Dynamic Analysis of a Closed Brayton Cycle Using Supercritical Carbon Dioxide

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Abstract: It is important to study the dynamic characteristics of closed Brayton cycle using supercritical carbon dioxide (S-CO₂) for the research of the automatic control method of the system operation. The dynamics characteristics of a split-flow recompression closed Brayton power conversion system with supercritical carbon-dioxide as the working-fluid was investigated. The mathematics model about the compressor, turbine, recuperator in a S-CO₂ Brayton cycle system was studied on the basis of the conservation principle and the basic equations of engineering thermodynamics, heat transfer theory and hydrodynamics. A complete system dynamic simulation model was built to study the dynamic response characteristics of the system under different disturbances by using the programming language in the computer. The results show that: the heating power of the heater is increased by 15.4%, the operating parameters are improved, the cycle efficiency is increased by 7.8%, the power of the pre-cooler is reduced by 50%, the cycle efficiency is reduced by 1.8%, and the opening of the valve is reduced by 10%, and the cycle efficiency is improved.

1. Introduction

With the development of economy and technology, gas turbines have become an important part of ship power systems. The increase in thermal efficiency and power of gas turbines plays a key role in the endurance and speed of ships. In addition, due to the limited space of the engine room, how to reduce the volume of the power plant by optimizing the gas turbine cycle is a concern of related scholars and technicians [1]. In recent years, the Brayton cycle using S-CO₂ as a working fluid has become a hot spot for scholars. In the S-CO₂ Brayton cycle, the working fluid is not in contact with the atmosphere, and a larger pressure ratio can be used to increase the power of the unit. And it is possible to improve the circulating operating parameters by adjusting the flow of the working fluid to increase the atmospheric, and a larger pressure ratio can be used to increase the power of the unit. And it is possible to improve the circulating operating parameters by adjusting the flow of the working fluid to increase the thermal efficiency. At the same time, due to the physical characteristics of S-CO₂, the S-CO₂ Brayton cycle has a huge advantage in compactness. In a suitable temperature range (400-750°C), the efficiency of the Brayton cycle is higher than that of the Rankine cycle [2], and it also has great prospects in the utilization of ship waste heat [3].

The research on the S-CO₂ Brayton cycle began in the 1960s. Feher et al. [4] conducted a study on the S-CO₂ Brayton cycle, it has great advantages compared with the traditional steam cycle. However, due to the limitations of mechanical design and production technology at that time, the research and application of the S-CO₂ Brayton cycle encountered great resistance. With the advancement of science and technology, the S-CO₂ Brayton cycle has gradually been favored by scholars in the field of power generation. Internationally, many institutions and scholars have carried out fruitful work around the design, performance calculation, control strategy, system optimization, and the characteristics of
S-CO$_2$ heat exchanger of the S-CO$_2$ Brayton cycle system. For example, Padilla R V [5] studied the influencing parameters of the performance of the S-CO$_2$ Brayton cycle system using a solar central receiver as a heat source; American Argonne National Laboratory Moisseytsev A et al. [6-7] studied the application of the S-CO$_2$ Brayton cycle in the field of nuclear energy, analyzed the influencing factors of cycle efficiency, and proposed an optimization plan for the cycle layout.

The Sandia National Laboratory in the United States and related research institutes have conducted in-depth research on the S-CO$_2$ Brayton cycle. Turchi CS et al. [8-9] studied the adaptability of S-CO$_2$ Brayton cycle in the field of renewable energy and coal-fired power plants, the results show that the cycle thermal efficiency is improved because S-CO$_2$ is used as the cycle working fluid; Conboy Tet al. [10] comparatively analyzed the thermal performance of S-CO$_2$ Brayton cycle and conventional steam Rankine cycle, and build a low-parameter Brayton cycle test system with different layouts [11]. EPS company (Echogen Power System) built a commercial EPS100 power machine [12], its net cycle power reached 7.5MW, and its performance is significantly better than the steam system. Compared with the gas turbine between 20 and 50 MW, its net power has increased by 35%. In short, the S-CO$_2$ Brayton cycle shows a huge advantage in the current research.

Chinese scholars have also carried out corresponding studies on the S-CO$_2$ Brayton cycle. Zhang Yifan [13] performed static thermodynamic calculations on the S-CO$_2$ Brayton cycle by using computer programming and Aspen Plus software to analyze and study the influencing parameters of cycle efficiency. Based on the laws of thermodynamics, Guo Jiaqi et al. [14] calculated and analyzed the factors affecting the cycle efficiency of the three cycle modes, and innovatively added other gases to the working fluid to reduce the critical parameters of the working fluid. However, there are few documents and research results about the dynamic characteristics of S-CO$_2$ system. Massachusetts Institute of Technology and some foreign advanced nuclear research institutions [15] modified GAS-PASS (a Fortran program code for Brayton cycle simulation and control) to make it suitable for CO$_2$ working fluid, establishing S-CO$_2$ Brayton Cyclic dynamic simulation model to simulate the response characteristics of the system under different working conditions.

Real-time dynamic simulation based on mathematical models is one of the classic methods for studying the dynamic characteristics of thermal systems. This thesis takes the S-CO$_2$ Brayton cycle as the object, using theoretical modeling and digital simulation methods, Study the real-time dynamic simulation model of the thermal system. The research on the dynamic response characteristics of the system under different disturbances is the focus. The system model is mainly composed of compressor, turbine, single-phase medium heat exchanger and auxiliary model. Under the step disturbance of different boundary conditions, the influence of system external conditions on the circulatory system is studied by analyzing the change law of system operating characteristics. It can lay the foundation for studying the automatic control method of S-CO$_2$ circulation system.

2. Overview of S-CO$_2$ Split Recompression Brayton Cycle System

Taking the S-CO$_2$ split and recompression Brayton cycle system as the simulation research object, the system mainly includes two types of equipment, rotating and heat exchange. The rotating equipment mainly includes compressor and turbine, and the heat exchange equipment mainly includes High Temperature Recuperator, (HTR), Low Temperature Recuperator (LTR), heater and precooler. During the operation of the system, the high temperature and low pressure S-CO$_2$ fluid at the outlets of the two sets of turbines are mixed (point 8), then turned into low-temperature and low-pressure S-CO$_2$ fluid (point 10) through HTR and LTR, and finally enters the low-temperature side of the cycle. Separate the flow of low-temperature and low-pressure CO$_2$ fluid, a part of it flows into the pre-cooler and then enters the main compressor (point 1-2), and the other part is directly boosted in the recompressor (points 2a-3a). The high-pressure and low-temperature CO$_2$ fluid from the main compressor is first heated by the LTR (point 4), mixed with the working fluid at the outlet of the recompressor, and then flows into the HTR to be heated again and enter the high-temperature side of the cycle. The S-CO$_2$ fluid is heated in the heater (points 6-7) to meet the turbine inlet temperature conditions. Split the flow into the two turbines and form a closed circuit. The system diagram is
shown in Figure 1.

![Schematic diagram of S-CO₂ split-flow recompression closed Brayton power conversion cycle](image)

Fig.1 flowchart of S-CO₂ split-flow recompression closed Brayton power conversion cycle

The main design operating parameters of the circulatory system in the steady state are shown in Table 1.

| Parameter          | Total flow/kg s⁻¹ | Compressor split ratio | Turbine split ratio | Main compressor outlet pressure/MPa | Recompressor outlet pressure/MPa | Turbine inlet temperature/℃ | Turbine outlet pressure/MPa | Outlet temperature of precooler/℃ |
|--------------------|-------------------|------------------------|---------------------|------------------------------------|---------------------------------|----------------------------|-----------------------------|----------------------------------|
| value              | 3.483             | 0.73                   | 0.5                 | 10.6                               | 10.6                            | 400                        | 7.8                         | 35.5                             |

3. Simulation Model of S-CO₂ Cycle System

The flow of S-CO₂ in the Brayton cycle system is continuous. The calculation and solution of the system requires the establishment of mathematical models and digital simulation models. Mathematical model based primarily on the energy balance, momentum balance, mass conservation equation is composed of differential equations and algebraic equations. Real-time dynamic simulation requires high calculation speed, so partial differential equations are rarely used. Instead of a continuous system model simulation model discrete difference equation by using Euler discrete mathematical equations to obtain. The main components of the system include three modules: turbine, compressor, and heat exchanger. Next, the modeling process will be introduced in detail.

3.1. Simulation Model of Recuperator

Recuperator must have the characteristics of adapting to a wide range of operating conditions and high heat exchange efficiency before it can be used in the S-CO₂ cycle system. The Printed Circuit Heat Exchanger (PCHE), which is one of the microchannel heat exchangers, has the characteristics of high integration and compact size. Compared with conventional shell-and-tube heat exchangers, PCHE has stronger adaptability to high-temperature and high-pressure fluids. PCHE is selected for HTR and LTR in the research system. Figure 2 is a schematic diagram of the regenerator model.

In order to simplify the recuperator model, the following assumptions are made:

1. Due to the extremely small diameter of the PCHE microchannel, it is assumed that the temperature and pressure of the fluid only change in the axial direction (flow direction);
2. The thermal resistance of metal is negligible compared with the thermal resistance of fluid heat transfer, and the metal temperature difference is close to zero. Therefore, the metal wall has no thermal resistance, and its temperature is uniform;
3. When the fluid passes through the heat exchanger, it is considered that the heat absorption in the pipe direction is certain. At the same time, it is reasonable to ignore the changes in the specific heat capacity and density of the working fluid; without considering leakage, the flow of the working fluid in the heat exchanger is related to the physical properties of the working fluid.
Fig. 2 Recuperator unit model

Continuity equation:

\[
\frac{dm}{dt} = m_{\text{in}} - m_{\text{out}} \tag{1}
\]

In the modeling process, it is necessary to ignore the influence of density on the flow rate and the inlet and outlet flow rates of the heat exchanger are considered equal.

\[
m_{\text{hot}}^{p+1} = m_{\text{hot-in}}^{p+1} = m_{\text{hot-out}}^{p+1} \tag{2}
\]

\[
m_{\text{cold}}^{p+1} = m_{\text{cold-in}}^{p+1} = m_{\text{cold-out}}^{p+1} \tag{3}
\]

In (1)-(3) equation: Subscript (hot, cold, in, out) respectively indicate the high temperature fluid, cold fluid, inlet parameters, and outlet parameters in the heat exchanger; Superscript(p, p+1) respectively represent two adjacent moments with a difference of one unit time; \(m\) — mass flow, kg/s.

Energy conservation equation:

\[
\frac{dE}{dt} = E_{\text{in}} - E_{\text{out}} \tag{4}
\]

Discretization of the energy equation on the thermal fluid side:

\[
\frac{m_{\text{hot}}^{p+1}U_{\text{hot}}^{p+1} - m_{\text{hot}}^{p}U_{\text{hot}}^{p}}{\Delta t} = m_{\text{hot-in}}^{p+1}h_{\text{hot-in}}^{p+1} - m_{\text{hot-out}}^{p+1}h_{\text{hot-out}}^{p+1} - Q_{\text{hot}}^{p+1} \tag{5}
\]

Heat transfer equation on the high temperature side:

\[
Q_{\text{hot}}^{p+1} = HTC_{\text{hot}}^{p+1}A(T_{\text{hot}}^{p+1} - T_{\text{wall}}^{p+1}) \tag{6}
\]

In (4)-(6) equation: E — Sum of fluid energy per unit time, kJ/s; \(U\) — The internal energy of the fluid in the heat exchanger, kJ; \(h\) — The enthalpy value of the fluid in the heat exchanger, kJ/kg; \(HTC\) — The heat transfer coefficient of the fluid, kJ/(m².K); \(T_{\text{hot}}^{p+1}\) — The average temperature of the thermal fluid in the heat exchanger at \(p+1\), °C; \(T_{\text{wall}}^{p+1}\) — The average temperature of the metal wall of the heat exchanger at \(p+1\), °C; \(u\) — The flow velocity of the fluid, m/s.

Momentum equation:

\[
P_{\text{in}} - P_{\text{out}} = K_{p}m_{\text{hot}}^{2} \tag{7}
\]

Calculate the outlet pressure through the pressure loss coefficient along the way:

\[
\Delta p_{\text{hot}}^{p+1} = K_{p}m_{\text{hot}}^{2} \tag{8}
\]

In the equation: \(P\) — Fluid pressure, Pa; \(\Delta P\) — The average fluid pressure drop, Pa; \(K_{p}\) — Coefficient of pressure loss along the way.

The calculation equation of the fluid on the low temperature side is similar to that on the high temperature side, and will not be repeated here.
3.2. Mathematical Model of Compressor

In order to simplify the actual process, here we only study the changing laws of the working fluid's physical properties on the axis of the compressor, and simplify the complex flow in the actual process into a one-dimensional fluid model.

Mass conservation equation:

\[
V \frac{d \rho_2}{dt} = \dot{m}_1 - \dot{m}_2
\]  

(9)

In the equation: \(\dot{m}_1, \dot{m}_2\) — The mass flow rate of inlet and outlet of working fluid in the compressor per unit time, kg/s; \(V\) — Volume of compressor flow channel, m³; \(\rho_2\) — The air density at the outlet of the compressor, kg/m³.

Energy balance equation:

\[
V \frac{d (\rho H - P)}{dt} = \dot{m}_1 H_1 - \dot{m}_2 H_2 + \dot{Q} - \dot{W}_s
\]  

(10)

In the equation: \(\dot{Q}\) — External heat transfer of the working fluid in the compressor per unit time, kW; \(H_1, H_2\) — The enthalpy value of the air at the inlet and outlet of the compressor, kJ/kg; \(\dot{W}_s\) — power consumption per unit time of the compressor, kW.

Momentum equation:

\[
V \frac{d (\rho u)}{dt} = \dot{m}_1 u_1 + P_1 A_1 - \dot{m}_2 u_2 - P_2 A_2 + F
\]  

(11)

In the equation: \(u_1, u_2\) — The axial velocity of the working fluid at the inlet and outlet of the compressor, m/s; \(A_1, A_2\) — The flow area of the compressor inlet and outlet, m²; \(P_1, P_2\) — The pressure of the working fluid at the inlet and outlet of the compressor, Pa; \(F\) — The force of the compressor blades on the working fluid, N.

Compressor outlet pressure:

\[
\frac{d P_2}{dt} = \frac{R \left( m_1 C_p T_1 - W_s \right) - C_p A_2 u_2}{V_2 \left( C_p - R \right)}
\]  

(12)

In the equation: \(C_p\) — Heat capacity of working fluid at constant pressure, kJ/(kg•K); \(R\) — Working fluid flow resistance coefficient.

Compressor outlet temperature:

\[
\frac{dT_2}{dt} = \frac{R \left( m_1 C_p T_1 - W_s \right) T_2}{V_2 \left( C_p - R \right) p_2} - \frac{R A_2 T_2 u_2}{V_2 \left( C_p - R \right) p_2} - \frac{R m_1 T_1^2}{V_2 p_2}
\]  

(13)

Actual consumption of shaft power by compressor:

\[
W_s = m_1 H_1 - m_2 H_2
\]  

(14)

The compressor's operating conditions are determined by the compressor's performance curve and pipeline characteristic curve. At present, there is no precise theoretical method to calculate the performance curve of the compressor, so the establishment of the simulation model is mainly based on the experimental performance curve provided by the manufacturer. The following simulates the dynamic pressure and flow characteristics of the compressor.

For a compressor with a certain structure, the actual total pressure is affected by the compressor inlet flow rate and the rotor speed of the unit. The calculation formula is:
\[ \Delta P_{\text{max}} = \alpha \left( \frac{n}{n_0} \right)^2 \] (15)

In the equation: \( \Delta P_{\text{max}} \) —Maximum total pressure of compressor, Pa; \( \alpha \) —Standardized guide vane angle, its value is 0~1, corresponding to the guide vane rotation angle range; \( n, n_0 \) —The actual speed of the compressor rotor and the rated speed of the compressor rotor, r/min.

The air flow rate in the compressor:

\[ G_c = c_0 \left( P_1 + \Delta P_{\text{max}} - P_2 \right)^{1/2} \] (16)

In the equation: \( c_0 \) —Compressor admittance, which is related to the compressor flow channel structure and air characteristics.

Compressor efficiency \( \left( \eta_c \right) \) is the ratio of ideal power consumption to actual compression work.

4. Dynamic Simulation Experiment of S-CO\(_2\) Split-flow Recompression Brayton Cycle System

4.1. Static simulation verification
The relevant structural parameters required for modeling come from literature. Compare the static simulation results after the simulation system is running stably with the experimental results of Sandia laboratory \(^7\) to verify the accuracy of the simulation system to simulate the S-CO\(_2\) cycle. Table 2 shows the comparison results of the power of the main components of the system and the cycle efficiency of the system.

| Parameter | Sandia laboratory experimental results | Simulation results | Relative error/% |
|-----------|---------------------------------------|--------------------|-----------------|
| Total flow/kg.h-1 | 12539 | 12539 | 0 |
| Turbine outlet pressure/MPa | 7.96 | 7.88 | 1.0 |
| Turbine inlet temperature/\(^\circ\)C | 400 | 403 | -0.7 |
| Turbine inlet pressure/MPa | 9.9 | 10.1 | -2.0 |
| Heater input heat/kW | 341.7 | 346.1 | -1.3 |
| Pre-cooler output heat/kW | 239.2 | 246.1 | -2.8 |
| Work done by Turbine A/kW | 36.4 | 36.3 | 0.3 |
| Work done by Turbine B/kW | 37.8 | 36.3 | 4.0 |
| Main compressor power consumption/kW | 39.2 | 38.1 | 2.9 |
| Recompressor power consumption/kW | 16.7 | 16.5 | 1.2 |
| Circulating net power/kW | 18.3 | 18.0 | -1.7 |
| Cycle efficiency/% | 5.4 | 5.2 | -3.8 |
It can be seen from the comparison between the simulation static results and the laboratory measurement results that the relative errors of the main parameters are all within 5%. The power error of the compressor and turbine is slightly higher. The main reason is the ideal assumption of the working fluid flow process in the modeling process, ignoring the heat loss and friction loss in the actual process. The absolute error of the cycle efficiency is only 0.2%. Considering that part of the pipeline loss is ignored during the simulation operation, and the measurement results of Sandia laboratory have measurement errors, the simulation model built is in good agreement with the experimental results, and the model can be used to analyze the dynamic characteristics of the system.

4.2. Heating Disturbance Experiment

When t=0s, the heating power of the heater is increased from 346.1kW to 416.7kW. The temperature response process of the main components of the system is shown in Figure 3(a): The heater outlet temperature responds quickly. At t=1000s, the heater outlet temperature increases by 8.2%, and the change trend is from rapid to slow; Due to pipe heat loss, the turbine inlet temperature is slightly lower than the heater outlet temperature, and the response is slightly slow. The temperature change of the HTR thermal fluid inlet is relatively gentle, and its value increases by 9.6% in the steady state; The temperature response time of the above components on the high temperature side of the circulatory system is different, corresponding to the position of the module, that is, the temperature response time of the heater at the disturbance end is the shortest, and the response time of each component is gradually extended along the way. There are two reasons for this. First, it takes a certain time for the S-CO2 fluid from the heater outlet to the heater inlet; second, HTR and LTR have strong thermal inertia. There are two reasons for this. First, it takes a certain time for the S-CO2 fluid from the heater outlet to the heater inlet; second, HTR and LTR have strong thermal inertia. And it was observed that the temperature at the outlet of the cold HTR fluid changed very little. This phenomenon was caused by the change of S-CO2. That is, under the same change in heat exchange, the temperature of the cold test fluid hardly changed due to the larger specific heat.

As the inlet temperature of the turbine increases, the power generation capability of the turbine is enhanced. As shown in Figure 3(b), when it reaches the steady state, the turbine work increases by 6.1%; As the temperature difference between the hot side and the cold side of the HTR increases, when t=3500 s, the heat transfer increases by 20.6%; From Figure 3(a), it can be seen that the temperatures on the low-temperature side of the cycle are almost unchanged. The compressor inlet and outlet parameters have not changed much, Therefore, the power consumption of the main compressor and the recompressor has only a minimal change.

The cycle efficiency calculation formula:

\[ \eta = \frac{W}{Q} \times 100\% \]  \hspace{1cm} (17)

In the above formula, W- the net circulating power, kW; Q -- the heater input heat, kW

The heating capacity of the heater increased from 346kW to 416kW in a short time, It can be seen from formula (17) that the cycle efficiency instantly drops from 5.1% to 4.4%. As the turbine work increases, the net cycle power increases, and the cycle efficiency continues to rise. At t=2000s, the cycle efficiency rises to 5.5%.
4.3. Disturbance of the Opening Degree of the Main Compressor Regulating Valve

When \( t = 0 \)s, the opening degree of the regulating valve at the outlet of the main compressor is reduced from 100% to 90%, and the heater heat remains unchanged at 416.7kW.

Because the valve opening of the main compressor is reduced, the pressure in front of the valve rises, and the pressure of the circulation system does not change much because the circulation is a closed loop arrangement. As shown in Figure 4(a), the inlet pressure of the main compressor rose from 7.70MPa to 7.73MPa during 0-300s, an increase of 0.4%; The fluid outlet pressure on the high-pressure side of the LTR rose from 10.10 MPa to 10.14 MPa, an increase of 0.4%.

Figure 4(b) shows the change of flow in the circulation system, The instantaneous decrease of valve opening reduces the flow rate to the lowest value in a very short time, that is, in 0-20s, the main compressor outlet flow rate drops from 9128kg/h to 8221kg/h. But in 20-240s, the outlet flow of the main compressor increased from 8221kg/h to 8273kg/h. This phenomenon is caused by the increase of the density of S-CO2 due to the slight increase in working fluid pressure.

The heating power of the heater does not change. Due to the decrease in the flow rate, the heater outlet temperature changes extremely fast, as shown in Figure 4(c). In a very short period of time, the temperature rose from 436°C to 446°C. The outlet temperature of the HTR cold fluid rises sharply at the beginning of the disturbance, from 314.2°C to 322.3°C, It is mainly caused by the rapid decrease in the flow rate under the condition that the heat exchange amount of the HTR fluid is almost unchanged at 0-20s (as shown in Figure 4(d)).

Due to the rapid decrease in the flow rate, the power consumption of the main compressor and the recompressor decreases instantaneously, as shown in Figure 4(d), and due to the rise in temperature and pressure, the output power and the heat exchange of the regenerator increase.

As the power consumption of the compressor is reduced in a very short time, it can be seen from equation (17) that the cycle efficiency is instantly increased from 5.1% to 6.5%. After that, the efficiency is slowly increased mainly due to the increase of the turbine output power. At \( t = 4000 \)s, The efficiency reaches 7.2%.

It can be seen from the analysis of the disturbance experiment results of the opening degree of the main compressor regulating valve, The valve opening of the main compressor is reduced from 100% to 90%, the cycle efficiency of the system is effectively improved (from 5.1% to 7.2%). That shows that the opening of the main compressor valve is slightly adjusted to be beneficial to the operation of the circulatory system.
5. Conclusions

1) A dynamic simulation model of the S-CO₂ shunt and recompression Brayton cycle system was built. Each module in the simulation system corresponds to each device in the system. The stable operation parameters of the model conform to the experimental results, which verifies the correctness of the modeling idea.

2) The simulation results show that the model can not only correctly reflect the operating characteristics of the S-CO₂ Brayton cycle system, but also reflect the change trend of the thermal performance of the main equipment in the system.

3) The dynamic disturbance experiment results show that the model has high static accuracy and self-stability, which can provide a good reference model for the research and development of the actual unit control system.

4) The results of the disturbance experiment show that the heating power has a significant effect on the cycle efficiency; a small valve opening can increase the cycle efficiency and improve the system performance; the cooling capacity of the precooler has a minimal effect on the high temperature side parameters of the cycle system and the cycle efficiency.

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