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Process Synthesis of an S-CO_2 Brayton Cycle Operating at an Ultra-Supercritical Steam Cycle Level

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Supercritical CO_2 (S-CO_2) Brayton cycle perceives the advantages of high cycle efficiency and compact turbine machinery. In certain temperature ranges, its cycle efficiency can be higher than that of a steam Rankine cycle. However, process synthesis for an S-CO_2 cycle operating at an ultra-supercritical steam cycle level, aiming at large-scale power generation, is still lacking, and systems analysis and comparison between these cycles are needed. In this paper, a double reheat recompression S-CO_2 Brayton cycle model for a coal-fired power plant is presented, with which various options of mass and energy integration amongst the process can be studied and optimized. Thermodynamic analysis of various process integration options is carried out focusing on impacts on key cycle parameters. Process synthesis results show that when turbine inlet temperature reaches 600 °C, thermal efficiency of the S-CO_2 cycle can reach 52.5 %, 4.5 percentage points higher than an ultra-supercritical steam cycle operating at the same level.

1. Introduction

At present, coal-fired power plants generally use steam Rankine cycle for power generation. In order to improve cycle efficiency, the most economical and effective method is to increase the steam temperature and pressure at the turbine inlet. In ultra-supercritical units, steam parameters near turbine inlet have reached 300 bar/600 °C /620 °C (Li et al., 2014). It raises high requirements for steam turbine material (Sun et al., 2014). Another method to improve the unit efficiency is to combine with the gas turbine cycle, with increase of system complexity (Madzivhandila et al., 2009).

One of the most promising methods to achieve higher efficiency at the same parameters is supercritical CO_2 Brayton cycle. When CO_2 functions as the working fluid and operates entirely above the critical point, it offers extremely effective performance for power generation (Kim et al., 2012). S-CO_2 cycle is characterized by high efficiency, low cost and compact structure. CO_2 has high density in the supercritical zone, which leads to a decrease of the volume size of mechanical parts.

In terms of process design, Feher (1967) revealed the Pinch Point problem in S-CO_2 cycle. The Pinch Point is the location in the recuperator where the minimum temperature difference exists. Normally, the Pinch Point appears at the recuperator inlet or outlet (Dostal et al., 2017). However, in the single heat recovery S-CO_2 cycle, due to the huge difference in specific heat capacity between the cold and hot fluids on both sides of the heat exchanger, the Pinch Point would appear locally inside the recuperator. This leads to a serious deterioration of the heat recovery conditions and a significant reduction in efficiency. Later, Angelino (1968) put forward the recompression cycle, which divided the CO_2 at the outlet of the low-temperature heat exchanger, and made the heat capacity of the fluid on both sides of the heat exchanger match each other by controlling the flow rate, thus overcoming the Pinch point problem. At present, research about S-CO_2 cycle mainly focuses on nuclear energy, solar energy and other fields (Ahn et al., 2015). Studies on S-CO_2 cycle applied in ultra-supercritical coal-fired power generation is still lacking. Problems in different process construction, system integration and thermal characteristics are remained to be investigated.

The purpose of this paper is to study the cycle performance and system characteristics of S-CO_2 cycle applied in an ultra-supercritical steam cycle level. Referring to the parameters of typical coal-fired generating sets, this paper gives a preliminary double reheat recompression S-CO_2 cycle model design. The optimum operating

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conditions of the system are obtained by analyzing cycle parameters. The influence of certain parameters and the reheat process on the efficiency is discussed.

2. Process Configuration

2.1 Double reheat recompression S-CO_2 cycle layout

Considering the typical characteristics of coal-fired power plants, this paper adopts the design of double reheat and combines with the recompression cycle. Schematic of overall cycle is shown as Figure 1a, and corresponding entropy diagram is shown as Figure 1b. S-CO_2 exhaust enters the high temperature recuperator (HTR) for heat exchange, and the temperature decreases. Then it enters the low temperature recuperator (LTR) for heat release. At the outlet of LTR, there is a split point of working fluid. Part of S-CO_2 enters the secondary compressor directly, and then enters the HTR for heat absorption. The other part of the split flow goes to the heat sink for cooling, which is then compressed by the main compressor, followed by the LTR and HTR. S-CO_2 experiences double reheat process, and then goes into turbines, so as to complete a thermal cycle.

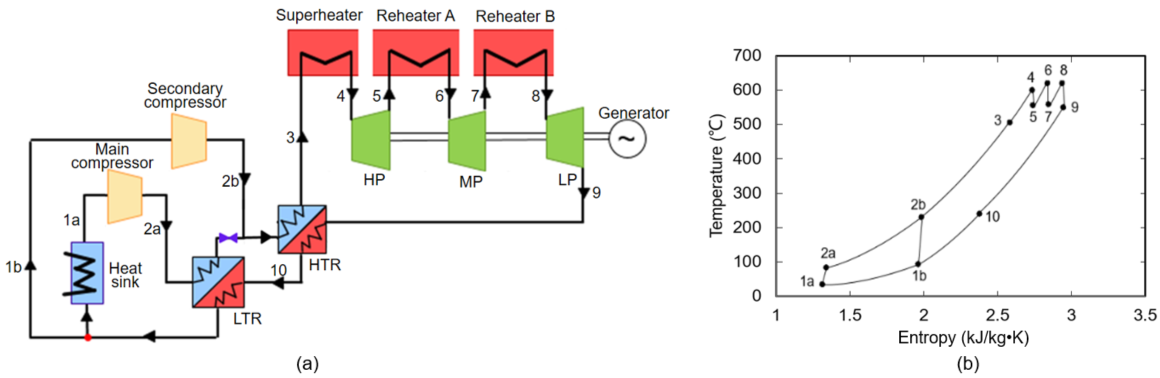

Figure 1: (a) Schematic of double reheat recompression S-CO_2 cycle and (b) Temperature–entropy diagram

2.2 Energy analysis

General assumptions in this analysis are as follows. Complex internal structure of coal-fired boilers is not considered, and the heat loss inside the boilers is ignored. Friction losses in pipelines are negligible, whilst pressure drop of working fluid in each components is 1 bar. Isentropic efficiency of compressors is 89 %, whilst that of turbines is 93 %. Mechanical efficiency of all compressors and turbines is 99.8 %. The cycle process is performed with EBSILON and properties of CO_2 refer to REFPROP-NIST.

Table 1: Nomenclature

Nomenclature		Subscripts	
h	enthalpy	Com, main	main compressor
m	mass flow rate	Com, s	secondary compressor
P	pressure	drop	pressure drop
Q	rate of heat	HP	high pressure turbine
S-CO_2	supercritical CO_2	HTR	high temperature recuperator
T	temperature	H, A	reheater A
W	rate of work	H, B	reheater B
α	split ratio	H, S	superheater
η_i	isentropic efficiency	LP	low pressure turbine
η_m	mechanical efficiency	LTR	low temperature recuperator
η_{th}	cycle thermal efficiency	MP	medium pressure turbine
		Pin	Pinch Point
		s	isentropic
		tot	total

Since the core components applicable to S-CO_2 Brayton cycle, such as heat exchangers and turbines, are still in the conceptual design and small-scale test stage, this paper adopts an ideal processing method to model the components. The energy analysis of several components is carried out as examples below.

Select high pressure turbine to introduce the expansion process, and the analysis of other turbines is similar:

$$\eta_{i,\mathrm{HP}} = \frac{h_4 - h_5}{h_4 - h_{5,s}} \tag{1}$$

$$\eta_{m,\mathrm{HP}} = \frac{W_{HP}}{m_{tot}(h_4 - h_5)} \tag{2}$$

For the main compressor:

$$\eta_{i,\mathrm{Com,main}} = \frac{h_{2a,s} - h_{1a}}{h_{2a} - h_{1a}} \tag{3}$$

$$\eta_{m,\mathrm{Com,main}} = \frac{m_{\mathrm{Com,main}}(h_{2a} - h_{1a})}{W_{\mathrm{Com,main}}} \tag{4}$$

When designing and manufacturing of heat exchangers, the temperature difference at the Pinch Point is a key design parameter, which is used to measure the heat exchange efficiency of heat exchangers. The temperature difference at the Pinch point of all heat exchangers in this paper is set 10 K.
For high temperature recuperator:

$$\Delta T_{Pin} = T_{10} - T_{2b} \tag{5}$$

$$Q_{HTR} = m_{tot}(h_9 - h_{10}) = m_{\mathrm{tot}}(h_3 - h_{2b}) \tag{6}$$

As the specific heat capacity of hot flow in LTR is much smaller than that on the cold side, there is a split point located at the outlet of LTR, to avoid the Pinch Point problem mentioned above. That is, only a part of flow that serves as the cold flow in LTR. The split ratio is defined as follows:

$$\alpha = \frac{m_{\mathrm{Com,main}}}{m_{tot}} \tag{7}$$

$$\Delta T_{Pin} = T_{1b} - T_{2a} \tag{8}$$

$$Q_{LTR} = m_{tot}(h_{10} - h_{1b}) = m_{\mathrm{Com,main}}(h_{2b} - h_{2a}) \tag{9}$$

The thermal efficiency of the cycle is calculated by:

$$\eta_{th} = \frac{W_{HP} + W_{MP} + W_{LP} - W_{\mathrm{Com,main}} - W_{\mathrm{Com,s}}}{Q_{H,S} + Q_{H,A} + Q_{H,B}} \tag{10}$$

2.3 Parameters adapted to coal-fired power plants and S-CO_2 cycle

Parameters at the inlet of turbines represent the maximum temperature and pressure of the whole cycle and also determine the scale of a cycle. Refer to the ultra-supercritical steam Rankine cycle, the highest temperature reaches 620 °C and the highest pressure reaches 300 bar (Li et al., 2014). Same values are selected in this paper to make the cycle adapted to coal-fired power plants.
Minimum temperature and pressure of the whole cycle occur at the inlet of compressors. Studies indicated that, with values of compressor inlet parameters approaching the critical point of CO_2, compression work consumed will be significantly reduced (Mecheri et al., 2016). In this paper, a certain safety interval is reserved to ensure that the whole cycle runs in the supercritical region. Specific values are presented in Table 2.

Table 2: Parameters adapted to coal-fired power plants

Parameter	Value(s)	Unit
Total mass flow rate	14,940	t/h
High pressure turbine inlet temperature	600	°C
High pressure turbine inlet pressure	30	MPa
Medium pressure turbine inlet temperature	620	°C
Low pressure turbine inlet temperature	620	°C
Main compressor inlet temperature	32	°C
Main compressor inlet temperature	7.9	MPa

2.4 Parameters design

Given values in Table 2, parameters of medium pressure turbine (MP) inlet and low pressure turbine (LP) inlet are remained uncertain. In this paper, 0.5 MPa is selected as a scale, and the relationship with cycle efficiency is obtained through hypothesis analysis, as shown in Figure 2a. It can be seen that when the cycle efficiency reaches maximum, the MP inlet pressure is 215 MPa, and the LP inlet pressure is 135 MPa.
The same method is adopted to determine the value of the main compressor inlet flow. Relationship between main compressor inlet flow and cycle efficiency is shown in Figure 2b. When the cycle efficiency reaches maximum, the split ratio is 0.669.

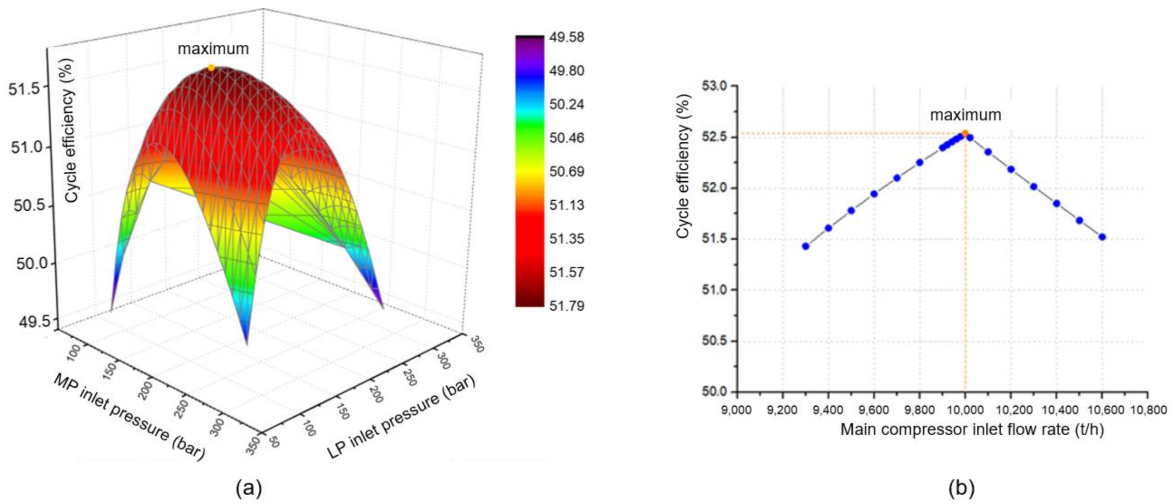

Figure 2: Cycle efficiency over (a) MP/LP inlet pressure and (b) main compressor inlet flow rate

3. Result analysis

3.1 Influence of main turbine inlet parameters

With parameter values given above, cycle efficiency of the double reheat recompression S-CO_2 cycle is 52.5 %. Compared with Rankine steam cycle with the same turbine inlet parameters, cycle efficiency increases 4.5 percentage points (Li et al., 2014). In this chapter, the influence of cycle parameters and different process designs on cycle efficiency is discussed.
As mentioned above, parameters at compressor inlet should be as close as possible to the critical point of CO_2, to make better use of the physical properties near the critical point to improve cycle efficiency. However, drastic changing physical properties put forward high requirements for the stability of the heat sink, and the extremely high energy density also brings challenges to the design of the compressor blade. It requires a balance between cycle efficiency and operating stability. This paper studies the relationship between these parameters and the cycle efficiency. Figure 3 shows cycle efficiency over main compressor inlet parameters.
Figures 3 indicates that S-CO_2 cycle provides a higher efficiency over steam Rankine's cycle (48 %) in a certain temperature range. It also shows that cycle efficiency is sensitive to main compressor inlet parameters. For a given main compressor inlet temperature, there is a unique minimum pressure that maximizes the cycle efficiency. And the optimal pressure value increases with the increase of main compressor inlet temperature.
For a specific main compressor inlet pressure, the cycle efficiency decreases with the increase of main compressor inlet temperature. With a decreasing pressure, there is a sudden drop of cycle efficiency near the critical temperature of CO_2. This phenomenon is caused by drastic change of CO_2 density near the critical point. Figure 4 shows the change of cycle efficiency and change of CO_2 density over temperature. When temperature increases from 31 °C to 35 °C, density of CO_2 is reduced by nearly 50 %, In the same range, cycle efficiency decreased by about 6 %. This can be explained by an increase in compression work. Based on these phenomena, this paper suggests a temperature range of 30 °C - 34 °C along with a pressure range of 86 bar - 94 bar when determining main compressor inlet parameters.

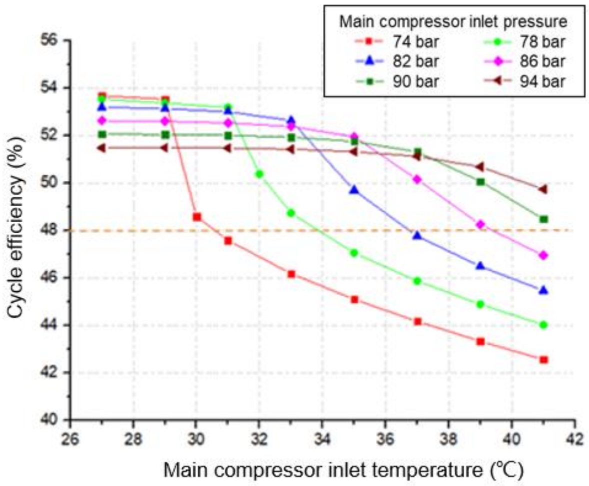

Figure 3: Cycle efficiency over main compressor inlet temperature

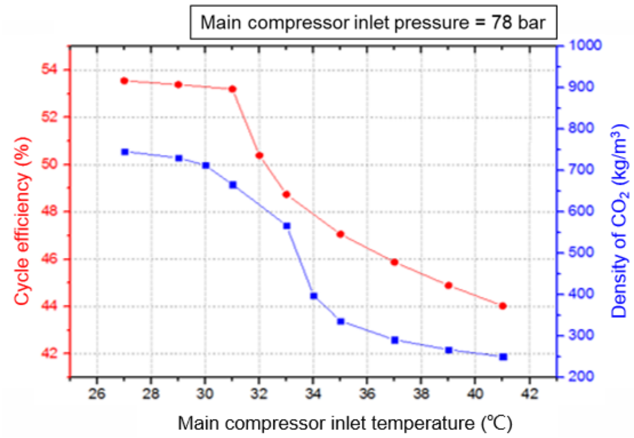

Figure 4: Cycle efficiency over main compressor inlet temperature at 78 bar and corresponding CO_2 density

3.2 Influence of reheat process

In steam Rankine cycle, the reheat process is an important measure to improve the cycle efficiency, but the construction cost and operation complexity will increase accordingly. In order to comprehensively analyse the benefits brought by reheat process to the cycle, another two different cycles are established. One of them is a recompression cycle without reheat process, and the other is a recompression cycle with single reheat process. Figure 5a shows the cycle efficiency of different cycles over HP inlet temperature. Result indicates that cycle efficiency increases nearly linearly with HP inlet temperature. The cycle efficiency can be noticeable increased with more than 2 percentage points by introducing single reheat design, whilst the gap shrinks with the increase of HP inlet temperature. And the cycle efficiency can be added another 0.4 percentage points by adopting double reheat design.

Figure 5b shows the cycle efficiency over HP inlet pressure. For a given HP inlet temperature (600 °C), there is only one HP inlet pressure that maximizes the cycle efficiency. Despite different designs of reheat process for each cycle, the optimal HP inlet pressure value occurs near 300 bar. The cycle efficiency can be effectively increased by 1.5 - 2.5 percentage point by introducing single reheat process, and the gap enlarge with the increase of HP inlet pressure. By adopting double reheat process, the cycle efficiency can be increased by another 0.4 percentage point. When HP inlet pressure is greater than the optimal value, the cycle efficiency of recompression cycle without reheat process decreases more sharply than the other cycles.

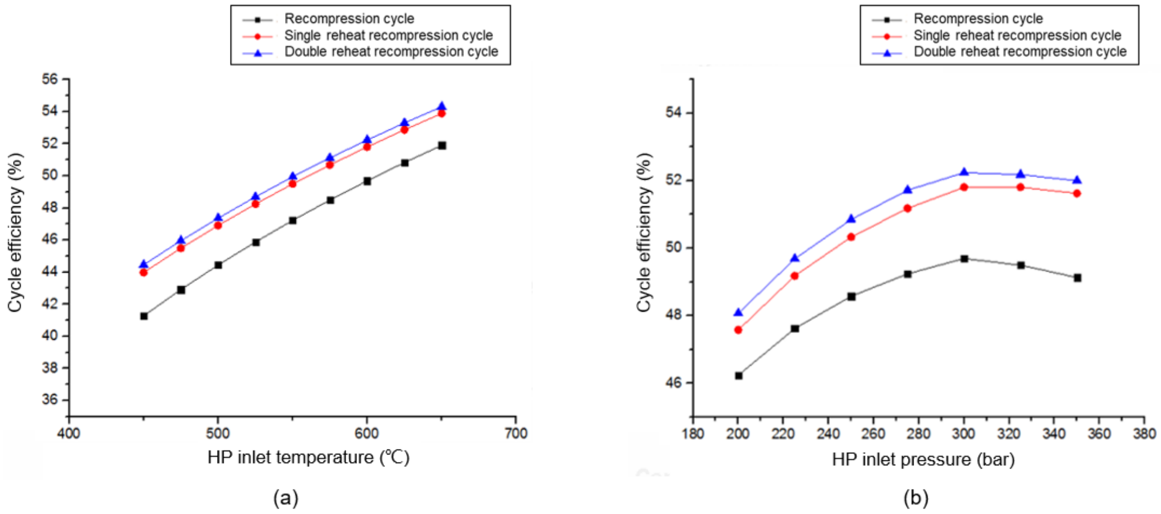

Figure 5: Cycle efficiency of different cycles over (a) HP inlet temperature and (b) HP inlet pressure

4. Conclusions

In this paper, a double reheat recompression S-CO_2 Brayton cycle is proposed. Cycle runs with same turbine inlet parameters of ultra-supercritical steam Rankine cycle. Results indicates that S-CO_2 Brayton cycle achieves an improvement on cycle efficiency of 4.5 percentage points. Moreover, the influence of main compressor inlet parameters is discussed. Temperature ranges of 30 °C - 34 °C and pressure range of 86 bar - 94 bar are suggested when determining compressor inlet parameters. The role of reheat process, which is widely used in coal-fired power plants nowadays, is investigated by comparing three different cycle designs. An improvement in cycle efficiency of nearly 2 percentage point can be observed when introducing a single reheat process. The contribution of a double reheat process is less obvious, with another 0.4 percentage point increase.

As a conclusion, this paper provides a novel concept for utilization of the S-CO_2 Brayton cycle in coal-fired power plants. For practical applications, specific design of cycle components is still in urgent need, specially heat exchangers and turbines. Heat transfer mechanism in and corresponding process integrations are key topics in the further research.

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