Preliminary analysis of supercritical CO$_2$ Brayton cycle applied to the solar thermal plant

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Abstract. The use of a highly compacted and efficient supercritical CO$_2$ (S-CO$_2$) Brayton cycle has become an effective way in utilizing solar energy. This work performs a comprehensive parametric study for S-CO$_2$ Brayton cycle applied to the solar thermal plant. Several different power cycle layouts including basic regenerative cycle, recompression cycle, intercool cycle and reheat cycle are analysed by the computational models consist of various components in this paper. Then, the optimization of parameters including the cycle turbine inlet temperature and so on, is also performed based on genetic algorithm to achieve the best cycle performance. The most promising cycle configuration is selected by comparison of the compromise between cycle and recuperator performance. This study provides some reference for the related research of the S-CO$_2$ Brayton cycle integrated with solar thermal systems.

Keywords: S-CO2 Brayton cycle; genetic algorithm; power output

1. Introduction

Due to fossil fuel depletion, environmental and climate threats, Concentrating Solar Power (CSP) is playing a leading role in energy scenario which can providing carbon-free, renewable and dispatchable electric energy. More and more ways are used to convert solar heat into electricity to reduce the Levelized Cost of Electricity (LCOE) of this technology. Supercritical carbon dioxide (S-CO$_2$) cycle has been identified as a promising technology to utilize solar energy [1,2]. Supercritical fluid is a state between the gas state and the liquid state, with low compressibility near the liquid state and high viscosity near the gas state [3]. The critical temperature of S-CO$_2$ is 31.1°C, and the critical pressure is 7.38MPa [4]. The research on the application of S-CO$_2$ in Brayton cycle has been widely concerned and promoted, including system design and core components analysis [5-9]. Most of the above studies have considered so far sCO2 for nuclear power plants, while there is a growing interest in deploying them in CSP applications [10-12]. For example, S-CO$_2$ Brayton cycles have been compared with trans and sub-critical working conditions of the same fluid [13]. S-CO$_2$ cycles in CSP plants have been considered in indirect configuration [14]. The main challenges of S-CO$_2$ application, especially coupled to CSP plants which are heat exchanger design and material selection have been preliminarily studied [15]. However, most studies only compare fixed parameters when comparing thermodynamic cycle performance. It must be clear that the optimization parameters of each thermodynamic system are different after optimization, and the corresponding contrast results will also be different. In this paper, several different power cycle layouts including basic regenerative cycle, recompression cycle, intercool cycle and reheat cycle are analysed by the computational models. Meanwhile, genetic algorithm is used to optimize the parameters of the cycles and a more impartial comparison is obtained.
2. Calculation model

The S-CO₂ Brayton cycle mainly consists of three kinds of equipment: turbines, compressors, and heat exchangers and the mathematical models are given as follows.

2.1. Turbine and compressor subroutine

The evaluation of the compression or expansion process starts from the machine inlet conditions that were specified in the input. The calculation procedure is the following:

\[
\begin{align*}
(h_{c, \text{out}} - h_{c, \text{in}}) &= \left(\frac{h_{c, \text{out, id}} - h_{c, \text{in}}}{\eta_c}\right) \\
(h_{t, \text{out}} - h_{t, \text{in}}) &= \left(\frac{h_{t, \text{out, id}} - h_{t, \text{in}}}{\eta_t}\right)
\end{align*}
\]

\(h\), the enthalpy; \(\eta\), the total to total efficiency; \(c\), the compressor; \(t\), the turbine; \(in\), the inlet conditions; \(out\), the outlet conditions; \(id\), the ideal state.

Given the total to total efficiency and inlet conditions of the turbine and compressor work, the rest of the state points can be determined.

2.2. Recuperator subroutine

In this paper, printed circuit heat exchangers (PCHE) are utilized to calculate the recuperator. In order to improve the effectiveness of recuperator and take into account the minimum temperature difference of recuperator, the minimum temperature difference of recuperator \(r\) between the hot and cold side is set as 10°C. The overall heat transfer coefficient is calculated from [1]:

\[
U_{i}=(\frac{1}{U_{i,h}} + \frac{k}{t} + \frac{1}{U_{i,c}})^{-1}
\]

\(U_{i,h}\) and \(U_{i,c}\) are the heat transfer coefficients on hot and cold sides respectively which are similarly to calculate. The parameter \(k\) is the thermal conductivity of the heat exchanger material and \(t\) is the plate thickness. \(U_{i,h}\) is calculated from:

\[
U_{i,h}=(\frac{f_h}{8}(Re_h-1000)\cdot Pr_h}{1+12.7(Pr_h^{2/3}-1)\sqrt{\frac{f_h}{8}(1+\frac{PD}{L})^{2/3}}}\cdot \frac{L}{PD}
\]

Where:

\[
f_h=(\frac{1.8\lg(Re_h)-1.5}{2})^2
\]

\[
PD=\frac{4\pi D^2}{8\pi D/2+D}
\]

In the heat, the correlations above are used again to calculate the parameters of heat. Then, the required heat exchanger area can be calculated by the parameter \(D\) and \(L\).

3. A brief introduction to the circulatory system

The circulatory system studied in this paper is shown in Figure 1. The basic regenerative cycle (Fig. 1, cycle (a)) comprises of recuperator, turbine, compressor, cooler and heater. The S-CO₂ after being cooled is compressed to an applicable pressure. Then, the cold S-CO₂ exchanges heat with the hot S-CO₂ exiting the turbine. The temperature of the pre-heated S-CO₂ is further elevated by the solar receiver before being expanded in the turbine.
The recompression cycle (Fig. 1, cycle (b)) has the same components as in the basic regenerative cycle but with different numbers and arrangements. Before the cooler, the S-CO$_2$ stream is splitted into two streams, one goes to the compressor directly while the other goes through the cooler before entering the compressor. Other arrangements are similarly to the basic regenerative cycle. The intercool cycle (Fig. 1, cycle (c)) utilizes a second cooler which helps to reduce the compression work by reducing the temperature of the compressed S-CO$_2$. Besides, the arrangements can make a further bigger pressure ratio in the cycle. The reheat cycle (Fig. 1, cycle (d)) utilizes a second heater which helps to enhance the turbine work by adding heat after the expansion.

![Figure 1. Supercritical CO$_2$ compound cycle layout](image)

To study the above circulatory system a calculation model is built which including the main part, the compressor subroutine, the turbine subroutine and the heat exchanger subroutine. Carbon dioxide properties are interpolated directly from the existing physical data points. Some assumptions are made to simplify the models:

1. All the circulatory system can reach a steady state in a short time.
2. The heat loss and the pressure drop in the circulatory system are neglected.
3. The CO$_2$ temperature at the outlet of the cooler can keep constant by changing the mass flow rate of the environmental water or air.

The mapping relation of the cost function from the input to the output can be obtained after the mathematical models above are built. Meanwhile, the efficiency of the circulatory system is taken as the output of the cost function, namely the optimization target. Table 1 summarizes the main design assumptions.

| Table 1 Main assumptions for the power cycle simulation |
|-----------------------------------------------|
| fixed parameters                         | value       |
| Solar thermal available                  | 16 MW       |
| Turbine isentropic efficiency            | 85%         |
| Compressor isentropic efficiency         | 80%         |
| Regenerator end temperature difference   | 10 °C       |
4. Results and discussion

4.1. Initial state and cycle results
After the establishment and construction of the equation above, the initial state parameters, which are not optimized, are calculated first and shown in Table 2.

| Term                        | Value       | Term                        | Value       |
|-----------------------------|-------------|-----------------------------|-------------|
| Cycle maximum pressure      | 20 MPa      | Turbine inlet temperature   | 600 °C      |
| Cycle minimum pressure      | 8 MPa       | Compressor inlet temperature| 35 °C       |
| Cycle intermediate pressure | 12.62 MPa   | (utilized in intercool and reheat cycle) |

According to the fixed variables mentioned above, we get the system parameters that only use the initial state parameters before the optimization, as are shown in Table 3.

| Circulatory system          | Cycle performance          | Net power output | Exergy efficiency |
|-----------------------------|----------------------------|------------------|-------------------|
| basic regenerative cycle    |                            | 5.57 MW          | 37.16%            |
| recompression cycle         |                            | 6.54 MW          | 43.64%            |
| intercool cycle             |                            | 6.30 MW          | 42.01%            |
| reheat cycle                |                            | 5.74 MW          | 38.23%            |

4.2. Optimization results
The genetic algorithm (GA) is used to optimize the circulatory system, with a population size of 50 and an iteration number of 50. As a result, the evolution curve of the best fitness value obtained by GA algorithm is shown in Figure 2.
The cycle efficiency of circulatory systems has reached 41.31%, 47.78%, 52.27%, and 42.55% respectively. It can be found that the cycle efficiency of the circulatory system has been effectively improved. After the optimization, the cycle efficiency of the intercool cycle exceeds the recompression system, which indicates that it is necessary to make a certain optimization and then compare the cycle in the process of selecting the circulatory system. What’s more, the change of the optimized input parameters is shown in Table 4.

Table 4 Optimization parameters table

| Term and optimization interval | Cycle(a) | Cycle(b) | Cycle(c) | Cycle(d) |
|-------------------------------|---------|---------|---------|---------|
| Cycle maximum pressure / (15,25) | 24.95 MPa | 24.91 MPa | 24.82 MPa | 24.96 MPa |
| Cycle minimum pressure / (7.5,10) | 7.64 MPa | 7.87 MPa | 9.42 MPa | 7.61 MPa |
| Compressor Inlet temperature / (32,40) | 32.37 °C | 32.21 °C | 32.07 °C | 32.42 °C |
| Cycle efficiency after optimization | 41.31% | 47.78% | 52.27% | 42.55% |

Then we calculate the required exchanging area of the circulatory system and the comparison of system efficiency and the required exchanging area among the above cycles is shown in Figure 3. It can be found that the demand for heat exchanging area in the recuperators increases while some structures have been added to the regenerative cycle layout. Among the above efficient cycles (a, b, c and d), the recompressed cycle (b) stands out to be the excellent choice because it can achieve high thermal efficiency under acceptable demand for heat exchanging area. Obviously, the recompression cycle can utilize advantages in previous cycles by combining both intercool and reheat within the allowable cost range.

5. Conclusions

In this paper, several different power cycle layouts including basic regenerative cycle, recompression cycle, intercool cycle and reheat cycle are analysed by the computational models and compared after optimization using GA algorithm. Conclusions are as follows:

1. The cycle efficiency of the circulatory systems has been effectively improved (Efficiency increased by more than 4%) after the optimization which shows that the optimization algorithm used in this paper is effective.
2. It can be found that the closer the inlet temperature of compressor is, the higher the cycle efficiency is, and the best circulating pressure ratio may exist in some thermodynamic systems through the system optimization.
3. The demand for heat exchanging area in the recuperators increases while some structures have been added to the regenerative cycle layout because of the increase in the complexity of the system. Therefore, it is necessary to consider the relationship between system efficiency and
cost in actual selection.

4. After calculation, it can be found that the recompression cycle is the most suitable thermal system in the process of solar energy utilization and adding some structure on this can still bring about the improvement of system efficiency. In the actual process, we should consider the power level and cost limitation of solar power and rationally choose the thermal cycle structure to achieve better economic returns.

In conclusion, the recompression cycle is a suitable thermodynamic cycle for solar energy utilization, and it is feasible to increase part of the structure to improve cycle efficiency. This research can help the development of residual heat utilization and other engineering fields.

6. References
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