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Effect of an IHX on energy and exergy performance of a modified double-evaporator transcritical CO_2 heat pump with ejector

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Abstract

At present, some studies have analyzed the effect of internal heat exchanger (IHX) on the performance of transcritical CO2 heat pump system with ejector (TCHSE). However, the effect of IHX on better-performing double-evaporator transcritical CO2 heat pump system with ejector (DE-TCHSE) have not yet been studied. In this paper, a modified DE-TCHSE simulation model based on variable efficiency ejector was established by Modelica, the model was tested and analyzed, and the energy and exergy performance of DE-TCHSE by IHX was conducted under different working conditions. The simulation results show there exists an optimal discharge pressure which make the coefficient of performance (COP) of TCHSE and DE-TCHSE maximum, and the COP of the latter was up to 21.5% higher than that of the former under ambient temperature of 5~20℃. The addition of an IHX results in lower optimal discharge pressure and higher COP, and optimal discharge pressure decreased by up to12.3% while the COP increased by up to 12.9%. With the larger ejector throat area and higher ambient temperature, the IHX becomes more effective in improving system COP and lowering the optimal discharge pressure, at the same time, the reduction of optimal discharge pressure also could be amplified by the increase of water outlet temperature and the decrease of water inlet temperature. In addition, the ejector efficiency decreased by up to 8.5% due to the supercooling effect of the IHX, and increasing the IHX area can weaken this effect. From exergy analysis point of view, the exergy destruction percentages of ejector significantly decreased with the application of IHX. Furthermore, the product exergy and the total exergy destruction decreased, especially the exergy efficiency of DE-TCHSE increased in all working conditions.

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Keywords: dual-evaporator, IHX, CO2, variable efficiency ejector, exergy analysis;

1. Introduction

The global demand for cooling and heating continues to rise due to population growth, economic development, and climate change, with the net-zero emissions target by 2050 driving the widespread adoption of heat pumps supported by relevant policies[1]. Concurrently, environmental regulations are phasing out CFCs and HCFCs in favor of natural refrigerants like CO_2[2]. CO_2 heat pumps are gaining prominence as CO_2 is a natural, A1-class refrigerant whose transcritical temperature glide well matches the high temperature lifts required for water heating, offering a sustainable and efficient solution[3], leading to the widespread application of the transcritical CO_2 cycle[4]. Studies, such as those by Saikawa et al.[5] has confirmed the superior heating performance of transcritical CO_2 systems.

However, a major challenge for transcritical CO_2 systems is the significant expansion loss. A common solution is to use an ejector to replace the expansion valve (Fig. 1a). Theoretical and experimental studies by Li and Groll[6] and Lee et al.[7] demonstrated COP improvements of 16% and 15%, respectively, while

Liu et al.[8] optimized performance via an adjustable ejector. Beyond component optimization, advanced cycles like the dual-evaporator system studied by Elakdhar et al.[9] (Fig.1b) and the two-stage evaporation cycle (EERC-TE) proposed by He et al.[10] have shown further performance gains. Zheng et al.[11] developed a dynamic model for the EERC-TC, achieving a 2.58 times higher COP under optimal control. Nonetheless, these advanced systems have been predominantly investigated for cooling, with their heating potential largely unexplored.

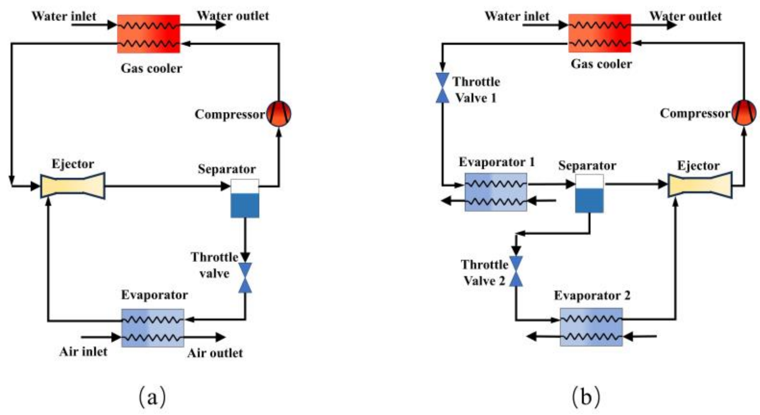


Fig.1. Schematic diagram of the TCHSE (a) and dual-evaporator ejector refrigeration system (b)

Another approach to recover expansion work is to reduce the gas cooler outlet temperature, such as through mechanical subcooling[12] or a thermoelectric subcooler, which increased COP by 11.3%[13]. A more widely adopted and simpler method is the internal heat exchanger (IHX), which subcools the refrigerant after the gas cooler, reducing throttling losses and ensuring compressor superheating. Since its introduction by Lorentzen and Pettersen[14], the IHX has been experimentally validated and optimized in transcritical CO_2 systems[15]. In ejector-based systems, Guruchethan et al.[16] reported a 12% COP increase and improved ejector efficiency with an IHX. However, its benefits are not universal; Zhang et al.[17] identified a transitional discharge pressure for positive IHX impact, while Sarkar et al.[18] found its effectiveness to be highly dependent on gas cooler outlet temperature.

In summary, while the IHX has been studied in conventional ejector-based heat pumps, its role in advanced, higher-performance dual-evaporator ejector systems remains unclear. Moreover, studies on the impact of IHX size and the use of variable-efficiency ejector models in such systems are scarce.

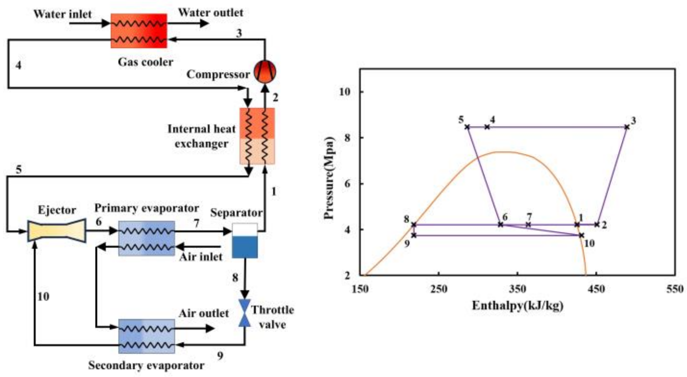
In this work, a modified double-evaporator transcritical CO_2 heat pump with ejector (DE-TCHSE) is proposed for heating. The main contributions are:

- A modified DE-TCHSE simulation model based on variable efficiency ejector established by Modelica, which is up to 21.5% higher than the COP of the conventional ejector-based heat pump system.
- Under the optimal discharge pressure, the effects of ejector throat area, ambient temperature, water inlet temperature and water outlet temperature on the system performance and ejector performance were studied, at the same time, the influence of these parameters on the role of IHX in the system is also analyzed.
- The influence of the IHX on the exergy destruction of each component in the DE-TCHSE under the optimal discharge pressure was studied, and the changes in system exergy efficiency under different operating conditions were obtained.

2. System description and assumptions

The modified double-evaporator transcritical CO_2 heat pump with ejector is illustrated in Fig. 2. This heat pump system consists of the following components: the gas cooler, double evaporators, compressor, expansion valve, variable efficiency ejector, internal heat exchanger, water pump, evaporator fan, separator, and double PI controllers, where PI controllers are adopted to maintain the water outlet temperature and degree of superheat of suction flow according to the setpoints by excuting the water pump and expansion valve.

The system cycle can be represented as the following: the high-temperature and high-pressure CO_2(state 3) flows out of the gas cooler and enters the internal heat exchanger for heat exchange, then flows into the ejector and is depressurized and accelerated inside the motive nozzle (state 6), meanwhile the low-temperature and low-pressure CO_2(state 13) from the secondary evaporator to the suction nozzle, then both of them are mixed and go through the diffuser. Next, the diffuser stream (state 9) is evaporator in the primary evaporator, then two-phase fluid (state 10) is sent into the separator and divided into saturated liquid (state 11) and saturated vapor (state 1). The saturated vapor enters the internal heat exchanger for heat exchange (state 2), and then is compressed and flows into the gas cooler. The saturated liquid leaving from the separator undergoes a pressure drop in the expansion valve (state 12) and then enters the secondary evaporator to transfer heat (state 13), which is then injected into the ejector to complete the cycle.



(a) (b)

Fig.2. Schematic of the DE-TCHSE and corresponding P-h diagram

The following assumptions have been taken into account for the proposed system:

- The system reaches a steady state, and pressure changes in the whole system only occur in the ejector, compressor and throttle valve, the rest of the pressure loss are neglected;
- The temperature of air and water remain unchanged when they pass through the fan and pump;
- Heat losses to the environments through heat exchangers, the compressor, valves, and separator are also neglected;

3. Modeling of system

3.1 Energy model

The system energy model, comprising the gas cooler, internal heat exchanger, evaporators, ejector, compressor, expansion valve, and separator, was developed using fundamental components from the TIL Library[19]. For detailed modeling procedures and validation, the reader is referred to the established methodologies in references[20]. The design specifications of models are listed in Table 1.

Table 1 Design specifications of major components of DE-TCHSE

Components	Design specifications
Compressor	Scroll compressor; displacement:15ml; variable speed
Evaporator	Fin-and-tube heat exchanger; staggered copper; tube diameter: 7 mm; wall thickness: 1.5mm; number of serial tubes: 5; number of parallel tubes: 8; thickness of fin: 0.2mm; number of parallel tube side flows: 4; tube length: 0.6 m. Number of evaporator: 2
Gas cooler	Tube-and-tube exchanger; counter-flow; tube length:20m; number of parallel tubes: 2; inner diameter: 1cm; wall thickness: 0.2mm; wall: copper.
IHX	Tube type; counter-flow; Variable IHX area: 0,0.3,0.6,0.9 m^2; wall: copper.
Water pump	Nominal water mass flow rate: 0.047 $kg \cdot s^{-1}$
Evaporator fan	Nominal air volume flow rate: 3600 $m^3 \cdot h^{-1}$
Expansion valve	Maximal effective flow area: $1.5\times10^{-6} m^2$;
Ejector	Adjustable nozzle throat area; Nominal ejector throat area: 0.4mm^2; ejector efficiency: 0.35
Separator	Nominal volume: 2L; initial filling level: 0.5

For the performance of the system, the thermodynamic parameters of the system are introduced to evaluate the system. The heating capacity $\dot{Q}$ is determined by the inlet and outlet water temperature and mass flow rate of the gas cooler and is calculated :

$$\dot{Q} = \dot{m}_{water} c_p \left(T_{water,out} - T_{water,in}\right) = \dot{m}_{ref}(h_5 - h_3) \tag{1}$$

The total power consumption is comprised of those for the compressor and evaporator fan and water pump, i.e.

$$P_{total} = P_{comp} + P_{pump} + P_{fan} \tag{2}$$

Then the COP is calculated as:

$$COP = \frac{\dot{Q}}{P_{total}} \tag{3}$$

Finally, the entrainment ratio and the pressure lift ratio are significant metrics for evaluating the performance of the DE-TCHSE. The entrainment ratio μ can be defined as the ratio of the CO_2 mass flow rate at the suction port to the drive port. The pressure lift ratio Π represents the ratio of the ejector outlet pressure to the suction flow pressure.

$$\mu = \frac{\dot{m}_{suc}}{\dot{m}_{dis}} \tag{4}$$

$$\Pi = \frac{p_{dis}}{p_{suc}} \tag{5}$$

3.2 Exergy model

Furthermore, an exergy analysis model was developed in Dymola. The modeling approach, detailed in reference[22], was employed to determine the exergy destruction of each component, as presented in Table 2.

Table 2 The exergy destruction for each component.

Components	Exergy destruction
Compressor	$E_{D,comp} = T_0(s_{comp,out} - s_{comp,in})$
Gas cooler	$E_{D,\mathrm{gc}} = h_{\mathrm{gc},in} - h_{\mathrm{gc},out} - T_0(s_{\mathrm{gc},in} - s_{\mathrm{gc},out}) - E_p$
IHX	$E_{D,IHX} = T_0[(s_{IHX,out,HS} - s_{IHX,in,HS}) + (s_{IHX,out,LS} - s_{IHX,in,LS})]$
Ejector	$Ex_{\mathrm{des,eje}} = t_0[(\dot{m}_\mathrm{p} + \dot{m}_\mathrm{s})s_2 - \dot{m}_\mathrm{s} s_1 - \dot{m}_\mathrm{p} s_6]$
Evaporator 1	$E_{D,\mathrm{evap}} = h_{\mathrm{evap},in} - h_{\mathrm{evap},out} - T_0(s_{\mathrm{evap},in} - s_{\mathrm{evap},out})$
Evaporator 2	$E_{D,evap} = h_{evap,in} - h_{evap,out} - T_0(s_{evap,in} - s_{evap,out})$
Expansion valve	$E_{D,\mathrm{exp}} = h_{\mathrm{exp},in} - h_{\mathrm{exp},out} - T_0(s_{\mathrm{exp},in} - s_{\mathrm{exp},out})$

3.3 Model validation

To verify the accuracy of the proposed DE-TCHSE model, the experimental results of Zhu et al.[23] are used to verify the overall system model. The experimental setup is based on TCHSE shown in Fig. 1(a), with the addition of an internal heat exchanger. The only difference from DE-TCHSE is that the experimental setup of Zhu[23] has only an evaporator. Therefore, to maintain consistent conditions for the reference experiment, the primary evaporator was removed for model validation. The water inlet temperature is fixed at 20°C and design parameters of the main components are consistent with the references[23]. The CO_2 temperature at the gas cooler exit ($T_{gc,out}$), heating capacity and COP are simulated for water outlet temperatures ranging from 50°C to 90°C. Comparing the simulation results with the experimental data, the errors are shown in Fig.3， and the relative errors of all the data are kept within 10%, which to some extent validates the effectiveness of the developed model.

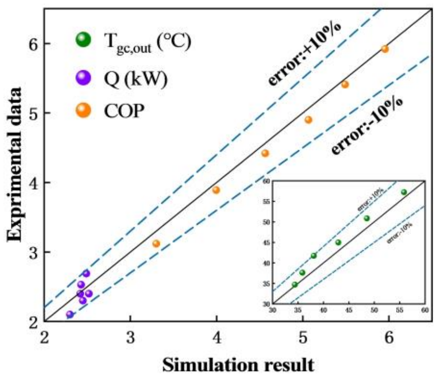


Fig.3. Validation of the DE-TCHSE model

4. Results and discussions

In this study, the simulation is performed based on the following operating conditions: the ejector throat area varies from 0.4 mm^2 to 1.0 mm^2 with a default of 0.8 mm^2, the ambient temperature ranges from 5 ℃ to 20 ℃ with a default of 10 ℃, the water inlet temperature varies from 5 ℃ to 20 ℃ with a default of 10 ℃, the water outlet temperature ranges from 55 ℃ to 70 ℃ with a default of 60 ℃.

Fig.4 shows the performance comparison between the DE-TCHSE and TCHSE under different operating conditions. As shown in the Fig.4(a), both systems exhibit an optimal discharge pressure, which makes the COP of the system maximum. Although the optimal discharge pressure of the DE-TCHSE is slightly higher than that of the system TCHSE, increasing from 94.2 bar to 97.4 bar under the default operating conditions, DE-TCHSE has a higher COP at all simulated discharge pressures and the COP corresponding to optimal discharge pressure is 7.2% higher. Fig.4(b) shows the variation of the systems COP and heating capacity COP and heat production corresponding to optimal discharge pressure with the change of ambient temperature. The results show that the COP and heating capacity of DE-TCHSE are better under all simulated conditions, and the COP enhancement effect is more obvious with higher ambient temperature.

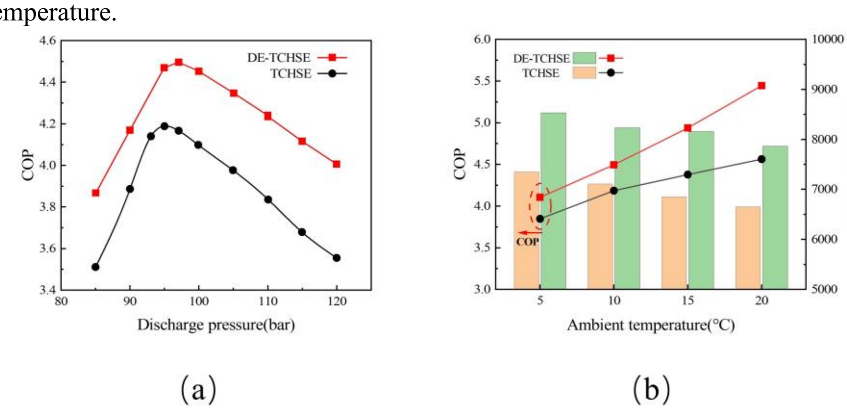


Fig.4 The performance comparison between the DE-TCHSE and TCHSE

Fig.5 shows the variation of COP and exergy efficiency with change of the discharge pressure for the DE-TCHSE with and without an IHX under the default operating conditions, where the IHX area is set to 0.3 m^2. The results show that regardless of whether an IHX is included in the DE-TCHSE, both systems exhibit an optimal COP and optimal exergy efficiency, which correspond to the system's optimal discharge pressure. It can also be observed that the inclusion of the IHX increases the system's optimal COP and significantly reduces the optimal discharge pressure. Under the default operating conditions, the optimal COP of the system with an IHX increased by approximately 3.5%, while the optimal discharge pressure decreased from 97 bar to 85 bar, a reduction of about 12.3%.

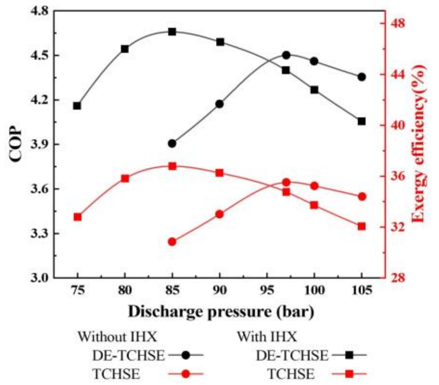


Fig.5 Variation of COP and exergy efficiency with discharge pressure for the DE-TCHSE with and without an IHX

4.1 Impact of ejector throat area

In order to analyze the impact of the ejector throat area on the role of the IHX in the system, Fig.6 displays variation the COP, heating capacity and optimal discharge pressure under different ejector throat area. The results show that regardless of whether the system includes an IHX, the system COP decreases and the heating capacity increases with rising ejector throat area, meanwhile, increasing the IHX area can enhance these effects, as shown in the Fig.6(a). This is primarily due to the increase in CO2 mass flow rate at the compressor inlet, which enhances the heating capacity but also leads to a greater rise in system power consumption, and the rate of power consumption increase surpasses the growth in heating capacity. In addition, with a larger ejector throat area, the internal heat exchanger becomes more effective in improving system COP and decreases heating capacity. The Fig.6(b) reveals that the optimal discharge pressure is essentially unaffected for change of the ejector throat area when the IHX is not used. However, when the IHX is integrated into the system, the optimal discharge pressure initially decreases and then stabilizes as the ejector nozzle area increases. Similarly, increasing IHX area can further reduce the optimal discharge pressure, but the reduction is limited to a certain maximum range. Furthermore, at higher ejector throat areas, the IHX has a more pronounced effect on lowering the optimal discharge pressure.

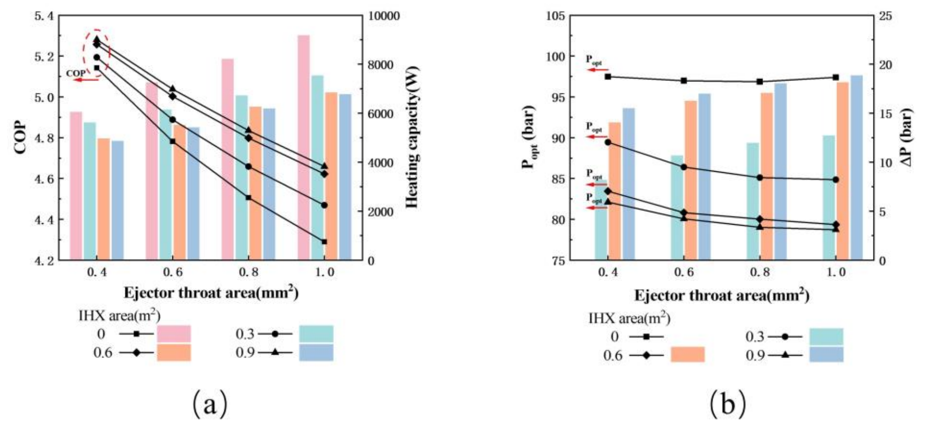


Fig.6. Variation the COP, heating capacity and optimal discharge pressure under different ejector throat area.

The CO_2 outlet temperature of the gas cooler is a key factor in evaluating the performance of the transcritical CO_2 system. Since the heat exchange between water and CO_2 in the gas cooler is countercurrent, the temperature difference between the inlet and outlet of the gas cooler directly reflects the heat exchange performance. Fig.7 shows the influence of the ejector throat area on the gas cooler exit temperature and discharge temperature. The results show that the discharge temperature at the optimal discharge pressure is essentially unaffected by the ejector throat area but is significantly influenced by the IHX. With the addition of the IHX, the system's discharge temperature exhibits a significant increase, which further intensifies as the IHX area increases, and the gas cooler exit temperature increases slightly with the ejector throat area at first and then gradually decreases to a stable temperature.

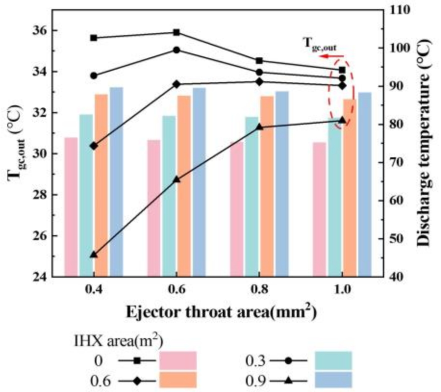


Fig.7. Variation the discharge temperature and gas cooler exit temperature under different ejector throat area

The ejector is one of the key components in the system, and its performance will impact the overall energy efficiency of the system. Fig.8(a) shows the influence of the ejector throat area on the entrainment ratio and pressure lift ratio under the optimal discharge pressure of the corresponding working condition. The results show that due to the supercooling effect of the IHX, the addition of IHX reduces both the ejector ejection ratio and pressure lift ratio. Moreover, increasing the area of the IHX can enhance this effect. Fig.8(b) shows the variation of ejector efficiency and IHX efficiency. It can be observed that both the ejector efficiency and IHX effectiveness increase as the ejector throat area rises. The trend of the ejector efficiency is similar to that of the ejection ratio and pressure lift ratio, and the difference is that increasing the IHX area will slightly increase the ejector efficiency.

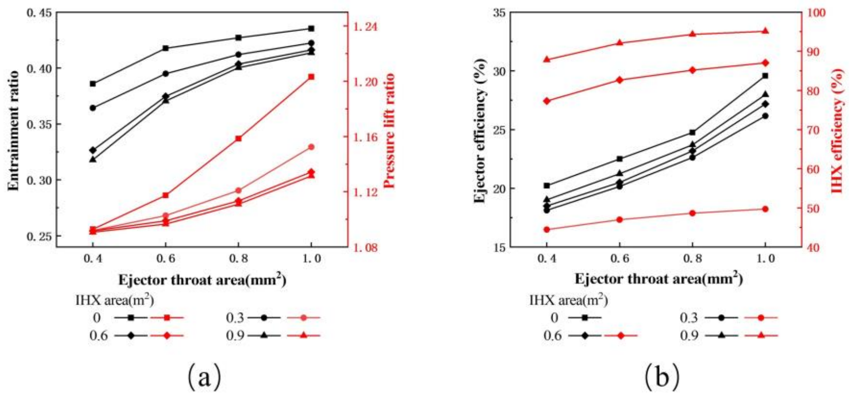


Fig.8. Variation the entrainment ratio, pressure lift ratio, ejector efficiency and IHX efficiency under different ejector throat area.

4.2 Impact of ambient temperature

This section analyzes the impact of ambient temperature on the performance of the DE-TCHSE with and without an internal heat exchanger. Fig.9 shows the influence of ambient temperature changes on the COP of system, heating capacity, and optimal discharge pressure. The results show that the increase of ambient temperature will increase the COP of the system, but will slightly reduce the heating capacity. At the same time, an increase in ambient temperature results in a corresponding increase in the optimal discharge pressure, this is mainly due to the increase in compressor suction pressure with rising the ambient temperature. Additionally, IHX is more effective in improving the COP and decreasing optimal discharge pressure of the system at higher ambient temperature.

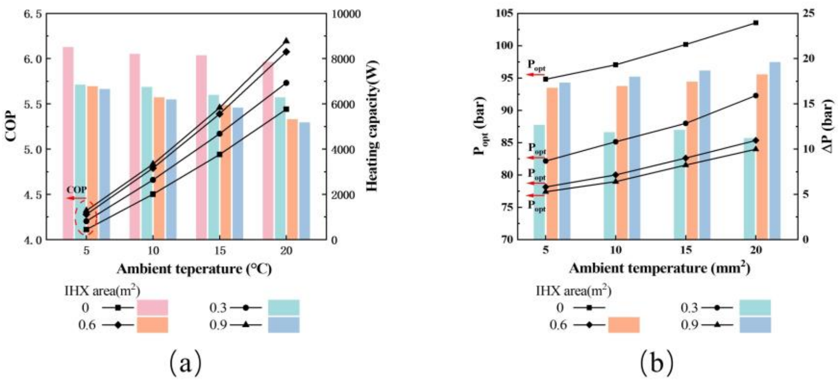


Fig.9. Variation the COP, heating capacity and optimal discharge pressure under different ambient temperature.

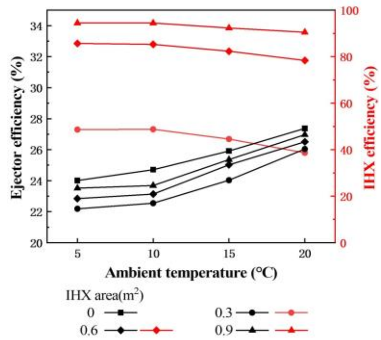


Fig.10. Variation the ejector efficiency and IHX efficiency under different ambient temperature.

Fig.10 shows the impact of ambient temperature on the ejector efficiency and IHX efficiency. It can be observed that with the increase of ambient temperature, IHX efficiency will decrease slightly. This is mainly due to the fact that the gas cooler exit temperature decreases to a certain extent. Meanwhile, the increasing ambient temperature causes the ejector efficiency to rise, which is determined by the gas cooler exit temperature and the optimal exhaust pressure. Although increasing the ambient temperature will lower the gas cooler outlet temperature, it also raises the optimal discharge pressure. The effect of the increase of discharge pressure is greater than the effect of the decrease of gas cooler exit temperature, which ultimately leads to an increase in ejector efficiency.

4.3 Impact of water inlet and outlet temperature

The water inlet and outlet temperatures on the gas cooler water side significantly influence the optimal discharge pressure, which in turn affects the role of the IHX in the system.

Fig.11 shows variation the COP, heating capacity, optimal discharge pressure, ejector efficiency and IHX efficiency under different water inlet temperature. It can be seen that as the inlet water temperature increases, the system's optimal COP gradually decreases, while the heating capacity and optimal discharge pressure slightly increases. This is mainly because the increase of inlet water temperature raises the system's optimal discharge pressure and power consumption. and the rate of increase is greater than the increase in heating capacity, resulting in a decrease in COP. Furthermore, at higher lower ambient temperatures, the IHX has a more pronounced effect on improving the COP. but the effect of IHX on reducing the system's optimal discharge pressure is quite similar across all simulated water inlet temperatures. In addition, the change of water inlet temperature has no effect on the IHX efficiency, but increasing the water inlet temperature will reduce the efficiency of the ejector.

Fig.12 shows the impact of the water outlet temperature on the system's performance and various parameters. It can be seen that, similar to the inlet water temperature, increasing the outlet water temperature will reduce the system's optimal COP. This is because the increase in outlet water temperature raises the system's optimal discharge pressure, leading to a rise in system power consumption, at the same time, the system's heating capacity increases. However, unlike the change of water inlet temperature, when IHX is not added to the system, increasing the water outlet temperature can significantly increase heating capacity. Additionally, at higher water outlet temperatures, the effect of IHX on improving COP and reducing the system's optimal discharge pressure becomes more pronounced.

This is primarily because the increase of water outlet temperature will raise the system's discharge temperature. Moreover, the water outlet temperature has a minimal effect on IHX efficiency, but increasing the water outlet temperature is beneficial for improving ejector efficiency.

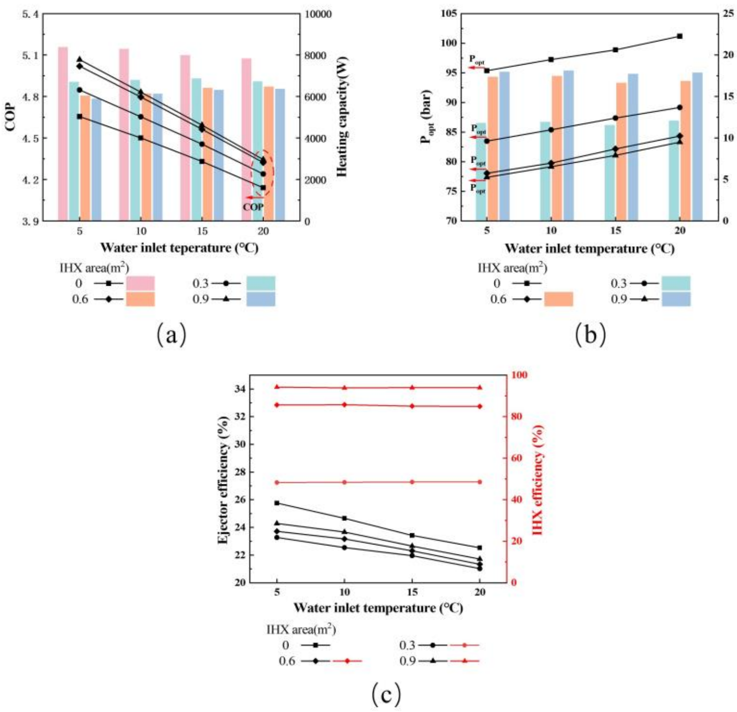


Fig.11. Variation the COP, heating capacity, optimal discharge pressure, ejector efficiency and IHX efficiency under different water inlet temperature.

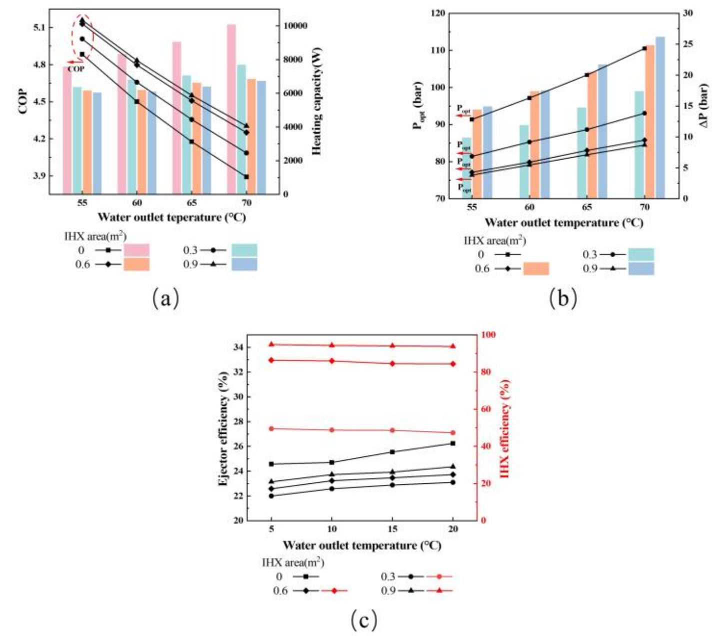


Fig.12. Variation the COP, heating capacity, optimal discharge pressure, ejector efficiency and IHX efficiency under different water outlet temperature.

4.4 Effect of IHX on the system exergy

In this section, the exergy analysis of the system under optimal discharge pressures at different operating conditions is presented. Fig.13 shows exergy destruction percentages of the components in different working conditions, it can be observed that after adding IHX, the exergy destruction percentages of the second evaporator and ejector are reduced, with the exergy destruction of the ejector showing a more significant decrease. When the ambient temperature increases from 10°C to 20°C, the ejector's exergy destruction percentage decreases. However, when the gas cooler water side inlet temperature increases from 10°C to 20°C or the outlet temperature increases from 60°C to 70°C, the ejector's exergy destruction percentage increases. Additionally, as the ejector throat area increases, the exergy destruction percentage of ejector increases after the IHX is added.

Fig.14 shows the variations of system product exergy, total exergy destruction and exergy efficiency of system in different working conditions. It can be seen that after adding IHX, both the system's product exergy and total exergy destruction decrease, while the exergy efficiency improves. However, higher ambient temperatures lead to a decrease in exergy efficiency. Moreover, under low ejector throat area conditions, the system exhibits higher entropy efficiency, whereas under high ejector throat area conditions, both product exergy and total exergy destruction increase.

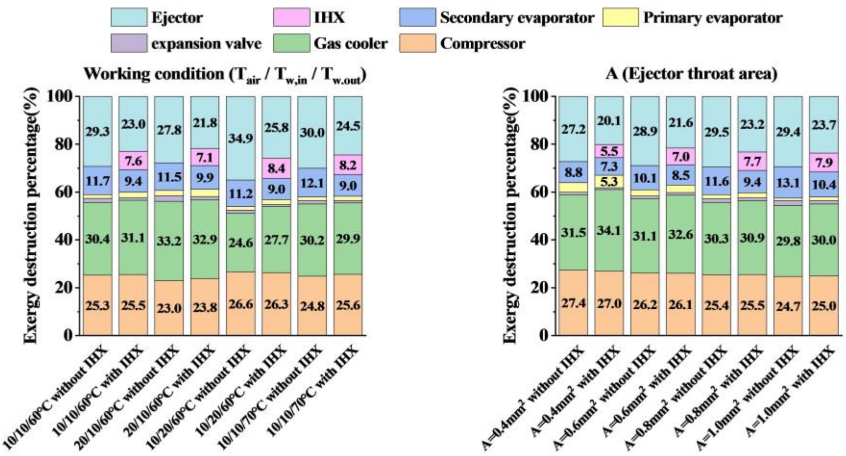


Fig.13. Exergy destruction percentages of the components in different working conditions

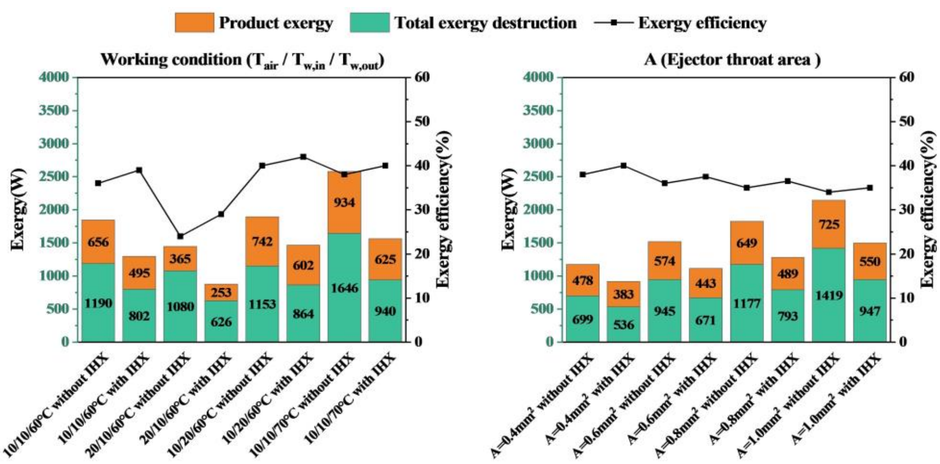


Fig.14. Product exergy, total exergy destruction and exergy efficiency of system in different working conditions

5. Conclusions

In this paper, a modified double-evaporator transcritical CO_2 heat pump with ejector is constructed and simulated. The significance of IHX, ejector throat area, ambient temperature, water inlet temperature and water outlet temperature are studied. The following conclusions are drawn from the study:

1. The addition of an IHX results in lower optimal discharge pressure, entrainment ratio, pressure lift ratio, ejector efficiency, as it subcools the gas donstream of the gas cooler while the COP will increase.

Under the default operating conditions, adding an IHX improves the maximum COP by 3.5% and reduces the optimal discharge pressure by 12.3%.

2. The increase in ejector throat area result in lower COP and optimal discharge pressure, while entrainment ratio, pressure lift ratio, ejector efficiency and IHX efficiency will improve. with a larger ejector throat area, the IHX becomes more effective in improving system COP and lowering the optimal discharge pressure.

3. With the increase of ambient temperature, the system COP, optimal discharge pressure and ejector efficiency rise, while the IHX efficiency will decrease, as the gas cooler exit temperature decrease to a certain extent. Furthermore, IHX is more effective in improving the COP and decreasing optimal discharge pressure of the system at higher ambient temperature.

4. The water inlet and outlet temperatures have similar impacts on system performance. The increase in either water inlet or outlet temperature result in lower system COP while higher the optimal discharge pressure, and increasing the inlet water temperature or decreasing the inlet water temperature is beneficial to the ejector efficiency, but both have little effect on the performance of IHX. Moreover, at lower water inlet temperature or higher water outlet temperature, IHX has a more pronounced effect on improving the COP.

5. Adding an IHX significantly reduces the exergy destruction percentages of ejector, and can decrease the system's total exergy destruction and increases the exergy efficiency at all simulated operating conditions.

Acknowledgments

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