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Parametric evaluation of S-CO_2 Brayton cycles for bottoming applications

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Abstract: Parametric design studies for the preliminary assessment of physical footprint of Supercritical CO_2 power plants are presented herein. The aim of the study is to quantify trade-offs between cycle efficiency and plant complexity for a range of S-CO2 cycle configurations.

1. Scope

With the drive towards higher cycle efficiency, the need for exploiting waste heat from thermal power systems is becoming increasingly critical. One of the main challenges when assessing novel bottoming power cycle layouts for waste heat recovery is the size and weight penalties imposed by the installation of the system.
In this work, a parametric evaluation of Supercritical CO_2 power plants is presented. Figures of merit include overall cycle performance indices such as net power, specific power and thermal efficiency as well as the design metrics of the heat exchangers required to implement the heat balance of the cycle. This work aims to establish a preliminary estimation of the overall physical footprint of the bottoming power plant.

2. Methodology

The cycle configurations considered for the design studies are the following (Figure 1):

- Simple recuperated cycle
- Recompression cycle
- Nested expansion cycle
- Dual split nested expansion with inter-cooling cycle

For the performance calculations a suite of computational tools was developed and validated as reported by Brighenti in [1]. CO_2 gas properties were calculated from NIST miniREFPROP 9.1 [2] using the Span and Wagner equation of state [3]. A stream at a flow rate of 100 kg/s at 740 K was used as heat input to the cycle. The input variables and constants used for the comparative cycle analysis are shown in Table 1.

Table 1- Reference of input variables and constants for all cycles

Variable [Unit]	Value
Inlet compressor pressure [MPa]	7.5
Inlet compressor temperature [K]	305
Compressor isentropic efficiency [-]	0.87
Compressor delivery pressure [MPa]	22
Turbine isentropic efficiency [-]	0.85
Recuperator effectiveness [-]	0.9
MHEX pressure loss coefficient [-]	0.02
High/low pressure recuperator pressure loss coefficient [-]	0.01
Cooler pressure loss coefficient [-]	0.02
Coolant inlet temperature [K]	300

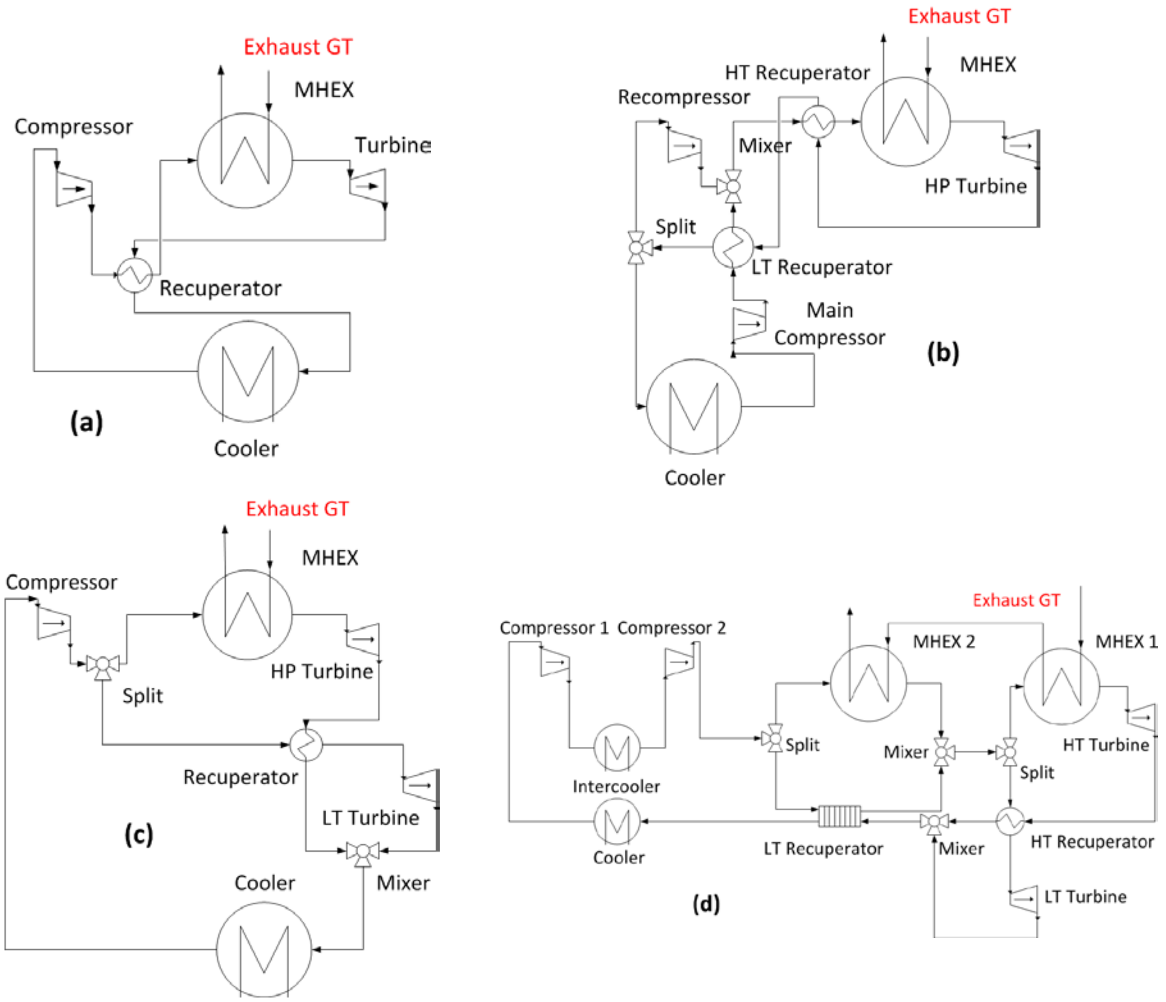

Figure 1 – Representations of the S-CO2 cycle configurations analysed. (a) Simple recuperated, (b) Recompressed, (c) Nested expansion and (d) Dual split nested expansion.

The design space of the cycles was explored in terms of total CO_2 mass flow and the CO_2 split mass flow ratio in the recompressed, nested and dual nested configurations. The split ratio of the recompressed cycle is defined as the ratio between the CO_2 mass flow that is compressed at low temperature and the mass flow that is compressed at high temperature bypassing the cooler. For the nested and dual split nested cycles (Figure 1c and d) the split ratio is defined as the ratio between the CO_2 mass flow heated by the gas turbine exhaust and the CO_2 heated by the recuperated heat within the cycle.

The maximum cycle pressure of the bottoming cycle was fixed at 22 MPa (overall pressure ratio was approximately 2.9), dictated by the mechanical robustness of the closed loop system. The effectiveness of the heat exchangers was set to 90%.

For the calculation of the required coolant flow rate, the water temperature was assumed at 15°C, with 20°C return temperature to comply with environmental limitations for water cooled power plants. For the heat exchanger UA calculations a first order backward difference discretization of the energy equation was used to account for aggressive changes in fluid proprieties variation.

3 . Results and discussion

Table 2 summarises the performance metrics of the S-CO_2 Brayton cycles under consideration. The size of the heat exchangers required for each cycle are shown in Table 3. The reported performance and size values correspond to the point of maximum net identified for each configuration. Figure 2 summarizes the requirements of coolant water mass flow for each cycle, noticing that the dual split nested cycle includes values from intercooling services.

Table 2- Performance comparison for maximum power output of S-CO_2 Brayton cycles for waste heat recovery

Cycle Layout	% Diff CO_2 mass flow rate	% Diff Waste Heat Recovered	% Diff Specific power	% Diff Net power	Efficiency [%]
Simple Recuperated	117 kg/s (reference)	27.2 MW (reference)	60.3 kJ/kg (reference)	7.0 MW (reference)	25.8 (reference)
Recompression	2.6%	-9.2%	-4.0%	-0.9%	28.1
Nested expansion	8.5%	30.5%	0.2%	10.0%	21.6
Dual split nested expansion w intercooling	45.3%	55.5%	-0.5%	45.7%	24.1

Table 3- Heat exchanger size expressed in UA [kW/K] for maximum power output of the S-CO_2 cycle configurations.

Cycle Layout	Main	HT Recuperator	LT Recuperator	Cooler	Intercooler	**% Diff Total**
Simple Recuperated	402	503	n/a	666	n/a	**1571(Ref)**
Recompression	345	538	558	560	n/a	**27.4%**
Nested expansion	592	508	n/a	1000	n/a	**33.7%**
Dual split nested expansion w intercooling	842	1660	420	654	532	**161.5%**

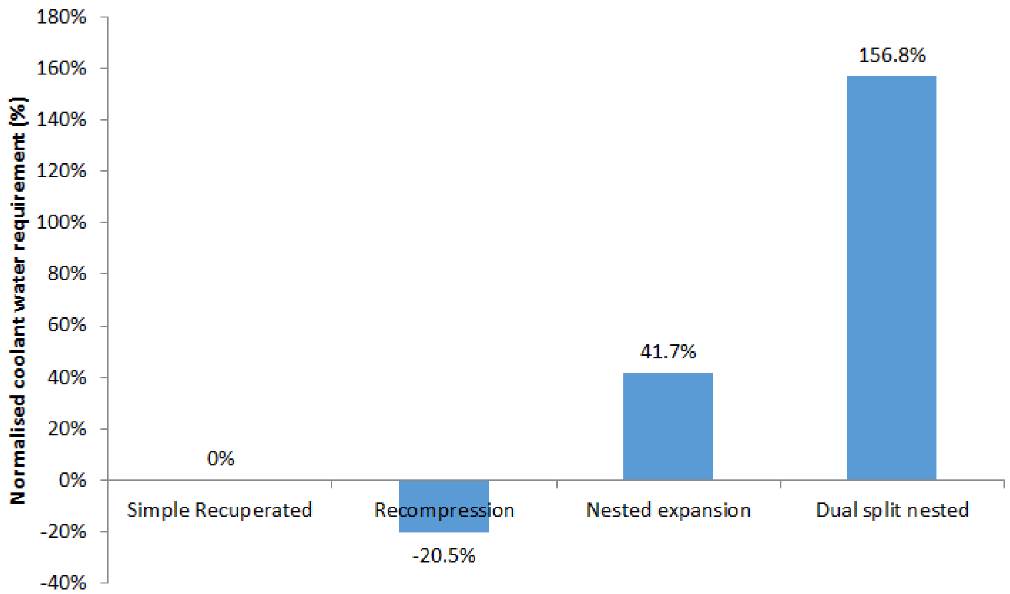

Figure 2- Normalised coolant water mas flow requirement for maximum power output of the four S-CO_2 Brayton cycles for waste heat recovery applications.

For the herein considered characteristics of the input heat stream (100 kg/s at 740 K), the simple recuperated S-CO_2 Brayton cycle is capable of achieving an output net power of 7 MW with a nominal required CO_2 mass flow of <120 kg/s. This is the lowest required CO_2 flow in the studied configurations and enables the design of compact heat exchangers in size, satisfying at the same time the prescribed pressure drop in both streams. For the simple recuperated cycle the recuperator UA size is 50% less than the two recuperators required for the recompression cycle (Table 3), and is the most compact of the four cycles.

The recompression cycle (Figure 1b) proposed by Angelino [4] is considered to be the best-suited cycle for nuclear applications [5]. The current study confirmed that for the range of CO_2 flow rates considered this is the configuration that features the highest thermal efficiency. However, the achieved heat recovery is 9.2% lower than the simple recuperated configuration (Figure 1a). Previous studies undertaken by Dostal *et al.* [6] and by Cho *et al.* [7] showed that the net power benefits from a recompression cycle (or its variations) may not always justify the increased complexity of the system.

The nested expansion cycle layout (Figure 1c), introduces an additional turbine to split the expansion of the simple recuperated configuration. This second turbine expands a stream heated after the first expansion at the high pressure turbine. This helps in reducing the stresses in the hot components of the system. This configuration shows a 10% increase in net power compared to the baseline case (Table 2). However, an approximately 40% larger heat exchange area and cooling water requirement is needed compared to the simple recuperated configuration (Table 3 and Figure 2 respectively).

Table 2 shows that the highest value of net power is achieved by the dual split nested expansion cycle reported by Cho *et al.* [7] and shown in Figure 1d. In this configuration the heat transfer from the main hot stream is controlled by a flow split. A reduced amount of CO_2 mass flow at high pressure and low temperature is introduced into the MHEX 2, whereas the remaining CO_2 mass flow at high pressure and high temperature is introduced into the MHEX 1. As a result the major part of the heat recovery occurs within MHEX 1 where the specific heat of CO_2 is lower. Additionally, the recuperation of heat after mixing the outlet streams from both turbines reduces the amount of heat rejected at the cooler. This approach allows for a large heat recovery and therefore a high net power generation that exceeds that of the nested cycle. However, the implementation of multi-stage heat transfer requires a notable increase in the total UA size of the heat exchangers used by approximately 160% compared to the baseline cycle configuration. High cooling mass flow, 156.8% higher than the simple recuperated configuration (Figure 2), is due to the bigger CO_2 mass flow rate of the cycle and the presence of an intercooler at the inlet of the second compressor, which increases the need for pumping power. The double flow splitting represents a challenge from the control point of view, as this will affect the rotational speed of both turbines and the recuperator performance.

4. Conclusions

In this work a comparative design study of four S-CO_2 power cycle configurations was conducted. Figures of merit such as cycle performance, the required CO_2 flow rate, cooling requirements and size of the heat exchangers involved were evaluated for the peak net power point of each configuration. Although the dual split nested expansion with intercooling, has the potential of the highest net power output at almost the highest possible efficiency of the studied configurations, it features notably higher cooling requirements than the simple recuperated configuration and consequently physically larger heat exchangers. On the other extreme, the simple recuperated cycle was found to require physically more compact heat exchanger configurations at modest net power output and efficiency values. A recompression cycle, despite being more efficient relatively to the baseline case by 3 percentage units, comes with around 10% less power output potential. Finally, the nested cycle should be considered for further investigation as it showed the best compromise between performance and physical footprint.

5. References

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6. List of Symbols

HT	High temperature
LT	Low temperature
UA	Coefficient of overall heat transfer [kW/K]
MHEX	Main heat exchanger

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