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Thermodynamic evaluation of the effect of internal heat exchanger and expander on the CO_2 trans-critical cycle cascade refrigeration system

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ABSTRACT

In order to study the influence of expander and internal heat exchanger on the performance of the CO_2 trans-critical cycle cascade refrigeration system, the effects of evaporation temperature (T_e) on discharge temperature of the compressor, recovery power of expander, input power of compressor, COP, exergy efficiency and exergy destruction of the system, and exergy destruction ratio of each component were carried out by thermodynamic analysis. The results show that expander can effectively decrease the optimum condensation temperature of low-temperature cycle (LTC) and optimum high pressure, while high-temperature internal heat exchanger can slightly reduce the optimum condensation temperature and optimum high pressure of LTC and significantly reduce COP and recovery work of the expander. The COP of CE, when R14 is used as refrigerant, is 89.7% ~ 91.4% of that of R23 / R22CT. The biggest exergy destruction component is the gas cooler. The results provide a solution to improve the efficiency of the CO2 trans-critical cascade refrigeration system.

Keywords: CO_2 trans-critical; cascade refrigeration system; internal heat exchanger; expander; refrigerant; thermodynamic analysis

NONMENCLATURE

Symbols	
T	temperature, °C
h	specific enthalpy, $kJ\cdot kg^{-1}$
s	specific entropy, $kJ\cdot kg^{-1}\cdot K^{-1}$
R_p	pressure ratio of compressor
m	refrigerant mass flow rate, $kg\cdot s^{-1}$
Q	heating or cooling capacity, kW
W	power or work, kW
η	efficiency
ξ	effective exergy loss per unit refrigerating capacity
Abbreviations	
COP	coefficient of performance
LTC	low-temperature cycle
HTC	high-temperature cycle
CRS	cascade refrigeration system
TCRS	three-stage cascade refrigeration system
Comp	compression process or compressor
TV	throttling process or throttle value
Eva	evaporation process or evaporate
CHX	cascade heat exchanger or heat transfer of condensing evaporator
IHX	internal heat exchanger

1. INTRODUCTION

Resource shortage and environmental pollution have become two major problems in the world. Synthetic refrigerants, such as CFCs, HCFCs and HFCs, are widely used in the field of refrigeration and aviation. However, they can seriously damage the environment, destroy the ozone layer and cause the greenhouse effect. With the emphasis on environmental protection and energy conservation, people have turned their attention to the environmentally friendly refrigerants. At present, there are two main directions for the new generation refrigerants. One is to find new refrigerants with zero ODP and low GWP, but the high fabrication cost limits its wide application; the other is to return to the first generation of refrigerants, which are natural refrigerants [1], such as CO_2.

Generally speaking, CO_2 can be used in various refrigeration, air conditioning and heat pump systems. As a natural working fluid, CO_2 does not destroy the ozone layer. Its Global Warming Potential(GWP) is equal to one, Ozone Depletion(ODP) is zero, and safety level is A1. Besides, its unit volume refrigeration capacity is larger than that of ordinary refrigerants while the compression ratio of refrigeration cycle is lower than that of conventional refrigerants so that the volumetric efficiency of the compressor can be maintained at a high level[2]. Therefore, CO_2 has become an important substitute for CFC and HCFC.

CO_2 is also widely used in the refrigeration and heating systems of supermarkets all over the world. However, the following weaknesses of the CO_2 refrigerating system limit its large-scale application: operating pressure of the CO_2 trans-critical cycle system is too high(> 7.38MPa), the higher pressure bearing capacity of the system is a safety factor in the operation process need to be carefully considered; The Large pressure difference in the thermodynamic cycle leads to irreversible destruction in the throttling process and a high discharge temperature, which eventually lowers operation efficiency of the CO_2 system[3][4]. Many experts and scholars have carried out theoretical and experimental studies on different forms of CO_2 trans-critical cycle. Lorentzen first proposed the method of using the expander to recover expansion work [5]. He argued that if the throttle valve is replaced by a full-flow expander, the efficiency of the system would be greatly improved[6]. Kim's study has proved that the cycle mode of two-stage compression and intermediate cooling is an effective way to reduce the compressor power consumption and improve the system efficiency [7]. Besides, Yang has studied two-stage compression system theoretically and experimentally, and concluded that the two-stage compression system with intercooler has higher COP than the common type [8]. Ma and others have studied the performance of the two-stage compression, and carried out a thermodynamic analysis on the CO_2 trans-critical cycle and found that the COP of single-stage compression regenerative cycle is only 70% - 80% of that of conventional working fluids, such as R22, R134a [9][10].

When the temperature is above -80° C, cascade refrigeration system is more effective than the two-stage compression system. Many researchers have proved that the use of expander and internal heat exchanger can significantly improve the system efficiency.

According to the present researches both at home and abroad, there are three main cycles, including single-stage compression refrigeration cycle, double-stage compression refrigeration cycle and cascade refrigeration cycle . Liu and others have analyzed and compared the operation conditions of cascade refrigeration system and the two-stage compression refrigeration system when the evaporation temperature is in the range of - 80 ° C ~ -60 ° C, and their results show that the two-stage cascade refrigeration system had more advantages in terms of performance indexes such as suction pressure and efficiency of compressor [11]. Besides, Hamed has calculated the COP of R507/R744 cascade refrigeration system with evaporation temperature of -42°C [12]. What's more, Wang and others have made an experimental comparison between R404A two-stage compression refrigeration system and R404A/R23 two-stage cascade refrigeration system. The results indicate that when the condensation temperature is 40 °C and the evaporation temperature is -65°C, the suction pressure of the low-pressure compressor in the two-stage compression system is negative, which is not conducive to the normal operation of the unit[13].

However, there is a lack of a comparative analysis of various types of CO_2 trans-critical cascade refrigeration system with the expander and internal heat exchanger. In this paper, five different CO_2 trans-critical cascade refrigeration cycle systems with expander and internal heat exchanger were established and the thermodynamic analysis was carried out to provide a theoretical basis for the optimum design of the CO_2 trans-critical cascade refrigeration system.

Based on the latest development of refrigerant substitution, R41 is selected as the low-temperature refrigerant, and the detailed parameters of refrigerant are shown in Table 1.

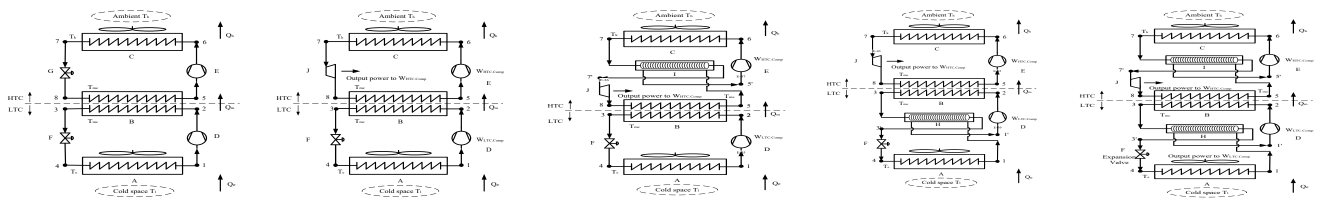
Tab.1 Basic physical properties of refrigerants

Refrigerant	Molecular weight /(kg/kmol)	Standard Billing Point/℃	Critical Temperature/℃	Critical Pressure/MPa	ODP	GWP	Safety Level
R170	30.1	-88.6	32.2	4.9	0	20	A3
R41	34	-78.1	44.1	5.9	0	97	A2
R23	70	-82.1	25.9	4.8	0	12000	A1
R744	44	-78.4	31.1	7.4	0	1	A1

2. THEORETICAL MODEL FOR THERMODYNAMIC ANALYSIS OF CASCADE REFRIGERATION SYSTEM

2.1 Introduction of cascade refrigeration system

The schematic diagram of cascade refrigeration system and *T-s* diagram of basic cascade refrigeration cycle are shown in Fig. 1 and Fig. 2, respectively. According to the application of HTC and LTCs IHX, HTC throttle valve and HTC expander, it is divided into five cascade refrigeration cycles. They show the flow chart of these five cycles respectively. Among them, Figure 1a is a cascade refrigeration system with high temperature cycle throttle valve, referred to as CT. Figure 1b is the system with high temperature cycle expander, which is CE for short. Figure 1c shows the system with HTC internal heat exchanger and high temperature cycle expander, referred to as CEI_{HTC}. Figure 1d shows a system with low temperature cycle internal heat exchanger and high temperature cycle expander, referred to as CEI_{LTC}. Figure 1E is the system with high and low temperature cycle heat exchanger and high temperature cycle expander, referred to as $CEI_{HTC-LTC}$. It should be noted that cascade refrigeration system is composed of two single-stage systems, namely high-temperature cycle and low-temperature cycle[14]. The two cycles are combined by the cascade heat exchanger, which works as condenser in LTC and evaporator in HTC. It can be seen that the condensation heat of the low-temperature cycle is equal to the evaporation heat of the high-temperature cycle.



(a)CT (b) CE (c)CEI_{HTC} (d)CEI_{LTC} (e) $CEI_{HTC-LTC}$

A-Evaporator; B-Cascade heat exchanger; C-Gas Cooler; D-LTC Compressor; E-HTC Compressor; F-LTC Throttle Valve; G-HTC Throttle Valve; H-LTC IHX; I-HTC IHX; J-HTC Expander;

Fig.1 Schematic diagram of cascade refrigeration system

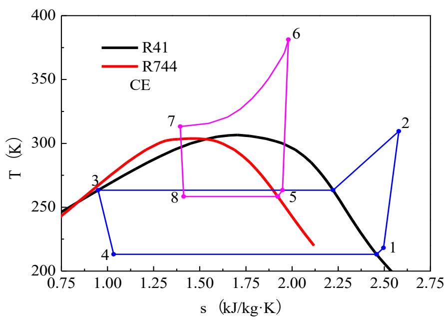


Fig.2 *T-S* diagram of CE

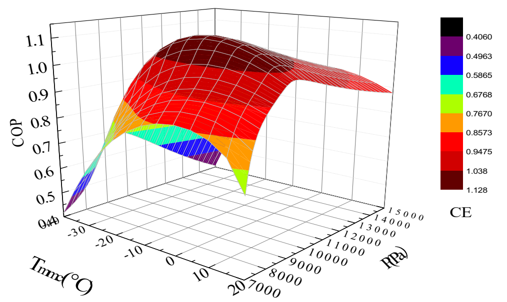


Fig.3 Variation of COP with LTC condensation temperature and HTC condensation pressure

2.2 theoretical model for thermodynamic analysis of cascade refrigeration cycle

According to the basic principle of cascade refrigeration system, the assumptions in the process of thermodynamic analysis are as follows:

The superheat occurred in the evaporator is effective overheating. The pressure drop and heat loss are ignored in the pipeline and heat exchanger. All components are assumed to be in stable state or steady flow process. The compression process, throttling process and expansion process are adiabatic processes. The isentropic efficiency of compressor and expander is closely related to the pressure ratio. Therefore, two polynomial functions are used to simulate the performance of compressor and expander separately.

Isentropic efficiency of compressor

$$\eta_{Comp} = 0.815 + 0.022\left(\frac{P_2}{P_1}\right) - 0.0041\left(\frac{P_2}{P_1}\right)^2 + 0.0001\left(\frac{P_2}{P_1}\right)^3 \quad (1)$$

Isentropic efficiency of expander

$$\eta_{Exp} = 1.0094 - 0.0504\left(\frac{P_2}{P_1}\right) \quad (2)$$

Where, P_1 and P_2 are evaporation pressure and discharge pressure respectively.

The general formula of CT and CE is as follows: Mass flow rate of LTC.

$$m_l = \frac{Q_e}{h_1 - h_4} \quad (3)$$

Actual compression work of LTC compressor.

$$W_{LTC.Comp} = m_l(h_2 - h_1) \quad (4)$$

3. RESULTS AND DISCUSSIONS

3.1 COP analysis of the systems

R170 is a low temperature refrigerant and R744 is a medium temperature refrigerant. A simulation study is carried out on five kinds of cascade refrigeration cycles when the cooling capacity Q_e is 10 kW, the heat transfer temperature difference △T in the condensing evaporator is 5 ° C, the gas cooler outlet temperature T_{gc} is 40°C, the isentropic efficiency of low-temperature cycle compressor is 80%, and the evaporation temperature T_e is - 85°C-- -30°C. Among them, the superheat degree of evaporator is 5°C, the superheater of recuperator in the low-temperature cycle is 25°C, the superheat of condensing evaporator and regenerator in high-temperature circle are 5°C and 25°C respectively. The theoretical analysis is programmed with Refprop 9.0 refrigerant data by MATLAB. All the results are obtained under the optimum interstage temperature and optimum high temperature stage condensation pressure.

Fig. 3 shows the variation of COP with low temperature cycle condensation temperature and pressure. As shown in Fig. 3, COP increases first and then decreases with the larger low-temperature cycle condensation temperature when the high-pressure pressure is fixed. Besides, when the condensing temperature is fixed, COP first increases and then decreases with the higher pressure. When the system high pressure deviates from the optimum pressure, COP will be reduced. Nevertheless, the change of COP is more gentle when the high pressure is higher than the optimum value than that in the opposite condition where the high-pressure is lower than the optimum-high-pressure. Therefore, there is an optimum low-temperature circulating condensation temperature and an optimum -high-pressure in the system to maximize the COP of the system.

Fig. 4 shows the variation of COP with evaporating temperature T_e for refrigerants R170 and R41. As shown in Fig.4, COP increases with the rise of evaporation temperature. Using the same cryogenic refrigerant, the COP of the five systems are CE, CEI_{LTC}, CEI_{HTC}, $CEI_{HTC\text{-}LTC}$ and CT, respectively. Among them, the difference between CE and CEI_{LTC}, difference between CEI_{HTC} and $CEI_{HTC\text{-}LTC}$ are both negligible, and the COP will be significantly reduced with the use of high temperature internal heat exchanger. What's more, the COP of R41 is higher than that of R170 in the three systems without low temperature internal heat exchanger. On the contrary, the effect of R170 is better with the use of internal heat exchanger, when the evaporation temperature is in the low and medium level(-85°C~-45°C); but when the evaporation temperature is high (-45°C~-30°C), R41 becomes a preferred refrigerant with the best effect, more specifically, the COP of CE is 23.5%~28.5% higher than that of CT when the refrigerant is R41 and 24.8%~31.9% higher than that of CT at R170.

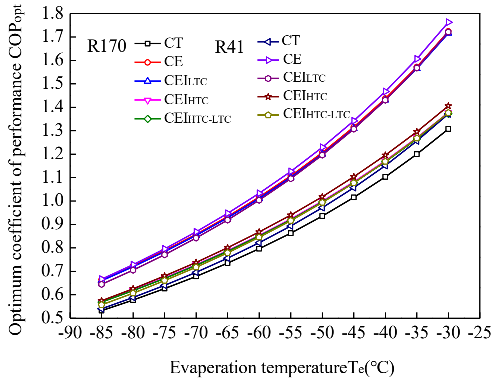


Fig.4 Variation of COP with evaporation temperature T_e

3.2 Pressure and Temperature analysis of the systems

Low and medium temperature circulating gas cooler is a evaporator as well as a gas cooler, which functions as a bridge in the process of cascade heat transfer in the tertiary circulation system. A lot of research have proved that intercooling temperature is of great significance to the optimization of system performance and the optimum pressure is also an important parameter to determine the operating conditions and to analyze the safety and reliability of the system.

Fig. 5 shows the variation of the optimum condensation temperature and optimum high-pressure with evaporation temperature T_e. As shown in Fig. 6, the optimum condensation temperature of LTC increases with the increase of evaporation temperature, and the optimum high- pressure decreases with the increase of evaporation temperature T_e. The five systems are sorted into CT、CEI_{HTC}、$CEI_{HTC\text{-}LTC}$、CEI_{LTC}/CE according to the descending order of optimum condensation temperature of LTC, among them, the value of CT system is obviously higher than that of other systems. While, the optimum high-pressure of CT, CEI_{HTC}/$CEI_{HTC\text{-}LTC}$, CEI_{LTC}/CE are sorted as descending order too, whose value are about 9.90

MPa ~ 10.35 MPa, 9.55 MPa and 9.45 MPa, respectively. From these different results of the five systems, it can be concluded that, the use of expander can effectively reduce the optimum condensation temperature and optimum high-pressure of LTC, but low-temperature internal heat exchanger does not have the similar effect, what's more, high-temperature internal heat exchanger also can slightly reduce the optimum condensation temperature and optimum high-pressure of LTC. The reduction of optimum high pressure has certain advantages in compressor selection and system leakage prevention, so it provides a new indicated way for the design of advanced refrigerating system in the future.

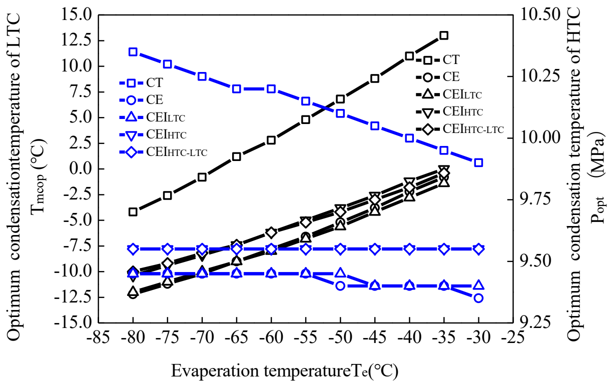


Fig.5 Variation of optimum condensation temperature in low temperature cycle and optimum high-pressure with evaporation temperature T_e

3.3 Exergy analysis of each component in system

The exergy destruction of vapor compression refrigeration device can be divided into heat transfer destruction and throttling (or compression) destruction. Figure 6 is the variation of exergy destruction and exergy efficiency with evaporation temperature T_e. When the evaporation temperature is in the range of -85°C to -30°C, the exergy efficiency first increases and then decreases, exergy destruction decreases with the increase of evaporation temperature T_e. The reason for this phenomenon is that the temperature difference between the evaporator and its environment decreases with the increase of evaporation temperature. The reduction of compressor pressure ratio and throttle valve pressure ratio can reduce the throttling destruction and compression destruction. From the above results, a conclusion can be drew that the use of the expander can reduce the system exergy destruction and the internal heat exchanger can increase the system exergy destruction.

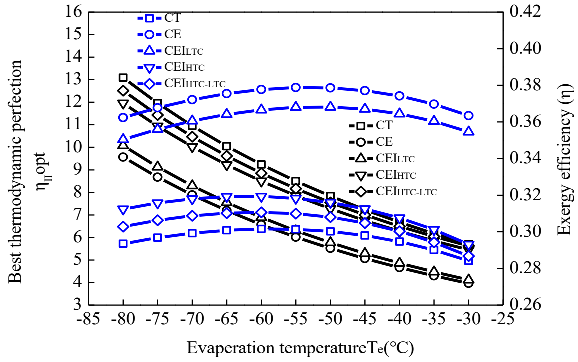


Fig.6 Variation of exergy destruction and exergy efficiency with evaporation temperature T_e

Figure7 shows the change of the proportion of exergy destruction of each component in the CE system with the evaporation temperature T_e. When the evaporation temperature is from - 85 ° C to - 30 ° C, the gas cooler has the least exergy destruction, while the gas cooler has the most exergy destruction, whose value increases from 29.0% to 39.9% with the increase of evaporation temperature. In order to reduce the heat destruction in the heat transfer process of the gas cooler, the condensing temperature and condensing pressure of refrigerant should be reduced, in this way, the exergy destruction of compressor and expansion valve can be reduced as well. Since the average temperature of refrigerant in the gas cooler mainly depends on the superheated steam temperature at the inlet of the gas cooler, reducing the discharge temperature of the compressor is also an important measure to reduce the heat destruction in the gas cooler.

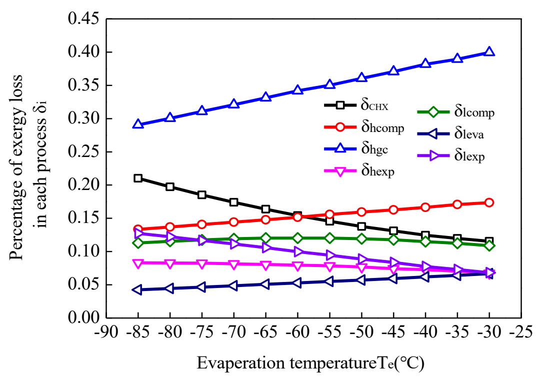


Fig. 7 Variation of the proportion of exergy destruction of each components in CE system with evaporation temperature T_e

4. CONCLUSION

According to the basic principle of cascade refrigeration cycle, the theoretical models of five kinds of cascade refrigeration cycle are established. The thermodynamic analysis was carried out on the variation law of the discharge temperature T_d of HTC and LTC, compressor input power W, expander recovery work W_{EXP}, COP of system, thermodynamic perfection degree η_2, system exergy efficiency η_x, exergy destruction x and its proportion of each component in the system δ_i, with the evaporation temperature T_e.

1. When the high pressure deviates from the optimum pressure, the COP will decrease. In other words, when the high pressure is higher than the optimum pressure, the COP will decrease more gently than that in the lower pressure. Meanwhile, there is an optimum condensation temperature of LTC and optimum high-pressure in the system, which makes the COP of the system maximum. The use of the expander can effectively reduce the optimum condensation temperature of LTC and the optimum high pressure. The use of high-temperature internal heat exchanger can slightly reduce the optimum cooling temperature of LTC and optimum high pressure. The reduction of optimum high pressure has certain advantages in the compressor selection and system leakage prevention.

2. The use of high-temperature internal heat exchanger will significantly reduce the recovery work of expander and the COP. In the system without low temperature internal heat exchanger, the COP of R41 is higher than that of R170. In the system using low temperature internal heat exchanger, R170 has good effect in the period of low and medium evaporation temperature (- 85°C~ -45°C) while R41 is the preferred refrigerant in a high-temperature period (-45°C ~ -30°C). More specifically, when R41 is used, the COP of CE is 89.7% ~ 91.4% of that of R23 / R22CT. What's more, using expander instead of throttle valve can reduce the throttling destruction to the greatest extent. However, the internal heat exchanger will reduce the performance of the system, which indicates that the increased partial cooling capacity due to the installation of the regenerator is less than the compression work increment caused by the overheated suction of the compressor.

3. The evaporator has the least exergy destruction, while the gas cooler has the largest value, which increases with the increase of evaporation temperature. Thus, optimizing the structure of the gas cooler, improving the heat transfer performance and reducing temperature difference are the effective ways to reduce the heat destruction in the heat transfer process of the gas cooler.

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