IOP Conference Series: Earth and Environmental Science

Study on Supercritical Carbon Dioxide Recompression Cycle System Employing Solar Energy

To cite this article: Chen Huiyong *et al* 2019 *IOP Conf. Ser.: Earth Environ. Sci.* **267** 022009

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.

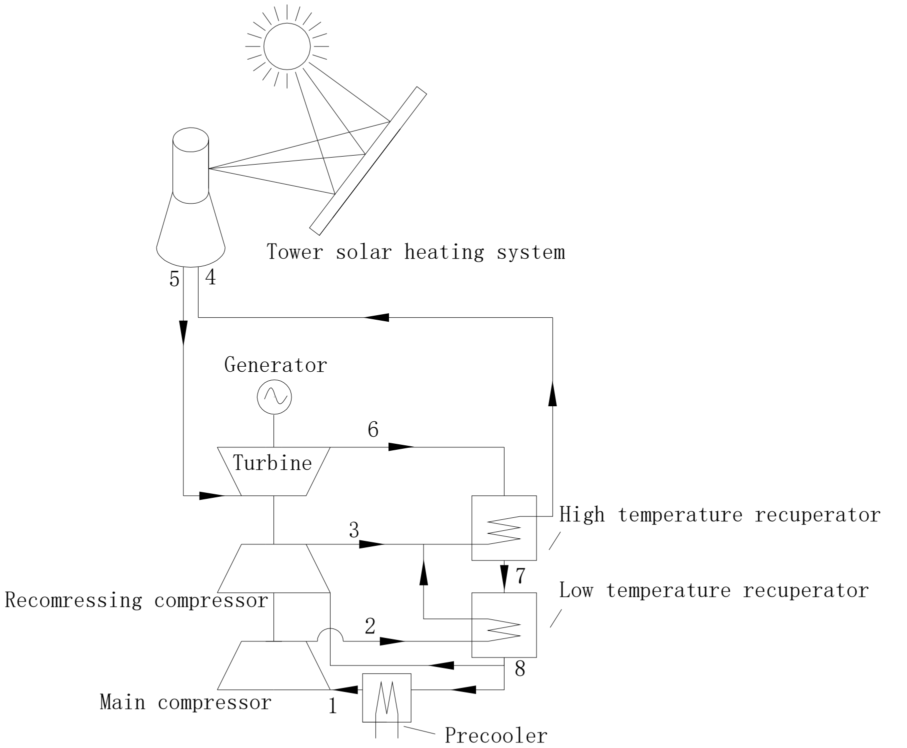

Figure 1. Schematic diagram of the S-CO_2 recompression cycle system employing solar energy.

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 Figure2.

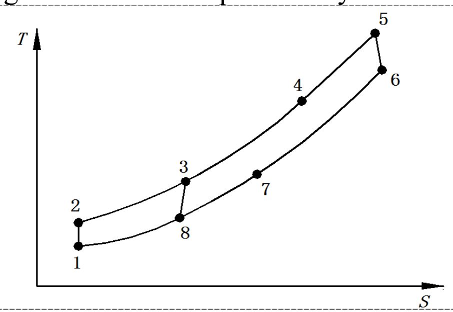

Figure 2. Temperature-entropy diagram of a recompression Brayton cycle.

3. Thermal analysis and thermal parameters selection of S-CO_2 recompression cycle system employing solar energy

The operating conditions selected for this analysis were 537℃ for the turbine inlet temperature, 32℃ 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℃ 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.

3.1Solar 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.

Table 1. Tower solar heating system parameters.

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	℃	292
Outlet temperature of medium	℃	565

3.2Turbomachinery 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.

3.2.1Compressor 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.

Table 2. Compressors parameters.

Compressor	Unit	Main Compressor	Recompressing Compressor
Inlet pressure $P_{com\text{-}in}$	MPa	7.69	7.755
Outlet pressure $P_{com\text{-}out}$	MPa	13.842	13.727
Pressure ratio r_{frac}		1.8	1.77
Inlet temperature $T_{com\text{-}in}$	℃	32	62
Outlet temperature $T_{com\text{-}out}$	℃	52	116
Mass flow rate of CO_2 $m_{CO2\text{-}com}$	$kg \cdot s^{-1}$	3.46	2.3
Efficiency η_{com}	%	0.7	0.7
Power W_{com}	kW	64	84

3.2.2Turbine 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.

Turbine	Unit	Main turbine	Recompressing turbine
Inlet pressure $P_{tur\text{-}in}$	MPa	13.499	13.499
Outlet pressure $P_{tur\text{-}out}$	MPa	7.885	7.885
Inlet temperature $T_{tur\text{-}in}$	℃	537	537
Outlet temperature $T_{tur\text{-}out}$	℃	477	477
Mass flow rate of CO_2 $m_{CO2\text{-}tur}$	$kg \cdot s^{-1}$	2.62	3.15
Efficiency η_{tur}	%	0.87	0.87
Shaft power $W_{tur\text{-}shaft}$	kW	174	210
Generator power $W_{tur\text{-}generator}$	kW	110	126

3.3Recuperator design

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

Table 4. Recuperator parameters.

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

3.4Precooler design

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

Table 5. Precooler parameters.

Precooler	Unit	value
Inlet pressure of CO_2 $P_{pre\text{-}in}$	kPa	7755
Inlet temperature of CO_2 $T_{pre\text{-}in}$	℃	62
Outlet pressure of CO_2 $P_{pre\text{-}out}$	kPa	7690
Outlet temperature of CO_2 $T_{pre\text{-}out}$	℃	32
Pressure drop of CO_2 $\triangle P_{CO2\text{-}pre}$	kPa	65
Mass flow rate of CO_2 $m_{CO2\text{-}pre}$	$kg \cdot s^{-1}$	3.47
Outlet temperature of cooling water $T_{out\text{-}water}$	℃	30.9

3.5Recompression Brayton cycle design

The selected design operating conditions were summarized in Table6. Table7 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 η_{net}	%	30.3
Net Electric Power W_{net}	kW	236
Main Compressor Outlet Pressure P_{cout}	MPa	13.8
Pressure Ratio r_a	/	1.8
Primary System Pressure Drop $\triangle P$	kPa	113

Turbine Inlet Temperature $T_{tur\text{-}in}$	℃	537
Main Compressor Inlet Temperature T_{cin}	℃	32
Cooling Water Inlet Temperature $T_{water\text{-}in}$	℃	27
Mass Flow Rate of CO_2 $m_{CO2\text{-}total}$	$kg \cdot s^{-1}$	5.77
Recompressed Fraction r_{frac}	/	0.40
Turbine Efficiency η_{tur}	%	87
Main Compressor Efficiency η_{cm}	%	70
Recompressing. Compressor Efficiency η_{cr}	%	70
Generator Efficiency $\eta_{generator}$	%	98
Mechanical Losses	%	1
Parasitic Losses	%	2

Table 7. Selected design cycle state points.

Point	Unit	1	2	3	4	5	6	7	8
Pressure	kPa	769	13842	13727	13612	13499	7885	7820	7755
Temperature	℃	32	51	116	425	537	477	145	62
Enthalpy	$kJ \cdot kg^{-1}$	304.6	322.95	501.31	888.74	1024.00	957.5	576.77	464.93
Entropy	$kJ \cdot (kg \cdot K)^{-1}$	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-CO_2) 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℃ and 52℃, 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℃ and 116℃, 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℃ and 477℃, respectively; the total carbon dioxide mass flow rate was 5.77 $kg \cdot s^{-1}$ 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