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Performance comparison of ultra-low temperature cascade refrigeration cycles using R717/R170, R717/R41 and R717/R1150 to replace R404A/R23

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ABSTRACT

This paper builds a cascade refrigeration cycle (CRS) system for the requirement of producing ultra-low temperature of −80 to −50 ℃. Taking R404A/R23 CRS as an example, the natural refrigerant NH_3 is used to replace the high temperature cycle (HTC) refrigerant R404A, and the environmentally friendly refrigerants R1150, R170, and R41 are used to replace the low temperature cycle (LTC) refrigerant R23. The influence of internal heat exchanger (IHX) on the performance of CRS when using new environmentally friendly refrigerants is analyzed. CRS with two-stage compression intermediate complete cooling method in HTC (CRS with TSC in HTC) is constructed, which not only reduces the discharge temperature of HTC compressor, but also improves the COP and exergy efficiency. According to the basic principles of thermodynamic energy and exergy balance, a system model is established and simulated, the CRS with TSC in HTC used NH_3/R1150, R170, R41 and CRS with IHX in HTC used R404A/R23 are comparative analyzed. The results show that optimal COP of the former is 15.79%, 18.58% and 16.17% higher than those of the latter under the given working conditions. The comparison analysis shows that the NH_3/R170 CRS had the best refrigeration performance. The research results will provide a reference to the application of cascade refrigeration system using environmentally friendly refrigerants for ultra-low temperature in considerations of efficiency and cycle configurations.

1. Introduction

With the rapid development of biomedical science, food science and electronic information technology, the requirement to maintain low temperature has risen sharply in many commercial and industrial applications. Scientific experiments in low temperature environments, preservation of organs, and processing and production of special foods (such as quick freezing of ice cream, preservation of tuna, etc.) all require temperatures below −50 °C. In order to realize the ultra-low temperature refrigeration system, the single stage compression and double stage compression refrigeration systems are difficult to meet the demand, so cascade refrigeration system (CRS) is generally used, that is, two conventional vapor compression refrigeration cycles are coupled through a cascade heat exchanger to achieve a larger temperature span, so as to achieve ultra-low temperature. The thermodynamic analysis of CRS and ordinary refrigeration system showed that: under a certain cooling capacity, although the COP of CRS was slightly reduced, it can produce a lower temperature, the discharge temperature of the compressor also reduced accordingly, and the volumetric efficiency was improved [1]. Bingming et al. compared the refrigeration performance of the NH_3/CO_2 CRS with the two-stage compression NH_3 refrigeration system and the single stage compression NH_3 refrigeration system with or without an economizer. The results showed that when the evaporation temperature is lower than −40 °C, the COP of CRS had the highest value and the best cooling performance [2]. Therefore, CRS is more promising than ordinary refrigeration systems in producing low temperature.

LTC and HTC of CRS can adopt different refrigerants. The refrigerants commonly used in HTC are R404A, R410A, R134A, NH_3, R290, etc. LTC refrigerants mainly include R23, R508B, R32, R41, R1270, CO_2, etc. Among them, NH_3 and CO_2 are widely used because they are natural refrigerants and environmentally friendly. Alberto Dopazo et al. analyzed the CRS with CO_2 and NH_3 as refrigerants, obtained the variation trend of COP, and found that each evaporation temperature corresponds to an optimal intermediate temperature

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Nomenclature

Q	refrigerating capacity (kW)
T	temperature (℃)
P	pressure (kPa)
h	specific enthalpy (kJ kg^{-1})
s	specific entropy [kJ $(kg\ K)^{-1}$]
m	mass flow rate (kg s^{-1})
P_r	compression ratio
W	energy (kW)
X	exergy destruction (kW)
Acronyms	
COP	coefficient of performance
CRS	cascade refrigeration system
ODP	ozone depletion potential
GWP	global warming potential
HTC	high-temperature circuit
MTC	middle-temperature circuit
LTC	low-temperature circuit
IHX	internal heat exchanger
TSC	two-stage compression
NBP	normal boiling point (℃)
Greek symbols	
ε	effectiveness of IHX
η_s	isentropic efficiency
η	exergy efficiency
Subscripts	
0	reference environment
E	evaporation
K	condensation
1,2	system location
S	isentropic process
cas	cascade refrigeration system
comp	compressor
eva	evaporator
con	condenser
exp	expansion valve
ec	economizer
opt	optimal
HTC	high-temperature circuit
LTC	low-temperature circuit

(condensation temperature of LTC) to make the system perform the best. Different refrigerant couples have different cooling effects [3]. Soni et al. selected several different refrigerant pairs for research. The results showed that refrigerant pair of NH_3/CO_2 is the best among all the selected low GWP refrigerant couple as NH_3/CO_2 pair has the highest value of COP and EER along with least power consumption [4]. In addition, some new optimization algorithms have been adopted to improve energy utilization and improve system cooling performance. Jeon proposed a calculation method of COP according to various conditions of R404A/CO_2 CRS. This paper proposed formulas that can be used to easily calculate the COP, the optimal cascade evaporation temperature to achieve the maximum COP, and the optimal amount of refrigerant to be charged in each cycle of the system [5].

With the rapid development of the refrigeration industry, it also brings a series of problems. The most commonly used refrigerants today are Freon refrigerants, which will destroy the ozone layer and cause global warming. According to the "Montreal Protocol" and "Kyoto Protocol", refrigerants with high ozone depletion potential (ODP) and global warming potential (GWP) will be phased out. Efficient new refrigerants are urgently needed. For different low-temperature needs, many researchers look for different combinations of environmentally friendly refrigerant pairs, and conduct theoretical analysis and experimental research. Sun et al. studied the use of R23, R41 and R170 as refrigerants for LTC in CRS, and the use of R32, R1234yf, R1234ze, R161, R1270, R290 and NH_3 as refrigerants for HTC in CRS. The results showed that the performance of R161 was better for HTC, while R41 and R170 were recommended for LTC [6]. However, conventional CRS was used in the analysis, and the cycle configuration and its influence on the pressure ratio and discharge temperature of compressor were not considered. Aktemur and Ozturk analyzed the refrigeration performance of several different refrigerants when CRS was applied at low temperature of −70 to −30 ℃. The results showed that the refrigerant pairs with high to low thermodynamic performance were R601/R41, cyclopentane/R41, R602A/R41 [7]. Sun et al. compared and analyzed the refrigeration performance of R404A/R41 and R404A/R23 refrigerant pairs, and found that refrigerant R41 was not only more environmentally friendly than R23, but also had higher cooling efficiency [8]. However, the GWP of HTC refrigerant R404A is 3800, which is not good for the environment, so HTC refrigerant is also in urgent need of replacement. Aktemur et al. studied the energy and exergy analysis of CRS using R41 for LTC, and environmentally friendly refrigerants R1243zf, R423A, R601, R601A, R1233zd(E) and RE170 for HTC. The results showed that RE170 can be used as an alternative refrigerant for R423A due to its low GWP and high cooling performance [9]. DĭKmen and ŞEncan ŞAhİN used R454C and CO_2 as LTC refrigerants in CRS, and R1234ze, R1234yf and NH_3 as HTC refrigerants. Thermodynamic analysis showed that the refrigerant combination NH_3/R454C has better cooling effect [10]. Roy and Mandal studied the refrigeration performance of two different refrigerant pairs R404A/R41 and R161/R170 in CRS under the same conditions, and the results showed that compared with R404A/R41 system, compressor power consumption and total exergy losses of R161/R170 system were reduced, and it has a higher optimal COP and exergy efficiency [11]. Yilmaz and Selbaş conducted thermodynamic analysis and comparison of the CRS using HFE7000/CO_2, R134a/CO_2, R152a/CO_2, R32/CO_2, R1234yf/CO_2 and 365mfc/CO_2 refrigerant pairs, and found that the 365mfc/CO_2 refrigerant pair has the best refrigeration performance [12]. Roy and Mandal used different refrigerant combinations to conduct numerical studies on the energy consumption, economic and environmental performance of CRS, and the results showed that the performance of the R161/R41 and R161/R170 system were better than that of the R404A/R41 system in all aspects [13]. Adebayo et al. compared the refrigeration performance of CRS using NH_3/CO_2, R134a/CO_2, HFE7000/CO_2 and HFE7100/CO_2 refrigerant pairs. The exergy efficiency were 47.01%, 45.84%, 44.30% and 41.06%, respectively, so HFE7000 can be used as an alternative refrigerant for R134a [14]. Sun Z et al. analyzed the application of low-GWP refrigerants in three-stage CRS, and the results showed that R1150 could replace R14 in LTC. In MTC, R41 and R170 can replace R23, and R170 was suggested to be preferred. In HTC the paper recommended NH_3, R152a, and R161 refrigerants. Considering environmental protection, NH_3 natural refrigerant was a better choice in large refrigeration systems [15]. The research results provide a basic theoretical analysis for the selection and replacement of refrigerants in three-stage CRS.

In addition, improving the performance of the refrigeration system is also a goal that people have been pursuing. Many studies have shown that building a new cycle system by adding internal heat exchanger (IHX) [16], subcooling [17], ejector [18], using two-stage compression [19] and other methods can improve the cooling performance of the system from different degrees, and the COP is significantly improved. The quality of the IHX to the system has a lot to do with the selected refrigerant and the range of system cycle conditions. S A Klein et al. found that IHX was beneficial to the performance improvement of the R404A refrigeration system, but it was unfavorable to the system using NH_3 [20]. Keshtkar took NH_3 and CO_2 as examples to study the effects of subcooling and superheating degrees on CRS based on thermos-economic optimization. Compared with the basic design, the maximum exergy efficiency of the system optimized by the genetic algorithm optimizer can be increased by 94.5%, and the total system cost can be reduced by 11% [17]. Rodrigo Llopis et al. evaluated the refrigeration performance of HFC134a/CO_2 CRS based on experiments, the experimental results showed that IHX slightly reduced the cooling capacity, but can increase the total COP by 3.7% [21]. Mota-Babiloni et al. used HFO-1234yf, HFO-1234ze(E), butane (HC-600), isobutane (HC-600a) and propane (HC-290) for LTC, and HCFO-1233zd(E), HFO-1336mzz(Z), HCFO-1224yd(Z) and pentane (HC-601) for HTC, discussed the CRS refrigeration performance with or without IHX. The results showed that the higher the heat recovery effectiveness of IHX at LTC and HTC, the higher the system COP, and the lower the intermediate pressure [22]. Jin et al. studied and compared the performance of the R404A air source heat pump system with or without IHX. The results showed that the performance of the system with IHX was much better than that without IHX [23]. Oruç and Devecioğlu experimentally proved that R404A and R442A single stage vapor compression refrigeration system with IHX can improve COP and exergy efficiency [24]. Bai et al. conducted a thermodynamic study on the structure optimization of the −80 °C freezer ejector-enhanced auto-cascade refrigeration cycle, using a natural mixture R1150/R600a as refrigerant. The simulation results showed that the addition of IHX can improve the overall performance of the system [25]. Chen et al. proposed a new type of CRS, which improved the cooling efficiency of the system by setting an auxiliary refrigeration cycle in the HTC. The results showed that when the subcooling degree was 10 °C, the maximum COP of the new system was increased by 4.58% compared with the original system [26]. Rodrigo Llopis et al. summarized the common means of supercooling, commonly used methods are dedicated mechanical subcooling and thermoelectric subcooling. Among dedicated mechanical subcooling, it includes IHX, IHX with ejector, two-stage compression with subcooler or economizer, etc [27]. Many scholars have studied this method. SYED M. ZUBAIR et al. added a mechanical subcooling loop to conventional refrigeration cycle. The results showed that the subcooling cycle can effectively reduce system power consumption and significantly improve refrigeration performance, and it reduced the refrigerant charge and saved energy [28,29]. But above system used single stage compression refrigeration cycle, in order to achieve lower temperature, the refrigeration cycle requires a higher temperature span, which will result in a higher compression ratio and affect system efficiency and safety. Adopting two-stage compression intermediate complete cooling system can effectively reduce the system compression ratio and improve the refrigeration performance of the system. Therefore, this article selects this loop method and discusses the advantages and feasibility of the system. In addition to simple subcooling, introducing an injector is also a good way to improve system performance. Chi et al. proposed a CRS with ejector subcooling, which significantly reduced the compressor discharge temperature compared with the original system, and improved the COP and exergy efficiency by 5.4% and 4.8%, respectively [18]. Liu et al. proposed an enhanced ejector refrigeration cycle to improve the performance of traditional ejector refrigeration cycles by introducing economizers and auxiliary ejectors. The calculation results showed that compared with the original system, the maximum COP of R134a and R600a increased by 10.7% and 9.9%, respectively [30]. Therefore, the introduction of the new enhanced injection system is conducive to the development of injection refrigeration technology and has a good prospect.

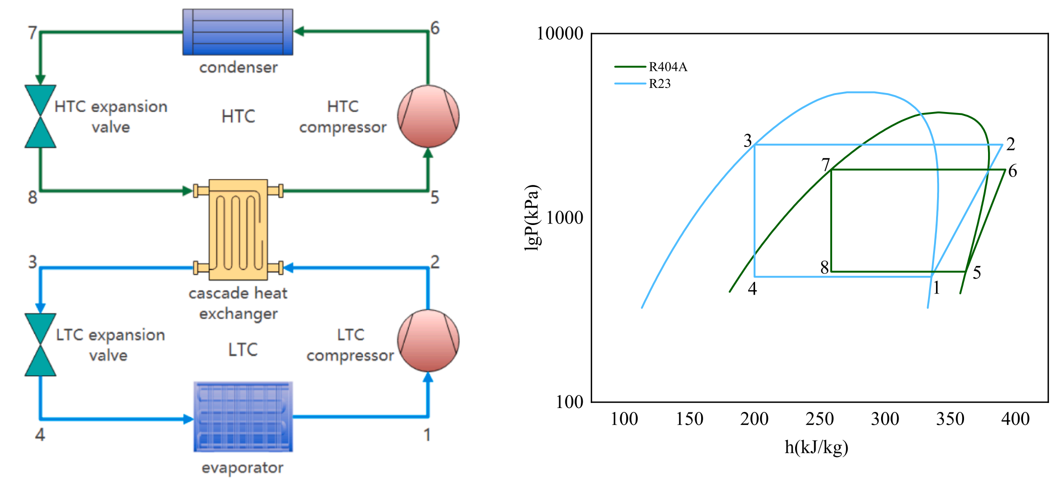
CO_2 cannot be used for ultra-low temperature application below −50℃ due to its triple point at −56℃. In order to produce a lower temperature, CO_2 mixtures can be used to participate in the LTC of CRS, which not only reduces the evaporation temperature but also improves the refrigeration performance of the system. ILK and Karlsruhe University of Applied Sciences have developed and built two cryogenic systems using a mixture of N_2O and CO_2 as refrigerants and successfully operated them at an evaporating temperature of −80 °C [31]. Massuchetto et al. evaluated the thermodynamic properties of three mixed refrigerants CO_2/R1270, CO_2/RE170 and CO_2/NH_3 in CRS LTC, and compared the thermodynamic properties with CRS using pure refrigerants. The results showed that CO_2/RE170 is the best combination [32]. Mota-Babilone et al. presents the mixture refrigerants in LTC for ultro-low temperature applications.[33]. Apart from those mainly based on CO_2, other refrigerants have been proposed for ultra-low temperature refrigeration. Liu et al. made an experimental test on the cascade system with the low temperature of −80℃ by using the environment-friendly refrigerant R290/R170 [34]. The results showed that the achievable range of COP was 0.511–0.526, which proved the feasibility of the device and put forward the method of optimizing the system. Mouneer et al. performed thermodynamic analysis on CRS and hydrocarbon refrigerant pair R290/R170 with R32/R170, R123/R170, R134a/R170, R404A/R170, R410/R170, etc [35]. The results show that R290/R170 has a good performance. At present, there are many researches on R290/R170 refrigerant pair, but there are few researches on NH_3/R170, so this paper adopts NH_3 as high-temperature refrigerant.

From the above literature review, it can be seen that for the ultra-low temperature refrigeration, more refrigerant pairs with higher GWP are used, such as R404, R134a, R23, etc. Among them, R23 is the most widely used, and its GWP is 12000, which is not friendly to the environment. Therefore, environmentally friendly refrigerants are required to replace them. In addition, the evaporation temperature of CRS such as NH_3/CO_2 and R290/CO_2, which are widely studied and used at present, is generally above −50℃, and there are few studies on CRS with lower temperature by using environmentally friendly refrigerants such as R1150, R170, and R41. For this reason, this paper takes NH_3 as HTC refrigerant and R1150, R170 and R41 as LTC refrigerants, researches refrigeration performance of CRS as low as −80℃, and compares it with the conventional R404A/R23 CRS, and discusses the effect of IHX on cycle performance, considering the large overall temperature span, CRS with two-stage compression and complete cooling in HTC (CRS with TSC in HTC) was constructed to reduce the HTC compressor pressure ratio and discharge temperature. A more comprehensive comparative analysis of the CRS of the new environmentally friendly refrigerants is carried out from the aspects of energy and exergy.

2. System descriptions

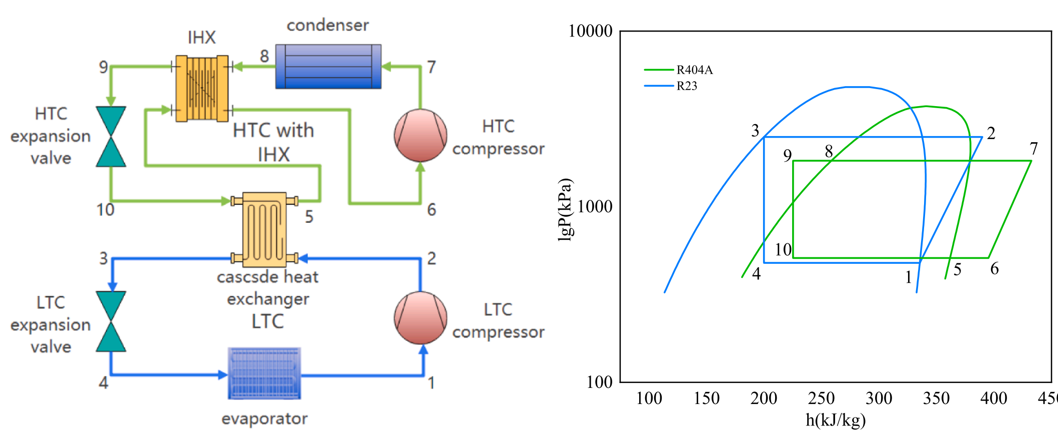
Fig. 1 is a schematic flowchart of a conventional CRS. The cycle formed by coupling a cascade heat exchanger in the middle of the two refrigeration cycles is a cascade refrigeration cycle, which is divided into HTC and LTC. The cascade heat exchanger acts as a condenser in LTC and acts as an evaporator in HTC. The refrigerant evaporates and absorbs heat in the LTC evaporator to achieve cooling effect. Fig. 2 shows the flow chart (a) and *p-h* diagram (b) of R404A/R23 CRS with IHX in HTC. 1–2–3–4 is the LTC process, 1–2 is the isentropic compression process of the compressor, and 2–3 is the condensation process in the cascade heat exchanger. 3–4 is the isenthalpy throttling of the expansion valve, and 4–1 is the evaporative process in the evaporator. 5–6–7–8–9–10 is the HTC process, 5–6 means that the refrigerant from the cascade heat exchanger enters the IHX for superheating, 6–7 is the isentropic compression process of the compressor, and 7–8 is the condensation process, 8–9 indicates that the refrigerant liquid from the condenser enters the IHX to exchange heat with the refrigerant gas in 5–6, and 9–10 is the isenthalpy throttling process. In practice, it is also possible to set IHX in LTC alone or at both stages.

Fig. 3 shows the flow chart (a) and *p-h* diagram (b) of CRS with TSC in HTC using NH_3/R170 as refrigerants. Similarly, 1-2-3-4 is the LTC process, and 5-6-7-8-9-10-11–12 is the HTC with TSC process. The



(a) Flow chart (b) *p-h* diagram

Fig. 1. R404A/R23 CRS.



(a) Flow chart (b) *p-h* diagram

Fig. 2. R404A/R23 CRS with IHX in HTC.

refrigerant is firstly isentropically compressed in the first stage compressor (5–6), and the compressed refrigerant gas enters the economizer to be cooled to a saturated state (6–7), and then enters the second stage compressor to continue isentropic compression (7–8), then enter the condenser to condense (8–9), the condensed liquid refrigerant is divided, and part of it is depressurized by the expansion valve (9–10), and then enters the economizer to absorb heat and reaches a saturated state (10–7), and then enters the second stage compressor, the other part enters the economizer for cooling (9–11), and then passes through the throttling valve for isoenthalpy throttling (11–12), and the throttled refrigerant enters the cascade heat exchanger for heat exchange (12–5), the whole cycle is completed.

The physical parameters of the selected refrigerants are shown in Table 1 [6,36,37].

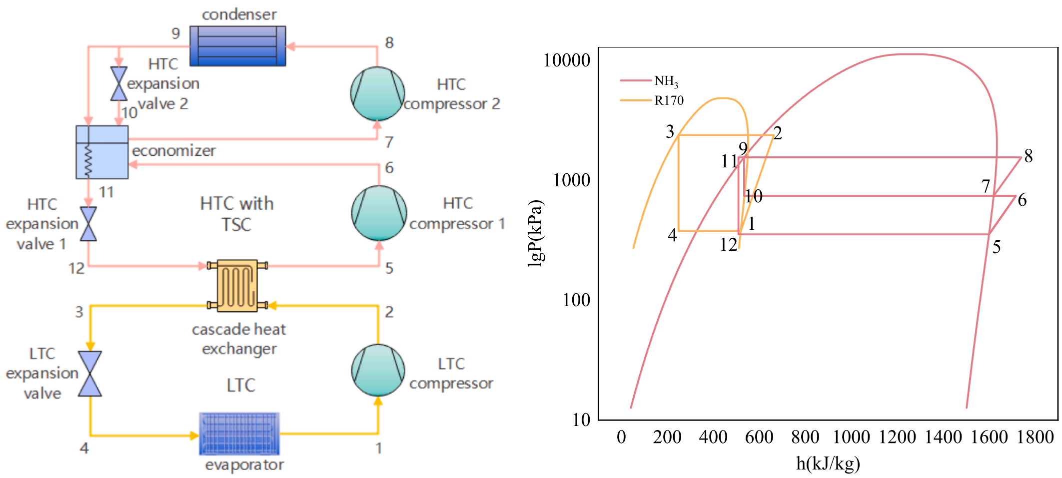
3. Mathematical model

According to the first law of thermodynamics and the second law of thermodynamics, the following assumptions are made to simply the simulation of the refrigeration system performance:

(1) The flow of refrigerant in the cycle is stable.
(2) Neglect heat loss and pressure drop in heat exchangers and connection pipes [15].
(3) The compression process is adiabatic and the isentropic efficiency is a function of the compressor pressure ratio.
(4) The evaporation and condensation processes are isobaric, and the throttling process is isenthalpic [10].
(5) The refrigerant at the outlet of the evaporator and the evaporative condenser is saturated gas, and the outlet of the condenser is saturated liquid [3].

In addition, the system also needs to assume some other environmental parameters and operating parameters in order to make the system more perfect, the assumptions made are shown in Table 2.

According to previous research [20], it can be known that setting IHX in the R404A refrigeration system can improve the system performance. The working principle of IHX is to exchange heat between the refrigerant gas at the outlet of the evaporator and the refrigerant liquid at the outlet of the condenser, preheat the gas entering the compressor and supercool the refrigerant at the condenser outlet. IHX has different effects on the cycle performance when using different refrigerants, and there are few



(a) Flow chart　　(b) *p-h* diagram

Fig. 3. NH_3/R170 CRS with TSC in HTC.

Table 1
Refrigerant physical parameters.

Refrigerants		Molecular weight	NBP (℃)	Critical temperature (℃)	Critical pressure (MPa)	ODP	GWP	Security Level
HTC	R404A	97.6	−46.2	72.04	3.73	0	3922	A1
	NH_3	17.03	−33.3	132.3	11.34	0	<1	B2
LTC	R23	70.01	−82.1	25.9	4.84	0	12,000	A1
	R41	34.0	−78.1	44.1	5.9	0	97	A2
	R170	30.07	−88.9	32.2	4.87	0	~20	A3
	R1150	28.05	−103.8	9.2	5.04	0	4	A3

Table 2
System design parameters.

Ambient temperature T_0(℃)	Ambient pressure P_0(kPa)	Condensation temperature T_k(℃)	Cooling capacity Q_e(kW)	Temperature difference T_d(℃)	Evaporation temperature T_e(℃)
25	100	40	9	5	−80 to −50

studies on the influence of refrigerants which are mentioned in this paper on IHX. In order to evaluate the effect of IHX in CRS such as NH_3/R1150, R170, R41, etc., the concept of IHX effectiveness was introduced [20].

IHX effectiveness is defined as follows:

$$\varepsilon = \frac{T_{v,out} - T_{v,in}}{T_{l,in} - T_{v,in}} \tag{1}$$

The numerator represents the temperature difference between the refrigerant gas entering and leaving the IHX, and the denominator represents the maximum degree of heat exchange that can be achieved.

The isentropic efficiency of the compressor varies with the operating conditions. In the theoretical analysis, it is assumed that the isentropic efficiency η_s is a function of the pressure ratio P_r, which is expressed by the following formula [38]:

$$\eta_s = 0.874 - 0.0135 \cdot P_r \tag{2}$$

According to the law of conservation of mass, the law of conservation of energy and the exergy balance relationship, the conservation equations of each component can be obtained.

Mass balance:

$$\sum_{in} m = \sum_{out} m \tag{3}$$

Energy balance:

$$Q - W = \sum_{out} m \cdot h - \sum_{in} m \cdot h \tag{4}$$

Exergy balance:

$$X = \sum_{out} \left(1 - \frac{T_0}{T}\right) \cdot Q - W + \sum_{in} m(h - T_0 \cdot s) - \sum_{out} m(h - T_0 \cdot s) \tag{5}$$

Taking the system shown in Fig. 3 (CRS with TSC in HTC) as an example, the calculation process and formula is listed as follows:

In the analysis, assuming a constant cooling capacity of 9 kW, when the evaporating temperature is given, the mass flow rate in evaporator of LTC can be obtained in Eq.(6):

$$m_1 = \frac{Q_e}{h_1 - h_4} \tag{6}$$

LTC compressor power consumption is defined as follows:

$$W_{comp,L} = m_1 (h_{2s} - h_1)/\eta_s \tag{7}$$

The heat load of cascade heat exchanger is calculated by:

$$Q_{cas} = m_1 (h_2 - h_3) \tag{8}$$

Then the refrigerant mass flow rate of HTC is expressed:

$$m_2 = \frac{Q_{cas}}{h_5 - h_{12}} \tag{9}$$

According to the conservation of mass, the refrigerant mass flow rate of HTC part 9–10 and part 9–11 in Fig. 3 can be calculated in Eq.(10) and Eq.(11) as follows:

$$m_{2a} = m_2 \frac{h_6 + h_9 - h_{11} - h_7}{h_7 - h_{10}} \tag{10}$$

$$m_{2b} = m_2 - m_{2a} \tag{11}$$

The value of the intermediate pressure of the two-stage compression is determined based on the empirical formula in Eq.(12):

$$P_m = P_6 = P_7 = \sqrt{P_5 \cdot P_8} \tag{12}$$

The power consumption of first stage compressor and second stage compressor in HTC is calculated by Eq.(13) and Eq.(14), respectively.

$$W_{comp1,H} = m_{2b}(h_{6s} - h_5)/\eta_s \tag{13}$$

$$W_{comp2,H} = m_2(h_{8s} - h_7)/\eta_s$$

The cooling load of condenser is defined in Eq.(15):

$$Q_{con} = m_2(h_8 - h_9) \tag{15}$$

The total power consumption of compressors is defined in Eq.(16):

$$W_{comp} = W_{comp,L} + W_{comp1,H} + W_{comp2,H} \tag{16}$$

The efficiency of the cascade refrigeration system can be estimated by the coefficient of performance (COP) in Eq. (17):

$$COP = \frac{Q_e}{W_{comp}} \tag{17}$$

When using a conventional refrigeration system or a system with IHX, the model of each component of the system can also be modeled and calculated as described above after simple processing.

For the exergy analysis, regardless of kinetic energy and potential energy, the exergy of each state point can be expressed as Eq. (18):

$$e = (h - h_0) - T_0(s - s_0) \tag{18}$$

The exergy destruction of each component is listed in Table 3 [3].

The total exergy destruction of the system can be obtained by Eq.(19):

$$X_{total} = X_{comp,L} + X_{eva,L} + X_{comp,L} + X_{comp1,H} + X_{comp2,H} + X_{con} + X_{exp1,H} + X_{exp2,H} + X_{cas} + X_{ec} \tag{19}$$

For the overall system, the exergy efficiency is the ratio of the total system's exergy destruction to the total power consumption for the overall system, as shown in Eq.(20):

$$\eta = \frac{W_{comp,L} + W_{comp1,H} + W_{comp2,H} - X_{total}}{W_{comp,L} + W_{comp1,H} + W_{comp2,H}} \tag{20}$$

According to the above conservation laws, MATLAB is used for

Table 3
Exergy destruction of each component.

Component	Exergy destruction(kW)
LTC compressor	$X_{comp,L} = W_{comp,L} + m_1 \cdot (e_1 - e_2)$
LTC evaporator	$X_{eva,L} = (1 - \frac{T_0}{T}) \cdot Q_e + m_1 \cdot (e_1 - e_2)$
LTC expansion valve	$X_{comp,L} = m_1 \cdot (e_3 - e_4)$
HTC first stage compressor	$X_{comp1,H} = W_{comp1,H} + m_{2b} \cdot (e_5 - e_6)$
HTC second stage compressor	$X_{comp2,H} = W_{comp2,H} + m_2 \cdot (e_7 - e_8)$
HTC condenser	$X_{con} = m_2 \cdot (e_8 - e_9)$
HTC expansion valve 1	$X_{exp1,H} = m_{2b} \cdot (e_{11} - e_{12})$
HTC expansion valve 2	$X_{exp2,H} = m_{2a} \cdot (e_9 - e_{10})$
Cascade heat exchanger	$X_{cas} = m_1 \cdot (e_2 - e_3) + m_2 \cdot (e_{12} - e_5)$
Economizer	$X_{ec} = m_{2b} \cdot (e_9 - e_{11}) + m_{2a} \cdot (e_{10} - e_7) + m_{2b} \cdot (e_6 - e_7)$

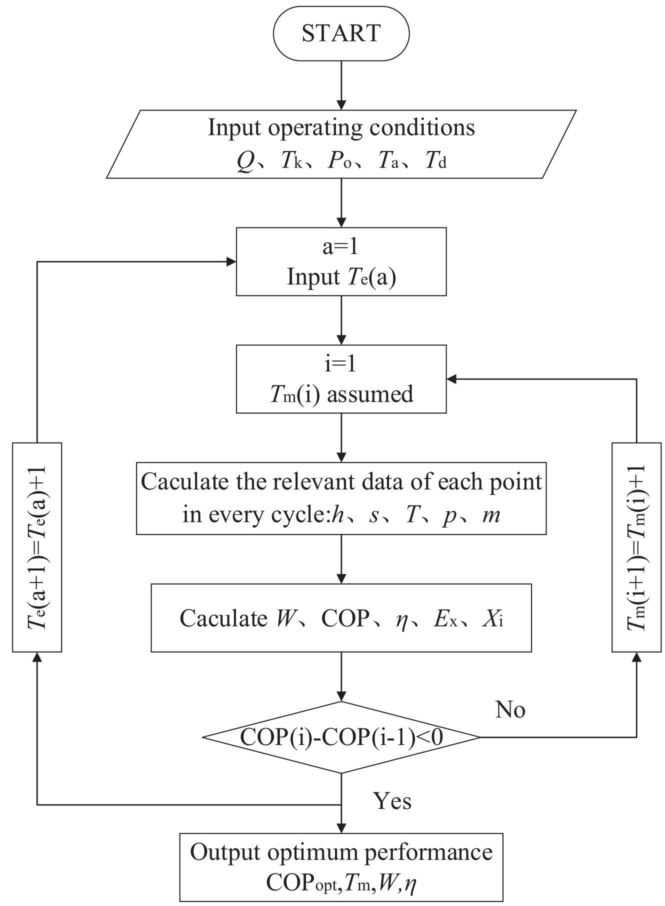
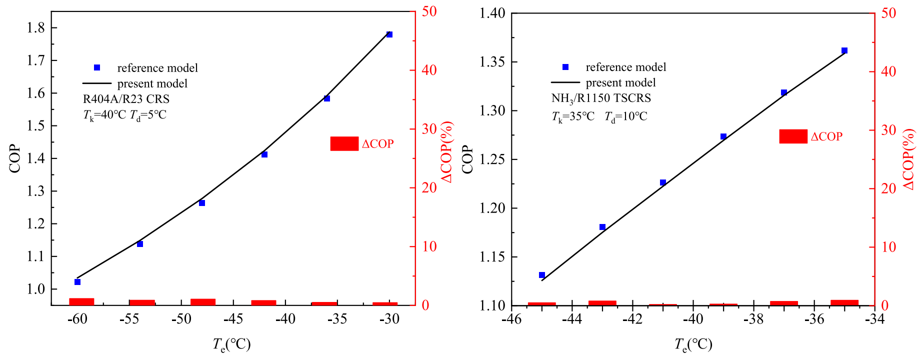


Fig. 4. Flow chart of calculation.

programming, and the parameters of each state point can be obtained by calling the physical property database Refprop. By changing the evaporation temperature or intermediate temperature, the system refrigeration performance is analyzed and compared in terms of COP, exergy efficiency, discharge temperature, compressor power consumption and exergy destruction. The specific calculation flow chart is shown in Fig. 4.

4. Model validation

Applying the operating parameters and design conditions of literature [8,39]to the mathematical model established in this article, the comparison diagram between the simulated results and literature data is shown in Fig. 5. In the calculation, the selected reference parameters are as follows: the ambient temperature is 25℃, the condensation temperature is 40℃, the temperature difference of cascade heat exchanger is 5℃, and the isentropic efficiency of compressors in LTC and HTC is 0.8. At the same time, the superheating of LTC is 5℃ and that of HTC is 12℃. Similarly, the reference parameters of TSCRS verification system are: the ambient temperature is 25℃, the condensation temperature is 35℃, the temperature difference of cascade heat exchanger is 10℃, and the isentropic efficiency of compressors in LTC and HTC are calculated according to empirical formulas separately. As shown in Fig. 5 (a), when T_e increases from −60℃ to −30℃, the condensation temperature T_k is 40℃, and the heat exchange temperature difference T_d is 5℃ as an example, it can be seen that the maximum error is 1.19%. Similarly, TSCRS was validated as shown in Fig. 5 (b). When T_e increases from −45℃ to −35℃, the condensation temperature T_k is 35℃, and the heat exchange temperature difference T_d is 10℃, error between the COP of the model in this article and reference model maximum only 0.93%, so the logic correctness of this program and model can be verified.



(a)CRS (b) TSCRS

Fig. 5. Comparison of simulation results with previous literature in COP with T_e.

5. Results and discussions

5.1. Influence of T_m

The COP of CRS shows a trend of first increasing and then decreasing with the intermediate temperature T_m (LTC condensation temperature), and there will be an optimal T_m, which maximizes the corresponding COP. This is because with the increase of T_m, the COP of LTC decreases, the COP of HTC increases, and the total COP will reach the maximum value at a certain point in the middle. Using the conventional CRS system shown in Fig. 1, taking the evaporation temperature T_e of −60 °C and the condensation temperature T_k of 40 °C as an example, system COP changes with the T_m as shown in Fig. 6. The R404A/R23 system has the highest COP when T_m is −10.0 °C, which is 1.018, while the systems corresponding to R1150, R170 and R41 reach the maximum COP at T_m of −16.2 °C, −12.1 °C and −10.9 °C, respectively are 1.088, 1.153 and 1.160. It can be seen that under this working condition, the R41 system has the best performance, R170 is the second, and R1150 is the last. The optimal T_m corresponding to the system composed of different refrigerants is also different. Therefore, in the following analysis, for different T_e or circulation systems, the obtained system performance is based on the optimal T_m.

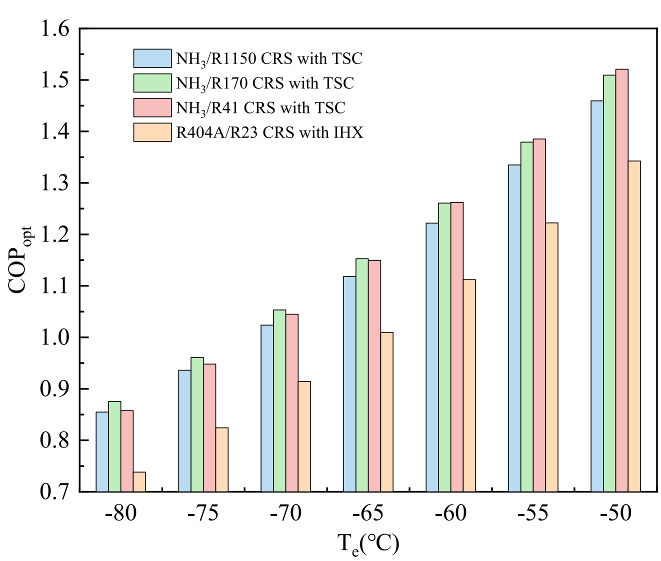


Fig. 6. Variation trend of COP of conventional CRS system with T_m.

5.2. Influence of IHX

A previous study showed that R404A refrigeration system with IHX is beneficial [20]. In order to further discuss the cooling effect of IHX on the use of R404A/R23 in CRS, several different systems with IHX were analyzed and compared, and the IHX effectiveness was taken as 1. Similarly, in order to study the influence of the CRS system with IHX, we use NH_3 for HTC refrigerant, R1150, R170, R41 for LTC refrigerant, analyze and compare system performance of the LTC and HTC with or without IHX. The results are shown in Fig. 7. It can be seen that the R404A/R23 CRS with IHX in HTC (IHCRS) have the best refrigeration performance, Therefore, the system is compared with the environmentally friendly refrigeration system next to discuss the possibility of refrigerant substitution. For CRS with NH_3/R1150, R170, R41, etc., when the T_e varies between −80 °C and −50 °C, CRS with IHX in LTC, CRS with IHX in HTC, and CRS with IHX in LTC and HTC all have lower COP than conventional CRS system. Taking T_e of −80 °C and T_k of 40 °C as an example, COP of the first three systems with NH_3/R1150 are respectively 7.10%, 6.23% and 12.68% lower than CRS.

And under fixed working conditions, with the increase of IHX effectiveness, the optimal COP of CRS used NH_3/R1150 etc decreases, as shown in Fig. 8. When the IHX effectiveness is 1, the COP of NH_3/R1150, R170, and R41 are reduced by 7.56%, 6.00% and 10.42% compared with CRS without IHX. Therefore, it can be seen that when HTC uses NH_3, and LTC uses R1150, R170 and R41 for refrigerants, it is disadvantageous to add IHX to LTC or HTC of CRS.

Other operating parameters of the system also determine the quality of the CRS system, such as the P_r should not be too high, and the discharge temperature T_d should be moderate. Variation of optimal T_m, COP, P_r and T_d of NH_3/R1150 conventional CRS system are shown in Table 4. It can be seen that when the T_e of −80 °C is produced, the P_r and T_d of the HTC compressor are too high, reaching 11.17 and 209.6 °C respectively, which is not good for the system operation and will reduce the compressor life. R170 and R41 the same. Therefore, to improve system performance, it is proposed to construct new CRS with two-stage compression in HTC (TSCRS), which can not only reduce the compressor P_r and T_d, but also reduce the power consumption of the compressor, which is beneficial to improve the system COP.

5.3. Cycle comparison

In order to prove that the two-stage compression in HTC is beneficial to the system, the NH_3/R1150 system is taken as an example to verify the performance improvement of the two-stage compression compared with the single stage compression system, as shown in Fig. 8. It can be seen from Fig. 9 (a) that the COP of the former system is greatly improved compared with the latter system. When the T_k is 40 °C and the T_e is changed from −80 °C to −50 °C, the COP increases by 10.90% to

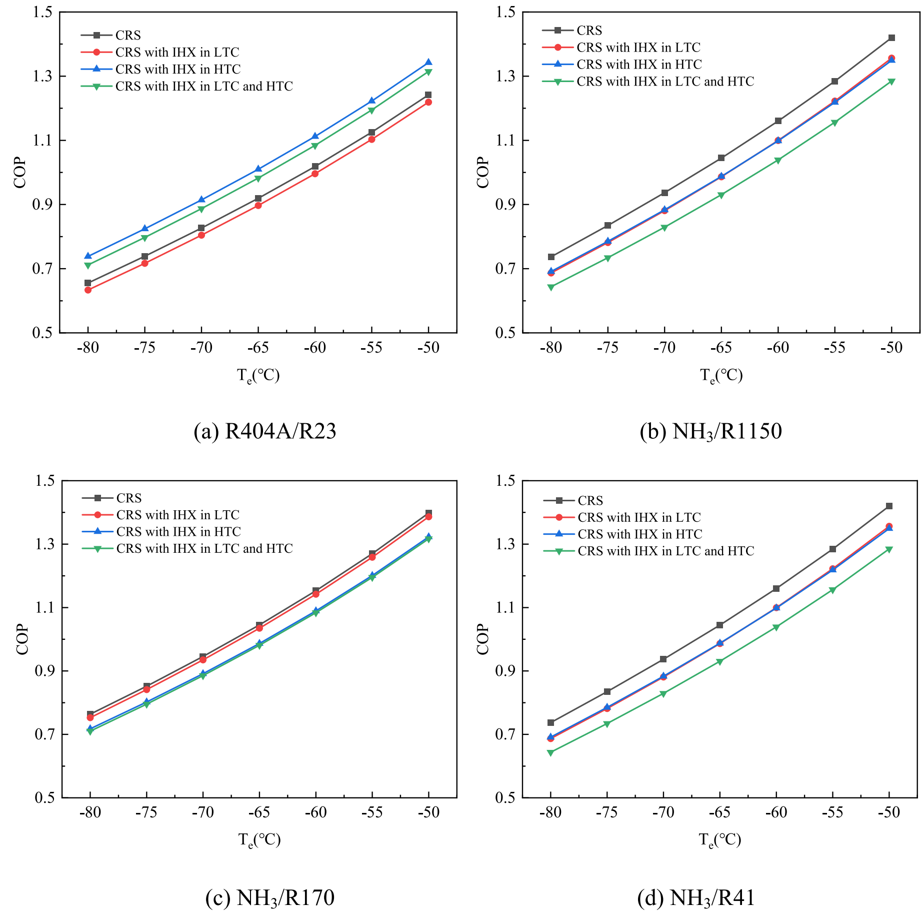


Fig. 7. Variation of maximum COP with T_e of CRS with or without IHX.

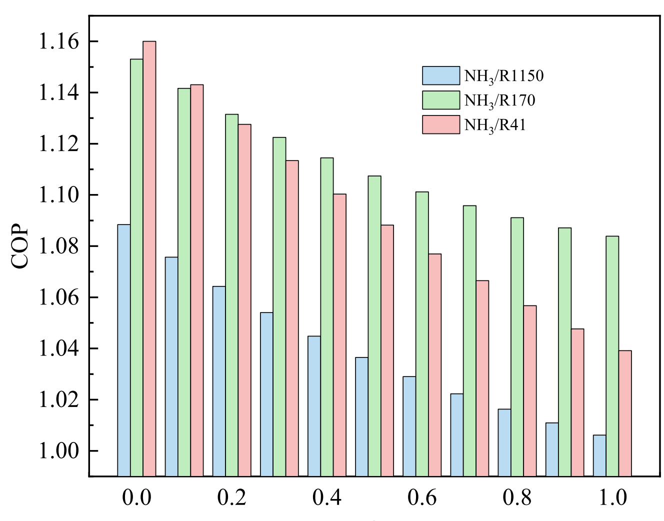


Fig. 8. COP with IHX effectiveness of CRS with IHX in HTC.

Table 4
NH_3/R1150 CRS system performance.

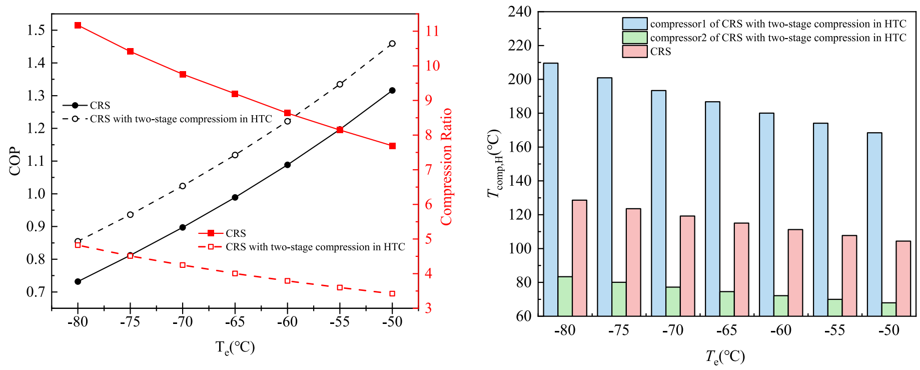
T_e(°C)	T_m(°C)	COP	$P_{r\text{-LTC}}$	$P_{r\text{-HTC}}$	T_{d1}(°C)	T_{d2}(°C)
−80	−21.8	0.731	7.09	11.17	51.0	209.6
−70	−18.9	0.897	5.03	9.76	37.2	193.4
−60	−16.2	1.088	3.69	8.63	27.0	180.0

16.82%. Fig. 9 (a) also shows the variation of the compressor pressure ratio P_r with T_e. The two-stage compression P_r of the HTC obtained by the empirical formula, so each stage P_r of HTC is equal, so the P_r of the two-stage compression system is significantly lower than that of the single stage compression system. When T_e varies between −80 °C and −50 °C, the P_r of each stage of the two-stage compression is between 4.8 and 3.4, which is greatly reduced from the original 11.1 to 7.7, which is reduced by 55.51% and 56.82%.

Moreover, the discharge temperature T_d of HTC compressors using two-stage compression is significantly lower than that of single stage compression system, as shown in Fig. 9(b). The T_d decreases with the increase of the T_e. In addition, the T_d corresponding to the two HTC compressors of two-stage compression is greatly lower than that of the single stage. When T_e varies between −80 °C and −50 °C, the T_d is 127.40 ~ 102.23℃ and 82.21 ~ 65.76℃ lower than that of the single stage compression system respectively, which is beneficial to the operation of the system. The R170 and R41 systems are similar. Therefore, the following uses NH_3/R1150, R170, R41 CRS with two-stage compression in HTC (TSCRS) and R404A/R23 CRS with IHX in HTC (IHCRS) to compare the refrigeration performance, and comprehensively analyze the refrigeration advantages of environmentally friendly refrigerants.

5.4. Influence of evaporation temperature T_e

The variation trend of the optimal COP corresponding to each system with the T_e is shown in Fig. 10. It can be seen from the figure that the optimal COP of each system increases with the increase of the T_e. The reason is that the T_k is constant. The smaller the temperature difference, the smaller the compressor *Pr* and the smaller the power consumption.



(a) COP and P_r with T_e (b) T_d with T_e

Fig. 9. Variation of parameters of NH_3/R1150 TSCRS with T_e compared with single stage compression system.

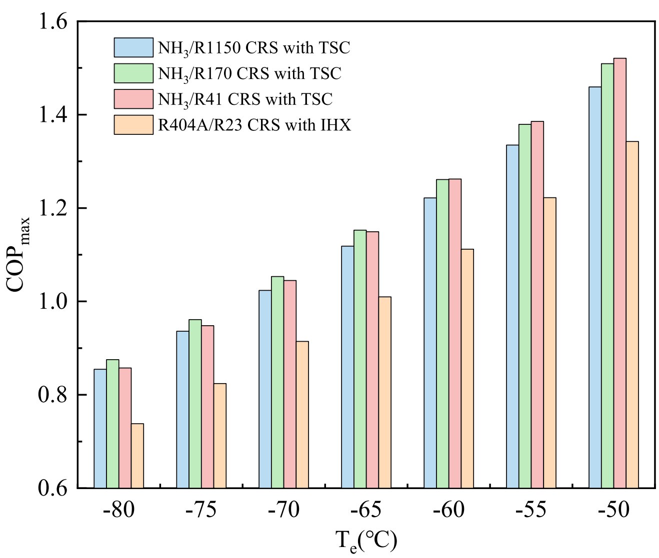


Fig. 10. Variation of COP_{max} with T_e of different CRS system.

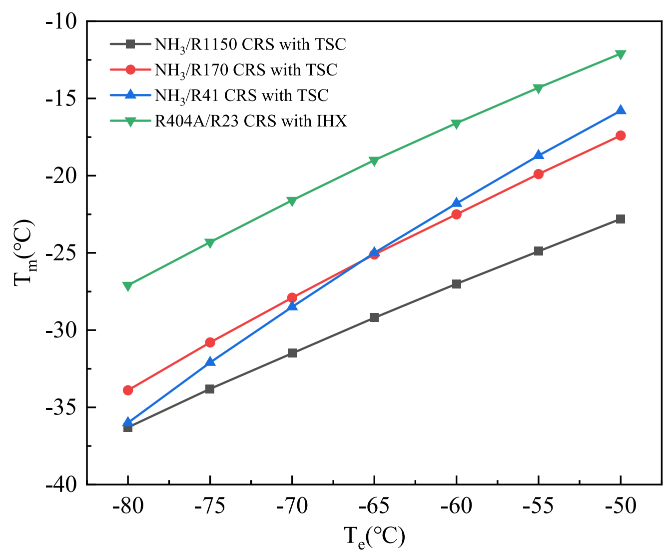


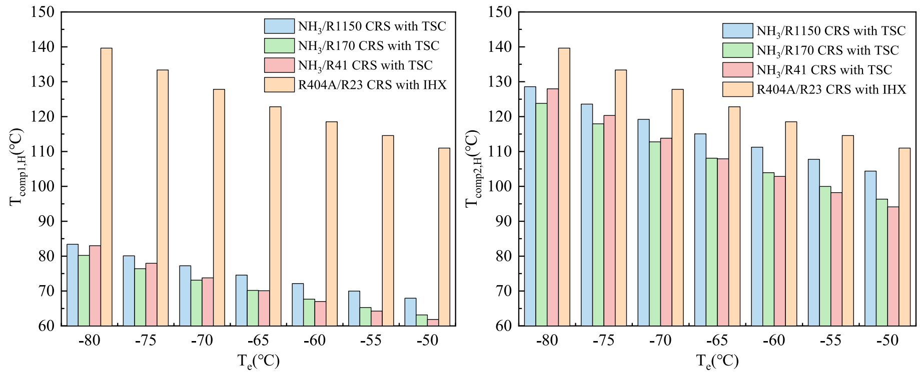
Fig. 11. Variation of optimal T_m with T_e of different CRS system.

Under the condition of a certain cooling capacity, the smaller the power consumption and the higher the COP. And the best COP of TSCRS with NH_3/R1150, R170, R41 is higher than that of IHCRS with R404A/R23. When the T_e changes from −80℃ to −50℃, the highest COP of R1150, R170 and R41 are increased by 15.79%, 18.58% and 16.17% respectively. The improvement rate of R1150 system is slightly lower than that of the R170 and R41 systems. When the T_e is less than −62℃, the R170 system has better cooling effect than the R41 system. When the T_e is higher than −62℃, the R41 system shows better cooling performance than the R170. Although R1150 is inferior to R170 and R41 systems, its cooling performance is also improved compared to IHCRS with R404A/R23. Therefore, the proposed TSCRS is advantageous, in which HTC uses NH_3, and LTC uses R170 for the best cooling effect.

Fig. 11shows the variation trend of the optimal T_m with the T_e. It can be seen from the figure that with the increase of T_e, the corresponding optimal T_m of each system increases. This is because when the T_k is constant, the higher the T_e, the smaller the temperature difference between stages, and the higher the corresponding optimal T_m. Among them, the optimal T_m of TSCRS is lower than that of IHCRS. When the T_k is 40 °C and the T_e is increased from −80 °C to −50 °C, the R1150, R170 and R41 systems decrease by 10.7 °C, 6.8 °C and 8.9 °C respectively. Overall, the T_m of R1150 is slightly lower than the other two refrigerants. And as the T_e decreases, the T_m of R41 decreases faster. When the T_e is less than −66 °C, the T_m of R170 is higher than that of R41, and when the T_e is higher than −66 °C, the optimum T_m corresponding to R41 is greater than that of R170. When the T_m is lower and the T_k remains constant, the compressor P_r of HTC will increase accordingly.

Compressor discharge temperature T_d is also a parameter to be concerned about in design. Over-high discharge temperature will damage the compressor, reduce its life, and be detrimental to the cycle. Considering that the T_d of LTC compressor is generally low, the T_d of HTC is mainly considered. The variation trend of T_d of the first stage and second stage compressors of HTC is shown in Fig. 12, the T_d of each system decreases with the increase of T_e. The reason is that when the T_k is fixed, with the increase of the T_e, the temperature difference between the stages decreases, and the P_r decreases, so the T_d decreases. Fig. 12 (a) shows the variation trend of the T_d of HTC first stage compressor with the T_e. It can be seen that the T_d of TSCRS is significantly lower than that of IHCRS, and the maximum reduction of R1150, R170 and R41 systems is 56.2 °C, 59.4 °C and 56.7 °C, And when T_e is less than −65℃, the T_d of the HTC first stage compressor of the R41 system is higher than that of R170 system. When the T_e is higher than −65℃, the T_d of the R170 system will be higher than R41 system. Fig. 12 (b) shows the effect of T_e on the T_d of HTC second stage compressor. It can be seen that the T_d decreases with the increase of the T_e. The T_d of TSCRS with NH_3/R1150, R170 and R41 is slightly lower than that of IHCRS with R404A/R23. When the T_e changes from −80 °C to −50 °C, the T_d decreases by 11.0 °C, 15.8 °C and 16.9 °C, respectively.

Fig. 13 shows the variation of compression ratio with T_e at LTC and HTC. The pressure ratio of LTC increases with the increase of T_e, while



(a) compressor 1 in HTC (b) compressor 2 in HTC

Fig. 12. Variation of optimal T_d of HTC compressors with T_e of different CRS system.

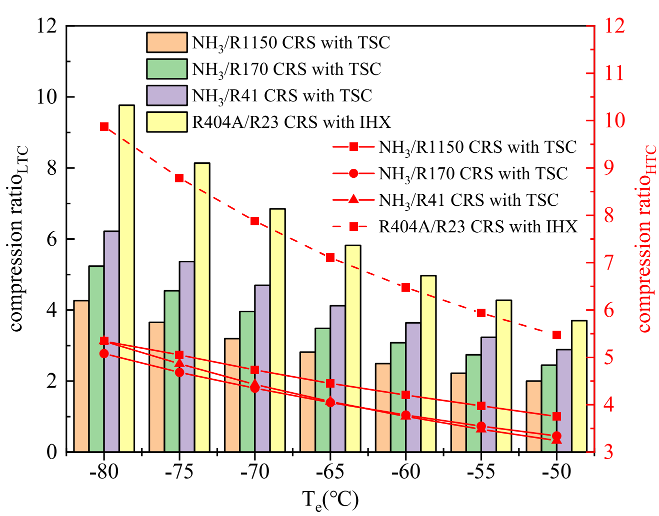


Fig. 13. Variation of compression ratio with T_e of different CRS system.

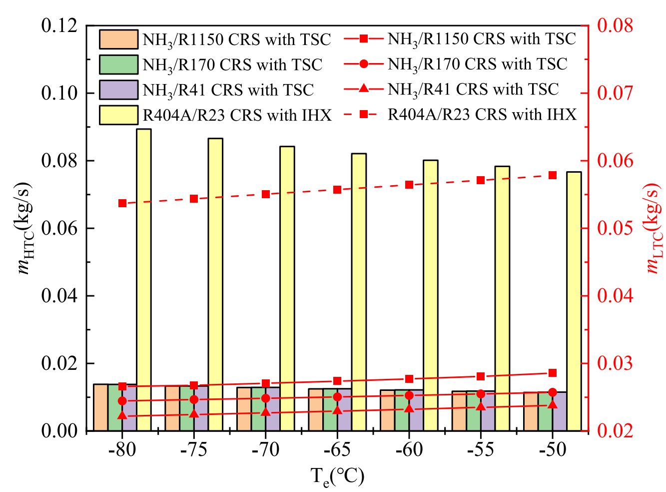


Fig. 14. Variation of mass flow rate with T_e of different CRS system.

that of HTC is the opposite. When the T_e increased from −80℃ to −50℃, the compression ratio of R404A/R23 IHCRS at LTC decreased from 9.8 to 3.7, and the compression ratios of NH_3/R1150, R170 and R41 TSCRS decreased by 45.95%~56.12%, 35.14%~46.94% and 21.62%~37.67%, respectively. Similarly, the HTC compression ratio is also obviously reduced. Compared with R404A/R23 IHCRS, the TSCRS of NH_3/R1150, R170 and R41 are reduced by 41.82%~46.46%, 40.40% ~48.48% and 46.46%~63.64%, respectively, The compression ratio of the improved system is obviously lower than that of the original system, and the system runs more safely.

Based on the above analysis of compression ratio and discharge temperature, and a large number of ultra-low temperature experiments conducted by scholars, all components of the system can operate normally at the low temperature as low as −80℃. Some companies have built refrigeration systems with low temperature as low as −100℃ using environmentally friendly refrigerants and put them into use, which confirms the possibility of heat exchangers and compressors operating at ultra-low temperature and has good development prospects.

Fig. 14 shows the variation of mass flow rate of HTC and LTC refrigerant with evaporation temperature. It can be seen that with the increase of T_e, the mass flow rate of HTC in each system decreases. When the T_e increases from −80℃ to −50℃, the HTC mass flow rate of TSCRS does not change obviously, and that of IHCRS decreases from 0.089 kg/s to 0.077 kg/s. The HTC mass flow rates of NH_3/R1150, R170 and R41 decrease by 84.53%~85.10%, 84.58%~85.01% and 84.56%~84.99%, respectively, compared with R404A/R23 IHCRS. Similarly, the mass flow rate of LTC increases with the increase of T_e. When the T_e increases from −80℃ to −50℃, the IHCRS increases from 0.054 kg/s to 0.058 kg/s, and NH_3/R1150, R170 and R41 are 50.48%~50.61%, 54.49% ~55.52% and 58.68%~58.87% lower than R404A/R23 IHCRS, respectively. The improved new system greatly reduces the refrigerant charge, which is beneficial to reduce the system cost.

The change of compressor power consumption W and total exergy destruction E_x with T_e is shown in Fig. 15. It can be seen that when the T_e increases from −80 °C to −50 °C, the optimal W of each system decreases. In addition, the W of TSCRS used NH_3/R1150, R170, R41 is lower than that of IHCRS used R404A/R23, with the highest reductions of 13.64%, 15.67% and 13.92% respectively. When the T_e is lower than −61 °C, the W corresponding to the optimal COP of the R41 system is higher than that of R170, and when the T_e exceeds −61℃, the W of the R170 system is higher than that of the R41 system.

As for total exergy destruction E_x, it can be seen from the figure that the optimal E_x of NH_3/R1150, R170, R41 TSCRS and R404A/R23 IHCRS decrease with the increase of T_e. The T_d is also reduced, the temperature difference between condenser and cascade heat exchanger is reduced, so the E_x in the heat transfer process is reduced. And the E_x of TSCRS with R1150, R170 and R41 is reduced to different degrees than that of IHCRS with R404A/R23, the highest reduction is 21.73%, 24.97% and 22.19% respectively. When the T_e is lower than −61℃, the E_x of R41 system higher than that of R170, and E_x of R170 is higher than that of R41 when the T_e is higher than −61 °C.

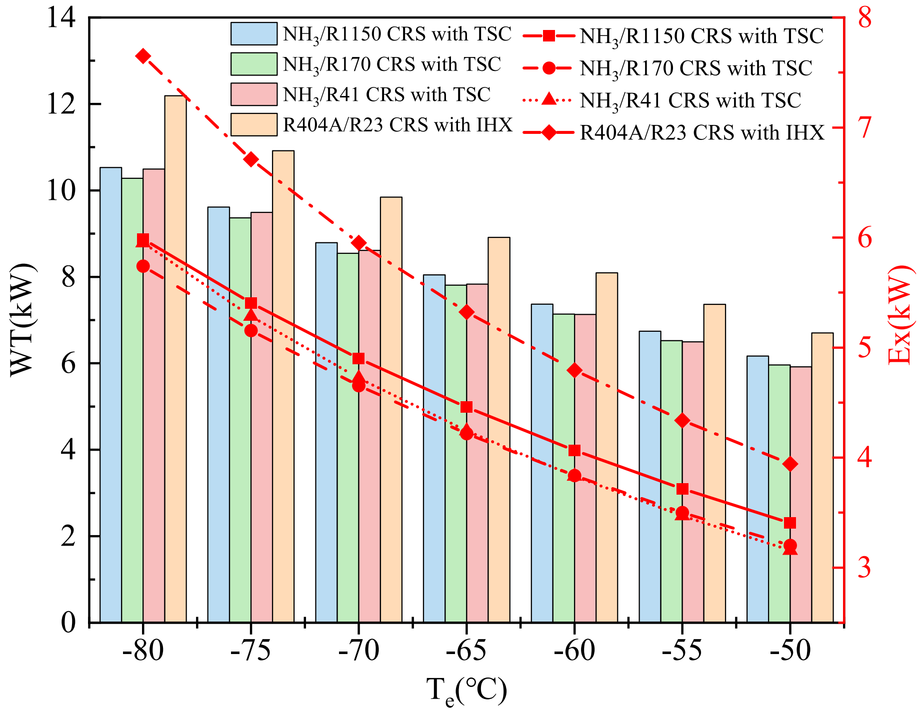


Fig. 15. Variation of optimal W with T_e of different CRS system.

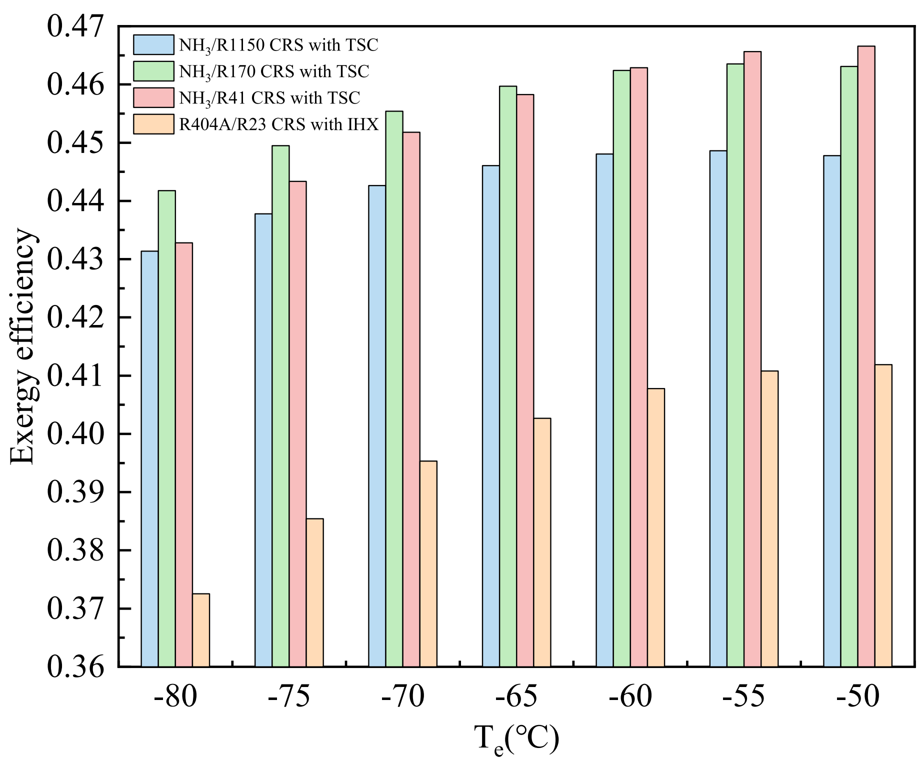
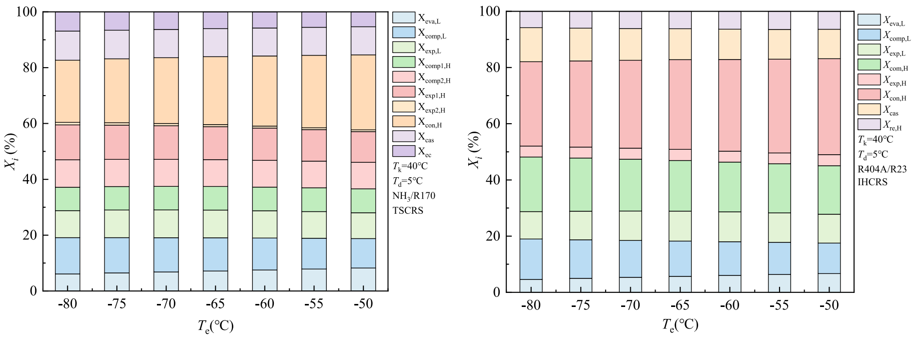


Fig. 16. Variation of optimal exergy efficiency with T_e of different CRS system.

In addition to COP, exergy efficiency is also a parameter to judge whether a refrigeration system has superior performance. Fig. 16 shows the variation trend of the optimal exergy efficiency corresponding to the optimal T_m of each system with the T_e. When T_e varies from −80 °C to −50 °C, the TSCRS of R1150, R170 and R41 are all improved compared with the IHCRS of R404A/R23, and the optimal exergy efficiency is increased by 15.79%, 18.57% and 16.17% respectively. It can be seen that as the T_e increases, the optimal exergy efficiency of each system first increases and then decreases. There is a maximum value. The maximum value corresponding to different refrigerant combinations is different. For example, when T_e of the NH_3/R1150 system changes from −80℃ to −50℃, the maximum optimal exergy efficiency is 44.86%, while that of the NH_3/R1150 system is 46.35%. When T_e is higher than −62℃, the exergy efficiency of the R41 system is the largest, while when T_e is lower



(a)NH_3/R170   (b)R404A/R23

Fig. 17. Variation of X ratio of different components with T_e of different CRS system.

than −62℃, the exergy efficiency of the R170 system is higher. When T_e decreases from −62 °C to −80 °C, the exergy efficiency of the R41 system decreases rapidly, from 0.461 to 0.433, while the R170 decreased more slowly, from 0.461 to 0.442.

The exergy destruction ratio of each component of the system is also very important for evaluating the system performance. Fig. 17 shows the variation trend of the exergy destruction ratio of different components of the NH_3/R170 TSCRS system and the R404A/R23 IHCRS system with T_e. As can be seen from Fig. 17 (a), for the TSCRS system of NH_3/R170, with the increase of T_e, the exergy destruction proportions of evaporator, HTC first stage compressor and condenser all increase, and the increase of condenser is the largest, when T_e increases from −80℃ to −50℃, the proportion of condenser increases from 22.3% to 26.9%. The proportion of exergy destruction of the LTC expansion valve shows a trend of first increase and then decrease with the increase of T_e. Among the components, the condenser exergy destruction accounts for the highest proportion, and the HTC second stage expansion valve accounts for the smallest proportion. When T_e changes between −80℃ and −50℃, the proportion only changes from 0.9% to 0.6%. The proportion of the rest of the components is relatively concentrated, the proportion of the evaporator has changed from 6.1% to 8.2%, the proportion of LTC compressor exergy destruction has been reduced from 13.0% to 10.5%, the LTC expansion valve, the HTC compressors, HTC first stage expansion valve, and cascade heat exchanger all change slightly, and the proportion of economizer exergy destruction changes from 6.9% to 5.3%. The same is true for the IHCRS system of R404A/R23 shown in Fig. 17 (b), in which the component with the highest proportion is the condenser, which increases with the increase of the T_e. When the T_e changes from −80 °C to −50 °C, the proportion of condenser exergy destruction increases from 30.01% to 34.15%, which is much higher than the proportion of condenser exergy destruction in NH_3/R170 TSCRS system, and the proportion of other components changes slowly with T_e. Therefore, it can be seen from the above two systems that the exergy destruction of the condenser components in the CRS accounts for the largest proportion, and the proportion gradually increases with the increase of the T_e. The reason is that the compressor T_d is much higher than the T_k, and there will be irreversible heat loss, and it accounts for the main part of the total exergy destruction of the system. Therefore, future research can focus on the improvement of compressor T_d to reduce the proportion of exergy destruction of condenser, thereby improving exergy efficiency and improving refrigeration performance.

5.5. Influence of superheating T_{sup} and subcooling T_{sub}

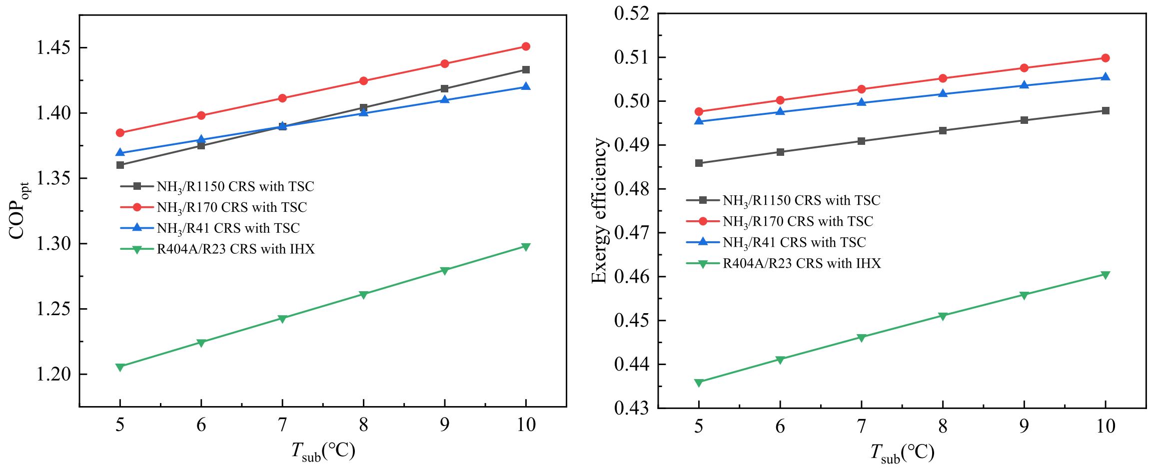
In actual operation, subcooling and superheating will affect the performance of the system. Fig. 18 shows the changing trend of COP and exergy efficiency of the system when the degree of superheat is from 2 to 8℃. Fig. 18 (a) shows that when the degree of superheat increases, the COP of each system decreases. When T_e is −60℃, T_k is 40℃, COP of NH_3/R170 CRS with TSC decreases from 1.292 to 1.218, reduced by 5.73%. Similarly, COP of R404A/R23 CRS with IHX decreases from 1.102 to 1.075, reduced by 2.45%. Fig. 18 (b) shows the changing trend of exergy efficiency with the degree of superheat, NH_3/R170, R1150, R41 CRS with TSC all decrease while R404A/R23 CRS with IHX increases·NH_3/R170 CRS with TSC decreases from 47.98% to 46.86%, reduced by 2.33%, R404A/R23 CRS with IHX increases from 40.85% to 41.00%, increased by 0.36%. It can be seen that overheating is unfavorable to the system performance on the whole.

Similarly, Fig. 19 gives the influence of the degree of supercool on the system performance. It is not difficult to find that with the increase of the degree of supercool, the COP and exergy efficiency of each system increase. Fig. 19 (a) shows that when T_e is −60℃, T_k is 40℃, T_{sub} increases from 5 to 10℃, COP of NH_3/R170 CRS with TSC increases from 1.385 to 1.491, increased by 4.78%. R404A/R23 CRS with IHX increases from 1.206 to 1.298, increased by 7.63%. In the same way, Fig. 19 (b) shows the changing trend of exergy efficiency with T_{sub}. When T_{sub} increases from 5 to 10℃, exergy efficiency of NH_3/R170 CRS with TSC increases from 49.76% to 50.98%, increased by 2.45%. To sum up, supercooling is beneficial to the system. Adding supercooling device can effectively improve the refrigeration performance of the system.

6. Conclusions

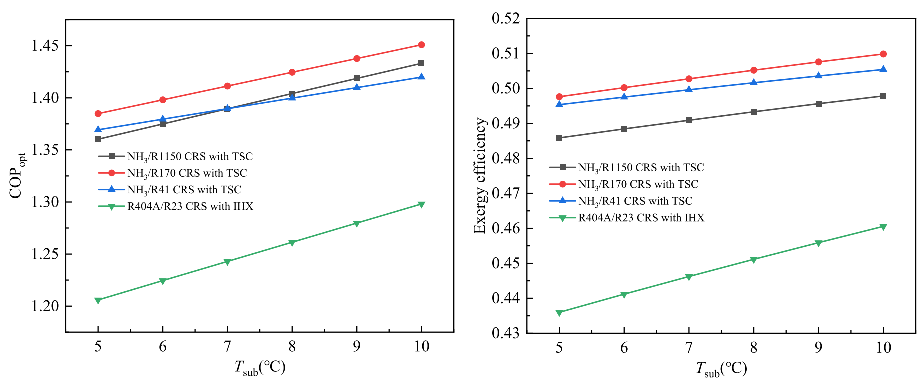
In this paper, aiming at the low temperature refrigeration requirement applied to the T_e of −80℃~-50℃, theoretically analysis are conducted on the performance of CRS using NH_3 as refrigerant in HTC and environmentally friendly refrigerants R1150, R170, R41 in LTC. And comparison analysis with the conventional R404A/R23 CRS system are also presented. The impact of IHX on system performance is discussed and a CRS with two-stage compression in HTC (TSCRS) are constructed to reduce pressure ratio and discharge temperature of compressors. According to the principle of thermodynamics and the law of conservation of energy, a mathematical model for system analysis is established to analyze the influence of working parameters on system performance. The main conclusions are as follows:

(1) The COP of each CRS increases first and then decreases with the intermediate temperature T_m, there is an optimal T_e, to make the COP reach the maximum value. For example, when the T_e is −60 °C and the T_k is 40 °C, taking the conventional CRS as an example, the optimal T_m corresponding to R404A/R23 is −10.0 °C, and the COP is 1.018 at this time. The optimal T_m of the NH_3/R1150, R170 and R41 CRS are −16.2 °C, −12.1 °C and −10.9 °C, and the corresponding maximum COPs are 1.088,



(a)COP (b)Exergy efficiency

Fig. 18. Variation of system performance with T_{sup} of different system.



(a)COP (b)Exergy efficiency

Fig. 19. Variation of system performance with T_{sub} of different system.

1.153 and 1.160, respectively, increase by 6.88%, 13.26% and 13.95%.

(2) When using R1150, R170 and R41 for LTC and NH_3 for HTC, it is not recommended to use IHX. It will reduce the COP of the system used environmentally friendly refrigerants. For example, when the T_e is −80 °C, the T_k is 40 °C, and the IHX effectiveness of HTC and LTC are 1, the COP of the R1150, R170 and R41 systems are reduced by 8.03%, 6.99% and 12.68% compared with CRS without IHX, all COP of CRS with R1150, R170 and R41decrease with increasing IHX effectiveness. While for the R404A/R23 system, CRS with IHX in HTC (IHCRS) can improve the cooling performance. When the T_e is −80 °C and the T_k is 40 °C, the COP is increased by 12.63% compared with the conventional CRS.

(3) The use of TSCRS can avoid the excessive pressure ratio and the high discharge temperature of HTC NH_3 compressor, at the same time improve the performance of the system. When the T_k is 40 °C, taking the NH_3/R1150 system as an example, when the T_e varies between −80 °C and −50 °C, compared with the HTC system with single stage compression, the discharge temperature of TSCRS are respectively decrease by 127.4 ~ 102.2℃ and 82.2 ~ 65.8℃, and the system COP increased by 10.90%~16.82%.

(4) Compared with the IHCRS using R404A/R23, the TSCRS with NH_3/R1150, R170 and R41 have different degrees of improvement in the optimal COP. When the T_e is changed between −80℃ and −50℃, R1170, R170 and R41 system has the highest increase of 15.79%, 18.58% and 16.17% respectively, so the R170 has the best performance.

(5) The exergy efficiency of TSCRS of NH_3/R1150, R170 and R41 is significantly higher than that of IHCRS with R404A/R23 as refrigerant. When the T_e is −60℃ and the T_k is 40℃, the COP of former increase by 9.87%, 13.39% and 13.51%, respectively. The TSCRS exergy destruction of the environmentally friendly refrigerant is lower than that of the IHCRS of R404A/R23. When the T_k is 40 °C and the T_e is changed in the range of −80 to −50 °C, the maximum exergy reduction is 21.73%, 24.97% and 22.19% respectively. The cooling effect to R170 is more advantageous than the other two refrigerants.

(6) With the increase of T_e, the proportion of exergy destruction of evaporator and condenser increase, and the proportion of condenser is the largest. Taking TSCRS using NH_3/R170 as an example, when the T_e changes from −80℃ to −50℃, the proportion of condenser exergy destruction changes from 22.3% to 26.9%, so in the future, research can focus on reducing condenser exergy destruction to improve system refrigeration performance.

The economic analysis and experimental research are very important step before the application of the TSCRS cycle using environmentally friendly refrigerants. Therefore, the economic analysis and experimental research on the proposed refrigeration cycle needs to be investigated in the further research.

CRediT authorship contribution statement

Muqing Chen: Methodology, Validation, Investigation, Writing – original draft, Writing – review & editing. **Qichao Yang:** Conceptualization, Investigation, Resources, Writing – review & editing, Supervision, Funding acquisition. **Benlin Shi:** Validation. **Xiaonan Chen:** Resources. **Weikai Chi:** Software. **Guangbin Liu:** Visualization. **Yuanyang Zhao:** Supervision, Funding acquisition. **Liansheng Li:** Project administration.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Data availability

Data will be made available on request.

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