Thermodynamic Analysis of CO_2 Transcritical Refrigeration Cycle with Expansion-Compressor

Qingping Wang, Paiwei Zhang, Li Chang, and Zhenying Zhang*

College of Civil and Architectural Engineering, North China University of Science and Technology, Tangshan 063210, China

*zzying30@126.com

Abstract

The parallel compression refrigeration cycle with expansion-compressor (P-C) and the two-stage refrigeration cycle with expansion-compressor as the main compressor (S-C) are analyzed thermodynamically and compared with the basic refrigeration cycle (BC) and the refrigeration cycle with expander (EC). The results show that the COP increases at initially before gradually decreasing with increasing the gas cooler pressure, and the COP of each system decreases with increasing gas cooler outlet temperature and decreasing the evaporation temperature. At the same evaporation temperature, the COP of S-C is 51.8%~104.9% higher than that of BC, 39.1%~81.5% higher than that of EC, and 11.9%~54.2% higher than that of P-C.

Keywords

Expansion-compressor; CO_2; Refrigeration Cycle.

1. Introduction

With the destruction of the ozone layer, the frequent occurrence of extreme weather and rising global temperatures, the refrigerants containing chlorofluorocarbons (CFC) are subject to many restrictions [1]. As a natural refrigerant CO_2 has attracted more attention due to its unique advantages of non-toxic and non-flammable [2]. The critical temperature of CO_2 is 31.1 °C, and its critical pressure is 7.38 MPa. Therefore, the CO_2 refrigeration cycle is mainly completed in the transcritical region. This makes that the operating pressure in the expansion process is high, the throttling loss is significant, resulting in COP lower than the traditional compression refrigeration cycle system under the same conditions [3]. However, due to the large throttling loss, the energy efficiency of the CO_2 transcritical refrigeration cycle is low [4]. Therefore, substituting the throttle valve for an expander has become a research hotspot.

Lorentzen [5]. first used expander instead of throttle valve in the late 20th century, and recovered the expansion work in the throttling process of CO_2 transcritical refrigeration cycle system. Xuan and Xie [6]. conducted a thermodynamic analysis of the CO_2 transcritical refrigeration system and showed that the efficiency of the refrigeration cycle using an expander is better. Wang et al. [2]. studied the CO_2 two-stage compression transcritical cycle with an expander. The results show that the COP of the refrigeration cycle with an expander is more over than 57.4% that of the basic transcritical cycle with a throttle valve. Tian et al. [7]. found that the expanders can generally increase the COP of the system by 6%~10%. Sun et al. [8]. proposed an innovative CO_2 transcritical refrigeration cycle that being the equal of the CO_2 basic refrigeration cycle and the CO_2 refrigeration cycle with expander, the COP of this innovative cycle increase by 20.17% and 10.59%, respectively.

Zhang et al. [9]. found that replacing the throttle valve with a CO_2 expander in the refrigeration cycle can effectively improve the refrigeration efficiency. In the CO_2 transcritical refrigeration cycle with

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expander, the work generated by the refrigerant through the expansion process can be recycled, and the kinetic energy of the refrigerant can be converted into mechanical work, which can be used for driving other equipment. Usually, the expander is connected to the generator, and the work of the expander can directly drive the generator to produce electricity, thereby improving the energy efficiency of the cycle. This energy recovery method not only reduce the waste of energy, but also reduce the operating cost of the entire refrigeration system. Along with recycling the power generated by the expansion process to drive the generator, this part of the power can also be used for other equipment that needs to drive power, such as driving compressors or pumps. Through reasonable design and installation, the efficient use of energy can be achieved, thereby improving the energy efficiency of the total refrigeration cycle. The expander also has stable working characteristics, which can smoothly expand the refrigerant from the high pressure area to the low pressure area, so that the pressure fluctuation of the system is small and the stability of the system is advanced. However, the expansion work is used for driving the compressor, that is, the expansion-compressor is the most ideal way to recycle the expansion work. Expansion-compressor can be linked in parallel or in series with the main compressor. In order to improve the COP of CO_2 transcritical two-stage compression cycle. The numerical simulation consequences show that the cycle efficiency is advanced in 23.5% after the application of the unit. COP increases with the decrease of the inlet temperature of the expander, but decreases with the increase of the suction pressure. The performance of the expander is comparatively insensitive to the change of the functioning pressure ratio, while the performance of the scroll compressor decreases rapidly as the operating pressure ratio is far away from the purpose pressure ratio. Yap et al. [10]. introduced a new type of vane expansion-compressor. CO_2 is used as the refrigerant, and the thermodynamic efficiency is as high as 95.9%. Compared with the basic vapor compression system, COP is increased by 36.6%. Erdinc [11]. simulated the performance of the expansion-compressor pressurized subcooled refrigeration system by numerical simulation. Six different refrigerants were used to simulate the evaporation temperature between-10°C and 5°C. It was showed that the optimal COP of R1234yf increased by 16.78%, 19.18% and 21.8%, respectively, dependent on the mechanical efficiency of the expander and the compressor and the isentropic efficiency of the expander. Erdinc [3]. introduced an expansion-compressor supercharger to replace the expansion device in the dual-evaporator refrigeration system to improve the COP of the dual-evaporator refrigeration system. The results show that this method could boost the COP of the system by 38%.

It can be seen from the above research that most of the research is on the expander system, and there is little research on the expansion-compressor. In this paper, the numerical simulation of CO_2 transcritical refrigeration cycle with expansion-compressor is carried out, and the COP of BC, EC, P-C and S-C is compared. The effects of evaporation temperature, gas cooler outlet temperature and gas cooler pressure on COP are analyzed.

2. Analysis of the System

2.1 Analysis of the Cycle

Figure 1 shows the principle and P-h diagram of CO_2 transcritical BC and EC. BC consists of compressor (C), gas cooler (g-c), throttle valve (TV) and evaporator (EV). The only difference between EC and BC in composition is that the throttle valve is replaced by an expander. In addition to generating more cooling capacity than the throttle valve, the expander could also recover the expansion work for the generator or compressor. The refrigerant 1 is compressed into a high temperature and high pressure fluid 2 by the compressor, and then becomes a low temperature and high pressure fluid 3 through the gas cooler, and the flow through the throttle valve can be regarded as an isenthalpic throttling process (the flow through the expander is regarded as a throttling process between isentropic and isenthalpic), and becomes a low temperature and low pressure fluid 4h (or 4), and then comes in the evaporator to soak up heat and becomes a high temperature and low pressure fluid 1, which is repeated.

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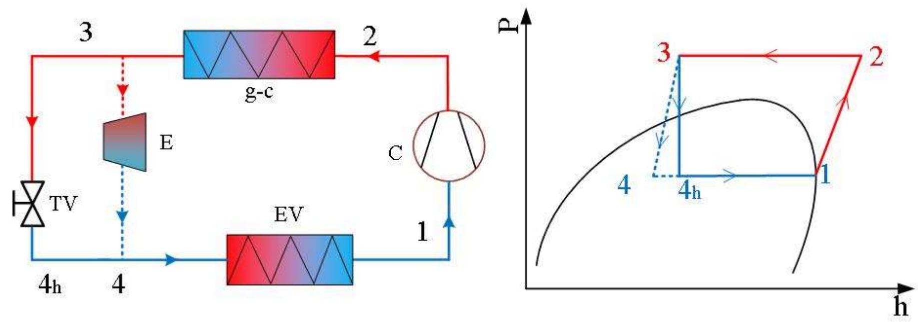

Figure 1. The schematic & P-h diagrams of BC and EC

Figure 2 shows the principle of P-C and P-h diagram. P-C is composed of main compressor (MC), gas cooler, expansion-compressor (E-C), gas-liquid separator (g-s), evaporator and throttle valve. The expansion-compressor is a compressor that uses the mechanical power output by the expander to drive the coaxial rotation to complete the compression of the refrigerant. The cycle process of P-C is divided into two parts. The expansion-compressor of P-C is connected in parallel with the main compressor to compress the refrigerant. The high-pressure refrigerant 3 flowing out of the gas cooler is expanded into an intermediate pressure fluid 4 with gas-liquid coexistence through the expander. A part of the saturated steam in the gas-liquid separator is directly expanded and compressed into a high-pressure fluid 8 by the compressor. The other part of the fluid 5 is throttled into fluid 6 by the throttle valve, and then absorbed by the evaporator to become a high-temperature and low-pressure fluid 1, which is then compressed into a high-pressure fluid 2 by the main compressor. Fluid 2 enters the gas cooler together with fluid 8 and flows out as refrigerant 3.

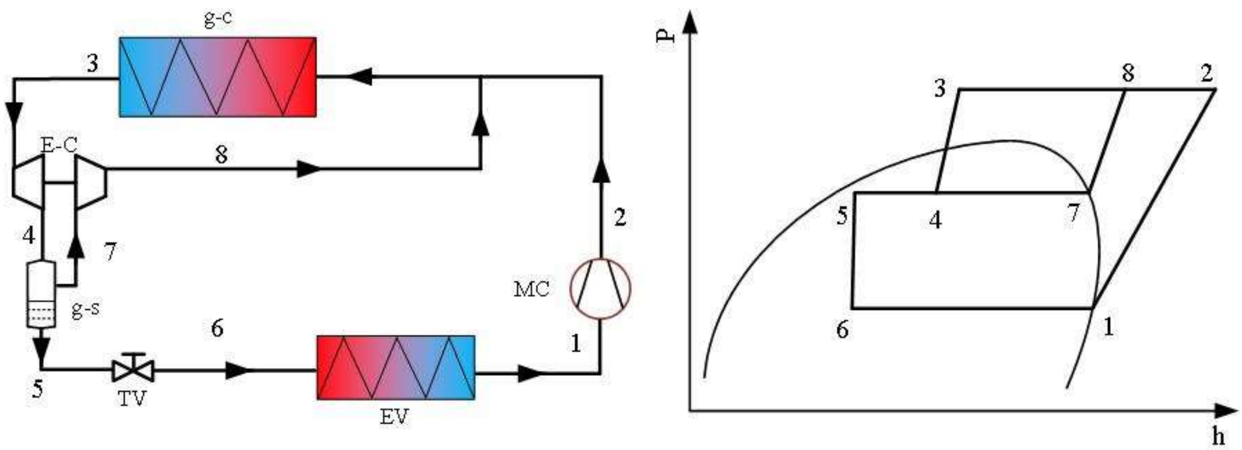

Figure 2. The schematic & P-h diagrams of P-C

Figure 3 shows the principle of S-C and P-h diagram. S-C is composed of main compressor, intermediate cooler, auxiliary compressor (AC), gas cooler, gas-liquid separator, throttle valve, motor (M) and evaporator. The expansion work recovered by the expander is utilized for the main compressor, and the insufficient power is then driven by the motor. The main compressor and the auxiliary compressor compress the refrigerant in series. The low-pressure refrigerant 1 is impacted into intermediate pressure fluid 2 by the main compressor, and then cooled to fluid 3 by the intermediate cooler, and then compressed into high-pressure fluid 4 by the auxiliary compressor. The high-temperature and high-pressure fluid 4 is cooled into low-temperature and high-pressure fluid 5 by the gas cooler, and then expanded into intermediate pressure fluid 6 by the expander. After the gas-liquid separator becomes liquid fluid 7, it is throttled into fluid 8, and then moves in the evaporator to absorb heat into high-temperature and low-pressure refrigerant 1. The steam 3 in the

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gas-liquid separator is directly sucked into the auxiliary compressor, compressed into fluid 4, cooled by the gas cooler into fluid 5, expanded by the expander into 6, and divided into fluid 3 by the gas-liquid separator.

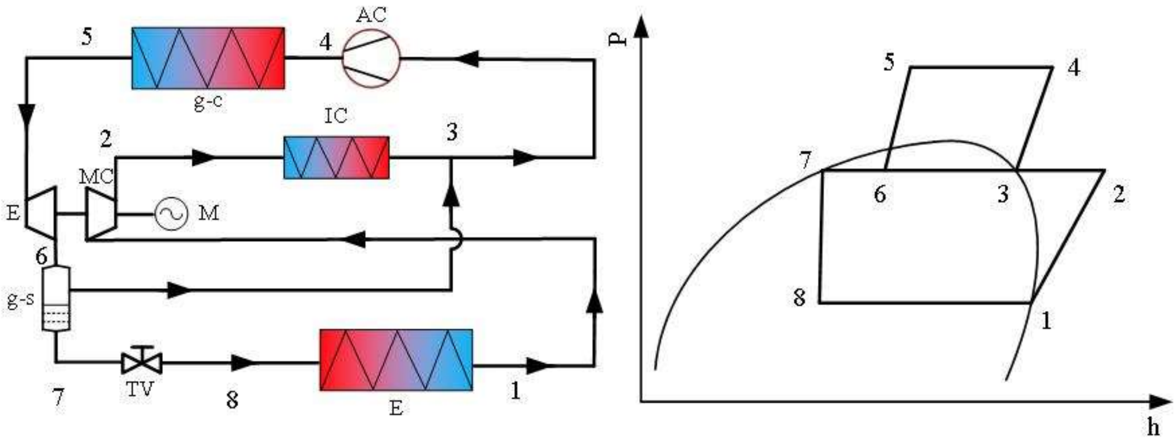

Figure 3. The schematic & P-h diagrams of S-C

2.2 Analysis of Thermodynamic System

In order to facilitate the analysis and calculation, the following suppositions are made:

1) All systems are in a stable state;

2) The pressure loss in the pipeline, gas cooler and evaporator of the system is neglected;

3) There is no heat leakage loss in the system;

4) The mechanical conversion efficiency of the expansion-compressor is 1, that is, the expansion work is completely converted into the compression work;

5) Without special instructions, the pressure of gas cooler is 9.0 MPa, the outlet temperature is 37 °C, the evaporation temperature is 0 °C, the value range of evaporation temperature is-15~5 °C.

The isentropic efficiency of compressor is:

$$\eta_{\mathrm{com}} = (h_{\mathrm{com,o,s}} - h_{\mathrm{com,i}}) / (h_{\mathrm{com,o}} - h_{\mathrm{com,i}}) \tag{1}$$

The isentropic efficiency of the expander is:

$$\eta_{\mathrm{exp}} = (h_{\mathrm{exp,i}} - h_{\mathrm{exp,o}}) / (h_{\mathrm{exp,i}} - h_{\mathrm{exp,o,s}}) \tag{2}$$

The theoretical calculation of the coefficient of performance and power of BC and EC is as follows:

$$w = h_2 - h_1 \tag{3}$$

$$q_{\mathrm{BC}} = h_1 - h_{4\mathrm{h}} \tag{4}$$

$$q_{\mathrm{EC}} = h_1 - h_4 \tag{5}$$

The theoretical calculation of the coefficient of performance and power of P-C is as follows:

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$$w = w_{\mathrm{mc}} + w_{\mathrm{exc}} - w_{\mathrm{exp}} = (1-i)(h_2 - h_1) + i(h_8 - h_7) - (h_3 - h_4) \tag{6}$$

$$q_{\text{P-C}} = (1-i)(h_1 - h_6) \tag{7}$$

The theoretical calculation of the coefficient of performance and power of S-C is as follows:

$$w = w_{\mathrm{mc}} + w_{\mathrm{ac}} - w_{\mathrm{exp}} = (1-i)(h_2 - h_1) + (h_4 - h_3) - (h_5 - h_6) \tag{8}$$

$$q_{\text{S-C}} = (1-i)(h_1 - h_8) \tag{9}$$

The refrigeration coefficient of the cycle is:

$$COP = q / w \tag{10}$$

In the formula: hcom,o,s--isentropic enthalpy of compressor outlet, kJ/kg; hcom,i--enthalpy of compressor intlet, kJ/kg; hcom,o--enthalpy of compressor outlet, kJ/kg; w--specific compression work, kW; q--specific refrigerating effect, kJ/kg; i--ratio of the refrigerant extracted from the gas-liquid separator to the total refrigerant.

Based on the above hypotheses, a steady-state simulation program of CO2 transcritical refrigeration cycle is developed by using Matlab software.

3. Result and Discussion

The variation of the specific compression work and unit cooling capacity of each system with the pressure of the gas cooler is shown in Figure 4a. The specific compression work of BC increases from 36.5 kW to 5.5 kW, with an average increase of 20.5% per MPa. The unit cooling capacity at 7.9 MPa is 50.4 kJ/kg, the unit cooling capacity at 9.1 MPa is 125.7 kJ/kg, and the average increase is 124.3%. The unit cooling capacity at 10.7 MPa is 143.0 kJ/kg, with an average increase of 8.6%. Before 9.1 MPa, the average increase of unit cooling capacity is better than the average increase of specific compression work, and then vice versa. The increase of specific compression work of EC is the same as that of BC. The unit cooling capacity of EC is 37.9 kJ/kg at 7.9 MPa, 117.5 kJ/kg at 9.1 MPa, with an average increase of 174.8%. The unit cooling capacity of EC is 134.8 kJ/kg at 10.7 MPa, with an average increase of 9.2%. Before 9.1 MPa, the average increase of unit cooling capacity is better than the average increase of specific compression work, and then vice versa. The specific compression work of P-C increases from 23.4 kW to 44.5 kW, with an average increase of 32.3%. The unit cooling capacity at 7.9 MPa is 50.2 kJ/kg. The unit cooling capacity at 8.9 MPa is 120.9 kJ/kg, with an average increase of 140.6%. The unit cooling capacity at 10.7 MPa is 142.9 kJ/kg, with an average increase of 10.1%. Before 8.9 MPa, the average increase of unit cooling capacity is better than the average increase of specific compression work, and then vice versa. The specific compression work of S-C increases from 23.6 kW to 48.9 kW, with an average increase of 38.1%. The unit cooling capacity at 7.9 MPa is 50.1 kJ/kg, the unit cooling capacity at 8.9 MPa is 120.8 kJ/kg, and the average increase is 141.0%. The unit cooling capacity at 10.7 MPa is 142.9 kJ/kg, and the average increase is 10.1%. Before 8.9 MPa, the average increase of unit cooling capacity is better than the average increase of specific compression work, and then vice versa. As the pressure of the gas cooler increases, the compression work of the compressor upsurges, and the cooling capacity also upsurges. The specific compression work of each system basically shows a linear growth trend, and the unit cooling capacity also shows an increasing trend. The growth rate is first rapid and then slow down.

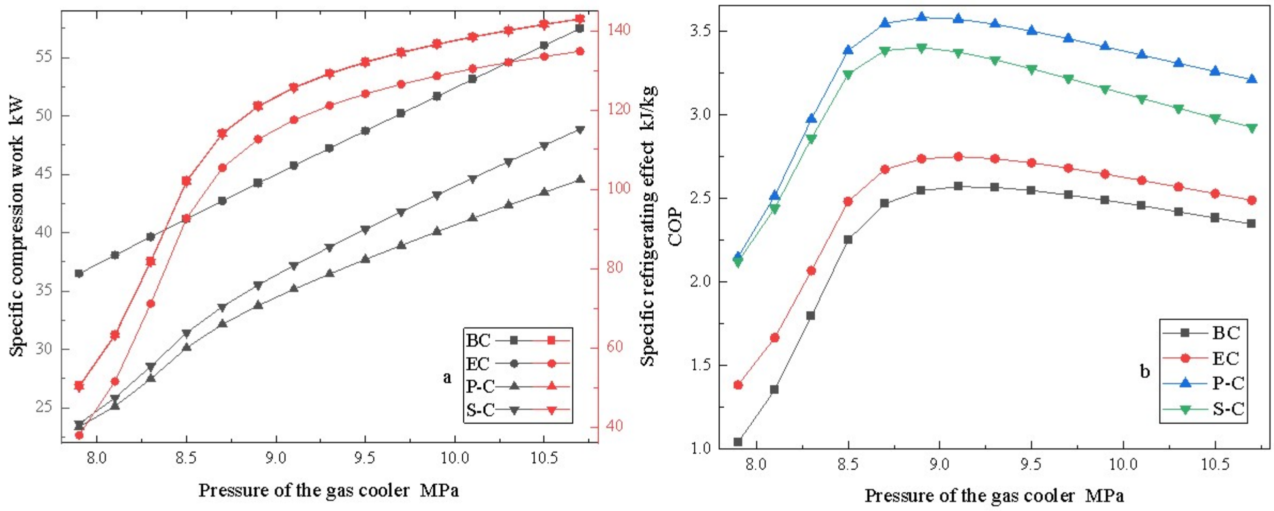

Figure 4. Effect of gas cooler pressure

Under different gas cooler pressures, the COP of the four refrigeration cycle systems is shown in **Figure 4**b. From the diagram, it can be seen that when BC is at 9.1 MPa, the maximum COP is 2.57, the COP at 7.9 MPa is 1.04, and the COP at 10.7 MPa is 2.34; when EC is 9.1 MPa, the maximum COP is 2.75, the COP at 7.9 MPa is 1.38, and the COP at 10.7 MPa is 2.49. When P-C is 8.9 MPa, the maximum COP is 3.58, the COP at 7.9 MPa is 2.15, and the COP at 10.7 MPa is 3.21. When S-C is 8.9 MPa, the maximum COP is 3.40, the COP is 2.12 at 7.9 MPa, and the COP is 2.92 at 10.7 MPa. Before reaching the optimum gas cooler pressure, each system is greatly affected by the pressure, and after reaching the optimum gas cooler pressure, it is less affected by the pressure ; before reaching the optimum gas cooler pressure, the average boost unit cooling capacity is better than the average increase of specific compression work, and then vice versa, so COP increases with the raise of gas cooler pressure. When it rises to a certain height, it presents a decreasing trend, and the maximum COP is obtained at 8.9 MPa or 9.1 MPa. Under the same gas cooler pressure, the COP of each system from small to large is BC, EC, S-C, P-C.

The specific compression work and unit cooling capacity of the system change with the outlet temperature of the gas cooler as displayed in **Figure 5**a. With the increase of the temperature of the gas cooler, the enthalpy of the outlet of the gas cooler increases, the enthalpy of the outlet of the expansion-compressor increases, and the enthalpy of the inlet of the evaporator also increases, which eventually leads to the reduction of the cooling capacity. The growth of the outlet enthalpy of the gas cooler in the BC and EC systems does not affect the inlet enthalpy and outlet enthalpy of the compressor, so the specific expansion work of the two systems does not change. With the increase of the outlet enthalpy of the gas cooler, the enthalpy and dryness of the fluid at the outlet of the expansion-compressor increase, so that more cooling capacity can be compressed by the expansion-compressor to the gas cooler pressure, and the compression work of the main compressor declines. As the temperature of the gas cooler increases, the specific compression work of BC, EC and S-C remains unchanged, the specific compression work of P-C progressively decreases, and the cooling capacity of all systems also decreases.

At different gas cooler temperatures, the COP of the four refrigeration cycle systems is presented in **Figure 5**b. It could be observed from the diagram that the COP of BC decreased from 3.60 to 1.86, with a decrease of 48.3%, the COP of EC decreased from 3.72 to 2.10, with a decrease of 43.6%, the COP of P-C decreased from 4.22 to 2.75, with a decrease of 53.7%, and the COP of S-C decreased from 4.25 to 2.77, with a decrease of 34.8%. The outlet temperature of the gas cooler has a great

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influence on the COP of the system. From 27 °C to 40 °C, the COP of each system decreases by more than 34%. With the increase of the outlet temperature of the gas cooler, the COP decreases with the increase of the temperature of the gas cooler, because the specific compression work of the system does not change or decreases slightly, while the unit cooling capacity decreases greatly.

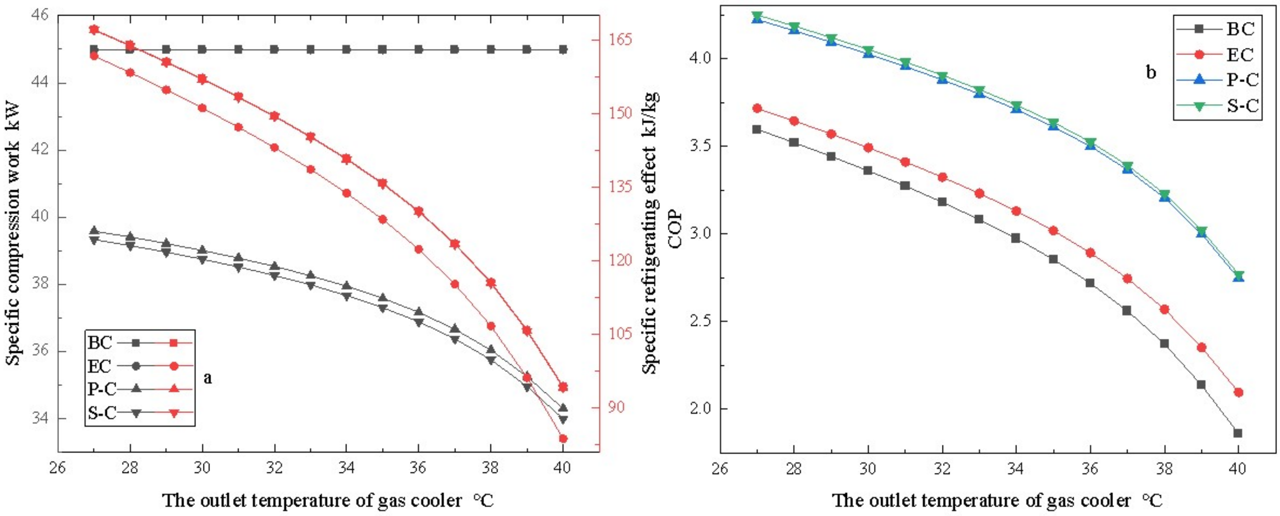

Figure 5. Effect of gas cooler outlet temperature

With the increase of evaporation temperature, the specific compression work and unit cooling capacity of each system change as shown in **Figure 6**a. The specific compression work of BC and EC decreased from 104.5 kW to 48.7 kW, with a decrease of 53.4%. The unit cooling capacity of BC decreased from 113.3 kJ/kg to 104.5 kJ/kg, with a decrease of 7.8%. The unit cooling capacity of EC decreased from 127.9 kJ/kg to 114.1 kJ/kg, with a decrease of 10.8%. The specific compression work of P-C decreases from 84.0 kW to 39.2 kW, with a decrease of 53.3%, and the unit cooling capacity decreases from 121.0 kJ/kg to 113.9 kJ/kg, with a decrease of 5.9%. The specific compression work of S-C decreases from 52.0 kJ/kg to 34.0 kJ/kg, a decrease of 34.6%, and the unit cooling capacity decreases from 115.5 kJ/kg to 110.6 kJ/kg, a decrease of 4.2%. The decline of specific compression work of each system is greater than that of unit cooling capacity. With the growth of evaporation temperature in each system, the enthalpy of refrigerant entering the compressor increases, the outlet temperature and pressure of the gas cooler are determined, the compression work of the compressor decreases, and the cooling capacity also decreases.

At different evaporation temperatures, the COP of the four refrigeration cycle systems is exposed in **Figure 6**b. It could be noticed from the figure that the COP of BC increased from 1.08 to 2.15, an increase of 99.1%, the COP of EC increased from 1.22 to 2.34, an increase of 91.8%, the COP of P-C increased from 1.44 to 2.91, an increase of 102.1%, and the COP of S-C increased from 2.22 to 3.26, an increase of 46.8%. The COP of each system has a positive linear correlation with the evaporation temperature, which improves with the increase of evaporation temperature, and the increase of COP is obvious. The evaporation temperature has a great influence on the COP of the system, and the COP of each system increases by more than 40% from-15 °C to 5 °C. At the same evaporation temperature, the COP of the expansion-compressor system is 32.9%~35.6% higher than that of the throttle valve system, and 17.7%~24.3% higher than that of the expander system. The COP of S-C is 51.8%~104.9% higher than that of BC, 39.1%~81.5% higher than that of EC, and 11.9%~54.2% higher than that of P-C.

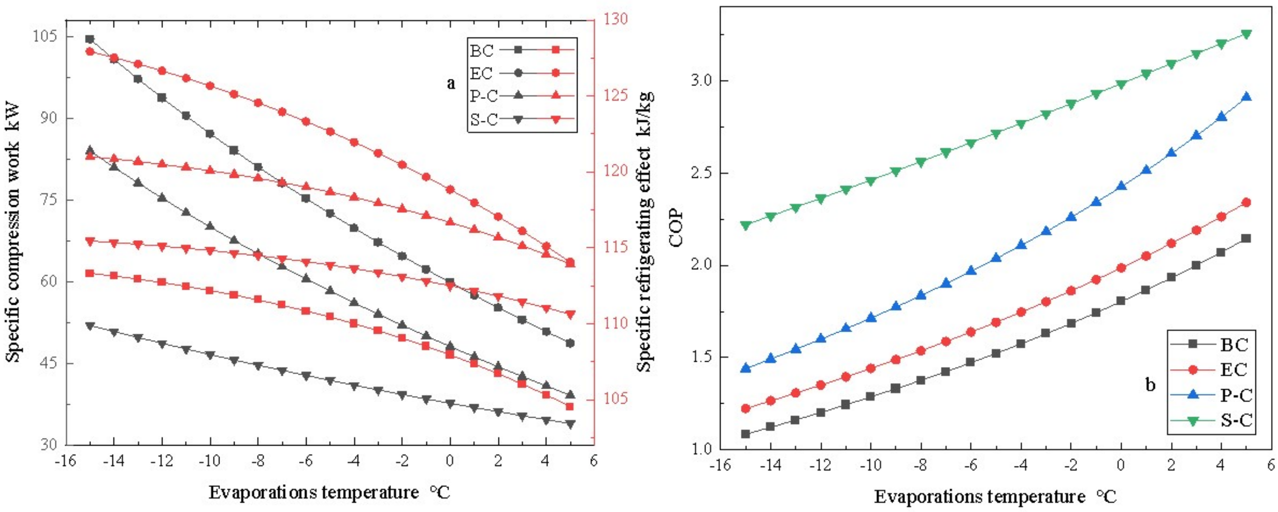

Figure 6. Effect of evaporation temperature

4. Conclusion

In this paper, the performance of four CO_2 transcritical refrigeration systems are calculated and analyzed. The main conclusions are as follows:

1) With increasing the gas cooler pressure, the compression work and cooling capacity increase, and the specific compression work shows a linear growth trend. The unit cooling capacity increases rapidly and then slows down, so the COP also increases and then declines. The optimal pressure is between 8.9 and 9.1 MPa.

2) With increasing the gas cooler temperature, the specific compression work of BC and EC remains unchanged, the specific compression work of the other two systems gradually decreases, and the cooling capacity of four refrigeration cycles also decrease. The COP decreases with the rise of the outlet temperature of the gas cooler, and is significantly affected by the temperature. As the temperature increases from 27 °C to 40 °C, the COP decreases more than 34%. The COP of each system increases with the increase of evaporation temperature and decrease of gas cooler temperature. At the same evaporation temperature or gas cooler temperature, the P-C system shows optimal performance.

3) Because S-C has an auxiliary compressor, the S-C system has higher COP and wider pressure range. At the same evaporation temperature, the COP of S-C is 51.8% ~ 104.9% higher than that of BC, 39.1% ~ 81.5% higher than that of EC, and 11.9% ~ 54.2% better than that of P-C.

Acknowledgments

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