Study on Supercritical Carbon Dioxide Recompression Cycle System Employing Solar Energy

Chen Huiyong, Xue Zhiheng, Wang Weifeng, Wu Tao, Fu Chang
Xi’an Thermal Power Research Institute Co., Ltd., Xi’an 710054, China
chenhuiyong@tpri.com.cn

Abstract. In this paper, a new kind of power generation system was built, which took solar energy as heat source of the system. All heat exchangers in the system adopted a new kind of heat exchanger type-Printed Circuit Heat Exchanger (PCHE). In order to take advantage of the abrupt property changes near the critical point of carbon dioxide, we selected recompression Brayton cycle. Then, given the operating conditions, the optimization progress was performed and the optimized parameters were obtained.

1. Introduction
The energy problem has become one of the three major problems in the world today. Energy greatly determines the status of human survival and social development degree, by the rapid development of social economy, more and more energy was needed [1]. Also, as the lack of fossil and serious environmental problems, clean renewable energy must be explored actively [2]. As a kind of renewable energy, solar energy is abundant, clean and renewable. With the development of solar thermal power generation, solar thermal power generation would have a potential in competing with conventional energy power generation [3]. Carbon dioxide was selected because of the moderate value of its critical pressure, its stability and relative inertness, sufficient knowledge of its thermodynamic properties, non-toxicity, abundance and low cost [4]. Taking the advantage of the abrupt property changes near the critical point of carbon dioxide could reduce the compression work, which resulted in a significant efficiency improvement [5].

2. Composition and working principle of the S-CO$_2$ recompression cycle system employing solar energy
The composition of the S-CO$_2$ recompression cycle system employing solar energy was presented in Fig.1. And the work process was as follows: this cycle layout improved efficiency by reducing the heat rejection from the cycle by introducing another compressor (a recompressing compressor) before the pre-cooler. The flow was split before entering the pre-cooler and heat was rejected only from part of the fluid flow. The outlet of the recompressing compressor was connected between the high and low temperature recuperators, which was another difference from the simple Brayton cycle where only one recuperator was used. Otherwise, the cycle was same.
Figure 1. Schematic diagram of the S-CO$_2$ recompression cycle system employing solar energy.

In the main compressor (points 1-2), a fraction of the fluid flow was compressed to high pressure. In the low temperature recuperator, the S-CO$_2$ fluid was preheated to the recompressing compressor outlet temperature (points 2-3). Then the fluid was merged with the rest of the fluid flow from the recompressing compressor (point 3). The entire fluid flow was then preheated in the high temperature recuperator to the heat transfer inlet temperature (points 3-4). The heat addition into the cycle took place in the heat transfer (points 4-5). The fluid left the heat transfer at the highest cycle temperature. At this temperature, it entered the turbine, where fluid expansion (points 5-6) generated rotational energy, which was converted into electricity in the generator. After leaving the turbine, the high temperature fluid was cooled in the high (points 6-7) and low (points 7-8) temperature recuperators, where the available heat was transferred to the cooler high-pressure side fluid flow. Before entering the precooler, the fluid flow was split (point 8). One part was recompressed to high pressure (points 8-3), and then the other was cooled in the precooler to the main compressor inlet temperature (points 8-1). The temperature-entropy diagram of the recompression cycle was shown in Figure 2.

Figure 2. Temperature-entropy diagram of a recompression Brayton cycle.
3. Thermal analysis and thermal parameters selection of S-CO\textsubscript{2} recompression cycle system employing solar energy

The operating conditions selected for this analysis were 537\degree C for the turbine inlet temperature, 32\degree C for the compressor inlet temperature and 13.8MPa for the compressor outlet pressure. The heat transfer power was 780kw. A cooling water inlet temperature of 27\degree C was used. The last two assumptions are the turbine and compressor efficiencies, which were taken at 0.7 and 0.87 respectively. Unless otherwise specified the values presented in this paragraph would be used for the optimization process of all cycle layouts.

3.1 Solar heating system design

In this paper, the solar heating system adopted tower solar system. And the tower solar heating system parameters are shown in Table 1.

| Item                              | Unit | Value |
|-----------------------------------|------|-------|
| Power                             | kW   | 780   |
| Height of heat absorber           | m    | 2.67  |
| Heat absorber diameter            | m    | 1.6   |
| Height of tower                   | m    | 20.6  |
| heliostat size                    | m×m  | 6.6×6.0 |
| Heliostat number                  |      | 120   |
| Inlet temperature of medium       | \degree C | 292 |
| Outlet temperature of medium      | \degree C | 565 |

3.2 Turbomachinery design

This section described the turbomachinery design. For both component types (turbine and compressors), axial flow machines were selected. The main reason was the necessity for employing multiple stage machines. The efficiency of centrifugal flow machines dropped significantly when multiple stages were used.

3.2.1 Compressor design. The cycle used two compressors. The main compressor operated close to the critical point and compresses at least 60\% of the total flow. Table 2 summarized the main compressor parameters. The recompressing compressor was considerably easier to design, since it operated much further away from the critical point. The recompressing compressor parameters were also shown in Table 2.

| Compressor       | Unit     | Main Compressor | Recompressing Compressor |
|------------------|----------|-----------------|--------------------------|
| Inlet pressure $P_{\text{com-in}}$ | MPa      | 7.69            | 7.755                    |
| Outlet pressure $P_{\text{com-out}}$ | MPa     | 13.842          | 13.727                   |
| Pressure ratio $r_{\text{frac}}$ | 1.8     | 1.77            |                          |
| Inlet temperature $T_{\text{com-in}}$ | \degree C | 32              | 62                       |
| Outlet temperature $T_{\text{com-out}}$ | \degree C | 52              | 116                      |
| Mass flow rate of CO\textsubscript{2} $m_{\text{CO2-com}}$ | kg·s\textsuperscript{-1} | 3.46            | 2.3                      |
| Efficiency $\eta_{\text{com}}$ | \%       | 0.7             | 0.7                      |
| Power $W_{\text{com}}$ | kW | 64              | 84                       |

3.2.2 Turbine design. Design of a turbine was in general simpler than compressor design, since the pressure gradient had the same direction as the fluid flow. The most important parameters of this design were summarized in Table 3.
Table 3. Turbine design.

| Parameter                        | Unit    | Main turbine | Recompressing turbine |
|----------------------------------|---------|--------------|-----------------------|
| Inlet pressure $P_{tur-in}$      | MPa     | 13.499       | 13.499                |
| Outlet pressure $P_{tur-out}$    | MPa     | 7.885        | 7.885                 |
| Inlet temperature $T_{tur-in}$   | °C      | 537          | 537                   |
| Outlet temperature $T_{tur-out}$ | °C      | 477          | 477                   |
| Mass flow rate of CO₂ $m_{CO₂-tur}$ | kg·s⁻¹ | 2.62         | 3.15                  |
| Efficiency $\eta_{tur}$         | %       | 0.87         | 0.87                  |
| Shaft power $W_{tur-shaft}$      | kW      | 174          | 210                   |
| Generator power $W_{tur-generator}$ | kW     | 110          | 126                   |

3.3 Recuperator design
The optimized design of recuperators were presented in Table 4. Both types were PCHE designs with straight channels.

Table 4. Recuperator parameters.

| Parameter                        | Unit    | High temperature | Low temperature |
|----------------------------------|---------|------------------|-----------------|
| Hot side inlet pressure $P_{hot-in}$ | kPa     | 7885             | 7820            |
| Hot side inlet temperature $T_{hot-in}$ | °C      | 477              | 145             |
| Hot side outlet pressure $P_{hot-out}$ | kPa     | 7820             | 7755            |
| Hot side outlet temperature $T_{hot-out}$ | °C      | 145              | 62              |
| Cold side inlet pressure $P_{cold-in}$ | kPa     | 13727            | 13842           |
| Cold side inlet temperature $T_{cold-in}$ | °C      | 116              | 51              |
| Cold side outlet pressure $P_{cold-out}$ | kPa     | 13612            | 13727           |
| Cold side outlet temperature $T_{cold-out}$ | °C      | 425              | 116             |
| Hot side pressure drop $\Delta P_{hot}$ | kPa     | 65               | 65              |
| Cold side pressure drop $\Delta P_{cold}$ | kPa     | 115              | 115             |
| Total power $W_{recuperator}$     | kW      | 2235             | 645             |

3.4 Precooler design
The pre-cooler was made of titanium. The optimized precooler design was summarized in Table 5.

Table 5. Precooler parameters.

| Parameter                        | Unit    | Value |
|----------------------------------|---------|-------|
| Inlet pressure of CO₂ $P_{pre-in}$ | kPa     | 7755  |
| Inlet temperature of CO₂ $T_{pre-in}$ | °C      | 62    |
| Outlet pressure of CO₂ $P_{pre-out}$ | kPa     | 7690  |
| Outlet temperature of CO₂ $T_{pre-out}$ | °C      | 32    |
| Pressure drop of CO₂ $\Delta P_{CO₂-pre}$ | kPa     | 65    |
| Mass flow rate of CO₂ $m_{CO₂-pre}$ | kg·s⁻¹ | 3.47  |
| Outlet temperature of cooling water $T_{out-water}$ | °C      | 30.9  |

3.5 Recompression Brayton cycle design
The selected design operating conditions were summarized in Table 6. Table 7 summarized the selected design cycle state points.

Table 6. Selected recompression Brayton cycle design parameters.

| Recompression Brayton cycle | Unit    | Value |
|-----------------------------|---------|-------|
| Cycle Thermal Power $W_{total}$ | kW     | 780   |
| Thermal Efficiency $\eta_{thermal}$ | %     | 34.0  |
| Net Efficiency $\eta_{net}$ | %       | 30.3  |
| Net Electric Power $W_{net}$ | kW     | 236   |
| Main Compressor Outlet Pressure $P_{out}$ | MPa | 13.8  |
| Pressure Ratio $r_a$ |       | 1.8   |
| Primary System Pressure Drop $\Delta P$ | kPa | 113   |
Turbine Inlet Temperature $T_{\text{tur-in}}$ °C 537
Main Compressor Inlet Temperature $T_{\text{cin}}$ °C 32
Cooling Water Inlet Temperature $T_{\text{water-in}}$ °C 27
Mass Flow Rate of CO₂ $m_{\text{CO₂-total}}$ kg·s⁻¹ 5.77
Recompressed Fraction $r_{\text{frac}}$ / 0.40
Turbine Efficiency $\eta_{\text{tur}}$ % 87
Main Compressor Efficiency $\eta_{\text{cm}}$ % 70
Recompressing Compressor Efficiency $\eta_{\text{cr}}$ % 98
Generator Efficiency $\eta_{\text{generator}}$ % 1
Mechanical Losses % 1
Parasitic Losses % 2

| Point | Unit | 1   | 2   | 3   | 4   | 5   | 6   | 7   | 8   |
|-------|------|-----|-----|-----|-----|-----|-----|-----|-----|
| Pressure | kPa  | 769 | 13842 | 13727 | 13612 | 13499 | 7885 | 7820 | 7755 |
| Temperature | °C  | 32  | 51  | 116 | 425  | 537  | 477  | 145  | 62  |
| Enthalpy | kJ·kg⁻¹ | 304.6 | 322.95 | 501.31 | 888.74 | 1024.00 | 957.5 | 576.77 | 464.93 |
| Entropy | kJ·(kg·K)⁻¹ | 1.341 | 1.3686 | 1.8786 | 2.6223 | 2.8036 | 2.8211 | 2.1530 | 1.8539 |

4. Summary

In this paper, a supercritical carbon dioxide (S-CO₂) recompression Brayton cycle employing solar energy was designed and its calculation model and thermodynamics were analysed. Given the operating conditions, the optimization progress was performed and the optimized parameters were obtained. The optimized parameters were acquired as following: the inlet and outlet temperature of main compressor were 32°C and 52°C, respectively; the inlet and outlet pressure of main compressor were 7.69MPa and 13.8MPa, respectively; the inlet and outlet temperature of recompression compressor were 62°C and 116°C, respectively; the inlet and outlet pressure of main compressor were 7.755MPa and 13.727MPa, respectively; the inlet and outlet temperature of turbine were 537°C and 477°C, respectively; the total carbon dioxide mass flow rate was 5.77 kg·s⁻¹ and the recompressed fraction was 0.4. Thus, the generation power of the system was 236kW, and the net efficiency was 30.3%.

References

[1] G.H.Qu, Evolvement of new energy policies in developed world and enlightenment to China, SINO-GLOBAL ENERGY.1(2010)29-33
[2] Z.M.Jiang, Reflection on energy issues in China, JOURNAL OF SHANGHAI JIAOTONGUNIVERSITY.42(2008)345-359
[3] Z.Wang, Y.Ren, Utilization and development of solar energy industry in China, RESOURCES AND INDUSTRIES.12(2010)89-92
[4] Hejzlar P, Dostal V, Driscoll M J, et al. Assessment of gas cooled fast reactor with indirect supercritical CO₂ cycle[J]. Nuclear Engineering and Technology, 2006,38(2):109-118.
[5] David C Wade. Optimizing economy of scale for the STAR energy supply architecture[C]. Miami: IAEA 2th CRP Meeting on Small Reactors without Onsite Refueling, 2007