OPTIMIZATION AND THERMAL ANALYSIS OF SUPERCRITICAL CARBON DIOXIDE BRAYTON CYCLE SYSTEM WITH ORGANIC RANKINE CYCLE

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Abstract

A supercritical carbon dioxide (S-CO2) Brayton cycle is recognized as a highly promising heat-to-power technology. Compared with the traditional Rankine cycle, the supercritical carbon dioxide Brayton cycle exhibits significant advantages in cycle efficiency and equipment size due to the unique physical properties of carbon dioxide near the critical point (31.1 °C, 7.39 MPa). In recent years, it has garnered extensive attention in fields such as solar energy, nuclear energy, and waste heat utilization. The majority of current studies focus on steady-state thermodynamic calculations and system layout design of the circulation system. However, substantial potential remains for the recompression S-CO2 power cycle to enhance its waste heat extraction and power output capabilities. Consequently, considerable research efforts are still required to optimize system characteristics. The novelty of this work lies in the development of a new type of S-CO2 power cycle with a recompression cycle aimed at thoroughly and efficiently reusing waste heat. The S-CO2 Brayton cycle system model is established using Aspen HYSYS software to analyze the performance of the cycle system. The model investigates the effects of turbine inlet temperature, inlet pressure, condenser outlet temperature, and compressor split coefficient on cycle thermal efficiency. The influence of compressor outlet pressure and turbine inlet temperature on the recompression Brayton cycle efficiency is significant. The optimum value of the main compressor split coefficient lies between 0.3 and 0.5. Moreover, the Organic Rankine Cycle can increase cycle thermal efficiency by 2–6% compared to other cycles.

Full Text

Preamble

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ABSTRACT

The supercritical carbon dioxide (S-CO_2) Brayton cycle is recognized as a highly promising heat-to-power technology. Compared with the traditional Rankine cycle, the supercritical carbon dioxide Brayton cycle offers significant advantages in cycle efficiency and equipment size due to the unique physical properties of carbon dioxide near the critical point (31.1 °C, 7.39 MPa). In recent years, it has attracted extensive attention in the fields of solar energy, nuclear energy, and waste heat utilization. Most current studies focus on steady-state thermodynamic calculations and system layout design of the circulation system. However, there remains substantial room for improvement in the recompression S-CO_2 power cycle regarding waste heat extraction and power output capability. Therefore, considerable research work is still needed to optimize system characteristics.

The novelty of this work lies in the development of a new type of S-CO_2 power cycle with a recompression cycle aimed at thoroughly and efficiently reusing waste heat. The S-CO_2 Brayton cycle system model is established using Aspen HYSYS software to analyze the performance of the cycle system. The model aims to study the effects of turbine inlet temperature, inlet pressure, condenser outlet temperature, and compressor split coefficient on cycle thermal efficiency. The influence of compressor outlet pressure and turbine inlet temperature on the recompression Brayton cycle efficiency is significant, with the main compressor split coefficient having an optimum value between 0.3–0.5. Moreover, the Organic Rankine Cycle can increase the cycle thermal efficiency by 2–6% compared to other cycles.

Keywords: Supercritical CO_2, Aspen HYSYS, Brayton cycle, Thermal performance, Cycle efficiency

INTRODUCTION

The sodium-cooled fast reactor is one of the important reactors in the fourth-generation nuclear energy system. Currently, the three-loop power generation systems of sodium-cooled fast reactors built domestically and internationally adopt the traditional steam Rankine cycle, which not only has low efficiency but also suffers from adverse effects such as sodium-water reaction. These problems have hindered the rapid development of sodium-cooled fast reactors to some extent. Therefore, it is particularly important to develop and study a power generation cycle system suitable for sodium-cooled fast reactors.

The supercritical carbon dioxide (SCO_2) Brayton cycle has the characteristics of high efficiency, compactness, and avoidance of sodium-water reaction, making it an ideal reactor power cycle configuration for sodium-cooled fast reactors [1]. At present, all countries in the world are carrying out research on the application of the SCO_2 power cycle to sodium-cooled fast reactors. The Massachusetts Institute of Technology (MIT) studied the design of different SCO_2 cycle configurations and analyzed the characteristic parameters of each cycle configuration [2]. In addition, MIT has also proposed three SCO_2 design schemes for SCO_2-cooled fast reactors (GFR), of which the high-performance design scheme has a net efficiency of 49% [3]. Argonne National Laboratory (ANL) has carried out SCO_2 heat exchanger experiments, sodium and SCO_2 reaction experiments, and SCO_2 cycle configuration design and analysis [4]. The Tokyo Institute of Technology (TIT) in Japan proposed an SCO_2 partial pre-cooling direct cycle configuration, intermediate compression, and intermediate cooling process to reduce the heat taken away by cooling to improve efficiency. The Korea Atomic Energy Research Institute (KAERI) has proposed a demonstration fast reactor power plant KALIMER-600, which applies the SCO_2 power cycle to the 600 MWe pool-type sodium-cooled fast reactor, and the net efficiency of the power plant reaches 40.3% [5].

In recent years, China has also carried out research on the SCO_2 power cycle and its application to various power systems. Duan et al. [6-7] from Tsinghua University studied the influence of various parameters on the cycle efficiency and the relationship between the parameters by establishing a supercritical carbon dioxide Brayton cycle recompression cycle. They pointed out that the SCO_2 power cycle can achieve satisfactory efficiency at a lower temperature than the helium cycle, and the recompression cycle is suitable for reactors with lower outlet temperatures. Guo Zhangpeng et al. [8] proposed a double turbine recompression SCO_2 cycle and pointed out that this cycle is suitable for fourth-generation reactors with low heat source working pressure and high heat source temperature. By summarizing the current research status, it can be found that although there are many studies on the configuration of the SCO_2 cycle system, there is a lack of matching research on the characteristics of the SCO_2 cycle system and the sodium side cycle of the sodium-cooled fast reactor. Different from the traditional heat source that only considers the maximum operating temperature, the cycle system with a sodium-cooled fast reactor as the heat source not only needs to consider the maximum operating temperature of the cycle but also needs to ensure that the minimum temperature of the sodium cycle system remains stable, which puts forward new requirements for the regenerative system. In addition, in the sodium-cooled fast reactor circulation system, the circulation system operates in a constant thermal power mode, rather than a traditional constant electrical power mode. Based on the above heat source characteristics, this paper will carry out research on the SCO_2 cycle system and parameter optimization design matching sodium-cooled fast reactors, and provide support for the subsequent design of a large-scale sodium-cooled fast reactor power station system.

2. ESTABLISHMENT OF CYCLE MODEL

The heat source is modeled as a sodium-cooled fast reactor with a thermal power output of 300 MW, whose intermediate heat exchanger demonstrates secondary side inlet and outlet temperatures of 350°C and 515°C respectively. Based on these parameters and taking into account the design requirements of the thermodynamic parameters of the working fluid and the heat exchanger in the SCO_2 power conversion system, the basic design boundary parameters of the sodium-cooled fast reactor SCO_2 cycle power conversion system are determined (Table 1).

The physical properties of sodium and CO_2 were obtained from the National Institute of Standards and Technology (NIST) physical property database. In the process of cycle simulation, the system is assumed to be in a stable state, without considering the heat loss and pipeline resistance loss in the cycle process. Equipment resistance loss is replaced by pressure drop, and the efficiency of the turbine and compressor is set to a fixed value.

2.1 Circulation System Configuration

Because the physical properties of the SCO_2 working fluid are very special in a supercritical state, the selection of the physical property calculation method is particularly important in the simulation analysis of the SCO_2 Brayton cycle, as it is related to the accuracy of the simulation. Carbon dioxide is a non-polar system. In the supercritical state, it should be considered as a real gas and described as a Henry component. Therefore, to ensure accuracy, the physical properties of the SCO_2 working fluid in this paper are calculated using the Peng-Robinson equation of state provided with Aspen HYSYS software.

In this paper, the cycle models are built with Aspen HYSYS, and the modules and input parameters are shown in Table 2 . Figure 1 [Figure 1: see original paper] shows the schematic diagram of the system configuration of a simple regenerative cycle. Compared with more complex regenerative systems, this equipment structure is simple and easy to install and maintain. However, due to the large power consumption of the main compressor during the simple regenerative cycle, the efficiency of the simple regenerative cycle is low. In addition, the physical properties of supercritical carbon dioxide on the cold and hot sides of the regenerator in the simple regenerative cycle are quite different. The specific heat of carbon dioxide on the low-temperature and high-pressure side is much larger than that on the high-temperature and low-pressure side, resulting in poor heat transfer matching of the regenerator and affecting the cycle efficiency.

In order to solve the problem of large power consumption of the simple regenerative cycle compressor and the mismatch of heat transfer on both sides of the regenerator, the intercooled simple regenerative cycle and the recompression cycle are designed respectively.

Recompression Cycle

According to the physical properties of supercritical carbon dioxide, the heat capacity of carbon dioxide varies greatly in the near-critical region. In the low-temperature regenerator, the specific heat of carbon dioxide on the low-temperature high-pressure side is much larger than that on the high-temperature low-pressure side, which seriously limits the outlet temperature of the high-pressure side of the low-temperature regenerator, thereby affecting the cycle efficiency. To match the heat transfer between the low-pressure side and the high-pressure side of the low-temperature regenerator, a supercritical carbon dioxide recompression cycle is designed. The configuration diagram of the recompression cycle is shown in Figure 3 [Figure 3: see original paper].

In the recompression cycle, the split coefficient is defined as the ratio of the flow through the recompressor to the total flow of the cycle. The split coefficient has an important influence on the cycle efficiency. On the one hand, the flow coefficient determines the flow rate of the main compressor and the recompressor, which in turn affects the power consumption of the two compressors. On the other hand, a reasonable split coefficient can achieve better heat capacity matching between the hot side and the cold side of the low-temperature regenerator, thereby improving the cycle efficiency. This arrangement achieves heat capacity matching of the low-temperature regenerator at the expense of some total power consumption of compressors, improves the heat exchange capacity of the regenerator, and ultimately improves the overall cycle efficiency.

Intercooling Cycle

The intercooling simple regenerative cycle is mainly designed to address the adverse characteristics of large power consumption and high manufacturing cost of the main compressor in the simple regenerative process. According to compressor theory, isothermal compression consumes the least work, so the compression process should be made as close to isothermal compression as possible. Multi-stage compression and intercooling can be used. On the one hand, this method can minimize the work of the compressor and improve the volumetric efficiency of the compressor. However, although inter-stage cooling reduces the power consumption of the compressor, the reduction is relatively low, and inter-stage cooling will lead to a decrease in the outlet temperature of the regenerator. Under certain conditions of regenerator efficiency, the inlet temperature of the sodium-supercritical carbon dioxide heat exchanger will be reduced, which leads to an increase in the heat absorption of the cycle and a decrease in the cycle efficiency. The loop model is shown in Figure 5 [Figure 5: see original paper].

2.2 Mathematical Model of Circulation System

Based on the parameters in Table 1 and the different cycle system configurations shown in Figure 1, the calculation model of the SCO_2 Brayton cycle system is built. By modeling key equipment including compressors, exchangers, turbines, high-temperature heat regenerators, and condensers, the equipment models can be coupled to obtain a circulatory system model.

The compressor work model is:

$$w_c = \frac{m_c(h_{c,s,out} - h_{c,in})}{\eta_c}$$

where w_c is the compressor power consumption (kW), m_c is the mass flow through the compressor (kg/s), $h_{c,s,out}$ is the specific enthalpy of the outlet working fluid of the isentropic process of the compressor (kJ/kg), $h_{c,in}$ is the specific enthalpy of the compressor inlet working fluid (kJ/kg), and η_c is the compressor efficiency (isentropic efficiency). The compressor design efficiency is 0.9.

The turbine work model is:

$$w_t = m_t(h_{t,in} - h_{t,out})\eta_t$$

where w_t is the work of the turbine (kW), m_t is the mass flow rate through the turbine (kg/s), $h_{t,in}$ is the specific enthalpy of the working fluid at the turbine inlet (kJ/kg), $h_{t,out}$ is the specific enthalpy of the outlet working fluid of the isentropic process (kJ/kg), and η_t is the turbine efficiency. Similar to the compressor, the turbine efficiency is designed to be 0.9 in this paper.

For heat exchangers, it is assumed that the working fluid flows in a counter-flow arrangement. The heat exchangers with the same function under different configurations are identical, and the pressure loss values on the hot and cold sides of each heat exchanger are given and remain unchanged. The heat balance method is then used to calculate the inlet and outlet parameters of the heat exchanger:

$$Q = m_h(h_{h,in} - h_{h,out}) = m_l(h_{l,out} - h_{l,in})$$

where Q is the heat transfer (kJ), m_h is the mass flow rate of the working fluid on the high-temperature side (kg/s), $h_{h,in}$ and $h_{h,out}$ are the specific enthalpy of the inlet and outlet of the working fluid on the high-temperature side (kJ/kg), m_l is the mass flow rate of the working fluid on the low-temperature side (kg/s), and $h_{l,in}$ and $h_{l,out}$ are the specific enthalpy of the inlet and outlet of the working fluid on the low-temperature side (kJ/kg).

In the calculation of the regenerator, the heat recovery degree of the regenerator is also considered. Different from the traditional definition of the heat recovery degree of water, the heat recovery degree χ of SCO_2 working medium is defined as:

$$\chi = \frac{h_{h,in} - h_{h,out}}{h_{h,in} - h^*_{l,in}}$$

where $h_{h,in}$ and $h_{h,out}$ are the specific enthalpy of the inlet and outlet of the high-temperature side of the regenerator (kJ/kg), $h_{l,in}$ and $h_{l,out}$ are the specific enthalpy of the inlet and outlet of the low-temperature side of the regenerator (kJ/kg), and $h^*_{l,in}$ is the specific enthalpy when the working medium temperature is the inlet temperature of the low-temperature side of the regenerator and the pressure is the outlet pressure of the high-temperature side of the regenerator (kJ/kg).

2.3 The Final Cycle Configuration Selection

After calculating and comparing the efficiency of different cycle system configurations, it is found that the intercooling recompression cycle with reheat has the highest cycle efficiency, followed by the intercooling recompression cycle, and the lowest efficiency is the intercooling simple regenerative cycle, as shown in Figure 16 [Figure 16: see original paper]. Table 3 shows the main node parameters of the supercritical carbon dioxide power conversion system.

2.4 Efficiency Comparison of Three Cycle Schemes Under the Same Working Conditions

To compare the efficiency of the three-cycle schemes, the design parameters such as temperature and pressure of the recompression cycle and the split recompression cycle are selected respectively, and the Peng-Robinson equation of state of Aspen HYSYS is used to simulate the three-cycle schemes. In the cycle, the efficiency of the compressor is set to 0.9, and the efficiency of the turbine is 0.9.

The simulation results are shown in Table 4 . It can be seen that the efficiency of the three cycles is higher than 35%, and the main equipment power and cycle efficiency of the recompression cycle and the split recompression cycle are the same. The heat absorption of the heat source of the simple cycle is significantly greater than that of the other two cycles. However, due to the small power consumption of the compressor, the net work of the cycle is larger, which has no significant effect on the cycle efficiency. The turbine power of the three cycles is the same, so it can be seen that when the turbine efficiency is the same, the split flow of the turbine part of the working fluid does not affect the cycle efficiency.

3. RESULTS AND DISCUSSION

Based on the operating parameters of the established SCO_2 recompression Brayton cycle model, the Peng-Robinson equation of state of Aspen HYSYS was used to analyze the compressor outlet pressure, turbine outlet pressure, heat source outlet temperature, and condenser outlet temperature under two regenerative conditions: fixed heat transfer power and fixed heat transfer temperature difference of the regenerator.

3.1 Pressure

Compressor Outlet Pressure

The outlet pressure of the compressor is the highest in the circulation system. The outlet pressure of the main compressor and the recompressor in the cycle is set to be the same value, and the influence of the outlet pressure of the compressor on the cycle efficiency is shown in Figure 9 [Figure 9: see original paper].

Under the fixed heat transfer power of the regenerator, the cycle efficiency gradually increases with the increase of the outlet pressure of the compressor, changing almost linearly, and gradually increases with the increase of the split coefficient. However, due to the limitation of heat transfer conditions, the variation range of compressor outlet pressure decreases when the split coefficient increases to 0.5. Under the fixed heat transfer temperature difference of the regenerator, the cycle efficiency increases first and then decreases with the increase of the outlet pressure of the compressor, and decreases with the increase of the split coefficient. There is a compressor outlet pressure corresponding to the highest cycle efficiency under the same split coefficient.

From the comparison of the power of the regenerator under the two regenerative conditions shown in Figure 10 [Figure 10: see original paper], it can be seen that the power of the regenerator gradually decreases with the increase of the outlet pressure of the compressor under the fixed heat transfer temperature difference of the regenerator. Increasing the outlet pressure of the compressor will increase the cost of the compressor and turbine equipment in the circulation system, but the cost of the regenerator will be reduced. Therefore, under the premise of ensuring high cycle efficiency, the matching of compressor outlet pressure and split coefficient should be comprehensively considered to obtain the best system economy.

Turbine Outlet Pressure

The turbine outlet pressure is the lowest in the circulation system. Changing the turbine outlet pressure in the cycle, the effect of the compressor outlet pressure on the cycle efficiency is shown in Figure 11 [Figure 11: see original paper].

Under the fixed heat transfer power of the regenerator, the cycle efficiency gradually decreases with the increase of the turbine outlet pressure, and gradually increases with the increase of the split coefficient. Under the fixed heat transfer temperature difference of the regenerator, the cycle efficiency increases first and then decreases with the increase of the turbine outlet pressure, and decreases with the increase of the split coefficient. Similar to the influence of compressor outlet pressure on cycle efficiency, there is also a turbine outlet pressure corresponding to the highest cycle efficiency under the same split coefficient.

The turbine outlet pressure corresponding to the highest cycle efficiency under the three split coefficients is the same. The calculation results show that this value is close to the critical pressure of carbon dioxide working fluid. From the comparison of the power of the regenerator under the two regenerative conditions shown in Figure 12 [Figure 12: see original paper], it can be seen that with the increase of the turbine outlet pressure, the power of the regenerator gradually increases under the fixed heat transfer temperature difference of the regenerator. The difference between the regenerator power at the optimal cycle efficiency and the regenerator power under the fixed regenerator power condition is small. Therefore, by setting the turbine outlet pressure close to the critical point of carbon dioxide working fluid, it is not necessary to consider the cost of the equipment.

3.2 Temperature

Heat Source Outlet Temperature

The outlet temperature of the heat source is the highest in the cycle system. Changing the outlet temperature of the heat source in the cycle, the effect of the outlet temperature of the heat source on the cycle efficiency is shown in Figure 13 [Figure 13: see original paper].

Under the fixed heat transfer power of the regenerator, the cycle efficiency gradually decreases with the increase of the outlet temperature of the heat source, and gradually increases with the increase of the split coefficient. This is due to the power limitation of the regenerator. The increase in the outlet temperature of the heat source will increase the inlet temperature of the recompression, resulting in an increase in its power consumption, which will adversely affect the cycle efficiency.

Under the fixed heat transfer temperature difference of the regenerator, the cycle efficiency increases with the increase of the outlet temperature of the heat source and decreases with the increase of the split coefficient. According to the comparison of the power of the two regenerative modes in Figure 14 [Figure 14: see original paper], under the fixed heat transfer temperature difference of the regenerator, the power of the regenerator increases with the increase of the outlet temperature of the heat source. Although a higher outlet temperature of the heat source can obtain higher cycle efficiency, it will greatly increase the power consumption of the heat source, have higher requirements for the manufacturing process of the turbine, and increase the cost of the regenerator. Therefore, for the setting of the outlet temperature of the heat source, it is necessary to comprehensively consider the cycle efficiency and the cost of the heat source, turbine, and regenerator.

Precooler Outlet Temperature

The outlet temperature of the precooler is the lowest in the circulation system. Changing the outlet temperature of the precooler in the cycle, the effect of the outlet temperature of the precooler on the cycle efficiency is shown in Figure 15 [Figure 15: see original paper].

Under the fixed heat transfer power of the regenerator, the cycle efficiency gradually increases with the increase of the outlet temperature of the precooler. This is due to the increase in the outlet temperature of the precooler, and the heat absorbed by the carbon dioxide working medium in the heat source is significantly reduced. The cycle efficiency increases with the increase of the split coefficient.

However, due to the limitation of the heat transfer power of the regenerator, the variation range is small when the split coefficient is high.

Under the fixed heat transfer temperature difference of the regenerator, with the increase of the outlet temperature of the precooler, the cycle efficiency does not change significantly near the critical point and gradually decreases away from the critical point. The cycle efficiency decreases with the increase of the split coefficient. From the power comparison of the two regenerative modes in Figure 16 [Figure 16: see original paper], it can be seen that under the fixed heat transfer temperature difference of the regenerator, with the increase of the outlet temperature of the precooler, the change of the regenerator power is similar to the changing trend of the cycle efficiency. The change is not obvious near the critical temperature, and gradually decreases away from the critical point. To obtain higher cycle efficiency, the outlet temperature of the precooler should be reduced to make the carbon dioxide working fluid close to its critical temperature, but the power of the regenerator will increase sharply, and the amount of coolant used in the precooler will also increase to a certain extent. Therefore, when setting the outlet temperature of the precooler, it is also necessary to comprehensively consider the cycle efficiency and the cost of the regenerator and coolant.

4. CONCLUSION

In this paper, the physical property calculation method provided by Aspen HYSYS software is used to simulate the SCO_2 Brayton cycle. After comparative analysis of multiple schemes, the conclusions are as follows:

1. Aspen HYSYS can effectively simulate the supercritical carbon dioxide cycle. Comparing and analyzing the simulation results of the three cycle structures, the split-recompression cycle has the highest efficiency compared with the other two structures.

2. Increasing the turbine inlet temperature and pressure can significantly improve the system cycle thermal efficiency.

3. The closer the outlet temperature and outlet pressure of S-CO_2 are to the critical point, the higher the cycle thermal efficiency.

4. There is an optimal cycle efficiency when the split flow coefficient is between 0.3 and 0.5 in the S-CO_2 recompression cycle.

5. Based on the recompression cycle layout, the waste heat recovery system can increase the cycle efficiency by 7%.

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Note: Figure translations are in progress. See original paper for figures.