COMPARATIVE ANALYSES OF TWO IMPROVED CO_2 COMBINED COOLING, HEATING, AND POWER SYSTEMS DRIVEN BY SOLAR ENERGY

by

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To make use of solar energy fully and efficiently, the two improved combined cooling, heating, and power systems (CCHP) are proposed by adding a gas heater and an extraction turbine, based on the transcritical CO_2 ejector refrigeration system. A relatively high pressure fluid is extracted by the extraction turbine as a primary stream of ejector to improve the ejector performance. In the meantime, the gas heater absorbs low temperature exhaust heat to increase the extraction turbine's output work. Comparative studies on the thermal efficiency and exergy efficiency of the two improved systems show they are more efficient alternatives for the transcritical CO_2 ejector refrigeration system. The CCHP-B system has relatively broad working condition, higher thermal efficiency and exergy efficiency than that of CCHP-A system.

Key words: CCHP, CO_2, supercritical, exergy efficiency, thermal efficiency

Introduction

In order to solve environmental and energy problems, as a kind of clean renewable energy, solar energy has a rapid progress in research and development. The combined cooling, heating, and power (CCHP) system driven by solar energy with higher system efficiency, more eco-friendly environment, and better economic feasibility, offers an effective way to realize sustainable development [1, 2]. More and more literature studies can be found using the natural working fluid CO_2 in the Rankine cycle or the ejector refrigeration cycle [3]. The lower critical temperature (31.1 °C) of CO_2 results in the transcritical or supercritical Rankine cycle and the transcritical ejector refrigeration cycle or in supercritical cogeneration system.

Kim *et al*. [4] presents the optimization processes for supercritical CO_2 Rankine cycles. Nami *et al.* [5] proposes and analyzes a novel co-generation system, including a gas turbine, a heat recovery steam generator and a supercritical CO_2 cycle (GT-HRSG/SCO_2). Padilla *et al.* [6] perform a comprehensive thermodynamic analysis and a multi-objective optimization to study the proposed supercritical CO_2 Brayton cycles integrated with an ejector. The transcritical CO_2 ejector system has been widely used in air conditioning in recent years. But, the major difference between the transcritical CO_2 cycle and conventional refrigerant cycle is that up to 80 °C ～150 °C is reached at the outlet temperature of CO_2 compressor with its specific thermal

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properties near the pseudo-critical point. This system ignores the fact that the exit heat from the compressor is cooled by the gas cooler, which means the heat would go to waste. Using low-grade heat sources to further raise the heat temperature of the compressor outlet is an attractive alternative solution, and the heat grade is enhanced for producing heating output, cooling output and power output simultaneously. Xu *et al*. [7] proposed two improved CCHP systems by adding a gas heater and an extraction turbine. The extraction turbine is to extract a high pressure stream as a primary fluid of ejector which is benefit to obtain large refrigeration output, and continues to expand the remaining fluid of the extraction turbine to a lower pressure for more power output. In addition, the supercritical CO_2 stream from the compressor outlet is further heated by the gas heater, which upgrades the heat quality.

Our improved CCHP systems using supercritical CO_2 have owned the authorized patent [8]. Based the previous study on parametric analysis and exergy analysis for one of the improved CCHP systems [7], this paper attempts to compare the two improved CCHP using supercritical CO_2 to investigate the influence of the thermodynamic parameters on the performance and exergy destruction of the CCHP systems.

Cycle description

Figures 1 and 2 show the schematic diagrams of the improved CCHP cycles, figs. 1(a) and 2(a), and the processes of the corresponding cycles in a temperature-specific entropy diagrams, figs. 1(b) and 2(b). As can be seen in figs. 1(a) and 2(a), the major difference between the two CCHP cycles is that the extracted fluid from the extracted turbine with the fluid exiting from the evaporator are fed to ejector in CCHP-A cycle, fig. 1, whereas the fluid exiting from the extracted turbine with the saturated vapor from the gas-liquid separator are fed to compressor directly in CCHP-B cycle, fig. 2.

Thermodynamics analysis

Ejector model and system model

The mathematical models for ejector and overall system are described and presented in authors' former paper [2]. The principle of the ejector model is introduced by Keenan *et al.* [9]. Three independent efficiencies are assumed as 0.9, 0.85, and 0.85 for the motive nozzle, mixing section, and diffuser, respectively.

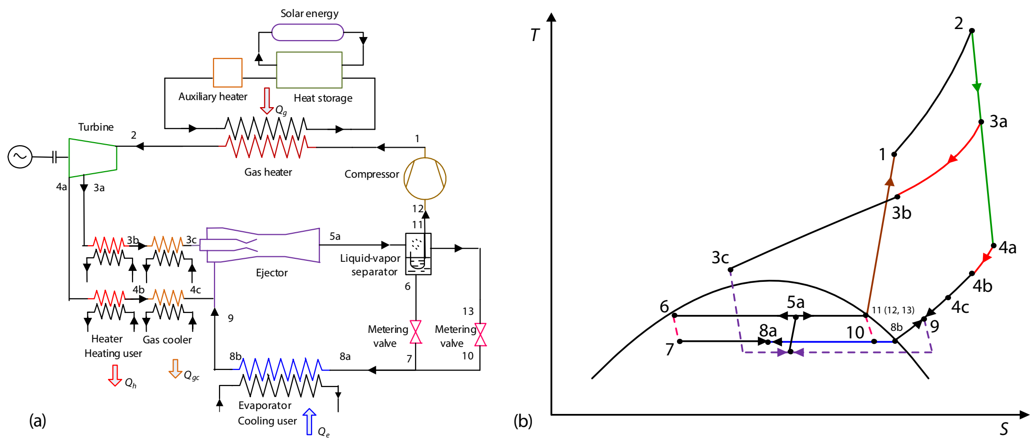

Figure 1. The CCHP-A cycle; (a) schematic diagram, (b) temperature-entropy chart

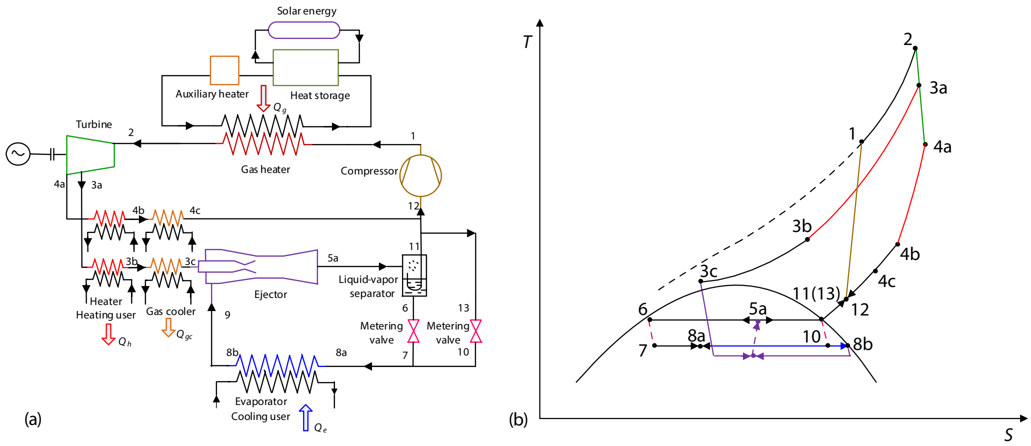

Figure 2. The CCHP-B cycle, (a) schematic diagram, (b) temperature-entropy chart

System stability conditions

Li and Groll [10] points out that the mass conservation constraint $x = 1/(1 + \mu)$ is used to control the stability of the transcritical CO_2 ejector refrigeration system. In this work, we introduce the gas heater and the extraction turbine into the Li's modified cycle. The equation $x > 1/(1 + \mu)$ is equivalent in function to $m_{13} > 0$, that is, the mass-flow rate of the bypass is larger than zero.

CCHP-A cycle

$$m_{13} = \left[e(1+\ \ )x_5\ -1\right]m_1 \tag{1}$$

The stable operation condition is written according to Li and Groll [10] ($m_{13} > 0$):

$$\left[e(1+\mu)x_{5a} - 1\right]m_1 > 0 \tag{2}$$

The ejector outlet quality must satisfy eq. (3) in order to maintain the system stability and realize the cycle:

$$x_{5a} > \frac{1}{e(1+\mu)} \tag{3}$$

Since the ejector outlet quality is related to the extraction ratio, it demonstrates that the extraction ratio of CCHP-A cycle can not be too low otherwise it would be unable to satisfy the eq. (3).

CCHP-B cycle

$$m_{13} = em_1\left(-1 + x_{5a} + \mu x_{5a}\right) \tag{4}$$

The stable operation condition is written according to Li and Groll .[10] ($m_{13} > 0$):

$$-1 + x_{5a} + \mu x_{5a} > 0 \tag{5}$$

The ejector outlet quality must satisfy eq. (6) in order to maintain the system stability and realize the cycle:

$$x_{5a} > \frac{1}{1+\mu} \tag{6}$$

Since the ejector outlet quality is not related to the extraction ratio, the CCHP-B cycle has relatively broad range and application compared to CCHP-A cycle.

System performance

Considering the improved CCHP cycles combined with the vapor compression cycle and the power cycle, is defined separately by the refrigeration *COP* and the energy utilization efficiency, η_{EUE}, are used to characterize the thermal efficiency. The ratio of *COP*, ξ, between the improved CCHP cycle and the ejector cycle is defined to indicate the system performance improvement:

$$\eta_{\mathrm{EUE}} = \frac{Q_{\mathrm{h}} + W_{\mathrm{t}} - W_{\mathrm{c}}}{Q_{\mathrm{g}}} \tag{7}$$

$$\xi = \frac{COP}{COP_{\mathrm{EJE}}} \tag{8}$$

The exergy efficiency, η_{EXG}, is a criterion for the system performance:

$$\eta_{\mathrm{EXG}} = \frac{E_{\mathrm{Q,e}} + E_{\mathrm{Q,h}} + W_{\mathrm{t}} - W_{\mathrm{c}}}{E_{\mathrm{Q,g}}} \tag{9}$$

$$E_{\mathrm{Q,i}} = Q_{\mathrm{i}}\left(1 - \frac{T_0}{T_{\mathrm{Q,i}}}\right) \tag{10}$$

$$E_{\mathrm{i}} = m_{\mathrm{i}}\left[\left(h_{\mathrm{i}} - h_0\right) - T_0\left(s_{\mathrm{i}} - s_0\right)\right] \tag{11}$$

$$I_{\mathrm{i}} = \sum_{\mathrm{i}} Q_{\mathrm{i}}\left(1 - \frac{T_0}{T_{\mathrm{Q,i}}}\right) + \sum_{\mathrm{i}} E_{\mathrm{in,i}} - \sum_{\mathrm{i}} E_{\mathrm{out,i}} - W_{\mathrm{i}} \tag{12}$$

Results and discussion

Comparison between the two improved CCHP cycles

The working conditions of the two CCHP cycles shown in tab.1 are reasonably assumed to maintain the cycles run stably.

Table 1. The simulation conditions of the improved CCHP cycles

Parameters	Amount (Type-A cycle)	Amount (Type-B cycle)
Mass-flow rate of working fluid	1.4 kg/s	1.4 kg/s
Environment temperature	15 °C	15 °C
Environment pressure	0.1013 MPa	0.1013 MPa
Turbine inlet temperature	220 °C	220 °C
Turbine inlet pressure	12 MPa	12 MPa
Turbine extraction pressure	8.2 MPa	8.4 MPa
Turbine extraction rate	0.8	0.55
Ejector inlet temperature	36 °C	36 °C
Ejector back pressure	4.4 MPa	4.6MPa
Heater outlet temperature	70 °C	70 °C
Evaporation temperature	5 °C	5 °C
Turbine isentropic efficiency	0.85	0.85
Compressor isentropic efficiency	0.8	0.8
Approach temperature difference of heat exchanger	10 °C	10 °C
Mass-flow rate of flue	5 kg/s	5 kg/s

Table 2 shows the performance of the improved CCHP cycles. As the turbine extraction ratio of CCHP-A cycle is higher than that of CCHP-B cycle, more high temperature refrigerant extracting from the turbine can supply more heating output in the CCHP-A cycle, whereas more refrigerant working in turbine results in more turbine output in the CCHP-B cycle. Although the difference of the refrigerant output in two cycles is very little, the *COP* and the exergy efficiency of CCHP-B cycle are higher than that of the CCHP -A cycle.

Table 2. The performance of the improved CCHP cycles

Parameters	Amount (CCHP-A cycle)	Amount (CCHP-B cycle)
Turbine power	49.80 kW	58.88 kW
Refrigerating output	71.44 kW	74.51 kW
Heating output	184.50 kW	166kW
Heat absorption from heat source	249.00 kW	218.2 kW
Compressor power	66.71 kW	71.82 kW
COP	4.225	5.755
Energy utilization efficiency	0.673	0.7013
Exergy efficiency	0.3449	0.3706

Figure 3 shows the influence of extraction rate on the both system performance. It notes that the ejector outlet quality of CCHP-A cycle and CCHP-B cycle should satisfy the condition listed in eq. (3) and eq. (6), respectively. The ejector outlet quality of CCHP-A cycle cannot be less than 0.68. As the extraction rate increases, the refrigeration output increase due to extracting more supercritical fluid CO_2 from turbine as the driving force of the ejector which entrained more stream from the evaporator. But, more supercritical fluid CO_2 extracted decrease the turbine power output, which far outpaces the growth rate of the refrigeration output. Thus, the *COP* decreases with the increasing of extraction rate. Compared to the CCHP-A cycle, the CCHP-B cycle is more suitability for the high demand of the refrigeration output.

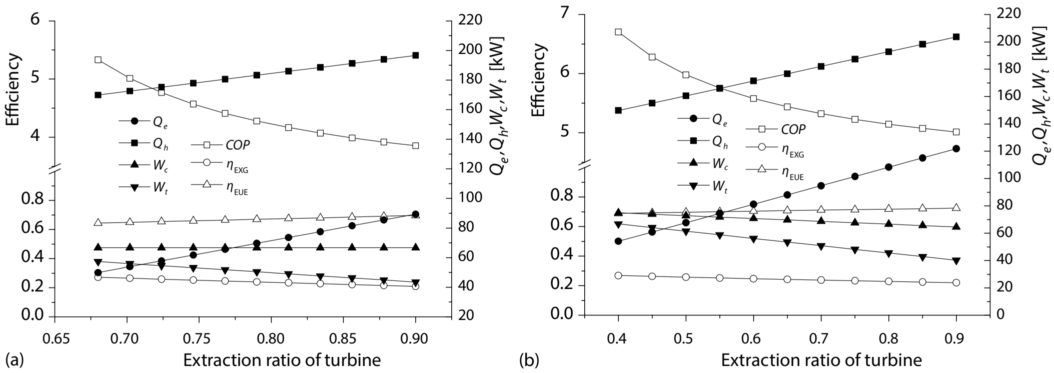

Figure 3. Effect of extraction rate on the system performance; (a) CCHP-A cycle (b) CCHP-B cycle

Based on the exergy destruction of components in the systems as shown in tab. 3, the irreversibility of heat transfer is the major source of exergy destruction in both systems. The exergy destruction ratio of three heat exchangers in CCHP-B system is 67.66% and that of CCHP-A system is 69.88%. The results reveal that the system exergy destruction could be reduced by decrease the heat transfer temperature difference in three heat exchangers. The most significant difference of the two systems lies in the exergy destruction of ejector outperformed by 4.6% than those of turbine in CCHP-A system, whereas, lowered by 1.51% in CCHP-B system. Ma *et al*. [11] points out that the exergy destruction of the turbine is lower than that of the ejector under general operating conditions. But, the result presents in CCHP-B system shows just the reverse. This could be explained by more working fluid CO_2 expanded in the turbine further increases the net power output, which also leads to the increase of the exergy efficiency.

Table 3. The exergy destruction of the two improved CCHP systems

Components	CCHP-A cycle exergy destruction, [kJ]	CCHP-A cycle exergy destruction ratio, [%]	CCHP-B cycle exergy destruction, [kJ]	CCHP-B cycle exergy destruction, [%]
Compressor	10.62	10.37	11.08	13.08
Ejector	10.62	10.37	5.58	6.59
Evaporator	1.307	1.27	1.36	1.61
Gas cooler	12.58	12.28	11.55	13.65
Gas heater	22.06	21.53	17.28	20.41
Heater	34.68	33.85	30.33	35.82
Turbine	5.893	5.75	7.09	8.37
Throttle	4.69	4.59	0.40	0.47
Total	102.45	100	84.67	100

Comparisons between the improved CCHP systems and the transcritical CO_2 ejector system

The previous two improved CCHP systems in this paper are based on the transcritical CO_2 ejector system proposed Li and Groll [10]. It is necessary to compare the improved CCHP systems with the transcritical CO_2 ejector system so as to estimate whether the expander will improve the system performance. The calculation is based on the same conditions including the ambient temperature 15 °C, the expander isentropic efficiency 0.85, the compressor isentropic efficiency 0.8, and the evaporating temperature 5 °C and the gas cooler outlet temperature 36 °C.

The *COP* of the combined cycle is:

$$COP_{A/B} = \frac{Q_\mathrm{e}}{W_\mathrm{c} - W_\mathrm{t}} \tag{13}$$

The *COP* of the conventional injection refrigeration cycle is:

$$COP_\mathrm{conventional} = \frac{Q_\mathrm{e}}{W_\mathrm{c}} \tag{14}$$

The ratio of CCHP-A and CCHP-B cycle to conventional transcritical ejector refrigeration cycle is expressed, respectively:

$$\mathrm{Ratio}_A = COP_A / COP_\mathrm{conventional} \tag{15}$$

$$\mathrm{Ratio}_B = COP_B / COP_\mathrm{conventional} \tag{16}$$

Figure 4 shows that there is optimum high-side pressure and maximum *COP* in the transcritical CO_2 ejector system. The *COP* maximizing high-side pressures of the improved CCHP systems are higher than that of the transcritical CO_2 ejector system. This is because the network equals the difference of compressor work and turbine power output, not equals compressor work. For CCHP-A cycle, the optimum high-side pressure is 14 MPa, for CCHP-B cycle is 12 MPa. Under a given condition in this paper, COP_B is the largest among three cycles.

It also can be seen that the *COP* of the improved CCHP systems are higher than that of the transcritical CO_2 ejector system with increase high-side pressure. The *COP* difference between the improved CCHP system and transcritical CO_2 ejector system becomes more and more large as the high-side pressure increases. The Ratio A increases from 0.713-2.016 and Ratio B from 1.103-2.433. To be mentioned that, the improved CCHP system has evident advantages over the transcritical CO_2 ejector system when they run under the conditions of the high-side pressure.

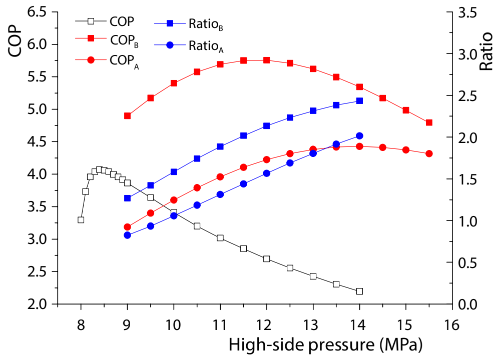

Figure 4. Comparison chart of the *COP*

Conclusions

- The two improved CCHP system using supercritical CO_2 are proposed. The CCHP-B system has higher thermal efficiency and exergy efficiency than that of CCHP-A system.
- The stability conditions of the two improved CCHP cycle are obtained. The CCHP-B system has relatively broad working condition than that of CCHP-A system.
- The improved CCHP systems exist optimal high-side pressures that give maximum *COP*, and the *COP* maximizing high-side pressure of the improved CCHP systems are higher than that of the transcritical CO_2 ejector system.
- Both extraction pressure and higher extraction rate are helpful to gain more refrigeration in the improved CCHP systems.

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Nomenclature

E – exergy, [kW]
e – extraction ratio, [–]
h – specific enthalpy, [kJkg^{-1}]
m – mass-flow rate, [kgs^{-1}]
p – pressure, [MPa]
Q – heat load, [kW]
s – specific entropy, [kJkg^{-1}K^{-1}]
T – temperature, [°C]
W – power, [kW]

Greek symbols

η – efficiency
μ – entrainment ratio
ξ – ratio

Subscripts

c – compressor
e – evaporator
g – gas heater
h – heater
Q – heating source
t – turbine
u – velocity, [ms^{-1}]
x – vapor quality, [–]

Abbreviations

COP – coefficient of performance, [–]
EJE – ejector cycle
EUE – energy utilization

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