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# Design and Optimization of a High-Efficiency Supercritical $CO_2$ Brayton Cycle System

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## ABSTRACT

In recent years, the challenges of energy utilization and environmental protection have become increasingly prominent due to rapid economic growth and technological advancement. As a result, the development of novel energy conversion systems that simultaneously improve efficiency and reduce environmental impact has remained a key research focus. Among these technologies, the supercritical carbon dioxide ($sCO_2$) Brayton cycle has garnered significant attention owing to its high thermal efficiency, reduced component count, and compact system configuration. However, the actual cycle efficiencies achieved by existing $sCO_2$ Brayton cycle units remain relatively low, hindering their practical application. The development of high-efficiency systems thus represents a major challenge in advancing this technology. This study focuses on practical $sCO_2$ Brayton cycle units and investigates strategies for improving system efficiency through cycle layout optimization, and provide insights and guidance for the design and development of advanced $sCO_2$ Brayton cycle systems.

Keywords: supercritical carbon dioxide; Brayton cycle; high cycle efficiency; thermodynamic performance;

## INTRODUCTION

The supercritical carbon dioxide Brayton ($sCO_2$) cycle has garnered increasing attention from researchers due to its superior thermal efficiency, reduced component size, and compact layout [1]. However, the actual efficiency of currently operational units falls significantly short of theoretical calculations. For instance, Sandia National Laboratories began constructing and researching an $sCO_2$ cycle test rig in 2008. By 2012, they had completed the installation and power-generation testing of a recompression cycle based on a simple regenerative configuration, producing the first $sCO_2$ test loop capable of generating electricity. However, the relevant operating parameters did not reach their design values. Its power-generation capacity was 12 kW, with the turbine inlet temperature of 399 °C and a maximum rotational speed of 65,000 rpm [2, 3]. Xi'an Thermal Power Research Institute Co., Ltd. has developed a 5 MW $sCO_2$ power-generation unit. Under compressor inlet conditions of 35 °C and 7.9 MPa, and turbine inlet conditions of 600 °C and 20 MPa, the maximum output power reached 3.866 MW, with a tested efficiency of 25.1% [4]. During actual operation, factors such as pipeline pressure losses, cooling losses, seal leakage, and other inefficiencies cause the operating efficiency to be lower than the ideal design value. The effects of these losses on system performance provide important theoretical guidance for the practical operation of $sCO_2$ generating units [5].

To explore more efficient designs for $sCO_2$ Brayton cycle systems, researchers have undertaken extensive studies. Among these, Xu et al. conducted research based on a 1000 MW unit. By employing designs featuring intercooling, split recompression, and secondary reheating, the system achieved a net efficiency of 48.37% under operating conditions of 32 MPa compressor outlet pressure and 620 °C turbine inlet temperature [6]. Ma et al. designed a 1000 MW net power-generation unit based on an intercooled, secondary-reheat, and recompression cycle configuration. Under operating conditions of 630 °C turbine inlet temperature and 35 MPa inlet pressure, the system achieved a thermal efficiency of 54.68% [7]. Zhang et al. designed a unit with a net power output of 1000 MW, employing a cycle configuration featuring recompression and single reheat. Under conditions of the turbine inlet temperature of 600 °C and the inlet pressure of 31 MPa, the system efficiency reached 50.71% [8]. However, in many studies, pipe pressure loss and boiler loss are insufficiently considered, and the energy consumption of cooler motors is sometimes overlooked. Therefore, to assess the efficiency achievable during actual operation, we modeled the system in gPROMS and optimized its design.

The efficiency of the system is defined as:

$$\eta = \frac{W_{net}}{Q}$$

$$W_{net} = W_G - W_M$$

where $W_{net}$ is the system's net electrical output, $W_G$ is the generator's net output, $W_M$ is the motor's power consumption, and Q is the heat input to the boiler.

The efficiency of the compressor is:

$$\eta_C = \frac{h_{outc,s} - h_{inc}}{h_{outc} - h_{inc}}$$

where $h_{outc,\ s}$ is the enthalpy of the outlet working medium for an isentropic compression; $h_{inc}$ is the enthalpy of the working medium at the compressor inlet; and $h_{outc}$ is the enthalpy of the outlet working medium in actual operation.

The efficiency of the turbine is:

$$\eta_{TU} = \frac{h_{intu} - h_{outtu}}{h_{intu} - h_{outtu,s}}$$

where $h_{outtu,\ s}$ is the enthalpy of the working medium at the turbine outlet for an isentropic expansion; $h_{intu}$ is the enthalpy at the turbine inlet; and $h_{outtu}$ is the enthalpy at the turbine outlet in actual operation.

The input heat of the heater is:

$$Q = m\left(h_{out} - h_{in}\right)$$

where m is the mass flow rate of the working medium through the heater; $h_{out}$ and $h_{in}$ are the outlet and inlet enthalpies of the working medium.

The calculation formula for the regenerator is:

$$T_{h,out} = T_{c,out} + \Delta T$$

$$m_h(h_{h,in} - h_{h,out}) = m_c(h_{c,in} - h_{c,out})$$

where ΔT is the terminal temperature difference of the regenerator; $T_{h,\ out}$ and $T_{c,\ out}$ are the outlet temperatures on the hot side and cold side; $m_h$ and $m_c$ are the mass flow rates on the hot side and cold side; $h_{h,\ in}$ and $h_{h,\ out}$ are the inlet and outlet enthalpies on the hot side; $h_{c,\ in}$ and $h_{c,\ out}$ are the inlet and outlet enthalpies on the cold side.

The formula for calculating the cooler is:

$$m_{CO_2}(h_{CO_2,in} - h_{CO_2,out}) = m_W(h_{W,in} - h_{W,out})$$

where $m_{CO2}$ is the mass flow rate of carbon dioxide, $m_W$ is the mass flow rate of the cooling water;

$h_{CO2,\ in}$ and $h_{CO2,\ out}$ are the inlet and outlet enthalpies of carbon dioxide; and $h_{W,\ in}$ and $h_{W,\ out}$ are the inlet and outlet enthalpies of the cooling water.

During the design and optimization of the system, the design parameters for the main components are as shown in Table 1.

Table 1 Main design parameters of the system

| Parameter (Unit) | Value |
|---|---|
| Efficiency of compressor (%) | 89 |
| Inlet temperature of compressor (°C) | 32.00 |
| Efficiency of turbine (%) | 92 |
| Efficiency of boiler (%) | 94.71 |
| Efficiency of pipe (%) | 99 |
| Outlet temperature of heater (°C) | 650 |
| Pressure loss of the hot side of the regenerator (MPa) | 0.1 |
| Pressure loss of the cold side of the regenerator (MPa) | 0.1 |
| Regenerator end difference (°C) | 8 |
| Pressure loss of $CO_2$ side of cooler (MPa) | 0.1 |
| Pressure loss of the water side of the cooler (MPa) | 0.2 |
| Cooling water temperature rise (°C) | 10 |

Although the proposed system configuration-featuring single intercooling, double reheating, and double recompression-introduces additional components and complexity compared to simple recuperated cycles, this design choice is driven by the stringent efficiency targets required for next-generation power plants. The system is modeled as a hundred-megawatt class unit, making the assumed turbomachinery efficiencies achievable. For such utility-scale applications, the thermodynamic benefits of improved heat source matching and reduced irreversibility in the heat exchange process significantly outweigh the increase in system complexity. The achieved system efficiency of 50.13% represents a substantial improvement over simpler configurations, translating to significant reductions in fuel consumption and operational costs over the plant's lifecycle.

## RESULTS AND DISCUSSION

Regarding the $sCO_2$ cycle, modelling and analysis were conducted using gPROMS. Following a systematic optimisation analysis, a cycle configuration featuring single intercooling, double reheating, and double recompression was selected. The system schematic is shown in Fig. 1.

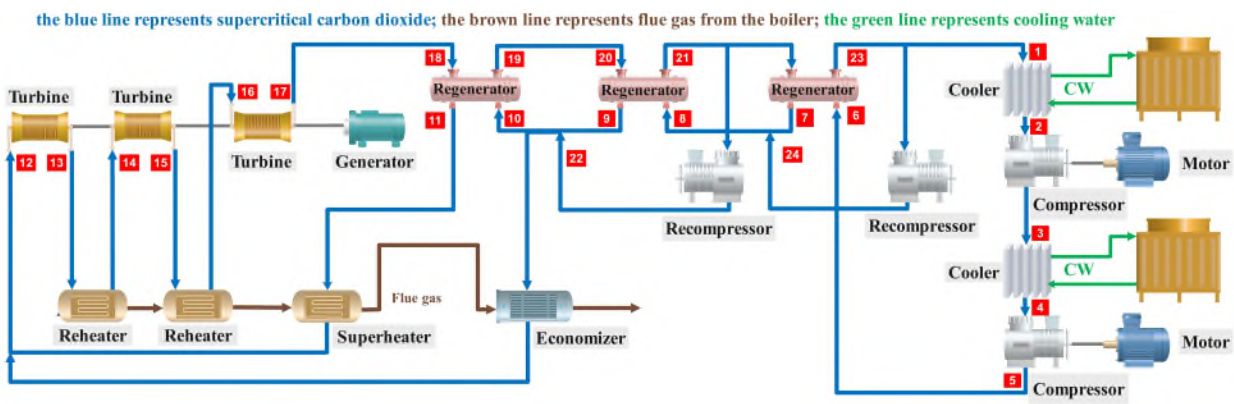


Fig. 1 Schematic diagram of the cycle

The optimization of system efficiency, targeting the highest possible cycle efficiency, yields parameters that achieve a high overall efficiency. After optimization, a system efficiency of 50.13% was obtained, with the remaining parameters listed in Table 2.

Table 2 Key parameters within the optimised system

| Point | Temperature/K | Pressure/MPa | Mass flow rate/(kg/s) |
|---|---|---|---|
| 1 | 334.73 | 7.90 | 1428.01 |
| 2 | 305.15 | 8.00 | 1428.01 |
| 3 | 318.73 | 13.64 | 1428.01 |
| 4 | 305.15 | 13.54 | 1428.01 |
| 5 | 325.48 | 32.85 | 1428.01 |
| 7 | 459.96 | 32.75 | 1428.01 |
| 8 | 459.96 | 32.75 | 2062.11 |
| 9 | 666.45 | 32.65 | 2062.11 |
| 10 | 660.72 | 32.65 | 2225.35 |
| 11 | 838.57 | 32.55 | 2225.35 |
| 12 | 923.15 | 31.90 | 2480.88 |
| 13 | 883.71 | 24.09 | 2480.88 |
| 14 | 923.15 | 23.94 | 2480.88 |
| 15 | 859.40 | 15.04 | 2480.88 |
| 16 | 923.15 | 14.89 | 2480.88 |
| 17 | 846.57 | 8.40 | 2480.88 |
| 19 | 675.45 | 8.30 | 2480.88 |
| 21 | 467.96 | 8.20 | 2480.88 |
| 22 | 467.96 | 8.20 | 418.77 |
| 23 | 334.73 | 8.10 | 2062.11 |
| 24 | 334.73 | 8.10 | 634.10 |

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