International Conference on Applied Energy 2020
Dec. 1 - Dec. 10, 2020, Bangkok / Virtual
Paper ID: 100

Comparative analysis of thermodynamic performance of three-stage cascade refrigeration system assisted with internal heat exchanger

Qifan Wang[1], Dandan Su[1*], Xuetao Liu[1], Liang Yao[1], Jingxuan Li[1], Pai Wang[1], Ning Zhang[1], Yao Yao[1]

1 Key Laboratory of Efficient Utilization of Low and Medium Grade Energy, MOE, Tianjin University, Tianjin, China

ABSTRACT

In this paper, the thermodynamic analysis of three kinds of three-stage cascade refrigeration systems (TCRS) with internal heat exchanger (IHX) are studied using R1150/R170/R717 and R50/R170/R717, including TCRS with LTC IHX ($TCRSI_L$), TCRS with MTC IHX ($TCRSI_M$) and TCRS with IHXs in MTC and LTC ($TCRSI_{L-M}$). The results indicate that when evaporation temperature is -120 °C ~ -80 °C and heat transfer efficiency of IHX is 60%, COP of TCRSs using R1150/R170/R717 is higher than that of TCRSs using R50/R170/R717. Under the same conditions, using IHX in TCRS will reduce the COP for R1150/R170/R717. Compared with TCRS without IHX, the average COP of $TCRSI_M$, $TCRSI_L$ and $TCRSI_{L-H}$ decreased by 0.6%, 1.5% and 3.6% respectively. For R50/R170/R717, when evaporation temperature is -115 °C ~ -100 °C and heat transfer efficiency of IHX is 60%, using IHX in LTC is helpful to improve COP, but the increase is less than 1% compared with TCRS without IHX. However, in other temperature range, using IHX will reduce the COP. And the use of IHXs increases the exergy destruction of cascade heat exchangers in MTC and LTC. Therefore, R1150 /R170/R717 is recommended for TCRS, and IHXs are not recommended in TCRS.

Keywords: three-stage cascade refrigeration system; internal heat exchanger; refrigerant; thermodynamic analysis, exergy analysis

NONMENCLATURE

Abbreviations	
T	temperature, °C
h	specific enthalpy, kJ·kg^{-1}
s	specific entropy, kJ·kg^{-1}·K^{-1}
R_p	pressure ratio of compressor
m	refrigerant mass flow rate, kg·s^{-1}
Q	heating or cooling capacity, kW
W	power or work, kW
COP	coefficient of performance
LTC or L	low-temperature cycle
MTC or M	medium-temperature cycle
HTC or H	high-temperature cycle
CRS	cascade refrigeration system
TCRS	three-stage cascade refrigeration system
BTCRS	basic three-stage cascade refrigeration system
TCRSI	three-stage cascade refrigeration system assisted with internal heat exchanger
Comp	compression process or compressor
TV	throttling process or throttle value
Eva	evaporation process or evaporate
Cond	condensation process or condenser
CHX	cascade heat exchanger or heat transfer of condensing evaporator
IHX	internal heat exchanger
Symbols	
a	environment
d	outlet of discharge
e	evaporation
i	system components
k	condensation
lc	condensation of LTC
mc	condensation of MTC
opt	optimum
s	isentropic
m	mechanical
E	electrical
vo	outlet of refrigerant gas in IHX
vi	inlet of refrigerant gas in IHX
li	inlet of refrigerant liquid in IHX
Greek symbols	
ξ	effective exergy loss per unit refrigerating capacity
χ	exergy destruction
II	second law efficiency
η	efficiency
Δ	difference

1. INTRODUCTION

Refrigeration technology with evaporation temperature below -80 ^{o}C is widely used in food processing, national defense and military, biomedicine, scientific research and so on [1, 2]. When the evaporation temperature is lower than -80 ^{o}C, the single-stage and two-stage compression refrigeration systems, and the cascade refrigeration system (CRS) are uneconomical due to the high temperature difference of evaporator and condenser. Because three-stage cascade refrigeration system (TCRS) can improve system performance and provide practical benefits, such as the use of a variety of combinations of refrigerants and lubricants, and preventing cold startup caused by liquid reflux, TCRS is considered to be an effective solution to the mentioned technical problem [3].

One of the challenges facing the TCRS is the substitution of refrigerants. Refrigerate in TCRS plays an important role to maintain low temperature for different purpose because it can provide high temperature rise and high system efficiency. Sun et al.[4] found that R1150 and R50 were recommended to replace R14 in LTC, and the thermodynamic performance of R1150 was better than that of R50. R170 and R41 are used at MTC, but the thermodynamic properties of R170 are better, and R717, R161 and R152a are used in HTC. R717 is an ideal choice for large-scale refrigeration systems due to its low price, easy to obtain and good thermodynamic performance. So this paper chooses R717 as HTC refrigerant, R170 as MTC refrigerant, R1150 and R50 as LTC refrigerant for thermodynamic analysis, that is, R1150/R170/R717, R50/R170/R717.

Another challenge facing the three-stage cascade refrigeration system is the improvement of performance. In addition to selecting the appropriate refrigerant group, the optimization of the base system can also improve the energy utilization of the system. Therefore, many experts and scholars have carried on the theoretical and experimental research on different forms of refrigeration cycle [5, 6], for example, the use of multi-stage compression can optimize the compression process, the use of the expander or the injector can recover the expansion work and improve the expansion process, or the thermoelectric cooling, and thermoelectric subcooling and mechanical subcooling to improve system performance, which is used in the basic system. However, these schemes inevitably have some limitations, and the auxiliary equipment will bring additional costs. The use of the internal heat exchanger (IHX) can not only increase the unit cooling capacity of the refrigeration system, but also ensure that the refrigerant inhaled by the compressor is in a gas state, thus avoiding the liquid impact in the compressor, which is an effective way to solve the inefficiency of the three-stage cascade refrigeration system. However, using IHX may be beneficial or harmful to the working of refrigeration system. This is closely relative to the thermodynamic properties of refrigerant and the operating conditions of system. Therefore, it is necessary to thermodynamic evaluation of the effect of IHX on the three-stage cascade refrigeration system.

At present, there is almost no research on the application of IHX in TCRS, but this paper fills up this gap. In order to investigate the effect of IHX on the performance of TCRS, three kinds of three-stage cascade refrigeration system with IHX (TCRSI), are established in this paper, respectively, which are TCRS with IHX in MTC ($TCRSI_{M}$), TCRS with IHX in LTC ($TCRSI_{L}$) and TCRS with IHXs in MTC and LTC ($TCRSI_{L\text{-}M}$). According to energy balance and exergy balance, the thermodynamic performance of TCRSI was compared with that of the basic TCRS (BTCRS). In order to determine the optimal system recommendation scheme, it provides a theoretical basis for the optimal design of TCRS with IHX.

2. THEORETICAL MODEL OF THERMODYNAMIC ANALYSIS OF THREE-STAGE CASCADE REFRIGERATION SYSTEM

2.1 Introduction of three-stage cascade refrigeration system

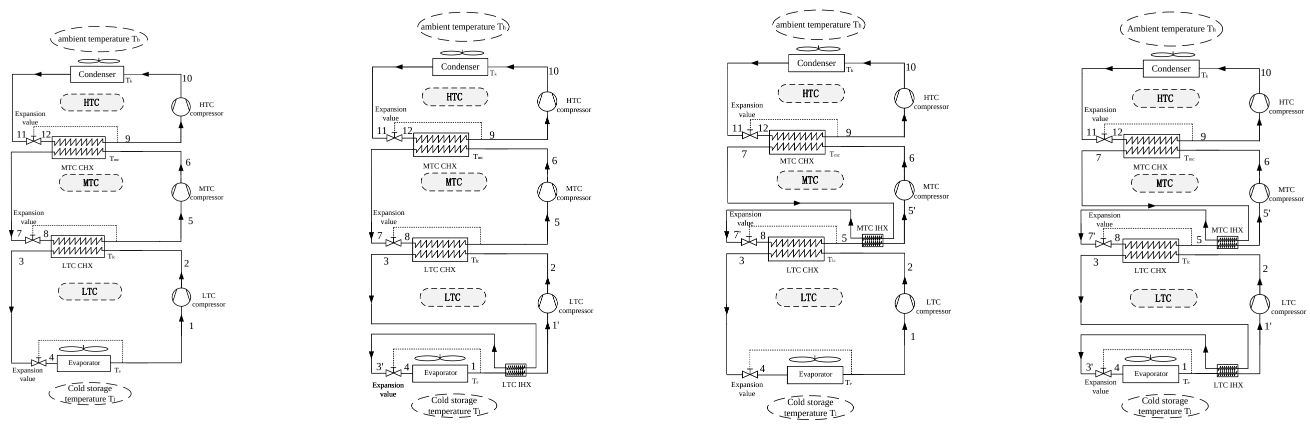
The three-stage cascade refrigeration cycle system is composed of three single-stage systems, which are high-temperature cycle (HTC), medium-temperature cycle (MTC) and low-temperature cycle (LTC), respectively. R717 is used in HTC, R170 is used in MTC, and R1150 and R50 are used in LTC. Fig.1 is schematic diagram of TCRS.

2.2 Theoretical model of thermodynamic analysis of three-stage cascade refrigeration system

According to the basic principle of cascade refrigeration cycle, a theoretical model for thermodynamic analysis of three-stage cascade refrigeration cycle is established. The following assumptions have been made to simplify the analysis. (1) The refrigerant is in a steady flow process in the circulating parts of the system.(2) The pressure drop and heat loss of the refrigerant in the pipeline and the heat exchange equipment are ignored [7]. (3) The degree of superheat of refrigerant gas at the outlet of evaporator, LTC CHX and MTC CHX is effectively overheated, and the refrigerant liquid at the outlet of the condenser, LTC CHX and MTC CHX is saturated liquid without undercooling. (4) The compression process is non-isentropic, and its isentropic efficiency can be described as a function of compression ratio [8].

As mentioned in the introduction, TCRS is widely used in the field of low-temperature refrigeration below -80 °C, so the range of evaporation temperature is -120 °C ~ -80 °C. Based on the above assumptions, the specific operating conditions of TCRS are as follows. Cooling capacity is 10kW. Heat transfer temperature difference in CHX is 5 °C [9]. Average ambient temperature is 25 °C. Condensing temperature is 40 °C. Heat transfer efficiency of Internal Heat Exchanger is 60% [10]. Mechanical efficiency and electrical efficiency are 0.93. Superheating in evaporator , LTC CHX, and CHX is 12 °C, 5 °C and 5 °C.

Based on the above assumptions and operating conditions, the mass balance equation, energy equation and exergy balance equation are established, as shown in Tab.1. MATALAB programming is used to solve the model. The enthalpy and pressure of each state point of refrigerant are all supplied by Refprop 9.1.



(a) TCRS (b) $TCRSI_L$ (c) $TCRSI_M$ (d) $TCRSI_{L-M}$

Fig.1 Schematic diagram of three-stage cascade refrigeration cycle system with or without IHX

Tab.1 Balance equation of three-stage cascade refrigeration system

System	Component	Energy balance and mass balance	Exergy balance
Thermodynamic theoretical model for three-stage cascade refrigeration system	HTC compressor	$W_{HTC.Comp}=\frac{m_{HTC}(h_{10s}-h_9)}{\eta_S\eta_m\eta_E}=\frac{m_{HTC}(h_{10}-h_9)}{\eta_m\eta_E}$	$\chi_{HTC.Comp}=T_a m_{HTC}(s_{10}-s_9)$
	Condenser	$Q_h=m_{HTC}(h_{10}-h_{11})$	$\chi_{HTC.Cond}=T_a\left[m_{HTC}(s_{11}-s_{10})+\frac{Q_h}{T_a}\right]$
	HTC expansion value	$h_{11}=h_{12}$	$\chi_{HTC.TV}=T_a m_{HTC}(s_{12}-s_{11})$
	MTC CHX	$Q_{MTC.CHX}=m_{HTC}(h_9-h_{12})=m_{MTC}(h_6-h_7)$ $m_{HTC}=\frac{Q_{MTC.CHX}}{(h_9-h_{12})}$	$\chi_{MTC.CHX}=T_a\left[m_{HTC}(s_9-s_{12})+m_{MTC}(s_7-s_6)\right]$
	MTC compressor	$W_{MTC.Comp}=\frac{m_{MTC}(h_{6s}-h_5)}{\eta_S\eta_m\eta_E}=\frac{m_{MTC}(h_6-h_5)}{\eta_m\eta_E}$	$\chi_{MTC.Comp}=T_a m_{MTC}(s_6-s_5)$
	MTC expansion value	$h_7=h_8$	$\chi_{MTC.TV}=T_a m_{MTC}(s_8-s_7)$
	MTC compressor	$W_{LTC.Comp}=\frac{m_{LTC}(h_{2s}-h_1)}{\eta_S\eta_m\eta_E}=\frac{m_{LTC}(h_2-h_1)}{\eta_m\eta_E}$	$\chi_{LTC.Comp}=T_a m_{LTC}(s_2-s_1)$
	LTC CHX	$Q_{LTC.CHX}=m_{LTC}(h_2-h_3)=m_{MTC}(h_5-h_8)$ $m_{MTC}=\frac{Q_{LTC.CHX}}{(h_5-h_8)}$	$\chi_{LTC.CHX}=T_a\left[m_{MTC}(s_5-s_8)+m_{LTC}(s_1-s_4)\right]$

	LTC expansion value	$h_3 = h_4$	$\chi_{LTC.TV} = T_a m_{LTC}(s_4 - s_3)$
	evaporator	$Q_e = m_{LTC}(h_1 - h_3)$ $m_{LTC} = \frac{Q_e}{(h_1 - h_3)}$	$\chi_{LTC.Eva} = T_a\left[m_{LTC}(s_1 - s_4) - \frac{Q_E}{T_e + \Delta T}\right]$
$TCRSI_M$	MTC compressor	$W_{MTC.Comp} = m_{MTC}(h_6 - h_{5'})$	$\chi_{MTC.Comp} = T_a m_{MTC}(s_6 - s_{5'})$
	MTC expansion value	$h_8 = h_{7'}$	$\chi_{MTC.TV} = T_a m_{MTC}(s_8 - s_{7'})$
	MTC IHX	$Q_{MTC.IHX} = m_{MTC}(h_{5'} - h_5) = m_{MTC}(h_7 - h_{7'})$	$\chi_{MTC.IHX} = T_a\left[m_{MTC}(s_{5'} - s_5) + m_{MTC}(s_{7'} - s_7)\right]$
$TCRSI_L$	LTC compressor	$W_{LTC.Comp} = m_{MTC}(h_2 - h_{1'})$	$\chi_{LTC.Comp} = T_a m_{LTC}(s_2 - s_{1'})$
	LTC expansion value	$h_4 = h_{3'}$	$\chi_{LTC.TV} = T_a m_{LTC}(s_4 - s_{3'})$
	LTC IHX	$Q_{LTC.IHX} = m_{LTC}(h_{1'} - h_1) = m_{LTC}(h_3 - h_{3'})$	$\chi_{LTC.IHX} = T_a\left[m_{MTC}(s_{1'} - s_1) + m_{lTC}(s_{3'} - s_3)\right]$

Isentropic efficiency of each compressor [7]:

$$\eta_s = 0.874 - 0.0135 R_p \quad (1)$$

The heat transfer efficiency of IHXs in MTC and LTC [28]:

$$\eta_s = \frac{T_{vo} - T_{vi}}{T_{li} - T_{vi}} \quad (2)$$

COP is one of the important economic performance indexes of refrigeration system, and the mathematical expression is as follows:

$$COP = \frac{Q_e}{W_{HTC.Comp} + W_{MTC.Comp} + W_{LTC.Comp}} \quad (3)$$

Exergy is the evaluation index of thermodynamic energy utilization efficiency, so the exergy destruction is analyzed by introducing the effective exergy destruction per unit refrigerating capacity. The mathematical expression is as follows:

$$\xi_i = \frac{\chi_i}{Q_e} \quad (4)$$

3. RESULTS AND DISCUSSIONS

3.1 *Optimal intermediate condensation temperature analysis*

The COPs of BTCRS and TCRSI are consistent with the change of condensation temperature of LTC (T_{lc}) and MTC (T_{mc}). Therefore, the BTRCS of R1150/R170/R717 is selected for research. As shown in Fig.2, when one intermediate condensation temperature is fixed, COP rises first and then decreases with the increase of the other intermediate condensation temperature. Therefore, there exists optimum condensing temperatures of LTC (T_{lcopt}) and MTC (T_{mcopt}), which results in the maximum COP value. Therefore, the following results and discussions are obtained at the optimum intermediate condensation temperature, that is, the maximum COP value of TCRS is obtained when the evaporation temperature is fixed.

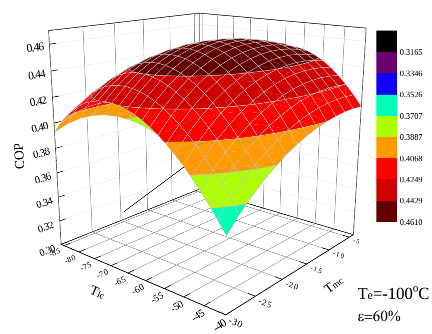


Fig.2 Effect of condensation temperature of LTC and MTC on COP

3.2 *Analysis of COP*

Fig.3 shows the effect of evaporation temperature on COP. As shown in Fig. 3, the COP increases with the increase of evaporation temperature, and the amplitude of the increase of COP increases with the increase of evaporation temperature. This is because the pressure ratio and isentropic efficiency of compressors decrease, and the magnitude of the decrease of total power consumption increases with the increase of evaporation temperature.

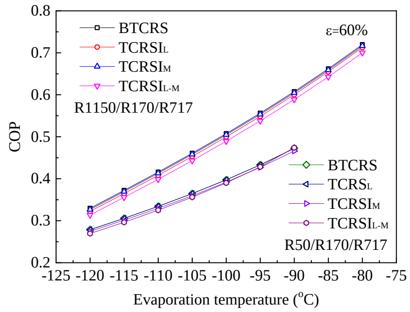
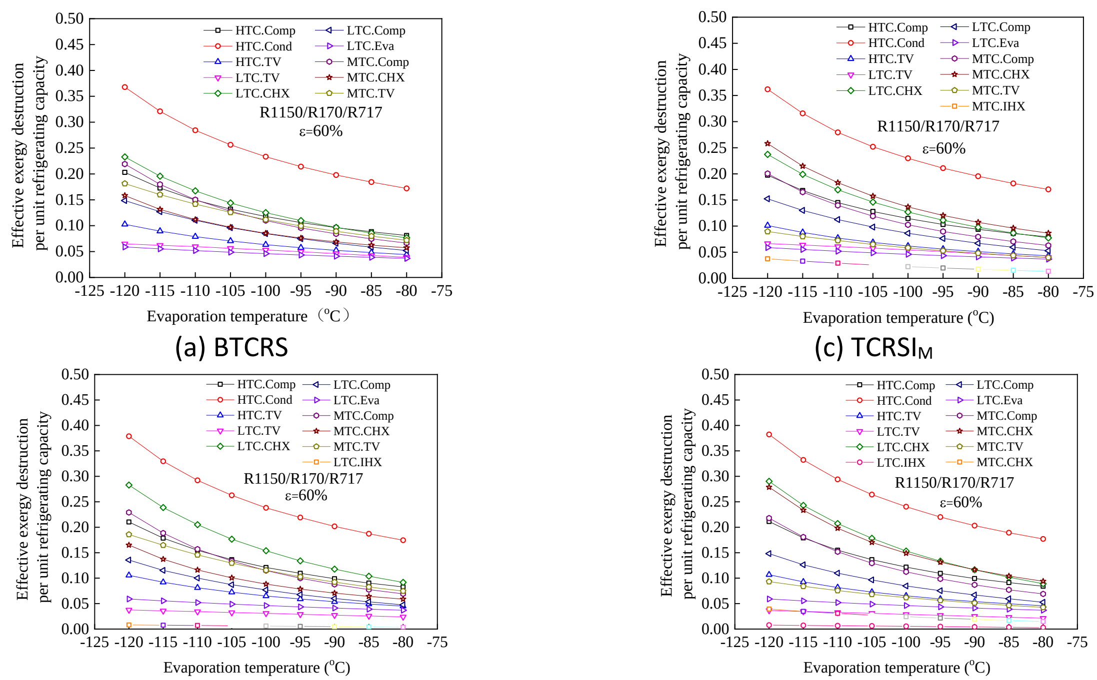


Fig.3 Effect of evaporation temperature on COP

When the evaporation temperature is in the range from -120 °C to -80 °C, the COP of the four TCRSs using R1150/R170/R717 follows in descending order by BTCRS, $TCRSI_M$, $TCRSI_L$ and $TCRSI_{L\text{-}M}$. Among them, the COP of BTCRS is 0.89%~0.43%, 2.25%~ 0.94% and 4.85%~2.89% higher than that of $TCRSI_M$, $TCRSI_L$ and $TCRSI_{L\text{-}M}$, respectively. For R50/R170/R717, when the evaporation temperature is in the range from -120 °C to -110 °C and about -95 °C, the COP of the four TCRSs follows in descending order by BTCRS, $TCRSI_L$, $TCRSI_M$ and $TCRSI_{L\text{-}M}$. When the evaporation temperature is -115 °C ~ -100 °C, the COP of the four TCRSs follows in descending order by $TCRSI_L$, TCRS, $TCRSI_M$ and $TCRSI_{L\text{-}M}$. It can be seen that for R1150/R170/R717, the use of IHXs in TCRS will reduce the COP of the system, and the effect of IHX in MTC on reducing system performance is greater. This indicates that the ratio of the partial unit cooling capacity increased by using the IHX to the compression specific power increment caused by the compressor suction superheat is smaller than the COP of the BTCRS. For R50/R170/R717, the use of IHX of LTC in TCRS can only improve the COP when the evaporation temperature is -115 °C ~ -100 °C, but the increase is less than 1% compared with the BTCRS. However, the IHX in MTC are not conducive to improving the COP. When the system is the same, the COP of TCRS using R1150/R170/R717 is higher than that of TCRS using R50/R170/R717, and the difference between the COP of TCRS using R1150/R170/R717 and the COP of TCRS using R50/R170/R717 increases with the increase of evaporation temperature. At the same time, the COP of worst performing system using R1150/R170/R717 — $TCRSI_{L\text{-}M}$ is greater than that of the optimal system of R50/R170/R717 —BTCRS or $TCRSI_L$. For example, the average COP of $TCRSI_{L\text{-}M}$ using R1150/R170/R717 is 34.43% higher than that of BTCRS using R50/R170/R717. Therefore, the refrigerant group R1150/R170/R717 is recommended in a TCRS, and IHX is not recommended to optimize system performance.

3.3 Exergy analysis of components



(a) BTCRS (c) $TCRSI_M$

(b) $TCRSI_L$ (d) $TCRSI_{L\text{-}M}$

Fig.4 Effect of evaporation temperature on effective destruction per unit refrigerating capacity

Exergy analysis provides a theoretical basis for improving the thermodynamic economy of the system and optimizing the efficiency design of each component. As analyzed above, the R1150/R170/R717 TCRS has better performance, so only R1150/R170/R717 TCRS is selected for exergy analysis. Fig.4 shows the effect of evaporation temperature on effective exergy destruction per unit refrigerating capacity. As shown in Fig.4, effective exergy destruction per unit refrigerating capacity decreases with the increase of evaporation temperature, which is contrary to the variation of COP (shown in Fig.3). Among the four systems, the condenser has the highest effective exergy destruction per unit refrigerating capacity, while the throttle valve or internal heat exchanger has the lowest exergy destruction. For $TCRSI_L$ and $TCRSI_{L-M}$, the use of IHX reduces the effective exergy destruction of LTC throttle valves and LTC compressor, while increases the exergy destruction of other components. Among them, the effective exergy destruction per unit refrigerating capacity of condensers in $TCRSI_L$ and $TCRSI_{L-M}$ is increased by 2.4% and 3.2% respectively. For $TCRSI_M$, the effective exergy destruction of HTC compressor, condenser, HTC throttle valve and MTC compressor, MTC throttle valve and evaporator are all reduced using IHX in MTC, in which the effective exergy destruction per unit refrigerating capacity of condenser is reduced by 1.5%. Compared with BTCRS, the effective exergy destruction per unit refrigerating capacity of LTC CHX and MTC CHX is increased to over 3.7% due to the use of the IHX in TCRS. The reason for this phenomenon is that the discharge temperature of HTC compressor decreases with the increase of evaporation temperature, which decreases the average temperature difference between the refrigerant in the condenser and the surrounding environment, and decreases exergy destruction during condensation and heat release process. It can be concluded that reducing the loss of condenser in TCRS can reduce the irreversible loss of cycle process.

4. CONCLUSION

The purpose of this paper is to discuss the effect of IHX on TCRS. Under the condition of evaporation temperature of -120 ℃ ~ -80 ℃, COP and exergy destruction of four TCRSs were compared and analyzed. The following conclusions are drawn:

1. For R1150/R170/R717, the use of the IHX does not increase the COP. For R50/R170/R717, when the evaporation temperature is -115 °C ~ -100 °C, the use of IHX in LTC can increase the COP, but compared with the BTCRS system, the increase is less than 1%.

2. The COP of R1150/R170/R717 is higher than that of R50 / R170 / R717 when the system is the same, and the difference between them is more than 0.13 with the increase of evaporation temperature.

3. For R1150/R170/R717, the exergy destruction of condenser is the largest, and the exergy destruction of throttle valve or internal heat exchanger is the lowest. In $TCRSI_M$, the use of internal heat exchangers reduced the effective exergy distribution per unit refrigerating capacity of condenser by an average of 1.5%. In $TCRSI_L$ and $TCRSI_{L-M}$, the use of internal heat exchangers increased the effective exergy distribution per unit refrigerating capacity of condenser by an average of 2.4% and 3.2%, respectively.

Therefore, R1150 /R170/R717 is recommended for TCRS, and IHXs are not recommended in TCRS.

REFERENCE

[1] Wang W, Zhou Q, Tian G, Hu B, Li Y, Cao F. The intermediate temperature optimization for cascade refrigeration system and air source heat pump via extreme seeking control. International Journal of Refrigeration. 2020;117:150-62.

[2] Ustaoglu A, Kursuncu B, Alptekin M, Gok MS. Performance optimization and parametric evaluation of the cascade vapor compression refrigeration cycle using Taguchi and ANOVA methods. Applied Thermal Engineering. 2020;180.

[3] Nagavarapu AK, Garimella S. High-flux thermal management at megawatt scale using a double-effect absorption/vapor-compression cascade refrigeration cycle. International Journal of Refrigeration. 2020;119:257-67.

[4] Sun Z, Wang Q, Dai B, Wang M, Xie Z. Options of low Global Warming Potential refrigerant group for a three-stage cascade refrigeration system. International Journal of Refrigeration. 2019;100:471-83.

[5] Sobieraj M, Rosiński M. High phase-separation efficiency auto-cascade system working with a blend of carbon dioxide for low-temperature isothermal refrigeration. Applied Thermal Engineering. 2019;161.

[6] Massuchetto LHP, Nascimento RBCd, Carvalho SMRd, Araújo HVd, d'Angelo JVH. Thermodynamic performance evaluation of a cascade refrigeration system with mixed refrigerants: R744/R1270, R744/R717 and R744/RE170. International Journal of Refrigeration. 2019;106:201-12.

[7] Sun Z, Wang Q, Xie Z, Liu S, Su D, Cui Q. Energy and exergy analysis of low GWP refrigerants in cascade refrigeration system. Energy. 2019;170:1170-80.

[8] Sánchez D, Cabello R, Llopis R, Catalán-Gil J, Nebot-Andrés L. Energy assessment and environmental impact analysis of an R134a/R744 cascade refrigeration plant upgraded with the low-GWP refrigerants R152a, R1234ze(E), propane (R290) and propylene (R1270). International Journal of Refrigeration. 2019;104:321-34.

[9] Cimsit C, Ozturk IT. Analysis of compression–absorption cascade refrigeration cycles. Applied Thermal Engineering. 2012;40:311-7.

[10] Yari M, Mahmoudi SMS. Thermodynamic analysis and optimization of novel ejector-expansion TRCC (transcritical CO2) cascade refrigeration cycles (Novel transcritical CO2 cycle). Energy. 2011.

