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Analysis and study on the thermodynamic performance of S-CO_2 simple Brayton cycle

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Abstract. The S-CO_2 Brayton cycle system has many characteristics such as low cost and compact structure, and is one of the hotspots in the fields of waste heat utilization, new energy and so on. In this paper, a mathematical model of the S-CO_2 simple Brayton cycle is constructed, and the cyclic characteristics are analysed. The relationship between the component parameters of the system and the calculation parameters of the cycle such as the cycle efficiency and the output net work are obtained under the design conditions. Meanwhile, calculation models of turbine and compressor under off-design conditions are established, and the characteristics of the whole cycle are analyzed. Thus, the characteristics of off-design conditions of the simple cycle are obtained. This study provides some reference for the related research of the S-CO_2 Brayton cycle and the practical engineering application.

1 Introduction

The supercritical carbon dioxide Brayton cycle system is a thermodynamic system using supercritical carbon dioxide as the working substance, which transforms the heat of the heat source into mechanical energy and finally outputs electric energy. CO_2 has good thermal stability, physical properties and safety, it is non-toxic, non-combustible and cheap. The critical temperature is 31.2 ℃, and the critical pressure is 7.38 MPa [1], which is far below the critical point of water, thus it is easy to achieve supercritical state. When CO_2 is in a supercritical state (S-CO_2), it has the density similar to liquid and the fluidity similar to gas, and there is no phase change during the cycle. This way, it can greatly improve the working capacity in turbine machineries, and under the same power generation capacity, the system volume is much smaller than other turbine machineries. In particular, there is no need for a large number of boiler piping equipment for steam making [2].

At present, research institutions in many countries or regions have carried out the research and prototype manufacturing of the S-CO_2 Brayton cycle system. The Sandy National Laboratory (SNL) has built a recompression Brayton cycle experimental system using two-stage compression, two regenerative structure, and a compact printed circuit plate heat exchanger (PCHE). The heat source power is 260kW in the initial design in 2010, and has been raised to 780W at present [3]. Tokyo Institute of Technology, Japan (TIT) also carried out the research on the S-CO_2 closed loop Brayton cycle. They improved the simple Brayton cycle by adding the process of flow rate, intermediate compression, and intermediate cooling, in order to improve the cycle efficiency [4]. On this basis, a complete set of supercritical carbon dioxide Brayton cycle experimental system was built by KAERI, in which the output power is 10kW, the single stage centrifugal compressor's pressure ratio is less than 2, and the turbine inlet temperature is 500 ℃ [5].

Iverson et al. studied the supercritical carbon dioxide Brayton cycle solar power generation system in the experimental system of 780kW. The results show that the cycle efficiency of the system can be improved by supercritical carbon dioxide Brayton cycle, especially when the turbine inlet refrigerant temperature is higher than 600℃ [6]. Sienicki et al. proposed a conceptual design of the supercritical carbon dioxide Brayton cycle system of 100MWe sodium cold fast reactor and pointed out that this system provides a cycle efficiency 1% or more higher than that of conventional steam circulation systems, and the turbine and reactor size ratio are also smaller [7]. In addition, Dyreby [8], Muto [9], Bae [10], Jeong [11], Moullec [12] and Ahn [13] et al. carried out detailed studies on the supercritical carbon dioxide Brayton cycle and achieved valuable results.

However, most of these studies are about the design of systems or components, there are few records on the research of variable conditions of the system. In the actual situation, the changes of the residual heat parameters are objective existence. Therefore, it is particularly important to study the change of the heat source of the system. In this paper, the design condition calculation model and the variable condition calculation

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model of S-CO_2 simple Brayton cycle are studied and constructed, and by changing the temperature of the heat source, the results are obtained and then compared and discussed.

2 Mathematical models and calculation process

The schematic diagram of a supercritical CO_2 Brayton cycle using afterheat is shown in Figure 1. The heat is provided by the gas turbine exhaust and transferred by the heat exchanger and some assumptions are made to simplify the mathematical models.

1、The system could reach a steady state in a short time.

2、The heat transfer between equipment and environment and the pressure drop in system are neglected.

3、The CO_2 temperature at the outlet of the pre-cooler can keep constant by changing the mass flow rate of the environmental water or air.

The system mainly consists of three kinds of equipment: turbines, compressors, and heat exchangers and the mathematical models are given as follows.

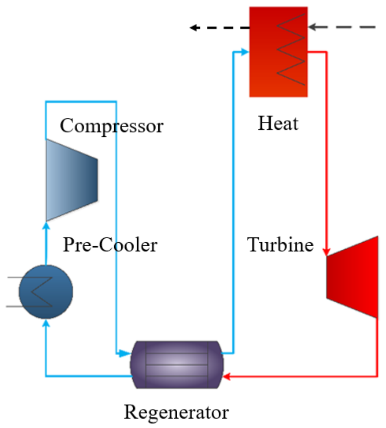

Figure 1. Supercritical CO2 simple cycle layout.

2.1 Turbine

Under the design conditions, an isentropic efficiency is used to describe the non-isentropic expansion process of turbines. The formula used follows:

$$\eta_{t\_d} = \frac{H_{t\_in}\text{-}H_{t\_out}}{H_{t\_in}\text{-}H_{t\_out,s}} \tag{1}$$

The power output of turbines is:

$$W_{t\_d} = m_{t\_d}(H_{t\_in} - H_{t\_out}) \tag{2}$$

When it operates under off-design conditions, the Stodola's ellipse method [14], which is established upon the flow characteristics of turbine, is employed to calculate the parameters of turbines:

$$\phi_{off} = m_{t\_off}\frac{\sqrt{T_{t\_in\_off}}}{P_{t\_in\_off}} \tag{3}$$

From the above two equations, it can be obtained the mass flow rate of working fluid through turbine under the off-design conditions once the turbine inlet pressure, turbine inlet temperature and turbine back pressure are determined.

$$m_{t\_off} = \frac{P^2_{t\_in\_off} - P^2_{t\_out\_off}}{T_{t\_in\_off}Y_d} \tag{4}$$

Where:

$$Y_d = \frac{P^2_{t\_in\_d} - P^2_{t\_out\_d}}{P^2_{t\_in\_d}\phi_d^2} \tag{5}$$

The turbine isentropic efficiency under the off-design conditions changes and could be determined by the following equation [14]:

$$\eta_{t\_off} = \eta_{t\_d}\sin\left[0.5\pi\left(\frac{m_{t\_off}\rho_{t\_in\_d}}{m_{t\_d}\rho_{t\_in\_off}}\right)^{0.1}\right] \tag{6}$$

2.2 Compressor

Under the design conditions, an isentropic efficiency is also used to describe the non-isentropic compression process of compressors. The formula used follows:

$$\eta_{c\_d} = \frac{H_{c\_out,s}\text{-}H_{c\_in}}{H_{c\_out}\text{-}H_{c\_in}} \tag{7}$$

The power compressed by compressor is:

$$W_{c\_d} = m_{t\_c}(H_{t\_out} - H_{t\_in}) \tag{8}$$

When it operates under off-design conditions, the performance curve, which is illustrated in Figure 2, is employed to calculate the parameters of compressors:

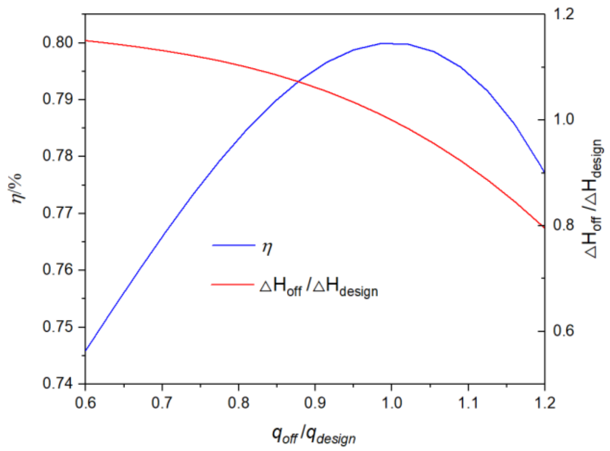

Figure 2. Compressor performance curve.

2.3 Heat exchanger

Since the heat exchangers are the largest components in the cycle, their careful design is an important issue. In order to reduce the total volume of heat exchangers, printed circuit heat exchangers(PCHE) are chosen in this article. The heat exchanger was divided into several axial nodes to simply the model.
The heat exchanger performance calculation can start from either the hot or cold end, therefore either hot or cold side operating conditions must be known. The heat exchanger calculation proceeds from the known end to the other one by sequentially evaluating the performance of all nodes.
The heat transferred in a node was calculated from:

$$q_i = U_i A_i (T_h - T_c) \tag{9}$$

The overall heat transfer coefficient is calculated from:

$$U_i = \left(\frac{1}{U_{i\_h}} + \frac{1}{U_{i\_c}}\right)^{-1} \tag{10}$$

U_{i_h} and U_{i_c} are the heat transfer coefficients on hot and cold sides respectively which are similarly to calculate.
U_{i_h} is calculated from:

$$U_{i\_h} = \frac{\frac{f_h}{8}(\mathrm{Re}_h - 1000)\,\mathrm{Pr}_h \frac{L_h}{PD_h}}{1 + 12.7(\mathrm{Pr}_h^{2/3} - 1)(\frac{f_h}{8})^{0.5}(1 + \frac{PD_h}{L})^{2/3}} \tag{11}$$

Where:

$$f_h = \left(\frac{1}{1.8\lg(\mathrm{Re}_h) - 1.5}\right)^2 \tag{12}$$

$$PD_h = \frac{4\pi D^2 / 8}{\pi D/2 + D} \tag{13}$$

In the heat, the correlations above are used again to calculate the parameters of heat.

2.4 Calculation process

After the mathematical models above are built, Figure 3 illustrates the flow chart of the calculation process of the system under off-design conditions. The calculation contains two iteration loops to determine CO_2 turbine inlet pressure and mass flow rate of water in the heater, respectively.
Once the above two thermodynamic parameters are obtained, all other parameters can be calculated by the mathematical models above, and then the off-design performance of the system is obtained.

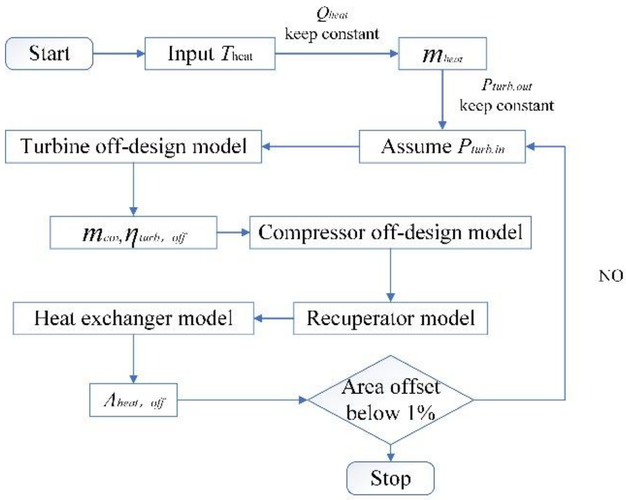

Figure 3 Flow chart of calculation process of the system under off-design conditions.

3 Results and discussion

3.1. Design condition results

The calculation models, including design and off-design conditions are both built by MATLAB software and the physical properties of carbon dioxide are directly retrieved by the database of National Institute of Standards and Technology. Before discussing the results under off-design conditions, several system settings and operation parameters under the design condition are provided to be compared to the results under the off-design conditions. Table 1 shows the operation parameters and some main thermodynamic parameters under the design condition. Table 2 illustrates corresponding system performance under the design condition.

Table 1. Operation and thermodynamic parameters of the system under design condition

Term	Value	Unit
Heat source temperature	500	℃
Heat source mass flow rate	80	kg s^{-1}
Heat source pressure	0.1018	MPa
Mass flow rate of CO_2	73.82	kg s^{-1}
Turbine inlet temperature	480	℃
Turbine inlet pressure	20	MPa
Turbine isentropic efficiency	85	%
Compressor inlet temperature	35	℃
Compressor inlet pressure	8	MPa
Compressor isentropic efficiency	80	%

Table 2. System performance under design condition

Term	Value	Unit
Power output of turbine	7.58	MW
Power consumption of compressor	2.12	MW
Net power output	5.46	MW
Exergy input	17.01	MW
Exergy efficiency	32.08	%

3.2 Off-design condition results and discussion

Figure 4 and Figure 5 illustrate the variations of off-design parameters and performance of the system with respect to the heat source temperature. It is noted that in each figure, the values of black points represent the values of design condition.

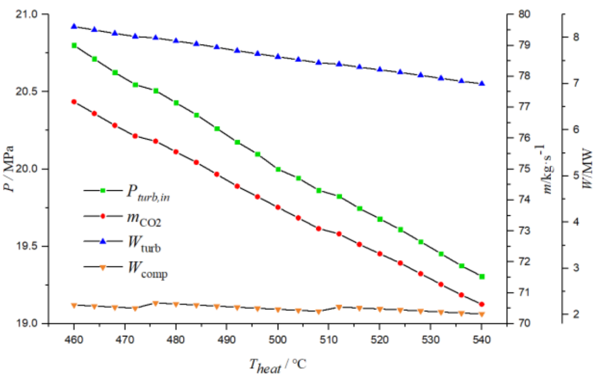

Figure 4. Variation of system parameters with respect to heat source temperature.

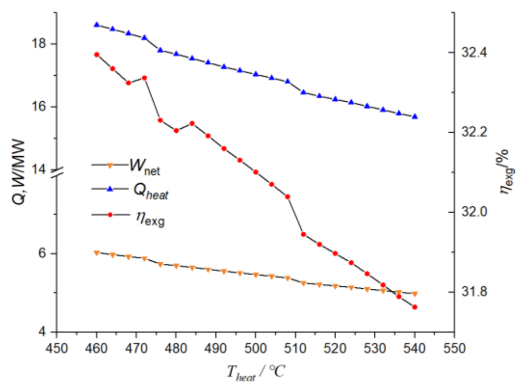

Figure 5. Variation of system performance with respect to heat source temperature.

Figure 4 shows the variation of some parameters with respect to the heat source temperature. Firstly, the heat source mass flow rate decreases as the temperature increases because the quantity of heat keeps constant. It can be seen that the turbine inlet pressure should decreases from 20.8 MPa to 19.30 MPa by regulation as the heat source temperature increases from 460 ℃ to 540 ℃. With the decreased heat mass flow rate, the CO_2 mass flow rate decreases from 77.18 kg·s-1 to 70.63 kg·s-1. Similarly, since the turbine inlet pressure decreases while the outlet pressure keeps constant and the CO_2 mass flow rate decreases, power output of turbine decreases from 8.24 MW to 7.00 MW. But the power consumption of compressor is not changed basically because the inlet temperature of compressor changes correspondingly.

Figure 5 presents the variation of total system performance with respect to the heat source temperature. The net power output of system decreases from 6.03 MW to 4.98 MW when the heat source temperature increases from 460 ℃ to 540 ℃. The quantity utilized by the system decreases from 18.61 MW to 15.69 MW. Finally, it is obtained that with the increase of heat source temperature, the system exergy efficiency decreases from 32.40% to 31.76%. It can be deduced easily that the heat source mass flow rate plays a more influential role than the heat source temperature in the system.

4 Conclusions

In this article, the off-design performance analysis of a supercritical CO_2 simple cycle was carried out. An off-design mathematical model for the system was established to examine the variation of system performance with the variations of heat source and temperature. Some main conclusions are summarized as follows.

(1) When the heat source temperature increases, the power output of turbine and the net power of whole system, and the system exergy efficiency decrease because of the heat source mass flow rate decreases.

(2) It can be deduced that the heat source mass flow rate plays a more influential role than the heat source temperature in the system by the results.

(3) Though the system parameters and system performance change when the heat source heat source temperature increases, they can be kept to be a certain range. So, we can come to the conclusion that supercritical CO_2 simple cycle is a good way to utilize the waste heat.

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