12 Total exergy breakdown for s-CO_2 as working fluid and pressure ratio of 2.7 vs. TIT

These results reveal the large exergy loss due to the solar block which is independent from the Brayton cycle operation. Exergy losses corresponding to the turbomachinery are greater when He is used as working fluid, whereas the significant recuperator losses in the case of s-CO_2, (as can be deduced from Figure 10) lead to an overall greater exergy destruction in the heat exchangers relative to the He cases, despite the comparatively lower precooler and intercooler exergy destruction and the elimination of the exchanger (air cycle is removed). Because of the higher adiabatic temperature rise when compressing He, an Organic Rankine Cycle (ORC) can be implemented in the place of the intercooler and precooler to maximize the efficiency. However, this would not be an option when s-CO_2 is used as working fluid since the available temperature is too low. The proportion in % of these exergy destruction contributions is shown in Figure 13 and Figure 14.

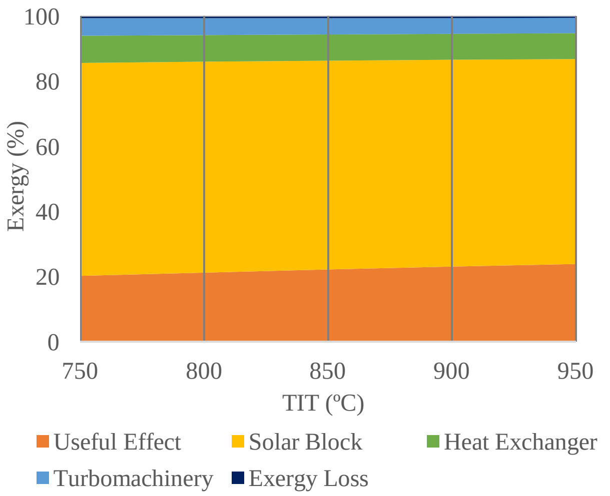

Figure 13 Exergy breakdown in % for He as working fluid at optimal pressure ratio vs. TIT

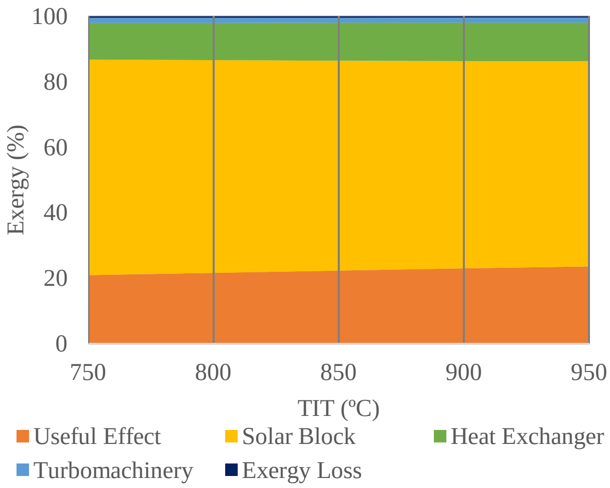

Figure 14 Exergy breakdown in % for s-CO_2 as working fluid and pressure ratio of 2.7 vs. TIT

5. Conclusions

The solar driven regenerative closed Brayton cycle with He or s-CO_2 as working fluids can be an attractive alternative for the solar thermal energy cycles. The maximum thermal and exergy efficiencies obtained were 22.4% and 24.0% for He, while values of 21.9% and 23.7% were obtained for s-CO_2 respectively, corresponding to the highest TIT of 950ºC. The difference in thermal and exergy efficiency of the s-CO_2 case with respect to He, together with the relative solar field size and net plant power output increase where calculated and are shown in Figure 15 at different TITs.

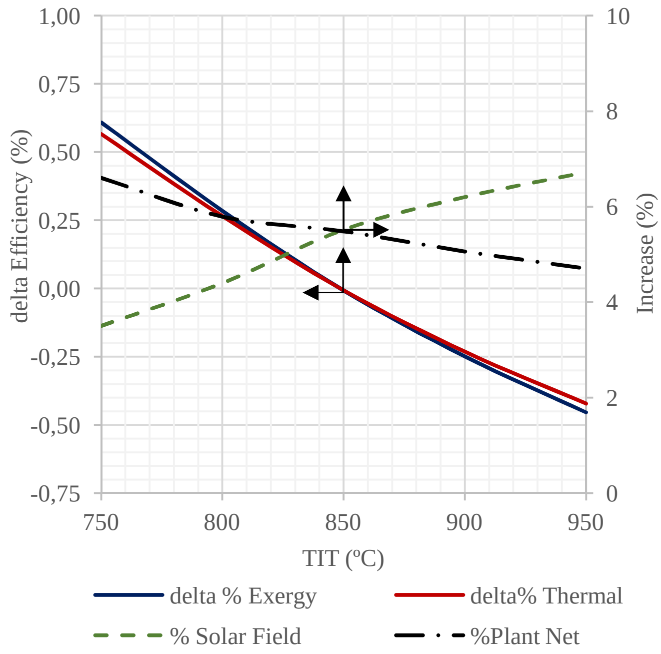

Figure 15 Plant performance comparison between closed regenerative Brayton cycle with s-CO_2 vs. He as working fluids

Figure 15 reveals that for TITs below 850ºC there is an efficiency benefit of s-CO_2 compared to He. In all cases the solar field size (in MW) is comparatively bigger, from 3.5% greater at TIT = 750ºC to 6.7% greater at TIT = 950ºC. Since He cycle have an auxiliary consumption due to the air cycle blower, the s-CO_2 cycle net plant output is 6.7% bigger at a TIT= 750ºC and only 4.7% bigger for a TIT = 950ºC. Cycles with He as working fluid show higher efficiencies than s-CO_2 for TITs above 850ºC.

When the closed regenerative Brayton cycle is designed, it is key to take into account these considerations. From an investment point of view, attention should be paid to the different requirements of both working fluids. The higher pressure and flow rate for s-CO_2 applications that will increase the cost have to be outweighed with the savings due to avoiding the air cycle and the exchanger. Also, higher outlet pressures from HPC will increase the total capital costs of the cycle but reduce the size of the solar field with an overall positive effect on efficiency. This work has demonstrated that from an efficiency point of view, s-CO_2 is preferable to He as working fluid for low TIT, with 850ºC as the break-even point given the model assumptions considered. The efficiencies obtained were in line with the references shown in Table II.

Table II. Thermal efficiency of some solar tower plants [9].

Power Plant	Thermal Efficiency
Solar Two	13 %
Solar Tres	19 %
Solar cuatro	22 %
Solar 100	22 %
PS10	17 %

6. Acknowledgment

The authors would like to acknowledge Honeywell for the free academic license of Unisim Design R451 with which the solar Brayton power plant was modelled.

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