THERMODYNAMIC ANALYSIS OF THE SUPERCRITICAL CO_2 POWER CONVERSION CYCLE AT LOW HEAT-SOURCE TEMPERATURE

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ABSTRACT

The supercritical CO_2 (sCO_2) cycle is being considered as one of the most promising options for future energy conversion systems. It potentially offers higher thermal efficiency and lower sensitivity to pressure losses than the Brayton gas cycles, smaller size turbomachinery and simpler system layout than the conventional Rankine steam system. Numerous studies have been conducted over the last decade to assess the benefit of its advantages for various applications. This paper presents a thermodynamic analysis, including an approach based on exergy concept, of the sCO_2 cycle at a turbine inlet temperature (TIT) of 310°C and a compressor inlet temperature of 32°C. Parametric studies are performed with various operating conditions. It has been found that the thermal efficiency changes slightly in a wide range of compressor inlet pressure for turbine inlet pressures above 200 bar. The compressor could be therefore designed to operate far from critical point, ensuring its safe operation in the region of smoother change of CO_2 properties. Nevertheless, working closer to critical point has been indicated to save cycle's recuperation power and thus heat exchangers volume.

Keywords: sCO_2 cycle, irreversibility, efficiency, operating conditions

INTRODUCTION

Thermal to mechanical energy conversion can be performed by utilizing a thermodynamic cycle. Today, the steam cycles based on the familiar Rankine cycle dominate power plants including all thermal solar, biomass, coal-fired and nuclear power plants. Rankine steam cycle offers a good efficiency due to low compression work of the pump in liquid phase and high expansion ratio in the turbine. The maximum temperature is limited due to the metallurgical consideration of turbine blades. On the other hand, the Brayton gas closed cycle allows operation at higher temperature, and thus higher cycle efficiencies. This cycle, however, requires large energy input for the compression process and is very sensitive to pressure losses.

Thermodynamic cycle using sCO_2 as working fluid was claimed to avoid most of problems of the Rankine steam and Brayton gas cycle and yet retains many of their advantages [1–4]: (i) potentially high efficiency thank to low compression work in the reduced compressibility region near critical point; (ii) smaller size of the turbomachinery resulting of the high density working fluid; (iii) simpler system layout; (iv) less sensitivity to pressure losses; (v) better match of temperature profile to that of the heat source providing by the fluid in the supercritical region.

The sCO_2 cycle was stated elsewhere [3] to be more efficient than the traditional Rankine steam cycle for high heat-source temperature applications, at TITs above 550°C. The present work provides a thermodynamic analysis of the sCO_2 cycle for a lower temperature range, specifically at a TIT of 310°C of the conventional Pressurized Water-cooled Reactors. The dependence of the cycle efficiency and the recuperation power on different operating conditions has been investigated. Exergy analysis, a technique based on the Second law of thermodynamics, will also be used to identify the locations and magnitudes of irreversibility.

BACKGROUND

S-CO_2 cycle configurations

The first proposed sCO_2 cycle is a simple regenerative cycle, operating entirely above the critical pressure with the compression performed in the liquid region [1]. Its configuration is similar to the regenerative Brayton gas closed-cycle (cf. Figure 1). Due to real gas behaviour of CO_2 near critical point, heat capacity depends not only on the temperature but also on the pressure. In the recuperator, heat capacity of the low pressure hot stream (2-3) is much smaller than that of the high pressure cold stream (5-6). Because of this imbalance, large temperature difference and thus large irreversibility are found in the recuperator.

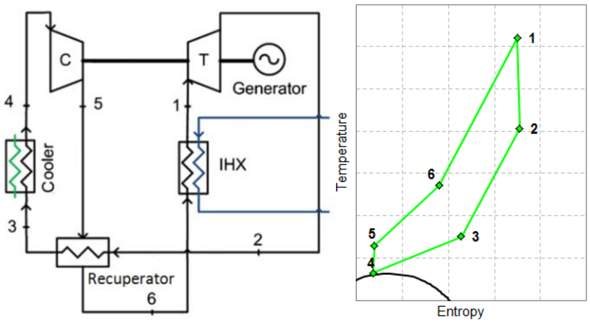

Figure 1: The simple regenerative sCO_2 cycle and its T-s diagram. Note that the compressor inlet conditions are shifted to near critical point in the T-s diagram compared to original cycle of Feher [1].

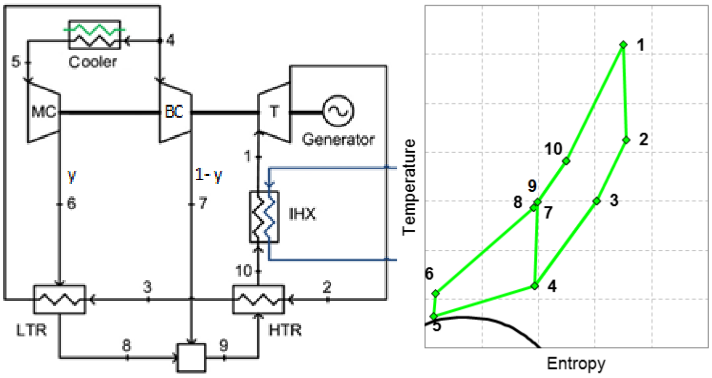

Figure 2: The recompression cycle proposed by Angelino [2] and its T-s diagram.

To overcome this so-called pinch-point problem resulting from heat capacity imbalance, Angelino [2] introduced the pre-compression, the recompression, and the partial cooling cycles. These cycles were initially proposed to operate in condensing mode, i.e. two phase fluid in the heat rejection process. Later, Dostal [3] suggested switching the compression to the supercritical region to adapt the cycles to worldwide heat-sink temperatures and to avoid the possible cavitation problems with pumps. Among others, the recompression one was seen as a good compromise between layout complexity and cycle performance.

In the recompression cycle, heat capacity imbalance between the hot and cold streams is overcome by sending only a fraction of flow through the cold side of the first recuperator (noted as LTR for Low Temperature Recuperator). The rest of the flow, split before entering the cooler (4), passes through a so-called recompressing compressor (or bypass compressor). These two high pressure streams are then mixed (9) before passing the second recuperator (noted as HTR for High Temperature Recuperator). Note that flow splitting is characterized by the Flow Split Ratio (FSR) which is the ratio of the mass flow rate across the Cooler to the total mass flow rate.

Exergy analysis of the CO_2 recompression cycle

The exergy concept developed in different parts of the world was brought together and integrated into a practical method of thermodynamic analysis in [5]. According to that, exergy represents the upper limit on the amount of work a system can deliver without violating any thermodynamics law as it comes to equilibrium with the reference environment. On the contrary to energy, exergy is not a conserved, but destroyed quantity, except in case of ideal, reversible processes. Exergy destruction is proportional to the entropy created due to irreversibility associated with any real process.

Exergy analysis measures the deviation of the considered processes from ideal ones, providing more insights and more meaningful evaluation of the system efficiency compared to the energetic approach. General methods for formulating criteria of performance for specific thermal plants are outlined in [6]. A comprehensible review of its application on thermal power plants can be found in [7] while a concrete example was given in [8]. Applying to the sCO_2 recompression cycle, specific irreversibility of the components, as well as energy balance used in traditional energetic approach, is given in Table 1.

Components	Energy balance	Specific irreversibility
Turbine	$w_T = h_1 - h_2$	$i_T = \psi_1 - \psi_2 - w_T$
Main compressor	$w_{MC} = h_6 - h_5$	$i_{MC} = y(\psi_5 - \psi_6) + w_{MC}$
Bypass compressor	$w_{BC} = h_7 - h_4$	$i_{BC} = (1-y)(\psi_4 - \psi_7) + w_{BC}$
HTR	$h_2 - h_3 = h_{10} - h_9$	$i_{HTR} = (\psi_2 - \psi_3) + (\psi_9 - \psi_{10})$
LTR	$h_3 - h_4 = h_8 - h_6$	$i_{LTR} = (\psi_3 - \psi_4) + y(\psi_6 - \psi_8)$
Cooler	N/A	$i_{CL} = y(\psi_4 - \psi_5)$
Mixer	$h_9 = (1-y)h_7 + yh_8$	$i_{MX} = (1-y)\psi_7 + y\psi_8 - \psi_9$
IHX	$q_{in} = h_1 - h_{10}$	$i_{IHX} = \psi_{in} + \psi_{10} - \psi_1$

Table 1: Energy balance and specific irreversibility of sCO_2 cycle components. *h* stands for specific enthalpy, *w* for specific work, *ψ* for specific exergy, *y* for FSR, *i* for specific irreversibility, and q_{in} for specific heat input to the cycle. State points are referred to Figure 2.

Specific exergy transferred to the cycle at the IHX and that at other state points are given by

$$\psi_{in} = q_{in}(1 - T_o / T_r)$$

$$\psi = h - h_o - T_o(s - s_o)$$

in which T_o and T_r are environment and heat-source temperatures, h_o and s_o are enthalpy and entropy of the environment. Exergy efficiency of the cycle is defined as the ratio of exergy output to the exergy input through the IHX

$$\eta_{II} = (w_T - w_{MC} - w_{BC}) / \psi_{in} = 1 - \left(\sum i\right) / \psi_{in}$$

Thermal efficiency of the cycle is defined as

$$\eta_{Thermal} = (w_T - w_{MC} - w_{BC}) / q_{in}$$

MATERIALS AND METHODS

Cycle modelling tool

In CYCLOP (for CYCLe Optimization), the CEA tool for power conversion cycle modelling [9], a cycle is represented by a set of fluid loops built from energetic components (reactor core, turbomachines, heat exchangers) which are described by macroscopic parameters such as efficiency, pressure losses, and mass flow. Components are connected by points where thermodynamic states (temperature, pressure) are stored. This tool solves automatically mass and energy balances based on first thermodynamics law for all components of the cycle from a minimum set of input data, allowing all cycle parameters to be quickly modified and optimized using the Nelder-Mead algorithm [10]. CYCLOP has been extensively used in previous studies for a wide range of cycles including Rankine steam and Brayton gas cycles for a Gas-cooled Fast Reactor [9], as well as the sCO_2 cycle for a Sodium-cooled Fast Reactor [11]. Note that CO_2 properties are referred to the Helmholtz free energy equation of state [12], which is seen as the most accurate one to describe thermodynamic properties for scientific and engineering applications.

Preliminary operating conditions

Referring to a previous study [13], the following conditions have been selected as the preliminary operating conditions of the sCO_2 recompression cycle (noted as cycle RC-0):

- Turbine inlet temperature: 310°C;
- Compressor inlet temperature: 32°C;
- Compressor outlet pressure (or maximum pressure):150 bar;
- Compressor inlet pressure (or minimum pressure): 77.082 bar;
- Flow split ratio: 0.7;
- Total pressure losses: 3.5 bar.

Concerning the components performance, a total-to-total isentropic efficiency of 93% has been used for the turbine while those for the main and bypass compressors are 88% and 89%, respectively. Each recuperator is set to 92.5% of effectiveness, with additional control on a minimum pinch-point temperature of 5°C. The cycle thermal efficiency at these preliminary conditions is 27.8%.

From the preliminary conditions, the thermal efficiency and the recuperation power, i.e. the power transferred through two recuperators, are parameterized onto several parameters of interest: the flow split ratio, the minimum and maximum pressures. For each set of these inputs, the cycle is re-optimized for highest efficiency. Exergy analysis will be performed with a reference temperature of 20°C and a heat-source temperature of 323°C.

RESULTS AND DICUSSIONS

The FSR is firstly investigated while other parameters are kept at their preliminary conditions. Thermal efficiency and component irreversibility of the sCO_2 recompression cycle, with those of the simple cycle, are illustrated in Figures 3 and 4. It is found that flow splitting is an effective way to reduce the irreversibility in the HTR, as well as in the IHX. However, too small FSR leads to a considerable increase of irreversibility in the LTR. Hence, there exists an optimal FSR which satisfies the compromise of these opposite trends. The preliminary cycle with the optimal FSR of 0.54, noted as cycle RC-1, has an efficiency of 30.5%.

The influence of the compressor inlet pressure and the maximum pressure on the thermal efficiency and recuperation power is illustrated in Figures 5 and 6. Note that FSR is re-optimized for highest efficiency for each point represented in these graphs. Higher maximum pressure cycle offers higher efficiency and requires smaller power to be transferred through two recuperators. The efficiency gain, however, seems to be very small above 200 bar. From this limit, the efficiency changes slightly with any increase of the compressor inlet pressure at a constant maximum pressure. Such behavior would favor the operation of the cycle in region of smother change of CO_2 properties, i.e. far from critical point, without considerable decrease of its performance. This could be viewed as a safe choice to avoid any possible implication on the stability of the cycle that might occurred in the region of sharp change of CO_2 properties. Nevertheless, cycle operating at low compressor inlet pressures (not below 77.5 bar) is a highly recommended option due to lower recuperation power, and thus smaller recuperator volume. If stable operation of the cycle of in this region of interest could be demonstrated, such a choice would fatherly lead to save components cost and system footprint.

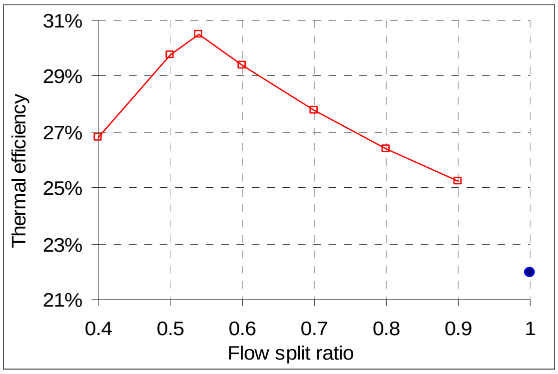

Figure 3: Dependence of the thermal efficiency on the FSR. Blue dot represents the efficiency of the simple cycle.

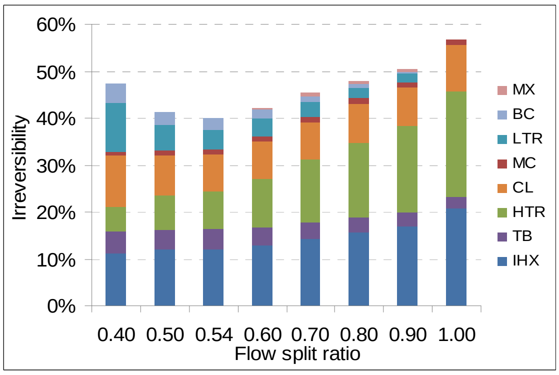

Figure 4: Dependence of the irreversibility of cycle components on the FSR. The column at FSR of 1.00 corresponds to the simple cycle.

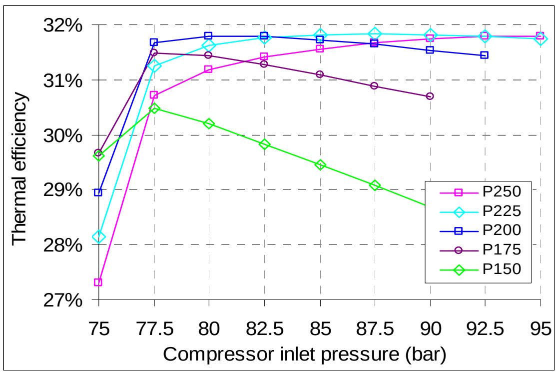

Figure 5: Dependence of the thermal efficiency on the compressor inlet and maximum pressures.

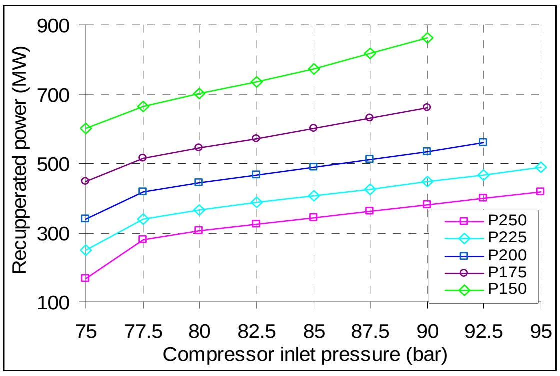

Figure 6: Dependence of the recuperation power on the compressor inlet and maximum pressures.

The optimization of cycle operating conditions leads to the highest efficiency of 31.8% at a maximum pressure of 225 bar and a compressor inlet pressure of 87.5 bar (noted as cycle RC-2). Details of components irreversibility of previously marked cycles are given in Figure 7. The optimization of both FSR and main compressor operating conditions significantly decreases irreversibility in the HTR to only a small fraction of that in the IHX and the Cooler. Note that part of irreversibility in these components could be claimed due the assumption of constant heat-source and heat-sink temperatures adopted in this analysis. To correctly represent it, the temperature profiles in these heat exchangers should be known. Unfortunately, such information is not available for the present study.

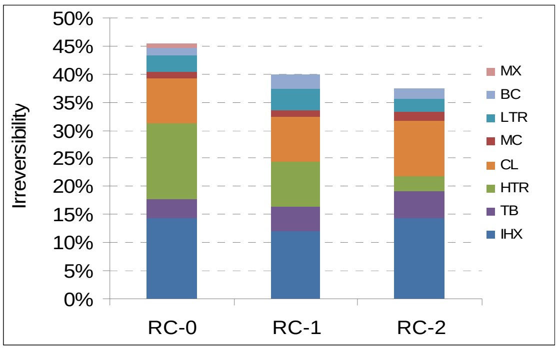

Figure 7: Irreversibility of the sCO_2 cycle components at different operating conditions.

	RC-0	RC-1	RC-2
Maximum pressure, bar	150	150	225
Minimum pressure, bar	77.082	77.082	87.5
FSR	0.700	0.540	0.657
Thermal efficiency	27.8 %	30.5 %	31.8 %

Table 2: Summary of the operating conditions and the efficiencies of the cycles RC-0, RC-1, and RC-2.

CONCLUSIONS

In the present work, thermodynamic analysis, including energetic and exergy approaches, of the sCO_2 cycle at a TIT of 310°C has been performed. The thermal efficiency and the recuperation power have been parameterized onto different operating conditions. It has been found that higher maximum pressure provides more flexible choice of the compressor inlet pressure, i.e. far from the region of sharp change of CO_2 properties to avoid any possible implication on the stable operation of the cycle, with only small decrease in cycle efficiency. Nevertheless, lower compressor inlet pressure has found to save recuperation power and thus HXs volume. Exergy analysis has highlighted how flow splitting in the sCO_2 recompression cycle and the optimization of operating conditions could overcome the inherent heat capacity imbalance between hot and cold sCO_2 streams and thus significantly decrease irreversibility in both HTR and LRT. Future work could focus on the correction of exergy losses in the IHX and the Cooler thank to available temperature profiles in these components, and the investigation of other cycle layouts in view of reducing the remaining irreversibility.

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