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Article

Effect of Suction Nozzle Pressure Drop on the Performance of an Ejector-Expansion Transcritical CO_2 Refrigeration Cycle

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Abstract: The basic transcritical CO_2 systems exhibit low energy efficiency due to their large throttling loss. Replacing the throttle valve with an ejector is an effective measure for recovering some of the energy lost in the expansion process. In this paper, a thermodynamic model of the ejector-expansion transcritical CO_2 refrigeration cycle is developed. The effect of the suction nozzle pressure drop (SNPD) on the cycle performance is discussed. The results indicate that the SNPD has little impact on entrainment ratio. There exists an optimum SNPD which gives a maximum recovered pressure and COP under a specified condition. The value of the optimum SNPD mainly depends on the efficiencies of the motive nozzle and the suction nozzle, but it is essentially independent of evaporating temperature and gas cooler outlet temperature. Through optimizing the value of SNPD, the maximum COP of the ejector-expansion cycle can be up to 45.1% higher than that of the basic cycle. The exergy loss of the ejector-expansion cycle is reduced about 43.0% compared with the basic cycle.

Keywords: refrigeration system; ejector; suction nozzle pressure drop; CO_2; transcritical cycle

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1. Introduction

Among the natural refrigerants, the carbon dioxide (CO_2) has received extensive attention owing to its zero ozone depletion potential, very low global warming potential, safety, and admirable thermal physical properties. However, the reference transcritical CO_2 refrigeration cycle suffers from the defect of its low energy efficiency due to the large throttling loss [1]. This loss can be reduced by using a two-phase ejector to replace the throttling valve [2]. A two-phase ejector utilizes the expansion work to lift the suction pressure of the compressor so as to enhance the performance of the refrigeration cycle.

Through experiments, Elbel and Hrnjak [3] found that the cooling COP of the ejector-expansion transcritical CO_2 refrigeration cycle was 7% better than that of the reference cycle. Lee *et al.* [4] experimentally discovered that the COP of the ejector-expansion cycle was approximately 15% higher than that of the reference cycle. Through the experiment, Liu *et al.* [5] found that the maximum COP gain by the introduction of the ejector in a transcritical CO_2 refrigeration cycle could reach 36%. Xu *et al.* [6] stated that the compression ratio was decreased by 5.6%–6.7% and 10%–12.1% for the ejector-expansion transcritical CO_2 systems with and without IHE, respectively. Nakagawa *et al.* [7] experimentally found that the mixing length of the ejector had a significant effect on the entrainment ratio and the recovered pressure, and improper sizing of the mixing length lowered the cycle COP by about 10%.

Through theoretical analysis, Li and Groll [8] found that the replacement of the throttle valve with an ejector could gain a 16% COP increase in transcritical CO_2 air conditioning systems. Deng *et al.* [9] developed a thermodynamic model of the ejector-expansion transcritical CO_2 refrigeration cycle, and concluded that its COP was 22% better than that of the reference system. Sun and Ma [10] investigated the ejector-expansion transcritical CO_2 refrigeration cycle based on the first and second laws of thermodynamics, and stated that the replacement of throttling valve by an ejector could decrease more than 25% exergy loss and increase COP by over 30%. Zhang *et al.* [11] concluded that the employment of an internal heat exchanger in the ejector-expansion transcritical CO_2 refrigeration cycle did not always improve the system performance.

Table 1. Summary of selected ejector SNPD values found in the literature.

Literatures	Year	Fluid	Selected Values of SNPD
Li and Groll [8]	2005	CO_2	SNPD = 0.01 MPa, 0.03 MPa, 0.05 MPa. The COP improvement of the ejector cycle increases with an increase in SNPD. SNPD was taken as 0.03 MPa during the cycle analysis.
Deng *et al.* [9]	2007	CO_2	SNPD = 0 MPa.
Sarkar [12]	2008	CO_2	SNPD = 0.03 MPa.
Bilir and Ersoy [13]	2009	R134a	The effect of SNPD on the performance of ejector-expansion refrigeration cycle was discussed. The calculated optimum SNPD was about 0.02 MPa.
Yari [14,15]	2009/2011	CO_2	SNPD = 0 MPa.
Sun and Ma [10]	2011	CO_2	SNPD = 0 MPa.
Cen *et al.*[16]	2012	CO_2	SNPD = 0.03 MPa.
Manjili and Yavari [17]	2012	CO_2	SNPD = 0 MPa.
Zhang *et al.* [11]	2013	CO_2	SNPD = 0 MPa.
Li *et al.*[18]	2014	R1234yf	The effect of SNPD on the performance of ejector-expansion refrigeration cycle was discussed. The calculated optimum SNPD was about 0.014 MPa.

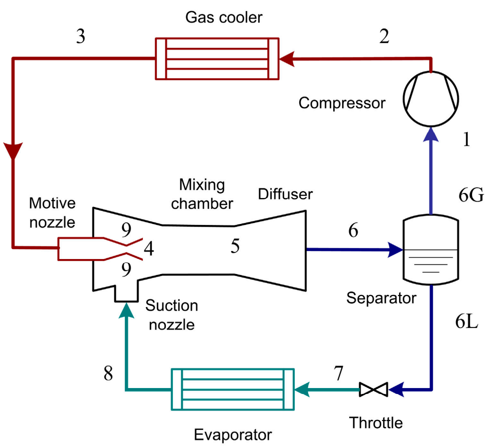
In the ejector-expansion transcritical CO_2 cycle, the suction nozzle pressure drop (SNPD) provides the power for the suction flow. Table 1 provides a summary of studies related to the SNPD values of the two-phase ejector during the simulation. It can be seen that the SNPD often is either ignored or is set to be a fixed value in the most accessible literature. Very few people investigate the impact of the SNPD on the transcritical CO_2 ejector-expansion cycle performance extensively. Whence, in the present paper, a thermodynamic model of the ejector-expansion transcritical CO_2 refrigeration cycle is developed. The effect of SNPD on the transcritical CO_2 ejector-expansion cycle performance is then analyzed.

2. Thermodynamic Modeling

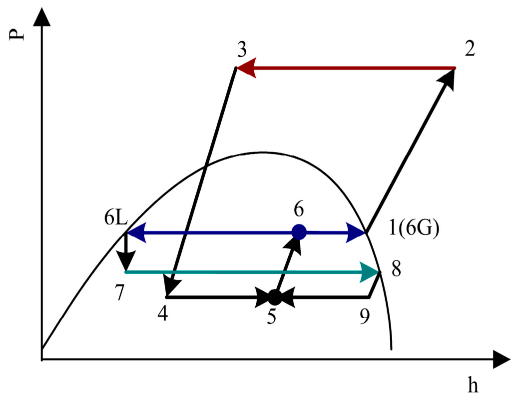
Figure 1 shows the schematic and the corresponding P-h diagram of the ejector-expansion transcritical CO_2 refrigeration cycle. The cycle is primarily comprised of a compressor, a gas cooler, an ejector, a gas-liquid separator, a throttle valve and an evaporator. The ejector is essentially comprised of a motive nozzle, a suction nozzle, a mixing chamber and a diffuser. In order to simplify the calculation, the following assumptions are made:

(1) One dimensional steady flow for the working fluid in the system.
(2) The pressure difference among the motive nozzle outlet, the suction nozzle outlet and the mixing chamber is negligible [9,10,16,18].
(3) Ignore the pressure loss in the heat exchangers and the pipes.
(4) Saturated vapor and liquid at the outlet of the gas-liquid separator.
(5) Expansion processes and compression processes are all adiabatic.

Figure 1. Schematic and P-h diagram of the ejector-expansion refrigeration cycle.



(a)



(b)

2.1. Energy Analysis

The ejector performance is usually described by two parameters: the mass entrainment ratio and the recovered pressure. The mass entrainment ratio is defined as a ratio between the suction mass flow rate and the motive mass flow rate. Thus the mass entrainment ratio of the ejector is:

$$\mu = m_8 / m_3 \tag{1}$$

The ejector recovered pressure is defined as:

$$p_{rec} = p_6 - p_8 \tag{2}$$

The ejector SNPD is expressed as:

$$SNPD = p_8 - p_9 \tag{3}$$

Therefore, for 1 kg refrigerant of the compressor, the suction mass flow rate is μ kg and the motive mass flow rate is 1 kg.

The isentropic efficiency of the motive nozzle is defined as:

$$\eta_{mot} = (h_3 - h_4)/ (h_3 - h_{4s}) \tag{4}$$

The energy balance equation in the motive nozzle is:

$$h_3 - h_4 = v_4^2/2 \tag{5}$$

The isentropic efficiency of the suction nozzle is defined as:

$$\eta_{suc} = (h_8 - h_9)/ (h_8 - h_{9s}) \tag{6}$$

The energy balance equation in the suction nozzle is:

$$h_8 - h_9 = v_9^2/2 \tag{7}$$

The momentum balance equation of the mixing chamber is:

$$v_5 = (v_4 + \mu v_9)/(1 + \mu) \tag{8}$$

The energy balance equation in the mixing chamber is:

$$h_3/(1 + \mu) + \mu h_8/(1 + \mu) = h_5 + v_5^2/2 \tag{9}$$

The isentropic efficiency of the diffuser is given as:

$$\eta_{dif} = (h_{6s} - h_5)/ (h_6 - h_5) \tag{10}$$

The energy balance equation in the diffuser is:

$$h_6 - h_5 = v_5^2/2 \tag{11}$$

The overall energy balance in the ejector can be written as:

$$\mu h_3 + h_8 = (1 + \mu)h_6 \tag{12}$$

The quality of the working fluid exiting from the ejector is:

$$x_6 = 1/(1 + \mu) \tag{13}$$

The isentropic efficiency of the compressor is:

$$\eta_{com} = (h_{2s} - h_1)/(h_2 - h_1) \tag{14}$$

The isentropic efficiency of the compressor is determined by [19]:

$$\eta_{com} = 1.003 - 0.121(p_2/p_1) \tag{15}$$

The compressor work consumption per unit mass flow rate is:

$$w_{com} = h_2 - h_1 \tag{16}$$

The evaporator refrigerating capacity per unit mass flow rate is:

$$q_{eva} = (h_8 - h_7)\mu \tag{17}$$

The cycle cooling coefficient of performance can be expressed as:

$$COP = q_{eva}/w_{com} \tag{18}$$

2.2. Exergy Analysis

The specific exergy of the refrigerant can be evaluated as:

$$ex = (h - h_0) - T_0(s - s_0) \tag{19}$$

For q at constant temperature T, the heat exergy rate ex_q can also be calculated by:

$$ex_q = (1 - T_0/T)\, q \tag{20}$$

Exergy loss equations for compressor, gas cooler, ejector, throttle valve and evaporator are given as follows:

$$I_{com} = T_0(s_2 - s_1) \tag{21}$$

$$I_{gc} = h_2 - h_3 - T_0(s_2 - s_3) \tag{22}$$

$$I_{ej} = T_0((1 + \mu)s_6 - s_3 - \mu s_8)) \tag{23}$$

$$I_{tv} = T_0(s_7 - s_{6L})\mu \tag{24}$$

$$I_{eva} = T_0(s_8 - s_7)\mu + T_0/T_r\,(h_7 - h_8)\mu \tag{25}$$

Therefore, the total exergy losses of the cycle are:

$$I_{tot} = I_{com} + I_{gc} + I_{ej} + I_{tv} + I_{eva} \tag{26}$$

The exergy efficiency of the ejector-expansion refrigeration cycle can be expressed as:

$$\eta_{ex} = 1 - I_{tot}/w_{com} \tag{27}$$

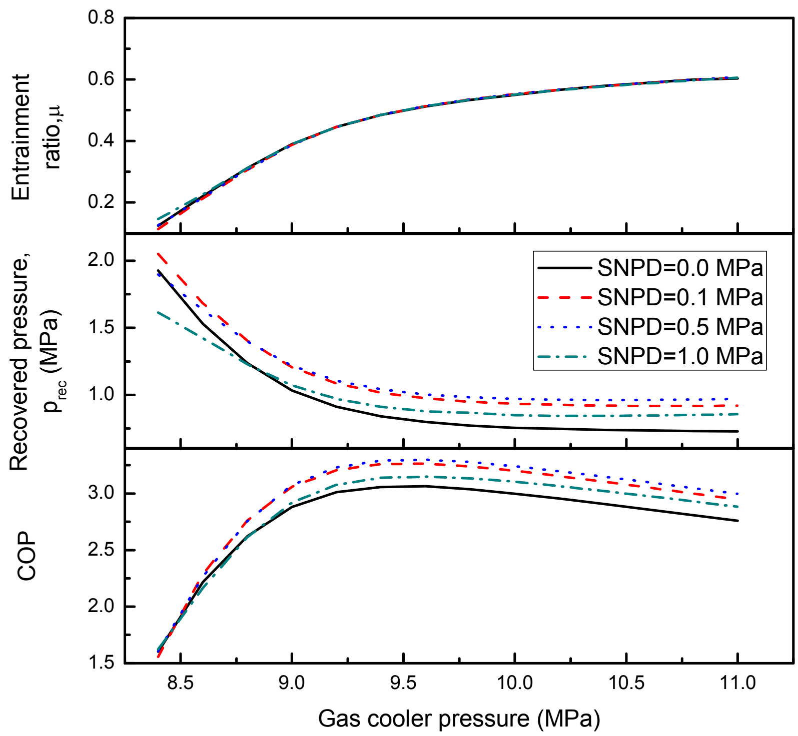
Based on the theoretical model, the simulation program using EES software [20] was developed to evaluate the performance of the ejetor-expansion refrigeration cycle.

3. Results and Discussion

In the following analyses, the results are obtained by varying one parameter according to its practical application, while other parameters are kept at designed values. Unless otherwise specified, the refrigerated object temperature is set at 20 °C, the reference state is defined as the environment temperature, 35 °C, the gas cooler outlet temperature is set at 40 °C, the evaporation temperature is set at 5 °C, and the ejector is assumed to have the following efficiencies: $\eta_{mot} = 0.9$, $\eta_{suc} = 0.9$, $\eta_{dif} = 0.8$.

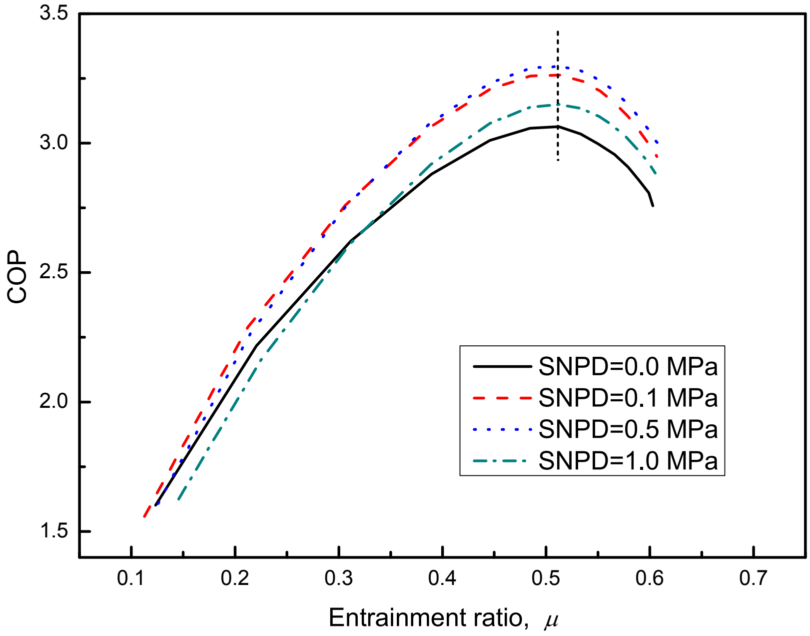
Figure 2 shows the variations of the entrainment ratio, the recovered pressure and COP with the gas cooler pressure at different SNPD values. It can be seen that the entrainment ratio grows rapidly at the outset and then slows down with increasing the gas cooler pressure. This rough inclination of the ejector entrainment ratio *versus* the gas cooler pressure shows a close correspondence with the published literature [9]. Furthermore, it can be observed that the entrainment ratio is basically independent of SNPD. At identical operating conditions, a higher recovered pressure means lower compressor pressure ratio, *i.e.*, higher efficiency of the compressor and higher cycle COP. With increasing the gas cooler pressure, the recovered pressure initially declines rapidly to a minimum value and then slightly increases as the gas cooler pressure surpasses around 10.4 MPa. However, the increase of SNPD does not necessarily result in the improvement of the recovered pressure. For example, recovered pressure values at SNPD = 0.5 MPa outperform the other three options when the gas cooler pressure is higher than 9.0 MPa. The cycle COP increases to the peak value rapidly and then declines slowly. The optimum gas cooler pressure is practically fixed at the investigated values of SNPD. The optimum COP value at SNPD = 0.5 MPa is the highest among the four options. This phenomenon implies that the effect of SNPD on the cycle COP is non-linear.

Figure 2. Variations of ejector entrainment ratio, pressure recovery and COP with the gas cooler pressure.



The COP values calculated for various ejector entrainment ratios are presented in Figure 3. It is seen that there exists an optimal entrainment ratio where COP peaks for the investigated ejector cycles. This result suggests that it is appropriate to suit the ejector to the given system, and that the system parameters should be coordinated with the ejector characteristics for a specified ejector with a certain entrainment ratio in order to keep its maximum COP. The optimum value of the ejector entrainment ratio is about 0.5. The value of SNPD has almost no effect on the optimum entrainment ratio.

Figure 3. Variation of COP with the ejector entrainment ratio.



Figures 4 and 5 depict the variations of the recovered pressure *versus* SNPD at different gas cooler exit temperatures and evaporator temperatures respectively, where the gas cooler pressure is kept at the calculated optimum value. As the evaporating temperature increases and the gas cooler exit temperature decreases, the recovered pressure of the cycle decreases. It is can also be seen that the recovered pressure of the cycle goes up initially to a maximum and then decreases with the increase of SNPD. At the investigated operating conditions of this study, the optimum SNPD is around 0.3 MPa.

Figure 4. Recovered pressure *versus* SNPD at different gas cooler exit temperatures.

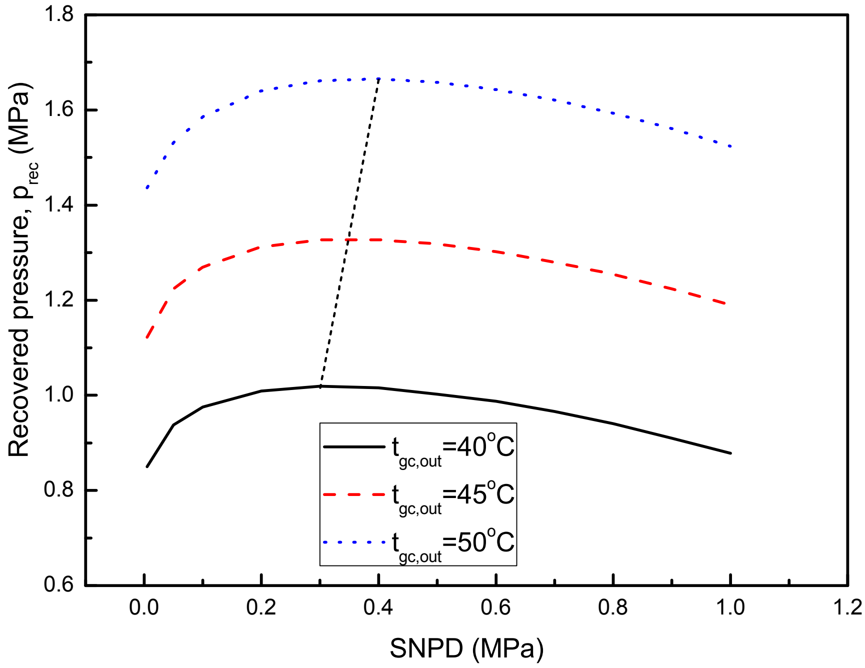
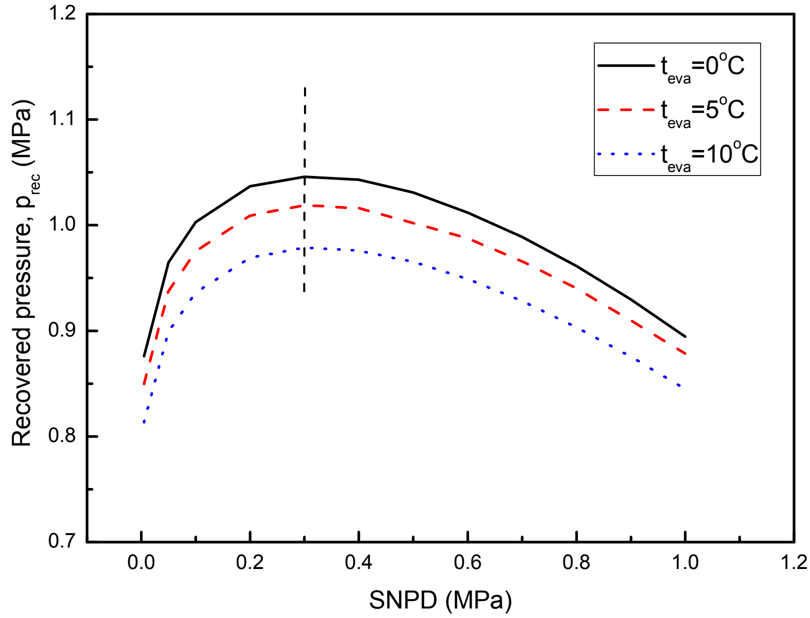


Figure 5. Recovered pressure *versus* SNPD at different evaporating temperatures.



Figures 6 and 7 show the variations of the COP *versus* SNPD at different gas cooler temperatures and evaporator temperatures respectively, where the gas cooler pressure is kept at the calculated optimum value. It is observed that COP of the ejector-expansion transcritical CO_2 cycle can be maximized by the variation of the values of SNPD. Furthermore, the optimum SNPD value is almost independent of evaporating temperature, but increases slightly with the gas cooler outlet temperature. The cycle COP achieves almost exactly the optimum value at the optimum recovered pressure. The cycle COP of SNPD = 0.3 MPa is 7.7%–8.7% higher than that of SNPD = 0.0 MPa for the given ranges of evaporator temperatures and gas cooler outlet temperatures. It can also be found that along with the decline of the gas cooler outlet temperature or the rise of the evaporating temperature, the peak COP value of the cycle increases.

Figure 6. COP *versus* SNPD at different gas cooler exit temperatures.

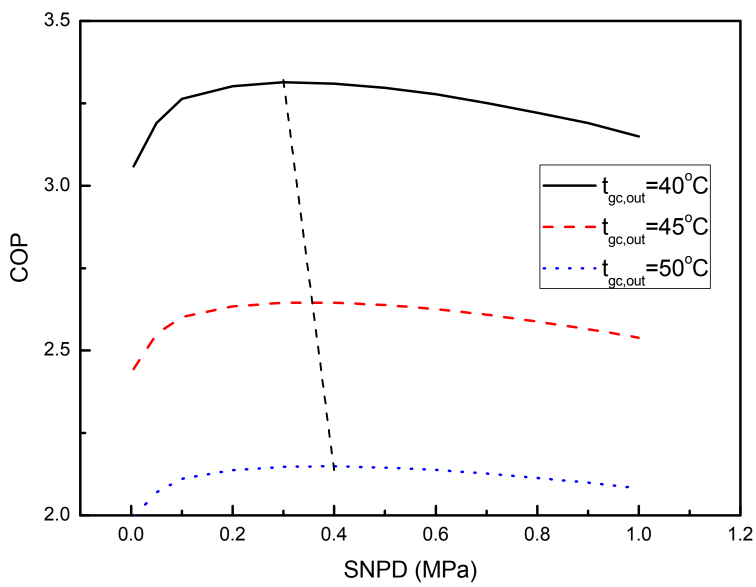


Figure 7. COP *versus* SNPD at different evaporator temperatures.

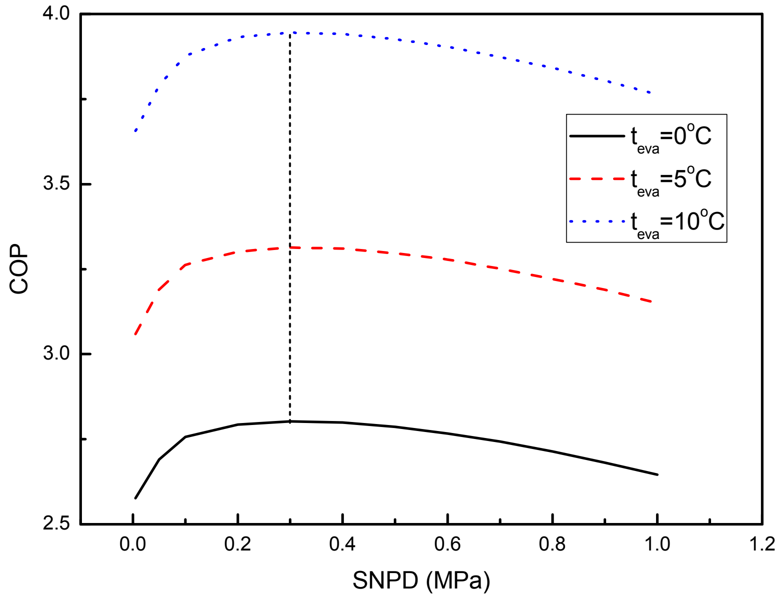


Figure 8 displays the variation of optimum SNPD with the efficiencies of the motive nozzle, the suction nozzle and the diffuser. It can be seen that the lower the efficiencies of the motive nozzle and the suction nozzle, the less the optimum SNPD. When the efficiency is less than 0.4, the optimum SNPD changes slowly. When the efficiencies of the two nozzles are higher than 0.5, the optimum SNPD is linearity to changes of efficiencies, whereas the efficiency of the diffuser has no effect on the optimum SNPD.

The performances of the transcritical ejector-expansion cycle and the basic cycle are compared in Figure 9, where SNPD is kept at 0.3 MPa. It can be seen that the optimum gas cooler pressure of the ejector-expansion cycle is lower than that of the basic cycle. The maximum COP of the ejector-expansion cycle is 45.1% higher than that of the basic cycle.

Figure 8. Variation of the optimum SNPD with the efficiencies of the nozzles.

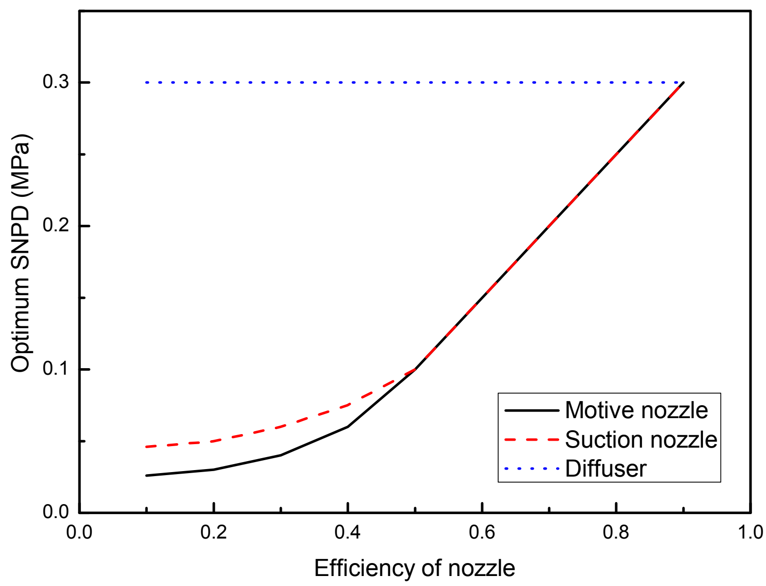


Figure 9. COP of the ejector-expansion cycle and the basic cycle for various gas cooler pressures.

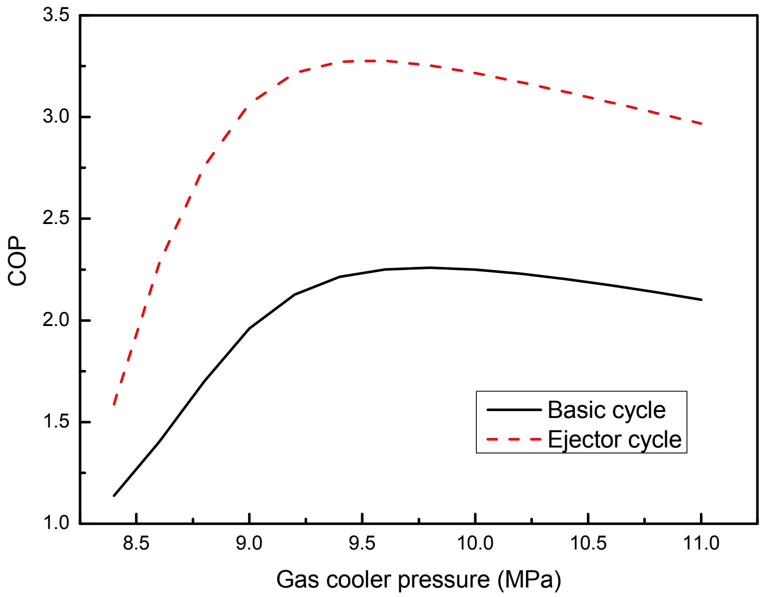


Table 2 illustrates the exergy losses in each component for both the basic cycle and the ejector-expansion cycle. The exergy losses in the ejector-expansion cycle are calculated based on one unit mass flow rate of the compressor. As shown in Table 1, the throttling exergy loss of the basic cycle is 15.03 kJ/kg, 34.5% of the total exergy losses. Nevertheless, the throttling exergy loss of the ejector-expansion cycle is only 0.8622 kJ/kg with the ejection exergy loss of 6.436 kJ/kg, the sum of the two losses is 29.4% of the total exergy losses. The exergy losses in the compression and heat rejection processes are also decreased in the ejector-expansion system. The total exergy loss of the ejector-expansion cycle reduces about 43.0% compared with the basic cycle.

Table 2. Exergy losses of the two cycles.

Process	Basic cycle		Ejector	
	Exergy loss(kJ/kg)	Percentage (%)	Exergy loss(kJ/kg)	Percentage (%)
Compression	12.79	29.3	6.3	25.3
Heat rejection	9.541	21.9	5.777	23.2
Ejector	-	-	6.436	25.9
Throttling	15.03	34.5	0.8622	3.5
Evaporation	6.253	14.3	5.496	22.1
Total	43.62	100	24.87	100
Exergy efficiency	0.1157		0.1678	

4. Conclusions

In this paper, a thermodynamic simulation of transcritical CO_2 ejector expansion cycle is presented. The effect of SNPD on the performance of the cycle is discussed theoretically. The simulation results show that SNPD has a significant impact on the cycle performance. The value of SNPD has little effect on the ejector entrainment ratio. There exists an optimum SNPD which gives a maximum recovered pressure and COP under a specified condition. The value of the optimum SNPD mainly depends on the efficiencies of the motive nozzle and the suction nozzle, but is virtually independent of evaporating temperature and gas cooler outlet temperature. The lower the efficiencies of the motive nozzle and the suction nozzle, the less the optimum SNPD value. Through optimizing the value of SNPD, the maximum COP of the ejector-expansion cycle can be up to 45.1% higher than that of the basic cycle. The total exergy loss of the ejector-expansion cycle is reduced by about 43.0% compared with the basic cycle.

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Author Contributions

Zhenying Zhang and Lili Tian did the theoretical work and wrote this paper. Both authors have read and approved the final published manuscript.

Conflicts of Interest

The authors declare no conflict of interest.

Nomenclature

COP	coefficient of performance in cooling condition
ex	exergy (kJ/kg)
h	enthalpy, kJ/kg
I	specific irreversibility (kJ/kg)
m	mass flow rate, kg/s
p	pressure, MPa
q	specific heat transfer rate, kJ/kg
SNPD	suction nozzle pressure drop, MPa
t	temperature, °C
T	temperature, K
v	velocity, m/s
w	specific power, kJ/kg
x	vapor quality
μ	entrainment ratio of ejector
η	efficiency

Subscripts

0	reference environment
com	compressor

dif	diffuser
eva	evaporator
gc	gas cooler
mix	mixing chamber
mot	motive nozzle
r	refrigerated object
s	isentropic process
suc	suction nozzle
tot	total
tv	throttle valve

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