THERMODYNAMIC PERFORMANCE ANALYSIS OF SUPERCRITICAL CARBON DIOXIDE BRAYTON CYCLE

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Abstracts: *S-CO_2 (supercritical carbon dioxide) is used as working fluid for power system cycle. This paper presents thermodynamic performance analysis results on S-CO_2 Brayton cycle. Based on the assumptions of the relevant initial parameters, the mathematical models of compressor, turbine, recuperator and heater are constructed, and the thermal efficiency of regenerative Brayton cycle and recompression Brayton cycle are calculated and analyzed. The results reveal that the efficiency of the recompression cycle is higher than that of the simple regenerative cycle. The effects of inlet temperature, inlet pressure of the main compressor and inlet temperature, inlet pressure of the turbine on the thermodynamic performance of the recompression cycle are studied, and the influencing mechanism is explained. The results show that the cycle efficiency decreases with the increase of the inlet temperature of the main compressor; there exists an optimum inlet pressure in the main compressor to maximize the cycle efficiency; and the cycle efficiency of the system increases with the increase of the inlet temperature and pressure of the turbine. When the inlet temperature of the turbine exceeds 600 ℃, the thermal efficiency of the cycle can reach more than 50%.*

Key words**:** *supercritical carbon dioxide; Brayton cycle; thermodynamic law; cycle thermal efficiency*

1. Introduction

When the pressure of carbon dioxide exceeds 7.38 MPa and the temperature exceeds 30.89℃, it enters supercritical state called supercritical carbon dioxide (S-CO_2). Supercritical carbon dioxide has good heat transfer performance, and its specific heat capacity changes greatly near the critical point. Its critical temperature and pressure are much lower than the critical point of water (22.1 MPa, 374.15℃), which is easy to achieve. S-CO_2 has liquid characteristics, high density, high heat transfer efficiency, high output power capability. S-CO_2 also has gas characteristics, low viscosity, strong fluidity, low cycle loss of the system, and there is no phase transition in the cycle. S-CO_2 also has the characteristics of non-toxicity, non-flammability, good chemical stability, environmental friendliness and low cost [1-3]. Another important feature of S-CO_2 Brayton cycle is its high efficiency in large temperature range. The efficiency of power generation at 550~750℃ is higher than that of cycle with helium as working fluid and Rankine cycle with steam as working fluid. Therefore, using supercritical carbon dioxide as working fluid of closed Brayton cycle can greatly reduce the power consumption of compressor and achieve at least 35% cycle efficiency [4].

$S\text{-}CO_2$ has attracted worldwide attention. Up to now, many scholars have made in-depth studies on $S\text{-}CO_2$ Brayton cycle in many aspects. The Brayton cycle of supercritical carbon dioxide was first proposed after the discovery of the special physical properties of supercritical carbon dioxide by scholar Feher in 1967 [5]. Then Angelino et al. studied the comparison and analysis of different Brayton cycles. They found that the supercritical carbon dioxide Brayton cycle at the inlet temperature of the turbine at 650℃ was superior to the steam-powered Rankine cycle which was widely used at that time [6]. At the same time, two kinds of layout designs are put forward, one is a simple cycle with low cost and compact structure; and the other is a recompression cycle, which has both advantages of the former and high efficiency. Many of these activities have been strongly supported by test facilities which have demonstrated the technical feasibility of the concept, even if still only at micro (<500 kWe) or mini scale (<1 MWe). Sandia National Laboratory in the United States has successfully built a small-scale supercritical carbon dioxide closed Brayton cycle test facility [7]. The flow split and recompression layout was used in the circle system, and the numerical simulation analysis of the circle system was carried out. The preliminary experiments were carried out and the results were obtained. There are also other supercritical carbon dioxide closed Brayton cycle test systems in the United States, which are the simple cycle system for Beckett marine propulsion plant [8], the commercial heat engine for Echogen waste heat recovery [9] and the 1MWe simple cycle test system of Southwest Research Institute of the United States. The European Union, Japan and Korea have successively carried out research on $S\text{-}CO_2$ power generation technology. Tokyo University of Technology has completed the design of a 48.2% $S\text{-}CO_2$ cycle system, which can be used for solar power generation [10]. In 2016, the Institute of Engineering Thermophysics of the Chinese Academy of Sciences made a strategic plan to focus on $S\text{-}CO_2$ power generation technology as a key cultivation project, and clearly proposed the development of MW $S\text{-}CO_2$ turbine power generation system and promoted its application in the field of new energy [11].

Many $S\text{-}CO_2$ Brayton power cycle configurations have been proposed and studied for nuclear applications [12] and thermodynamic cycle analyses indicate that the $S\text{-}CO_2$ power cycle can also be a good option for the CSP application [13]. Importantly, the cycle offers high efficiency at temperatures that are readily achievable by CSP collector technologies. Many studies have been conducted on the performance of $S\text{-}CO_2$ Brayton cycles [14]. The comparison of various $S\text{-}CO_2$ layouts was presented in terms of cycle performance by Ahn [15] and the recompression cycle showed the best efficiency. Dyreby provided a numerical modeling method for $S\text{-}CO_2$ Brayton cycles [16]. A 10 MWe recompression supercritical carbon dioxide Brayton cycle has been analyzed using in-house cycle design point and off-design codes [17, 18]. A dynamic model of the supercritical CO_2 recompression Brayton cycle was developed. The effects of high-temperature and low-temperature recuperators on the dynamic characteristics of the cycle were examined [19]. Monthly exergy destruction analysis was conducted to find the effects of different ambient and water temperatures on the performance of the system by Salim [20].

The thermodynamic performance of the $S\text{-}CO_2$ Brayton cycle system is highly sensitive to inlet temperature and inlet pressure of compressor and turbine, respectively, though few studies are available in literature covering that. In this paper, $S\text{-}CO_2$ Brayton cycle is taken as the research object, and the optimum basic parameters are selected. The thermal efficiency of regenerative Brayton cycle and recompression Brayton cycle is calculated by the laws of thermodynamics. The system efficiency of simple cycle and recompression cycle is compared and analyzed. By changing the initial parameters, the efficiency of recompression cycle is calculated and analyzed. Influencing factors of system thermal efficiency are analyzed and optimized scheme is obtained.

2. Regenerative S-CO2 Brayton cycle efficiency analysis

2.1. Basic principles of regenerative Brayton cycle

Fig.1 is the regenerative S-CO_2 Brayton cycle flow chart. It draws the key components and gives some parameters. The cycle is mainly composed of compressor, turbine, recuperator, precooler and heater. Low-temperature and low-pressure carbon dioxide is compressed and pressurized by a compressor (1-2 process). It is then heated by the carbon dioxide from the high temperature side of the regenerator (2-3 process), then continues to be heated by the heater from the heat source to the highest temperature (3-4 process), and then enters the turbine to do work and drives the alternator to work (4-5 process). The heat of S-CO_2 after turbine work is recovered and cooled by the carbon dioxide from low temperature side of the regenerator (5-6 process). The gas is cooled by the precooler to the inlet temperature required by the compressor, and then enters the compressor to form a closed cycle to do work.

Because the working fluid used in the circulating system is carbon dioxide, in order to prevent the leakage of working fluid, the whole system adopts closed cycle. Fig. 2 is a simple regenerative temperature entropy diagram of S-CO_2 Brayton cycle, which shows the change process of temperature and specific entropy in the cycle. Process 1-2 can be regarded as isentropic compression process, process 2-4 as isobaric heating process, process 4-5 as isentropic expansion process and process 5-1 as constant pressure exothermic process.

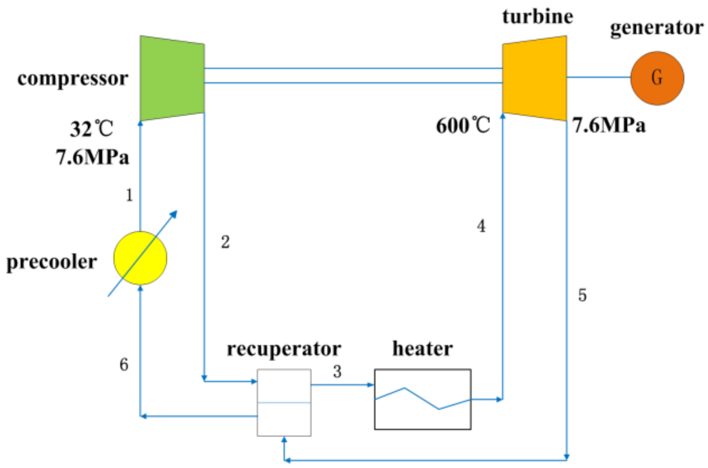

Fig. 1 Regenerative S-CO_2 Brayton cycle flow chart

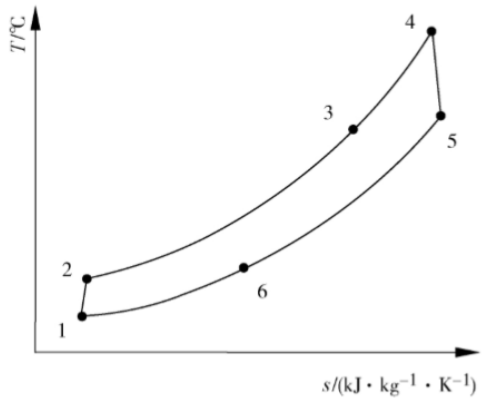

Fig. 2 Temperature entropy diagram of regenerative Brayton cycle

2.2. Parameter setting of regenerative Brayton cycle

In order to carry out the thermodynamic calculation of S-CO_2 Brayton cycle, it is necessary to assume the initial conditions of calculation and the characteristic parameters of the equipment. The parameter setting is considered from the point of view of reducing the compression power consumption of equipment and improving the system efficiency. Because the physical properties at the critical point are unstable and change dramatically, which is not conducive to the stable operation of the compressor, the inlet temperature and pressure of the compressor should be slightly higher than the critical point of carbon dioxide (7.38 MPa, 30.89℃). Considering the above conditions, the inlet temperature of the compressor is set to 32℃, and the corresponding inlet pressure is set to 7.6 MPa. The influence of the performance of system components on cycle efficiency is mainly related to the efficiency of turbines and compressors. The isentropic reference efficiency of compressor and turbine is 89% and 90% respectively. The inlet temperature of the turbine is 600℃, and the corresponding inlet pressure is 22.8 MPa. The pressure loss in the connecting pipe of the components is neglected. The initial calculation conditions and equipment parameters in this paper are shown in Table 1.

Table 1. Design parameters for the cycle

Design parameter	Parameter value	Design parameter	Parameter value
compressor efficiency	89%	turbine efficiency	90%
compressor inlet temperature	32℃	compressor inlet pressure	7.6MPa
turbine inlet temperature	600℃	turbine inlet pressure	22.8MPa
temperature difference in low temperature section of the recuperator	25℃	heat source power	600MW
compression ratio	3	recuperator efficiency	90%

2.3. Calculation and analysis of regenerative Brayton cycle efficiency

The corresponding enthalpy and entropy can be obtained according to the inlet temperature and inlet pressure of the compressor (h_1, s_1). In a simple S-CO_2 Brayton cycle, compression from the inlet to the outlet of the compressor is regarded as an adiabatic process, and the output entropy of the compressor is equal to the inlet entropy. The isentropic enthalpy of outlet ($h_{2,i}$) can be obtained when the outlet entropy and pressure are known. According to the known isentropic efficiency of the compressor 89%, the actual outlet enthalpy ($h_2 = 341.3\mathrm{kJ/kg}$) of the compressor can be obtained by the following formula.

$$\eta_c = \frac{h_{2,i} - h_1}{h_2 - h_1} = 89\% \tag{1}$$

According to the actual outlet enthalpy h_2 and the outlet pressure p_2 of the compressor, the outlet temperature of the compressor can be determined to be 69.8℃.

The mathematical model of a turbine is similar to that of a compressor. The inlet temperature, inlet pressure and outlet pressure of the turbine are known (Table 1). According to the inlet temperature and inlet pressure of the turbine, the corresponding enthalpy and entropy (h_4, s_4) can be obtained. The expansion from the inlet to the outlet of the turbine is regarded as an adiabatic process, so that the outlet entropy of the turbine equals the inlet entropy. The isentropic enthalpy of exit ($h_{5,i}$) can be obtained when the outlet entropy and pressure are known. According to the 90% isentropic efficiency of the turbine, the actual outlet enthalpy ($h_5 = 952.6\mathrm{kJ/kg}$) of the turbine can be obtained by the following formula.

$$\eta_t = \frac{h_4 - h_5}{h_4 - h_{5,i}} = 90\% \tag{2}$$

According to the actual outlet enthalpy h_5 and the outlet pressure p_5 of the turbine, the outlet temperature of the turbine can be determined to be T_5=472.4℃.

According to the temperature difference in low temperature section of the recuperator at 25℃ and the efficiency of the recuperator at 90%, the outlet temperature at the low temperature side T_6 and the outlet temperature at the high temperature side T_3 can be obtained.

$$\eta_{rec} = \frac{h_3 - h_2}{h_5 - h_6} = 90\% \tag{3}$$

According to the above formula, h_3=734.2kJ/kg and T_3=309.5℃ can be obtained.

Combining the above mathematical model analysis of compressor and turbine with the setting of temperature difference of recuperator, the operating parameters of each state point of S-CO_2 Brayton simple cycle can be obtained. Table 2 shows the calculation results of operating parameters at each state point of a simple cycle.

Table 2. Operating parameters of each state point

State points	Temperature (℃)	Pressure (MPa)	Enthalpy (kJ/kg)
1	32	7.6	315.1
2	69.8	22.8	341.3
3	309.5	22.8	734.2
4	600	22.8	1095.8
5	472.4	7.6	952.6
6	94.8	7.6	516.1

Based on the first law of thermodynamics, the cycle thermal efficiency of the system is expressed as:

$$\eta_{\text{cyc}} = \frac{W_t - W_c}{Q} = \frac{\dot{m}(h_4 - h_5) - (h_2 - h_1)}{\dot{m}(h_4 - h_3)} = \frac{117}{361.6} = 32.4\% \quad (4)$$

Compressor power consumption $W_c = h_4 - h_5$ and turbine expansion work $W_t = h_2 - h_1$, Q is the heat power of the system heat source, h_1, h_2, h_3, h_4 and h_5 are the enthalpy values of the corresponding state points, respectively.

3. Efficiency analysis of S-CO2 Brayton cycle with recompression

3.1. Basic principles of recompression cycle

The physical properties of supercritical carbon dioxide will change significantly with the change of temperature. When the supercritical carbon dioxide is used as working fluid in simple Brayton cycle, the high and low temperature side of the recuperator will have the problem of "pinch temperature" because of the great difference of specific heat of working fluid, thus affecting the efficiency of the recuperator, so that the efficiency of the whole simple cycle will not exceed 40% even if the inlet temperature of the turbine is increased. Therefore, in order to improve the efficiency of the whole cycle, the recompression Brayton cycle compound cycle is generally adopted.

Fig. 3 is the basic flow chart of the recompression Brayton cycle, which shows the structure of the cycle. Compared with the simple cycle, the recompression Brayton cycle adds a recompressor, and the recuperator is divided into low-temperature and high-temperature recuperators. The recompression cycle consists of a main compressor, a recompressor, a low-temperature recuperator, a high-temperature recuperator, a heater exchanger, a turbine and a precooler. Its working principle and process are as follows: after heat release from the precooler, the fluid enters the main compressor (8-1 process), part of the fluid is compressed and pressurized (1-2 process), and is heated to the outlet temperature of the recompressor (2-3 process) in the low-temperature recuperator, and directly enters the high-temperature recuperator for heating (3-4 process) after confluence with the fluid compressed to high pressure in the recompressor(8-3 process). The fluid absorbs heat from the heat source to reach the inlet temperature of the turbine (4-5 process). After reaching the temperature, it directly enters the turbine for expansion work (5-6 process). After doing work, the high-temperature fluid passes through the high-temperature recuperator (6-7 process) and the low-temperature recuperator to release heat (7-8 process), respectively. After heat release, one part of the fluid directly enters the recompressor and compresses to high pressure (8-3 process), the other part enters the main compressor after the precooler and compresses to form a closed cycle (8-1 process). Compared with the simple regenerative Brayton cycle, the advantages of the recompression Brayton cycle are that it can reduce the end temperature difference of the low-temperature recuperator, improve the recuperator efficiency of the system, reduce the heat needed by the cooler and improve the efficiency of the whole cycle.

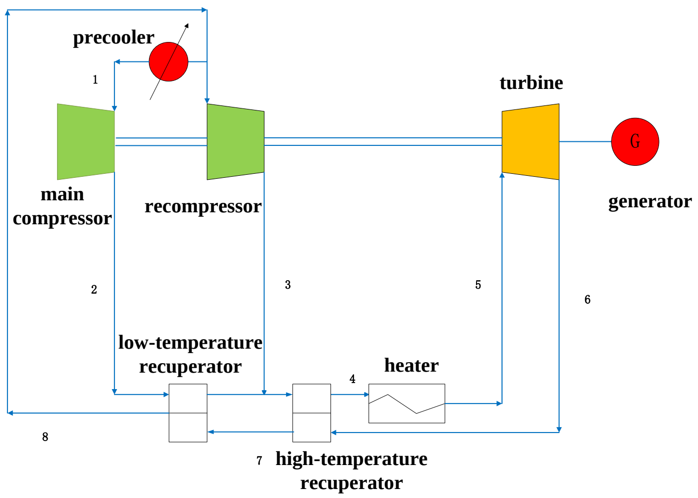

Fig. 3 Flow chart of the recompression Brayton cycle

3.2 Parameter setting of recompression cycle

In the process of calculating the efficiency of the recompression cycle, the basic parameters selected are the same as those of the simple cycle. The selected cycle parameters are as follows: the minimum temperature of the cycle is T_{min}=32℃, the maximum temperature of the cycle is T_{max}=600℃, the maximum pressure of the cycle is p_{max}=22.8 MPa, and the cycle compression ratio is ε=3.0.

Table 3 sets the equipment performance parameters for calculating S-CO_2 recompression Brayton cycle efficiency, and analyses the variation trend of recompression cycle efficiency according to the changes of parameters such as compressor inlet temperature, inlet pressure, turbine inlet temperature and inlet pressure. The calculation neglects the pressure drop in the pipeline, the environmental heat leakage loss of equipment and pipeline and other irreversible losses.

Table 3. Recompression parameter setting

Design parameter	Parameter value	Design parameter	Parameter value
main compressor efficiency	89%	turbine efficiency	90%
recompressor efficiency	89%	compression ratio	3
temperature difference in low temperature section of the recuperator	25℃	flow split ratio	0.2
compressor inlet temperature	30-50℃	compressor inlet pressure	7.5-8.5MPa
turbine inlet temperature	350-750℃	turbine inlet pressure	15-25MPa
efficiency of high-temperature recuperator	90%	efficiency of low-temperature recuperator	85%

3.3 Calculation and analysis of recompression cycle system efficiency

The mathematical model of the main compressor and turbine in the recompression cycle is the same as that of the simple cycle. The actual outlet enthalpy h_2 of the main compressor and the actual outlet enthalpy h_6 of the turbine can be calculated respectively by the isentropic efficiency of the main compressor and the turbine.

$$h_2 = h_2(69.8℃, 22.8\text{MPa}) = 341.3\text{kJ/kg} \tag{5}$$

$$h_6 = h_6(472.4℃, 7.6\text{MPa}) = 952.6\text{kJ/kg} \tag{6}$$

The temperature difference at the end of the low-temperature recuperator is 25℃. The corresponding enthalpy values h_4 and h_8 can be obtained from the physical parameters.

$$h_4 = h_4(447.4℃, 22.8\text{MPa}) = 905.8\text{kJ/kg} \tag{7}$$

$$h_8 = h_8(94.8℃, 7.6\text{MPa}) = 516.1\text{kJ/kg} \tag{8}$$

The recuperator efficiency of low-temperature recuperator is as follows:

$$\eta_{\text{L}} = \frac{T_7 - T_8}{T_7 - T_2} = 85\% \tag{9}$$

T_7, T_8, and T_2 in the formula are the temperatures of corresponding points 7, 8, and 2 in the cycle. The corresponding temperature T_2 can be found by the actual outlet enthalpy h_2 of the compressor, thus T_7 = 236.5℃ can be obtained.

The recuperator efficiency of high-temperature regenerator is as follows:

$$\eta_{\text{H}} = \frac{T_6 - T_7}{T_6 - T_3} = 90\% \tag{9}$$

$T_3 = 210.3℃$ can be calculated, h_3 and h_7 can be obtained from the physical parameters.

$$h_7 = h_7(236.5℃, 22.8\text{MPa}) = 682.2\text{kJ/kg} \tag{11}$$

$$h_3 = h_3(210.3℃, 22.8\text{MPa}) = 603.8\text{kJ/kg} \tag{12}$$

Combining the above formulas and the initial parameters, the physical parameters of each state point of S-CO_2 Brayton recompression cycle can be obtained. Table 4 is the calculation results of the operating parameters of each state point of the cycle.

Table 4. Working condition parameters of each state point in the cycle

State points	Temperature (℃)	Pressure (MPa)	Enthalpy (kJ/kg)
1	32	7.6	315.1
2	69.8	22.8	341.3
3	210.3	22.8	603.8
4	447.4	22.8	905.8
5	600	22.8	1095.8
6	472.4	7.6	952.6
7	236.5	7.6	682.2
8	94.8	7.6	516.1

Cycle efficiency can be expressed as:

$$\eta_{\text{cyc}} = \frac{W_t - W_c}{Q} = \frac{(h_5 - h_6) - x(h_2 - h_1) - (1 - x)(h_3 - h_8)}{h_5 - h_4} = 35.7\% \quad (13)$$

h_5, h_6, h_3, h_8, h_2, h_1 and h_4 are the enthalpy values of corresponding points 5, point 6, point 3, point 8, point 2, point 1 and point 4 in kJ/kg. x is the flow split ratio, and the selected flow split ratio in the parameter x=0.2. It is concluded that the cycle efficiency of the recompression system is 35.7%, which is higher than that of the simple cycle (32.4%).

4. Initial parameters influence on recompression cycle efficiency

The influencing factors of S-CO_2 Brayton recompression cycle can be divided into two situations: one is the loss of the fixed part of the cycle system, i.e. the performance efficiency of the components and equipment, the other is the process of the cycle system operation, i.e. the influence of the thermodynamic parameter setting at each state point. In this paper, the thermodynamic parameters of the recompression cycle are analyzed, and the cycle efficiency of the system is compared, that is, the influence of the inlet temperature and pressure of the main compressor, the inlet temperature and pressure of the turbine on the cycle efficiency are analyzed.

4.1. Effect of inlet pressure of main compressor

Fig.4 is the diagram of cycle thermal efficiency of recompression cycle under different inlet pressure of main compressor. The main compressor inlet temperature of the cycle is $T_{\min}$=32℃, the turbine inlet temperature is different, and the maximum cycle pressure $p_{\max}$=22.8MPa. The system efficiency is calculated by changing the inlet pressure of the main compressor. The graph shows the change curve of the overall efficiency of the cycle with the increase of the pressure when the inlet pressure of the main compressor is from 7.6 MPa to 8.4 MPa. The system efficiency increases first and then decreases with the increase of the inlet pressure of the main compressor. When the inlet pressure of the main compressor is near 7.8 MPa, the cycle efficiency of the system is highest. When the inlet pressure of the main compressor is lower than the critical pressure, the cycle efficiency increases; when the inlet pressure exceeds the critical pressure, the cycle efficiency will gradually decreases with the increase of pressure. With the increase of the inlet pressure of the main compressor, the output power of the turbine decreases gradually, and the heat absorption power of the working fluid in the heat exchanger increases obviously, which leads to the decrease of the cycle efficiency.

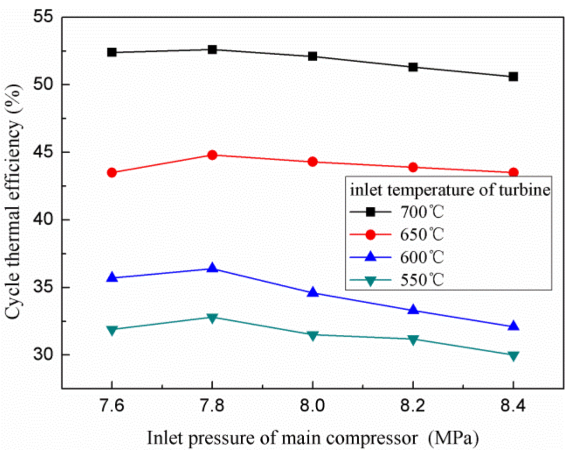

Fig. 4 Cycle thermal efficiency diagram under different inlet pressure of main compressor

4.2 Effect of inlet temperature of main compressor

Fig.5 shows the cycle efficiency with different inlet temperature of main compressor. At this time, the inlet pressure of the main compressor is 7.6 MPa, and the maximum cycle pressure is p_{max}=22.8 MPa. At different inlet temperature of the turbine, the thermal efficiency of the system is calculated at different inlet temperature of the main compressor (32-48°C). It can be seen that the thermal efficiency of the system is significantly affected by the inlet temperature of the main compressor. With the increase of the inlet temperature of the main compressor, the overall efficiency of the cycle shows a significant downward trend. When the inlet temperature of the turbine increases, the cycle efficiency also increases. Therefore, the optimum inlet temperature parameters of the main compressor should be set near the critical temperature, which can be set at about 32°C. As the inlet temperature of the main compressor rises, the power of the main compressor increases gradually, while the output power of the turbine changes slightly, resulting in a decrease in the net output power of the whole cycle and a decrease in the heat absorption of the working medium from the heat source, but the net output power of the cycle decreases more, so the overall cycle efficiency decreases with the increase of the inlet temperature.

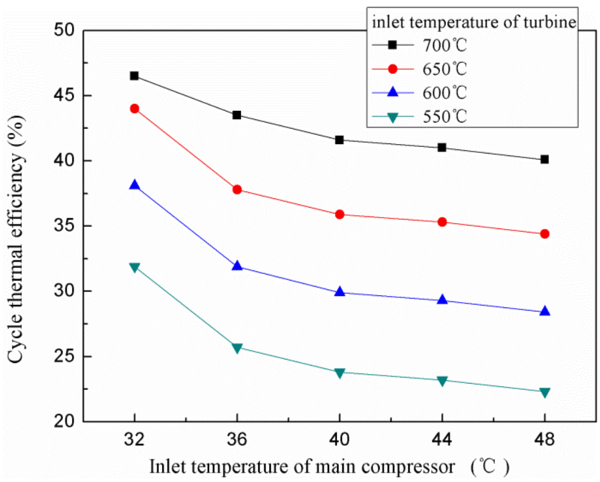

Fig. 5 Relationship between cycle thermal efficiency and inlet temperature of main compressor

4.3. Effect of turbine inlet temperature and inlet pressure

Fig.6 shows how the thermal efficiency of the recompression cycle varies with the inlet temperature and pressure of the turbine. At this time, the inlet temperature of the main compressor is T_{min} =32°C, the inlet pressure of the main compressor is p_{min} =7.6 MPa, and the inlet temperature of the turbine is 550°C, 600°C, 650°C and 700°C, respectively. Fig.6 shows that the overall cycle thermal efficiency increases with the increase of turbine inlet pressure, but the upward trend is gradually flat. With the increase of turbine inlet pressure, the total power of the compressor and the output power of the turbine gradually increase, and the cycle output power also increases. The increase of the cycle output power is larger than that of fluid heat absorption power, which leads to the increase of the overall cycle efficiency.

In addition, it can be seen from Fig.6 that the efficiency of the whole cycle system can be effectively improved with the increase of the inlet temperature of the turbine, and the system efficiency is approximately linear with the inlet temperature of the turbine. When the inlet temperature of the turbine exceeds 600°C, the thermal efficiency of the cycle can reach more than 50%. Therefore, increasing the inlet temperature of the turbine can improve the efficiency of the circulating system, but the inlet temperature of the turbine cannot be increased indefinitely. Excessive temperature will aggravate the corrosion of the system and high material requirements.

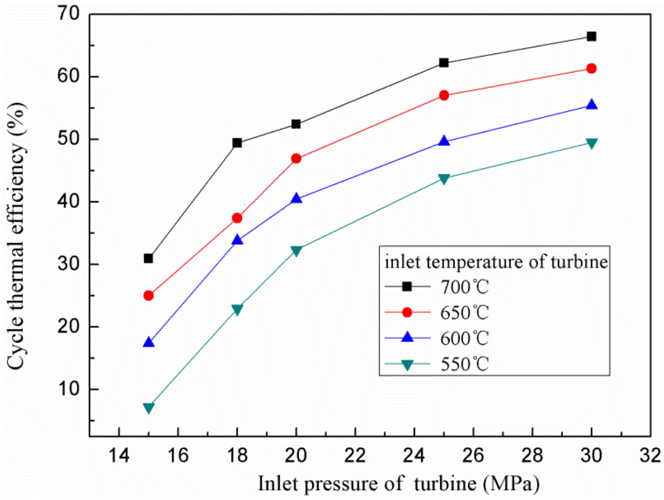

Fig. 6 Relation of cycle efficiency with turbine inlet temperature and inlet pressure

5. Conclusion

Based on the basic laws of thermodynamics, the thermodynamic performance of S-CO_2 Brayton cycle is analyzed in this paper. By calculating the system efficiency of regenerative cycle and recompression cycle, the inlet temperature and inlet pressure of the main compressor and turbine are studied. By assuming the same initial parameters to calculate the system efficiency of the simple regenerative cycle and the recompression cycle respectively, it can be seen that the efficiency of the recompression cycle is higher than that of the simple regenerative cycle. There is a maximum inlet pressure of the main compressor to maximize the cycle efficiency of the system. When other conditions are selected, the inlet pressure of the main compressor can get a higher efficiency near 7.8 MPa. The cycle efficiency of the system decreases with the increase of the inlet temperature of the main compressor, but because the inlet temperature of the main compressor is limited by the environmental conditions, it can be taken near the critical temperature of 32°C. The inlet temperature of the turbine is approximately linear with the cycle efficiency, so increasing the inlet temperature of the turbine can effectively improve the efficiency of the cycle system. When the inlet pressure of the turbine increases, the efficiency of the system will also increase, when the inlet pressure is too high, the efficiency of the system tends to rise gently.

Nomenclature

h	– specific enthalpy, [kJ kg^{-1}]	***Subscripts***	
$\dot{m}$	– mass flow rate, [kg s^{-1}]	i	– idealization
p	– pressure, [Pa]	c	– compressor
Q	– heat transfer power, [kW]	cyc	– cycle
s	– specific entropy, [kJ kgK^{-1}]	rec	– recuperator
		t	– turbine

T – temperature, [K]
W – power, [kW]
x – flow split ratio

Greek letters

η – efficiency
ε – compression ratio

min – minimum
max – maximum
L – low-temperature recuperator
H – high-temperature recuperator
1 to 8 – states on thermodynamic cycle

Acknowledgement

This work was funded by Teaching Quality and Teaching Reform Project of Dongguan university of Technology (No. 201802015, 201902088 and No. GC100113-21).

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Received: 14.03.2020.

Revised: 24.08.2020.

Accepted: 01.09.2020.