OPTIMIZING EXERGY EFFICIENCY IN SOLAR-DRIVEN REGENERATIVE BRAYTON CYCLES WITH HELIUM: A COMPARATIVE STUDY OF FOUR CONFIGURATIONS FOR 100 MW POWER PLANTS

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Solar thermal energy is a promising renewable energy source due to its low CO_2 emissions and cost-effective thermal storage, which surpasses electric batteries used in photovoltaic and wind systems. Despite facing challenges such as lower efficiency, high capital costs, and the intermittent nature of solar resources, advancements in manageability, storage systems, solar collection optimization, and power cycles are underway. Traditionally, subcritical steam Rankine cycles have been used in solar thermal plants but have limitations in adapting to solar resource variability and electrical demand. Recent proposals focus on Brayton cycles with Helium as the working fluid, benefiting from Helium's high thermal conductivity, specific heat, and inert properties.

This study explores four configurations of regenerative Brayton cycles powered by solar energy to optimize the performance of a 100 MW power plant. The systems include a solar block with heliostats and a solar receiver, and a power block utilizing a Helium Brayton cycle with components such as compressors, turbines, and recuperators. Simulation models for each configuration are developed using Equation Engineering Solver, with detailed mass, energy, and exergy balances. The study aims to identify the most efficient configuration and optimize overall plant performance. The results contribute to the design and development of next-generation solar-driven Brayton cycle power plants, enhancing renewable energy systems' efficiency and sustainability.

Key words*: Brayton cycles, Solar power plants, Helium*

1. Introduction

Solar thermal energy is emerging as one of the most promising renewable energy sources for the forthcoming years. This is largely due to its low CO_2 emissions and the potential for energy management through thermal storage, which is more cost-effective compared to alternatives such as electric batteries used in photovoltaic and wind systems. However, solar thermal energy faces certain limitations, including lower efficiency compared to other technologies, high capital costs, and the intermittent nature of solar resources. Despite these challenges, thermal storage remains a more economical and straightforward solution compared to other options [1].

In response to these challenges, significant resources are being devoted to improving the manageability of solar thermal power plants. This involves advancing storage systems, optimizing solar collection systems, and enhancing the power cycles used. Traditionally, solar thermal power plants have employed subcritical steam Rankine cycles. Although this technology has matured in the context of concentrated solar thermal plants, it faces limitations in adapting to the nature of solar resources and electrical demand compared to other technologies [2].

In recent years, alternative power cycles have been proposed to enhance the performance of solar thermal plants. Notably, Brayton cycles, which utilize various configurations of compressors, heat exchangers, and reheaters with different fluids, have gained attention [3].

Brayton cycles with Helium are considered a promising alternative due to Helium's advantageous thermal properties—an inert gas with high thermal conductivity and specific heat —and its operational experience in demanding thermal systems, such as nuclear environments [4,5]. Kusterer K. et al. [6] calculated the energy efficiency of different layouts for Helium Brayton cycles driven by a solar tower. Assuming a constant solar receiver efficiency of 85%, they concluded that the best configuration features a single turbine and two compressors. Zare V. et al. [7] proposed a combined Helium–ORC cycle for solar tower power plants. They carried out an energy and exergy optimization considering that the solar receiver temperature is constant and independent of the heat transfer fluid temperature. Their results show that the proposed system presents a higher performance than Rankine and supercritical CO_2 cycles. Habbi et al. [8] compared six working fluids for Brayton cycles and Brayton-ORC combined cycles driven by a molten salts solar tower from an energetic and exergetic perspective. The results indicate that cycles using Helium as the working fluid demonstrate higher net power output and efficiency. Tesio et al. [9] compare a Brayton cycle using SCO_2 and Helium, both integrated into a thermochemical storage system utilizing calcium looping driven by a solar tower. The results show that the Helium cycle is more efficient, while SCO_2 has lower investment costs. Li et al. [10] conducted a triple optimization of three different layouts of Helium Brayton cycles for ultra-high temperature solar power towers operating at temperatures greater than 1300 ℃. The objective functions are specific work, thermal efficiency, and temperature difference of the thermal energy storage, assuming the influence of the heliostat field and the receiver is negligible. Their study concludes that simultaneously optimizing the three objective functions is impossible. Zixiang et al. [11] propose a novel He-CO_2 cascade Brayton cycle coupled with a solar tower. They conducte thermodynamic and exergoeconomic analyses. The results demonstrate good performance from both perspectives, indicating a significant influence of the heliostat and the receiver on the results. Khan, Y. et al. [12,13] explore different layouts of combined cycles, where the topping cycle is a Brayton cycle working with Helium and the bottom cycle is a tCO_2 cycle. The studies conduct an energy, exergy, and exergoenvironmental analysis. They conclude that the different combined configurations improve performance compared to the stand-alone configurations.

This study focuses on the use of a closed Brayton cycle with Helium as the working fluid in solar thermal power plants. This research explores four different configurations of regenerative Brayton cycles driven by solar energy, each designed to optimize the performance and efficiency of a 100 MW power plant. Real power generation facilities can be more complex, although studies indicate that increasing the number of compressors, turbines, and regenerators leads to higher efficiency under nominal conditions. However, transient behaviour is worse, and costs are also higher compared to simpler configurations, [14]. A value of 100 MW has been assumed as a characteristic value for a commercial plant, as indicated in the reference [15].

The objective of this study is to develop and compare simulation models for each proposed configuration using Equation Engineering Solver [16]. The thermodynamic models for both the power block and the solar block are based on established references and include detailed mass, energy, and exergy balances. By evaluating these models, the study aims to identify the most efficient configuration and optimize the overall performance of the power plants. The novelty of this research is that comprehensive exergy analysis of the system is conducted, which examines the effect of the receiver temperature on the efficiency of the solar receiver and the overall efficiency

The paper is structured as follows: the first section describes the four plant configurations under study. The subsequent section details the development of the thermodynamic models, the parameters used for performance evaluation, the optimization problem addressed and concludes with a validation of the model. The final sections analyse the results from the simulations and present the conclusions of the study. This comprehensive approach provides a robust framework for advancing the design of solar-driven Brayton cycle power plants and contributes valuable insights for the development of next-generation renewable energy systems.

2. Methodology

2.1. System description

The four cycles studied in this work are represented in Fig. 1. All cycles are divided into two subsystems: solar block (SB) and power block (PB). Both subsystems are connected through a heat exchanger (IHE) that transfers heat power from the (SB) to the (PB).

The SB consists of a heliostat field (HF) and a solar receiver (R) on top of a tower (ST). The solar radiation strikes the heliostat field and is reflected to the solar receiver. The heat power absorbed by the receiver raises the temperature of the heat transfer fluid (HTF) and thermal power is transferred to the PB in the intermediate heat exchanger. In this work, air is used as HTF.

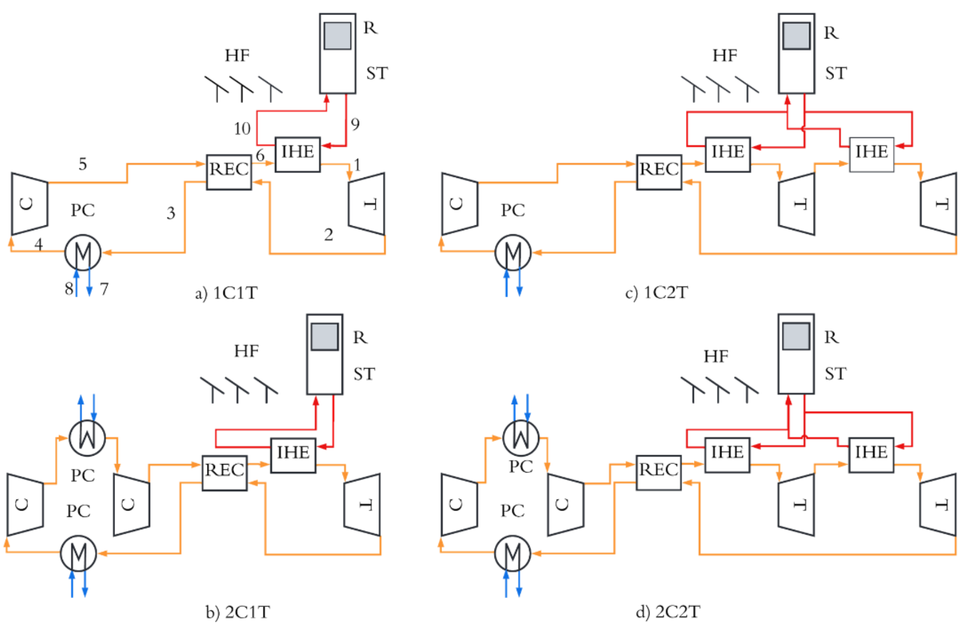

Fig. 1 The four-cycle studied in this work.

The PB consists of a regenerative closed Brayton cycle with Helium as the working fluid. The PB include a recuperator (REC) and different numbers of compressors (C), turbines (T), precoolers (PC) and the intermediate heat exchangers (IHE).

In the 1C1T cycle (Fig. 1a), the Helium is initially heated in the IHE and then expanded in the turbine. After the expansion, the Helium circulates through the recuperator and is cooled in the precooler before entering the compressor. The compressed Helium is preheated in the recuperator before entering in the IHE. Cycles 2C1T (Fig. 1b), 1C2T (Fig. 1c), and 2C2T (Fig. 1d) perform compression and expansion processes in one or two stages to improve efficiency concerning the 1C1T configuration. To compare the four power plants, it has been determined that all of them have a net power of 100 MW.

2.2. Thermodynamic modelling and optimization

2.2.1 Thermodynamic model

The following assumptions are considered:

- The steady-state conditions are considered.
- Kinetic and potential energy variations are negligible.
- Ambient temperature and pressure are 25 ºC and 100 kPa.
- The pressure losses on all heat exchangers are set as 2 % [12].
- The effectiveness of recuperator is 90% [7].
- Inlet and outlet water cooling temperatures are 25 °C and 35 °C [7].
- Minimum pressure of the cycle is 2500 kPa and compressors inlet temperature is 30 °C [12].

A simulation model for each system has been developed in Equation Engineering Solver [16] to compare the different layouts proposed. Two models have been developed: one for the power blocks of each cycle and one for the SB. The PB model is based on the model described in reference [8]. In this reference, a thermodynamic model for a regenerative Helium Brayton cycle with two compressors is developed. The SB is based on the model described in reference [10]. In both models, the mass, energy and exergy balances are applied to each component. The equations resulting from these balances to each component for the 1C1T (Fig. 1a) cycle are given in Tab. 1. Similar equations have been obtained for the rest of the layouts.

Tab. 1 Energy and exergy balances for each component (1C1T).

Component	Energy	Exergy
Compressor	$\dot{W}_C = \dot{m}_4 h_4 - \dot{m}_5 h_5$	$\dot{E}_{D,C} = \dot{m}_4 e_4 - \dot{m}_5 e_5 - \dot{W}_C$
Turbine	$\dot{W}_T = \dot{m}_1 h_1 - \dot{m}_2 h_2$	$\dot{E}_{D,T} = \dot{m}_1 e_1 - \dot{m}_2 e_2 - \dot{W}_T$
Recuperator	$\dot{Q}_{REC} = \dot{m}_6 h_6 - \dot{m}_5 h_5$	$\dot{E}_{D,REC} = \dot{m}_2 e_2 - \dot{m}_3 e_3 + \dot{m}_5 e_5 - \dot{m}_6 e_6$
Precooler	$\dot{Q}_{PC} = \dot{m}_4 h_4 - \dot{m}_3 h_3$	$\dot{E}_{D,PC} = \dot{m}_3 e_3 - \dot{m}_4 e_4 + \dot{m}_8 e_8 - \dot{m}_7 e_7$
Intermediate heat exchanger	$\dot{Q}_{IHE} = \dot{m}_1 h_1 - \dot{m}_6 h_6$	$\dot{E}_{D,IHE} = \dot{m}_6 e_6 - \dot{m}_1 e_1 + \dot{m}_9 e_9 - \dot{m}_{10} e_{10}$
Solar Block	$\dot{Q}_s = \dfrac{\dot{Q}_{IHE}}{\eta_{hel} \cdot \eta_R}$	$\dot{E}_{D,s} = \dot{E}_s + \dot{m}_{10} e_{10} - \dot{m}_9 e_9$

Compressors and turbines have a polytropic efficiency calculated as a function of the pressure ratio by eq. (1) and (2) respectively [7].

$$\eta_{PC} = 0.916 - 0.0175\ln(PR_C) \tag{1}$$

$$\eta_{PT} = 0.932 - 0.0117\ln(PR_T) \tag{2}$$

The efficiency of the heliostat field is assumed as a value 0.6 while the solar receiver performance has been calculated by eq. (3) [17]. The values of the parameters of eq. (3) are listed in Tab. 2.

$$\eta_R = \alpha - \frac{\varepsilon \cdot \sigma \cdot (T_R^4 - T_0^4) + h_{con} \cdot (T_R - T_0)}{DNI \cdot C} \tag{3}$$

Tab. 2 The values of the parameters of eq. 6 [17].

Parameter	Value
Receiver solar absorptance α [-]	0.9
Thermal emittance of the receiver ε [-]	0.8
Stefan-Boltzmann constant σ [W/m^{2}K^4]	5.67E-8
Effective solar flux concentration ratio C [-]	500
Receiver convective heat transfer coefficient h_{con} [W/m^{2}K]	8
Direct normal irradiance DNI [W/m^2]	1000

The temperature of the receiver T_R is equal to the outlet receiver temperature, T_9 in the cycle 1C1T. Two assumptions are made to calculate this temperature [18]:

- There is a difference of 10 K in the cold terminal of the IHE.

$$T_{10} = T_6 + 10\text{ K} \tag{4}$$

- The entropy generation is equal at both terminals of the IHE. This condition is expressed by eq. (5)

$$\mathrm{d}Q\left(\frac{1}{T_9} - \frac{1}{T_1}\right) = \mathrm{d}Q\left(\frac{1}{T_{10}} - \frac{1}{T_6}\right);\ T_9 = \frac{T_{10}T_6T_1}{T_6T_1 + T_{10}T_6 - T_{10}T_1} \tag{5}$$

The solar exergy $\dot{E}_s$ is calcuted by the well known Petela's formula [19]

$$\dot{E}_s = \dot{Q}_s\left(1 + \frac{1}{3}\left(\frac{T_0}{T_s}\right)^4 - \frac{4}{3}\frac{T_0}{T_s}\right) \tag{6}$$

2.2.2 Perfomance evaluation

The overall energy efficiency and the overall exergy efficiency are calculated by eq. (7) and (8), respectively.

$$\eta = \frac{\dot{W}_N}{\dot{Q}_s} \tag{7}$$

$$\varphi = \frac{\dot{W}_N}{\dot{E}_s} \tag{8}$$

2.2.3 Optimization

To determine the configuration with the highest exergy efficiency, an optimization of this performance parameter will be conducted based on the pressure ratio (PR) and the turbine inlet temperature (TIT). TIT refers to the temperature of the working fluid as it enters the turbine in a thermodynamic cycle. This parameter is critical for the turbine's performance and efficiency, as it directly impacts the energy conversion process and the overall system efficiency. In the context of Brayton cycles or other turbine-based systems, a higher TIT generally improves performance but may require more advanced materials and cooling technologies to handle elevated temperatures. Additionally, the pressure ratio (PR) is defined as the ratio between the maximum and minimum pressure within the PB. Optimizing both TIT and PR will help identify the configuration that maximizes the system's exergy efficiency.

The formulation of the optimization problem is expressed by eq. (9-11). The pressure ratio and turbine inlet temperature limits are established based on the reference [5].

$$\text{Maximize}\ \varphi = \varphi(PR, TIT) \tag{9}$$

$$1.5 \le PR \le 5 \tag{10}$$

$$700 \le TIT \le 900 \tag{11}$$

2.2.4 Model Validation

The PB model has been validated using data from reference [4]. The reference analyses a simple regenerative closed Helium Brayton cycle. Tab. 3 compares the results obtained from the model presented in this work with the results from reference [4].

Tab. 3 Validation of Results from This Work with [4].

Parameter	Ref [4]	This work	Deviation
Turbine power	552.8 MW	546.6 MW	-1.12 %
Compressor power	248.3 MW	250.4 MW	0.85 %
Pre-cooler thermal power	295.6 MW	297.8 MW	0.74 %
Recuperator thermal power	970.7 MW	988.1 MW	1.79 %
Helium mass flow rate	401.1 kg/s	404.0 kg/s	-0.73 %

The model used to calculate the efficiency of the solar receiver has been validated in the same way as in the reference [17]. The values obtained by substituting the values of Tab. 2 in eq. (3) have been compared with the experimental values of references [20-22]. According to Fig. 2, the results provided by the model are close to the experimental data within the range to be used (700-900 °C) in this article.

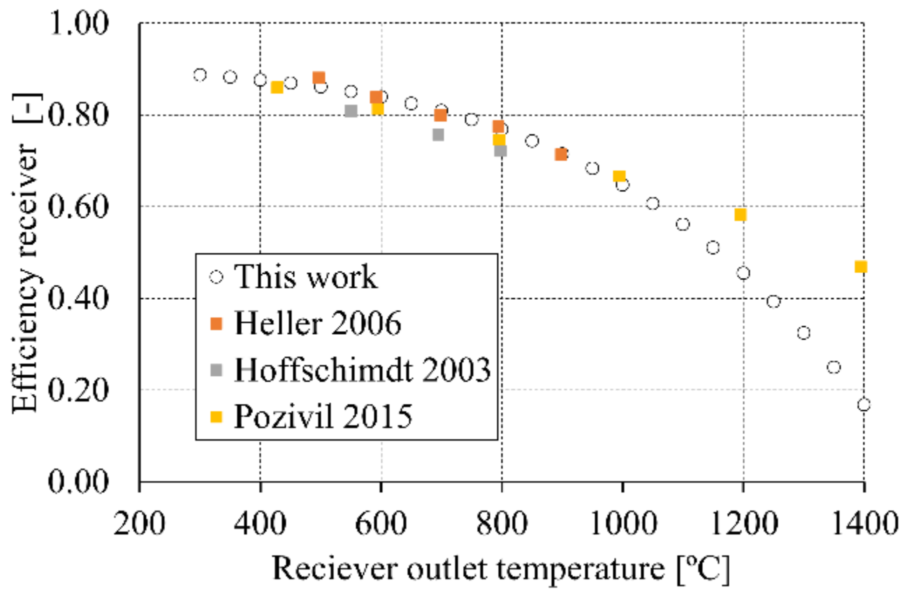

Fig. 2 Validation of solar receiver efficiency model with experimental data from [20-22].

3. Results

3.1. Parametric analysis

A parametric analysis is carried out to examine the influence of the pressure ratio and the turbine inlet temperature on performance of each cycle. As base values for the pressure ratio and turbine inlet temperature, values of 3 and 800 ºC respectively, as specified in reference [7], were selected.

3.1.1 Effect of the pressure ratio

Fig. 3a and 3b depict the changes in overall energy and exergy efficiency with the pressure ratio. The 2C2T cycle has the highest energy and exergy efficiencies, followed by the 2C1T cycle.

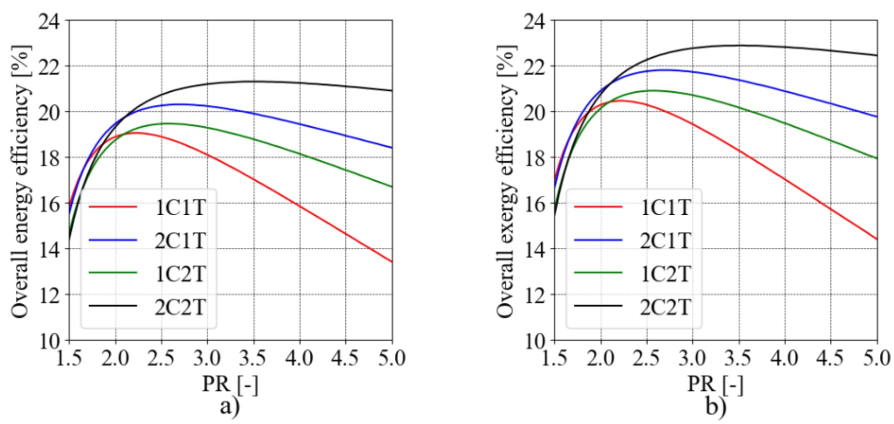

Fig. 3 Influence of the pressure ratio on the overall energy efficiency (a) and overall exergy (b) efficiency.

The cycles 1C2T and 2C2T present lower performance when the pressure ratio is approximately less than 2. There exists an optimal pressure ratio that maximizes both energy and exergy efficiency at the same time. When the pressure ratio is lower than optimal, efficiency increases sharply. On the other hand, when the pressure ratio is higher, efficiency decreases significantly for all cycles except for the 2C2T case, where the decrease is slow.

Fig. 4a and 4b illustrate how the pressure ratio affects the exergy efficiency of the PB and SB, respectively. Exergy efficiency of the PB is very similar trend that the overall system. However, the pressure ratio that optimizes the overall system is lower than the pressure ratio that optimizes the PB.

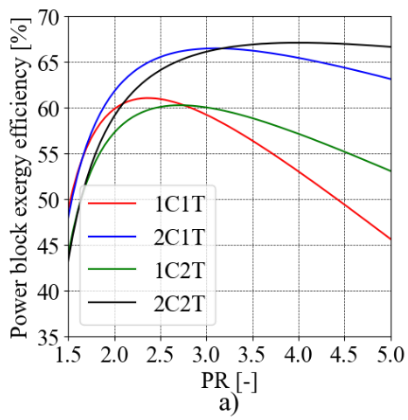

Fig. 4 Influence of the pressure ratio on the PB exergy efficiency (a) and SB exergy efficiency (b).

This is due to the behaviour of the efficiency of the SB (Fig. 4b). As can be seen, in the 1C1T and 2C1T cycles, the efficiency of the SB decreases as the pressure ratio increases. For the cases of 1C2T and 2C2T, the efficiency of the SB increases very slightly for very small pressure ratios but decreases as the pressure ratio increases. The decrease in SB exergy efficiency with pressure ratio is less pronounced in cases with two turbines.

3.1.2 Effect of the turbine inlet temperature

Fig. 5a and 5b show the variation of energy and exergy efficiency with turbine inlet temperature. For all the cycles, the optimum point is around 850 ºC. The layouts with two compressors present a higher efficiency that the cases with a single compression stage. As can be seen, the influence of the turbine inlet temperature is less significant than the influence of the pressure ratio.

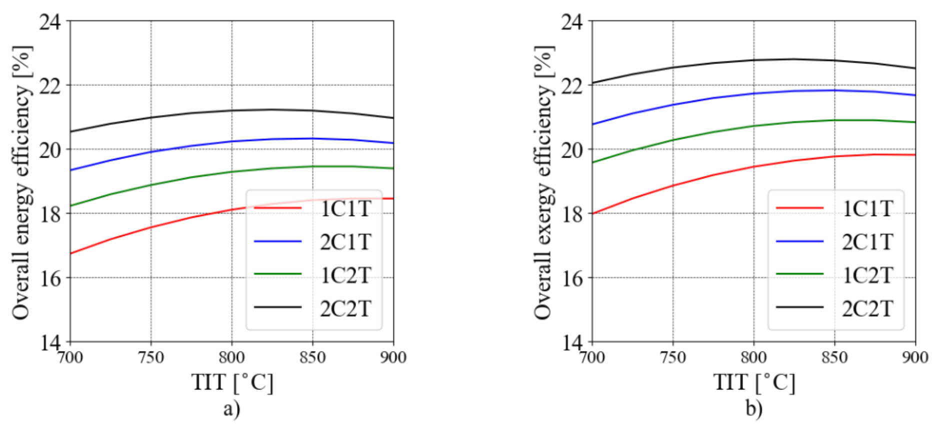

Fig. 5 Influence of the turbine inlet temperature on the PB exergy (a) and SB exergy (b) efficiencies.

Fig. 6a illustrates how the exergy efficiency of the PB varies with turbine inlet temperature. As the turbine inlet temperature increases, the PB efficiency increases for all cycles. The positioning of the cycle is determined by the chosen pressure ratio, as illustrated in the Fig 4a. As can be shown in Fig. 6b, the efficiency of the SB decreases as the temperature increases. The layouts are grouped according to the number of turbines. The layouts with two expansion stages present higher efficiency than the cases with one turbine.

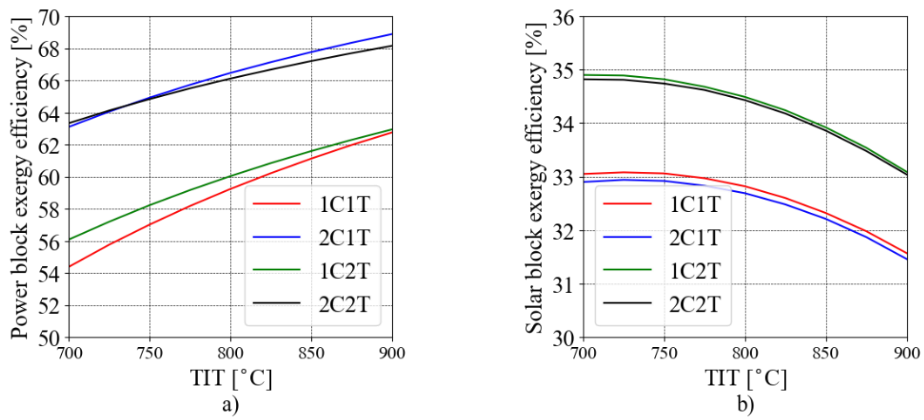

Fig. 6 Influence of the turbine inlet temperature on the PB exergy efficiency (a) and SB exergy efficiency (b).

3.2. Optimization results

The optimization results are displayed in Tab. 4. The 2C2T cycle exhibits the highest overall efficiency at 21.35 %. On the other hand, the simplest 1C1T cycle has the lowest efficiency. All power cycles present an exergy efficiency higher than 60%. However, the overall efficiencies are considerably lower due to the solar block's low efficiencies, which are between 32.56 % and 34.1 %.

In all cases, the optimal turbine inlet temperatures range between 832.9 °C (2C2T) and 849.9 °C (1C2T). It is interesting to note that in the 1C2T cycle the SB is most efficient while the PB presents the lower efficiency. These results indicate that the optimal turbine inlet temperature and pressure ratio balance the optimization of the PB and the SB.

Tab. 4 Optimum values of the main performance parameters for each cycle.

	1C1T	2C1T	1C2T	2C2T
Pressure ratio PR [-]	2.28	2.756	2.651	3.593
Turbine inlet temperature TIT [°C]	848.9	838.6	849.9	832.9
Overall energy efficiency [%]	19.15	20.36	19.56	21.35
Power block exergy efficiency [%]	62.27	67.16	61.63	67.79
Solar block exergy efficiency [%]	33.02	32.56	34.1	33.82
Overall exergy efficiency [%]	20.56	21.87	21.01	22.93
Solar exergy [MW]	493.16	463.49	482.53	442.01

The variations in exergy efficiency are reflected in the solar exergy (exergy input) required to generate the same net power (exergy output). Fig. 7 illustrates the distribution of the solar exergy across different categories: exergy destruction of the SB and PB, exergy output, and exergy loss. The SB exergy destruction represents approximately 67% of the exergy input in all cases. This result is mainly attributed to the low performance of the solar receiver. On the other hand, the PB present more noticeable differences between cycles. The PB with the lowest exergy destruction are the cycles with two compressors. Finally, exergy loss is minimal in all cycles. This exergy loss reflects the exergy gained by cooling water in various PCs due to the heat rejected.

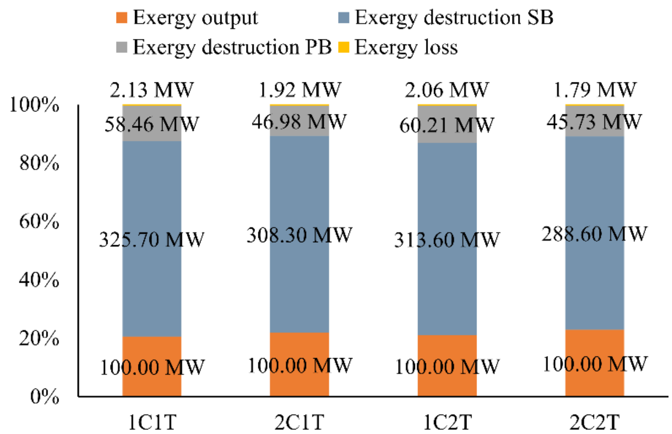

Fig. 7 Distribution of the solar exergy across different categories.

To complete the analysis, the exergy destruction of the PB is broken down in the exergy destruction contributed by each component (compressors, turbines, recuperators, precoolers, and intermediate heat exchangers) to identify differences between the studied cycles, as well as which equipment presents the largest amount of irreversibility. The results are shown in Fig. 8. Across all configurations, the precoolers present the highest exergy destruction. In addition, the exergy destruction in the precoolers shows the most significant variation between cycles. These values result from the significant temperature difference between Helium and cooling water streams. The second equipment with higher exergy destruction is the recuperator. This exergy destruction is influenced by the effectiveness of the heat exchanger used. If a higher effectiveness value is assumed, this exergy destruction could be reduced. It is noticeable that in cycles with two compressors, this exergy destruction is significantly reduced. The cases with two turbines (1C2T and 2C2T) present higher exergy destruction in the intermediate heat exchangers. This exergy destruction is affected by the hypothesis of equal entropic generation at both terminals of the intermediate heat exchanger. This assumption helps minimize exergy destruction in the heat exchanger while satisfying the temperature difference condition. If a different assumption were used, exergy destruction would increase. The exergy destruction in compressors (between 7.2 and 8 MW) and turbines (between 4.43 and 5.21 MW) is similar all cases.

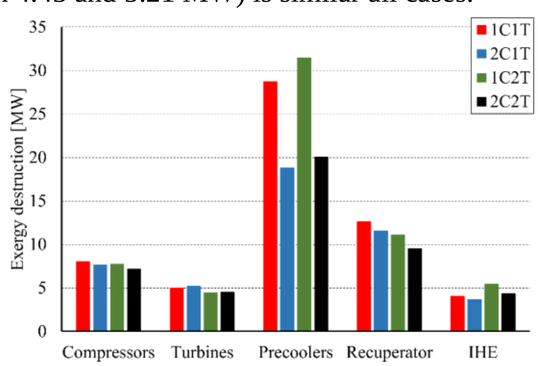

Fig. 8 Exergy destruction of the power block by each component.

4. Conclusions and future works

Brayton solar tower cycles using Helium as the working fluid can be a viable alternative to Rankine and supercritical SC-CO2 cycles due to Helium's favourable thermal properties. The main objective of this work is to develop the comprehensive energetic and exergetic analysis of four different configurations of a 100 MW regenerative Brayton cycle with Helium driven by solar tower. The methodological framework involves analyzing two systems for each configuration: the Brayton cycle and the solar system, which are connected by an intermediate heat exchanger. This approach allows to consider how the temperature of the (HTF), air in this case, affects the performance of the solar receiver.

The four Brayton cycle configurations were simulated by Equation Engineering Solver [16] based on established references [5, 17]. Through the evaluation of these simulations, the most efficient configuration has been identified for the considered boundary conditions.

In order to reach the most efficient configuration, several analyses have been previously carried out to the optimization itself: pressure ratio analysis and turbine inlet temperature analysis for the two coupled systems (Brayton cycle and solar system). These two variables, pressure ratio and turbine inlet temperature, are the ones considered for the optimization process. In relation with pressure ratio analysis, configuration two compressors and two turbines shows higher robustness to changes on this value. In relation with turbine inlet temperature, its influence is less significant than the influence of the pressure ratio on the overall system. Furthermore, opposing trends are observed in both parameters for the solar and power blocks. Consequently, achieving optimal values requires a balance between the efficiency of both subsystems.

The configuration with two compressors and two turbines is the most complex but offers the highest exergy efficiency, followed by the configuration with two compressors and one turbine. The turbine number effect follows similar tendency; two turbines give better efficiency than one turbine, but the effect is small in comparison of the change from one compressor to two, where appear the main improvement. Overall efficiency is more sensible to compression stages than to the expansion ones, due to the compression itself and to the required precoolers irreversibilities.

Taking into account obtained results, it is proposed to develop further analysis in relation with other effects that have relevant influence in the optimization results, as pressure drop. Also, the overall dynamic behaviour is a key point: higher number of sub-system can reduce the dynamic response decreasing the global efficiency.

In conclusion, obtained results can be altered taking into account pressure drop and dynamic analysis, but it is clear that the best results will be related with several compression stages.

Nomenclature

$\boldsymbol{DNI}$ – Direct normal irradiance [W/m^2]	$\dot{\boldsymbol{m}}$ – mass flow rate [kg/s]
$\boldsymbol{C}$ – effective solar flux concentration ratio [-]	TIT – turbine inlet temperature [°C]
$\boldsymbol{e}$ – specific exergy [kJ/kg]	$\dot{W}$ – electric power [MW]
$\dot{\boldsymbol{E}}$ –exergy [MW]	T – temperature [K]
$\boldsymbol{h}$– specific enthalpy [kJ/kg]	$\dot{Q}$ – thermal power [MW]
$\boldsymbol{h_{con}}$ – Convective heat transfer coefficient [W/m^2 K]	PR – pressure ratio [-]
ACRONYMS	
$\boldsymbol{IHE} -$ intermediate heat exchanger	$SB -$ solar block

$\boldsymbol{PB}$ − power block	HF − heliostat field
$\boldsymbol{REC}$ − recuperator	HTF − heat transfer fluid
GREEK SYMBOLS	
$\boldsymbol{\alpha}$ – receiver solar absorptance [-]	ε – Thermal emittance [-]
$\boldsymbol{\eta}$ – Overall energy efficiency [-]	η_{hel} − efficiency of the heliostat field [-]
$\boldsymbol{\eta_{PC}}$ – Compressor polytropic efficiency [-]	η_{PT} – Turbine polytropic efficiency [-]
$\boldsymbol{\varphi}$ – Overall exergy efficiency [-]	σ – Stefan-Boltzmann constant [$W/m^2 K^4$]

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