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Particular characteristics of transcritical CO_2 refrigeration cycle with an ejector

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Abstract

The present study describes a theoretical analysis of a transcritical CO_2 ejector expansion refrigeration cycle (EERC) which uses an ejector as the main expansion device instead of an expansion valve. The system performance is strongly coupled to the ejector entrainment ratio which must produce the proper CO_2 quality at the ejector exit. If the exit quality is not correct, either the liquid will enter the compressor or the evaporator will be filled with vapor. Thus, the ejector entrainment ratio significantly influences the refrigeration effect with an optimum ratio giving the ideal system performance. For the working conditions studied in this paper, the ejector expansion system maximum cooling COP is up to 18.6% better than the internal heat exchanger cycle (IHEC) cooling COP and 22.0% better than the conventional vapor compression refrigeration cycle (VCRC) cooling COP. At the conditions for the maximum cooling COP, the ejector expansion cycle refrigeration output is 8.2% better than the internal heat exchanger cycle refrigeration output and 11.5% better than the conventional cycle refrigeration output. An exergy analysis showed that the ejector expansion cycle greatly reduces the throttling losses. The analysis was also used to study the variations of the ejector expansion cycle cooling COP for various heat rejection pressures, refrigerant temperatures at the gas cooler exit, nozzle efficiencies and diffuser efficiencies.

Keywords: Refrigeration system; Transcritical cycle; CO_2; Ejector

1. Introduction

During the expansion of a refrigerant in a throttling process, much friction heat is dissipated to the refrigerant due to the large kinetic energy increases as the refrigerant pressure decreases. The process is then not isenthalpic, and this throttling loss reduces the refrigeration effect. In a CO_2 transcritical vapor compression refrigeration cycle, the supercritical CO_2 is expanded to a subcritical state. The throttling loss is greater than with conventional refrigerants owing to the higher pressure change during the expansion.

The throttling loss can be reduced by staged expansion, internal heat exchangers or a work-generating expansion. In principle, a low cost ejector with no moving parts is also an attractive alternative for the expansion device in the transcritical CO_2 cycle. The Denso Corporation in Japan in 2004 stated that the coefficient of performance (COP) of a CO_2 transcritical automotive air conditioning with an ejector was 25% better than the cooling COP of a convention vapor compression refrigeration cycle in their experiments [1].

Relatively little information is available on the use of ejectors as expansion devices in a vapor compression refrigeration cycle. Kornhauser [2] theoretically analyzed the performance of an ejector expansion refrigeration cycle using *R*-12 as the refrigerant. He found a theoretical cooling COP improvement of up to 21% over the conventional

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Nomenclature

COP	coefficient of performance in cooling condition
ex	exergy (kJ kg^{-1})
h	enthalpy (kJ kg^{-1})
m	relative exhaust mass flow rate or relative motive mass flow rate
q	specific refrigeration output per unit mixture mass flow rate (kW kg^{-1})
s	entropy (kJ kg^{-1} K^{-1})
t	temperature (°C)
T	temperature (K)
u	velocity (m s^{-1})
V	compressor volume displacement
w	specific power (kW kg^{-1})
x	vapor quality
EERC	the ejector expansion refrigeration cycle
VCRC	the conventional vapor compression refrigeration cycle
IHEC	the internal heat exchange cycle

Greek symbols

μ	ejector entrainment ratio
η	isentropic efficiency/exergy efficiency
ρ	density (kg m^{-3})

Subscripts

$1, 2, \ldots, i$	cycle states
$3', 5x, 5L$	cycle states
$12, 23, \ldots$	processes between two cycle states
1s, 3s, 4s	locations downstream of the isentropic process
a	referenced zero state in the exergy analysis
c	compressor
d	discharge pressure (high-side) of the compressor or ejector diffuser
e	evaporator
eje	ejector
ex	exergy
exp	expansion valve
H	heat sink
loss	total exergy loss
L	heat source
n	ejector nozzle
s	vapor–liquid separator

vapor compression refrigeration cycle for an evaporator temperature of −15 °C and a condenser temperature of 30 °C. Harrell and Kornhauser [3] found that the cooling COP was improved by 3.9% to 7.6% with R-134a as the refrigerant with a two-phase ejector. Menegay and Kornhauser [4] developed a bubbly flow tube installed upstream of the nozzle to reduce the non-equilibrium thermodynamic losses in the ejector nozzle. The cycle cooling COP with an ejector using the bubbly flow tube was improved by 3.8% over the conventional cycle for standard conditions with R-12 as the refrigerant. Menegay and Kornhauser [4] suggested that this result was not as good as expected so they anticipated more studies of the ejector expansion refrigeration cycle. Domanski [5] pointed out that the ejector efficiency significantly influences the cooling COP of the ejector expansion refrigeration cycle. Fan et al. [6,7] and Wu et al. [8] studied the modified ejector expansion refrigeration cycle with two heat sources. Nakagawa and Takeuchi [9] showed that a longer diverging section in the nozzle increased the nozzle efficiency. Disawas and Wongwises [10] experimentally investigated the performance of the ejector expansion refrigeration cycle without the expansion valve upstream of the evaporator so that the evaporator is flooded with the refrigerant. Their tests showed an improved cooling COP at low heat sink temperatures relative to the convention cycle with *R*-134 a as the refrigerant. The motive mass flow rate in the ejector, which is the flow rate entering the ejector from the gas cooler, was found to be strongly dependent on the heat sink temperature and independent of the heat source temperature. Liu et al. [11] analyzed the influence of an ejector on a transcritical CO_2 cycle to show that the new cycle effectively improved the transcritical CO_2 cycle performance. The results illustrated the effects of ejector entrainment ratio and efficiency on the cycle performance.

In the ejector expansion refrigeration cycle, the refrigerant leaving the ejector is divided into a saturated liquid stream and a saturated vapor stream in the vapor–liquid separator. The ejector entrainment ratio is equal to the mass ratio of the two streams in a stable system while the mass percent of saturated vapor is equal to the vapor quality at the ejector exit. Therefore, the ejector entrainment ratio is related to the vapor quality, which will be the focus of the analysis in this paper. The optimum ejector entrainment ratio will then give the optimum system performance. Previous studies have used both theoretical thermodynamic analyses and experimental research but have not clearly established the relationship between the ejector entrainment ratio and the vapor quality.

This work presents a theoretical analysis of the performance of the CO_2 transcritical ejector expansion refrigeration cycle (EERC) to identify the thermodynamic relationship between the ejector entrainment ratio and the vapor quality at the ejector exit. The overall performance of the ejector expansion cycle is then compared with that of the conventional vapor compression refrigeration cycle (VCRC) and the internal heat exchange cycle (IHEC) as a function of the significant parameters.

2. Model

Figs. 1 and 2 present a schematic of the ejector expansion refrigeration cycle with a *P–h* diagram illustrating the transcritical cycle. The ejector expansion system includes a compressor, gas cooler, ejector, vapor–liquid separator, expansion valve and evaporator.

The subcritical CO_2 enters the compressor at pressure P_s at state (1) and is compressed isentropically to the high-side pressure P_d at state (1s). The real CO_2 compression process to the high-side pressure P_d with an isentropic efficiency, η_c, ends at supercritical state (2). The supercritical CO_2 is then cooled in the gas cooler to temperature T_3 at state (3).

The flow at state (3) enters the ejector nozzle and expands to a mixture at state (3′) with a nozzle efficiency of $\eta_n = 0.7$, with the corresponding isentropic state (3s). The saturated secondary vapor stream enters the ejector at pressure P_e corresponding to state (7). The two streams mix at constant pressure in the ejector with the final state of the mixture corresponding to state (4). The mixture then flows through the ejector diffuser where it recovers to pressure P_s at state (5x). The diffuser is assumed to have a diffuser efficiency of $\eta_d = 0.8$ with the isentropic outlet at state (4s).

The stream leaving the ejector flows into the vapor–liquid separator where it is divided into saturated liquid and saturated vapor streams corresponding to states (5L) and (5). The saturated liquid enters the expansion valve and expands to pressure P_e at state (6). The saturated vapor (5) is superheated, then enters the compressor.

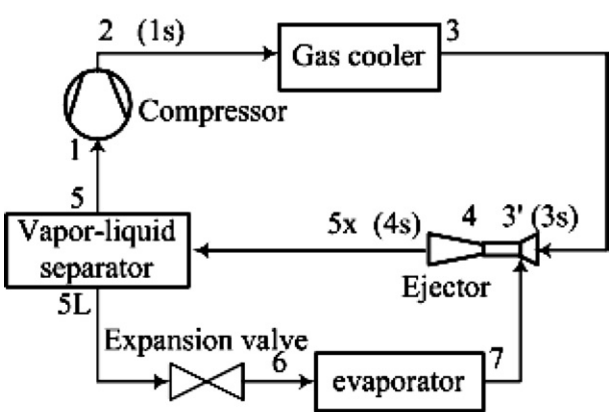

Fig. 1. Schematic of the ejector expansion system.

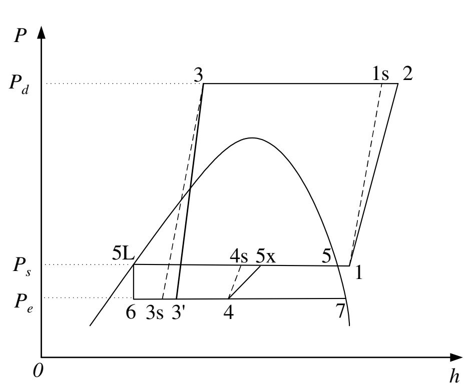

Fig. 2. Schematic of the ejector expansion system *P–h* diagram.

The ejector nozzle efficiency and the diffuser efficiency were given by Alexis and Rogdakis [12] who assumed that the mixture pressure in the ejector at state (4) differed from the evaporator pressure P_e. The present analysis assumes that the mixture pressure in the ejector is equal to the evaporator pressure P_e since this small pressure difference can be neglected in the transcritical CO_2 cycle.

The three cycles are compared based on the following assumptions:

1. Kinetic energies of the refrigerant at the ejector inlet and outlet are negligible.
2. Flow losses in the pipes and heat exchangers are negligible.
3. Unless stated otherwise, the evaporating temperature is 5 °C. The refrigerant leaves the evaporator as saturated vapor.
4. Unless stated otherwise, the refrigerant temperature at the gas cooler exit, t_3, is 36 °C.
5. Mixing occurs at constant pressure in the ejector mixing region with the assumption that the fluid momentum is conserved. The pressure is assumed to be equal to the evaporator pressure, P_e.
6. The heat sink temperature is 35 °C, while the heat source temperature is 27 °C.
7. The system is in thermodynamic equilibrium. All flow processes are analyzed based on their average velocities and temperatures.
8. The vapor is superheated 15 °C in the internal heat exchanger before entering the compressor in the internal heat exchanger cycle.
9. The vapor is not superheated before entering the compressor in the ejector expansion cycle and the conventional cycle.

3. Computer modeling

3.1. Thermodynamic modeling

The modeling of the ejector expansion cycle is based on one unit of mixture refrigerant mass in the ejector at state (5x). The ejector entrainment ratio, μ, is defined as the ejector suction mass flow rate at (7) divided by the motive mass flow rate at (3). Therefore, for 1 kg refrigerant mixture in the ejector, the suction mass flow rate is $\mu/(1+\mu)$ kg and the motive mass flow rate is $1/(1+\mu)$ kg.

The motive stream enters the ejector and expands to evaporator pressure P_e with a nozzle efficiency defined as:

$$\eta_n = (h_3 - h_{3'})/(h_3 - h_{3s}) \tag{1}$$

The energy balance between states (3) and (3′) is:

$$\frac{1}{2}u_{3'}^2 = h_3 - h_{3'} \tag{2}$$

The analysis further assumes that fluid momentum is conserved in the mixing section,

$$u_{3'}/(\mu+1) = u_4 \tag{3}$$

The overall energy balance equation can be written as:

$$h_3/(1+\mu) + h_7\mu/(1+\mu) = h_{5x} \tag{4}$$

The energy balance equation between states (4) and (5x) is:

$$\frac{1}{2}u_4^2 = h_{5x} - h_4 \tag{5}$$

The refrigerant mixture recovers pressure in the ejector diffuser with a diffuser efficiency of:

$$\eta_d = (h_{4s} - h_4)/(h_{5x} - h_4) \tag{6}$$

The adiabatic compressor efficiency is [13]:

$$\eta_c = 1.003 - 0.121 \times (p_d/p_s) \tag{7}$$

$$\eta_c = \frac{h_{1s} - h_1}{h_2 - h_1} \tag{8}$$

The compressor power consumption per unit mixture flow mass is:

$$w_c = (h_2 - h_1)/(1+\mu) \tag{9}$$

The refrigeration output per unit mixture flow mass is:

$$q_e = (h_7 - h_{5L})\frac{\mu}{1+\mu} \tag{10}$$

The cooling coefficient of performance is:

$$\mathrm{COP} = q_e/w_c \tag{11}$$

3.2. Exergy efficiency modeling

The reference zero state (a) is defined as the environment temperature, 35 °C, and the high-side pressure. The exergy values of all states were calculated based on one unit refrigerant mixture mass in the ejector:

$$\mathrm{ex}_i = [(h_i - h_a) - T_a(s_i - s_a)] \cdot m_i \tag{12}$$

while $i = 1, 2, 3, 3', 5$, $m_i = 1/(1+\mu)$
while $i = 5\mathrm{L}, 6, 7$, $m_i = \mu/(1+\mu)$
while $i = 4, 5\mathrm{x}$, $m_i = 1$

The exergy losses in the processes were calculated using:

Compression

$$\mathrm{ex}_{12} = (\mathrm{ex}_1 - \mathrm{ex}_2) + w_s \tag{13}$$

Heat rejection

$$\mathrm{ex}_{23} = \mathrm{ex}_2 - \mathrm{ex}_3 \tag{14}$$

Ejection

$$\mathrm{ex}_{eje} = \mathrm{ex}_3 + \mathrm{ex}_7 - \mathrm{ex}_{5x} \tag{15}$$

Throttling

$$\mathrm{ex}_{exp} = \mathrm{ex}_{5L} - \mathrm{ex}_6 \tag{16}$$

Evaporation

$$\mathrm{ex}_{67} = \mathrm{ex}_6 - \mathrm{ex}_7 - (T_a/T_L - 1) \cdot q_e \tag{17}$$

The total exergy loss for a unit refrigerant mixture mass in the ejector was calculated using:

$$\mathrm{ex}_{loss} = \mathrm{ex}_{12} + \mathrm{ex}_{23} + \mathrm{ex}_{eje} + \mathrm{ex}_{exp} + \mathrm{ex}_{67} \tag{18}$$

The ejector expansion cycle exergy efficiency was calculated using:

$$\eta_{ex} = 1 - \mathrm{ex}_{loss}/w_c \tag{19}$$

4. Results and discussion

4.1. Matching of the ejector entrainment ratio

When the ejector expansion cycle is running at steady state, the mass flow rates of the two refrigerant streams flowing into the ejector do not vary. The refrigerant leaving the ejector is divided into a saturated liquid stream and a saturated vapor stream in the vapor–liquid separator. Therefore, the mass flow ratio of the two streams must be equal to the ejector entrainment ratio in a stable operating system and a given set of system working conditions will have a unique ejector entrainment ratio. If the ejector does not have an entrainment ratio that produces the proper refrigerant quality at the ejector exit, then an unsteady system results and either liquid will accumulate with liquid entering the compressor or the separator will be full of refrigerant vapor. In this study, the ejector entrainment ratio is always specified as the ratio giving a stable running ejector expansion system.

The computer model predicts a unique ejector entrainment ratio for the specified conditions. The variation of the ejector entrainment ratio and the ejector exit quality with changes in the high-side pressure for three evaporator temperatures is shown in Fig. 3. The ejector entrainment ratio initially rapidly increases and then levels off with

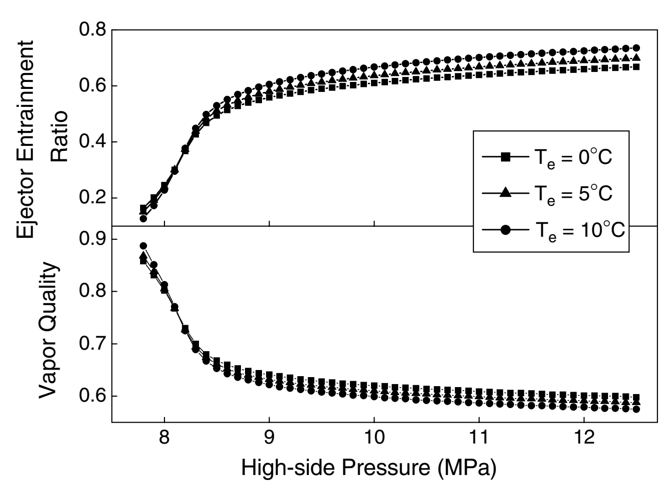

Fig. 3. Entrainment ratio and vapor quality variations for various compressor discharge pressures and evaporator temperatures in the ejector expansion cycle.

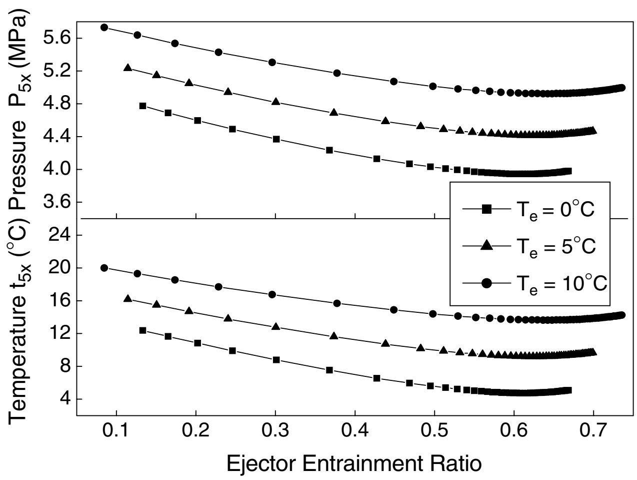

Fig. 4. Influence of ejector entrainment ratio on the pressures and temperatures at the ejector exit in an ejector expansion cycle.

increasing high-side pressure. The quality moves in the opposite direction due to the relation:

$$x = \frac{1}{\mu + 1} \tag{20}$$

The variations of the pressure and temperature in the vapor–liquid separator for various ejector entrainment ratios for three evaporator temperatures are shown in Fig. 4. The pressure and temperature in the vapor–liquid separator decrease slowly with increasing ejector entrainment ratio to minimums and then slowly increase for ejector entrainment ratios greater than about 0.6. Therefore, considering the results in both Figs. 3 and 4, for higher high-side pressures, lower pressures and temperatures can be more easily maintained in the vapor–liquid separator because a higher ejector entrainment ratio means more saturated vapor flow at the evaporator pressure and temperature exhausted into the ejector to facilitate the mixing and recovery process. Disawas and Wongwises [10] found the same phenomenon in their tests for compressor speeds over 450 rpm which increased the high-side pressures resulting in a higher ejector entrainment ratio, Fig. 3. This higher ratio then leads to a lower vapor–liquid separator pressure, Fig. 4.

Fig. 5 shows the variation of the cooling COP with the ejector entrainment ratio for the various evaporator temperatures. The results show an optimum value of the cooling COP at an ejector entrainment ratio between 0.5 and 0.6. If the entrainment ratio is too low, the COP will decrease to near 1.0 even though the system is stable. The results in Fig. 5 show that the ejector must be carefully matched with the ejector expansion system. For a given ejector with a fixed entrainment ratio, the system parameters must be adjusted to fit the ejector characteristics which may not give the optimum COP. Disawas and Wongwises [10] adjusted the compressor speed to fit the ejector. He maintained that the appropriate compressor speed is 450 rpm by referring to former operating experience on the ejector expansion cycle. At compressor speeds lower than 450 rpm, the amount of liquid in the separator gradually increased. Eventually, the liquid refrigerant flooded the separator outlet and flowed to the compressor, which results in compressor failure.

4.2. Comparison of the three refrigeration cycles

The performances of the CO_2 transcritical ejector expansion cycle, internal heat exchanger cycle and a conventional cycle were compared using the model described in Section 3. Fig. 6 shows the variation of the COP with high-side pressure in the three cycles. The three cycles all have an optimum high-side pressure corresponding to a maximum COP. However, the maximum COP for the ejector expansion system is 18.6% higher than the maximum COP for the internal heat exchanger system and 22.0% higher than that

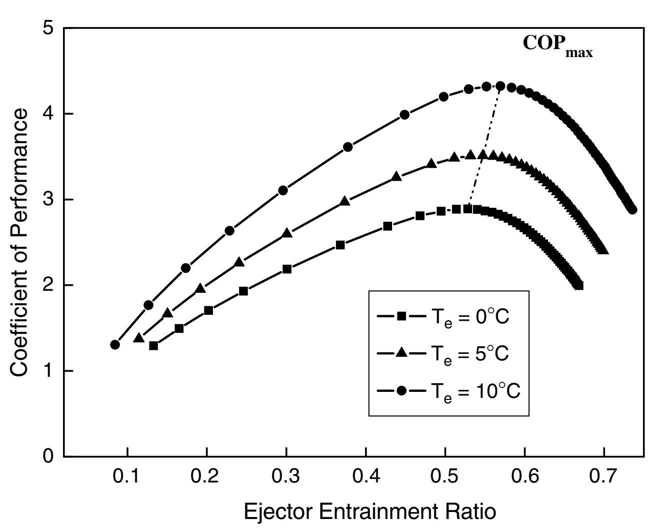

Fig. 5. Cooling COP for various entrainment ratios and evaporator temperatures.

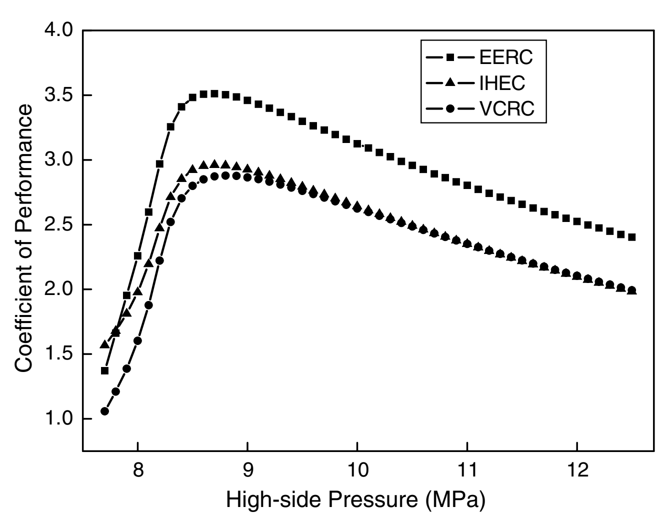

Fig. 6. Cooling COP of the ejector expansion cycle, internal heat exchanger cycle and conventional cycle for various high-side pressures.

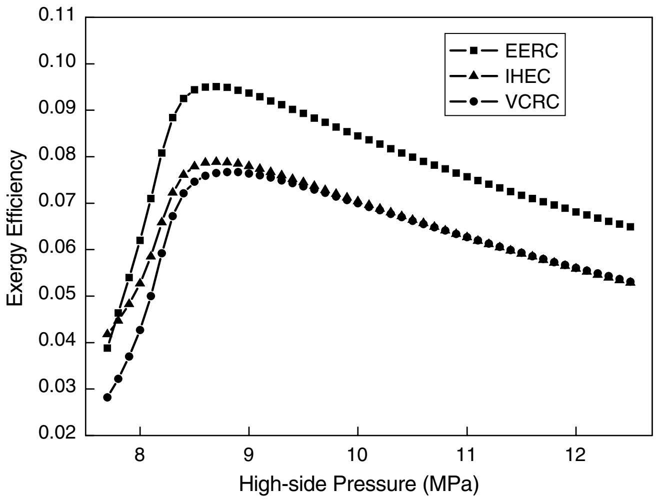

Fig. 7. Exergy efficiency of the ejector expansion cycle, internal heat exchanger cycle and convention cycle for various high-side pressures.

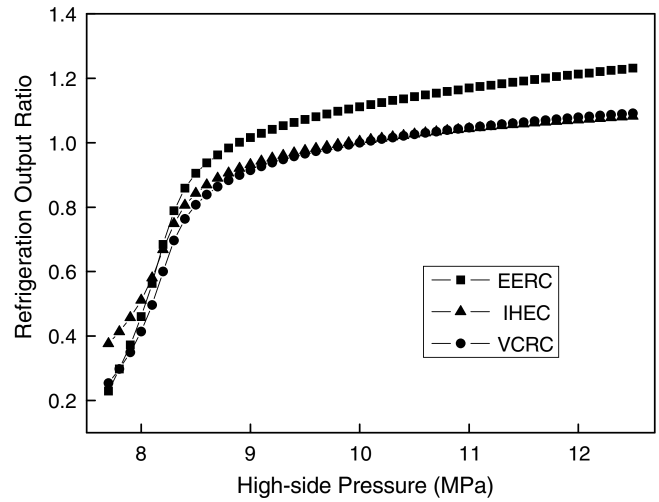

Fig. 8. Refrigeration output of the ejector expansion cycle, internal heat exchanger cycle and conventional cycle for various high-side pressures.

for the conventional system. Fig. 7 shows that the ejector expansion system has highest exergy efficiency for most of the high-side pressure range.

The exergy losses in each process in the three cycles are listed in Table 1 for a high-side pressure of 8.7 MPa which results in the maximum COP for all three cycles. The other parameters were given in Section 2. The ejector entrainment ratio is 0.547 for the ejector expansion cycle. The exergy losses in the ejector expansion cycle are based on a unit CO_2 mass flow rate compressed by the compressor (instead of the unit mass flow of mixture in the ejector).

As listed in Table 1, the throttling exergy loss in the conventional cycle is 12.8 kJ kg^{-1}, 34.3% of the total exergy loss. However, the throttling exergy loss in the ejector expansion system is only 0.39 kJ kg^{-1} with the ejection exergy loss of 8.1 kJ kg^{-1}, the sum of the two losses is 29.7% of the total system exergy loss. The exergy loss in the ejection process in the ejector expansion system is due to a process that is equivalent to the throttling losses in the other cycles. The exergy losses in the compression and heat rejection processes are also reduced some in the ejector expansion system. The exergy loss in the evaporation process is the largest loss in the ejector expansion system.

The refrigeration output is the important performance for the refrigeration system. In the ejector expansion system, the refrigerant into the compressor is at P_s which is higher than the evaporator pressure, P_e, so the CO_2 density entering the compressor is higher than that entering the compressor in the other two cycles. Therefore, in the ejector expansion system, the compressor compresses more mass. More importantly, the refrigerant compressed by the compressor is not directly providing cooling because only the liquid fraction of the refrigerant flow from the ejector flows through the evaporator. Therefore, for the three cycles can be compared for the same compressor conditions by comparing the refrigeration output:

$$Q = V\rho w \cdot \mathrm{COP} \tag{21}$$

The equivalent refrigeration output of the three cycles for various high-side pressures are showed in Fig. 8 with the refrigeration output ratio defined as the refrigeration output relative to the capacity of the conventional cycle at an evaporator temperature of 5 °C and a high-side pressure of 10 MPa. For refrigeration output defined at an evaporator temperature of 5 °C and a high-side pressure of 10 MPa, where the density at the compressor inlet is 114.4 kg m^{-3}, the specific power is 51.1 kJ kg^{-1}, and the

Table 1
Exergy losses in all three cycles for a high-side pressure of 8.7 MPa

Process	VCRC		IHEC		EERC	
	Loss (kJ kg^{-1})	(%)	Loss (kJ kg^{-1})	(%)	Loss (kJ kg^{-1})	(%)
Compression	9.56	25.69	10.74	24.38	6.79	23.73
Heat rejection	5.50	14.76	9.75	22.13	4.30	15.02
Ejection	–	–	–	–	8.12	28.38
Throttling	12.77	34.29	10.03	22.78	0.39	1.36
Evaporation	9.40	25.25	11.50	26.10	9.02	31.51
IHE	–	–	2.03	4.60	–	–
Total	37.229	100.0	44.04	100.0	28.64	100.0
Specific power	40.31	–	47.82	–	31.64	–

cycle COP is 2.62. The compressor volume displacement is assumed to be constant. The ejector expansion cycle refrigeration output is distinctly greater than the other two cycles for high-side pressures greater than 8.5 MPa. For example, for the high-side pressure of 8.7 MPa corresponding to the maximum COP, the ejector expansion refrigeration output is 8.2% higher than the internal heat exchanger cycle refrigeration output and 11.5% higher than the conventional cycle refrigeration output.

4.3. EERC performance

Fig. 9 shows the variation of the ejector expansion cycle cooling COP for various high-side pressures and evaporator temperatures of 0, 5 and 10 °C. The COP increases with evaporator temperature, with a maximum COP at 5 °C that is 20.6% higher than the maximum at 0 °C.

Fig. 10 shows the variation of the ejector expansion cooling COP with high-side pressure for various CO_2 temperatures at the gas cooler exit. The COP increases and the optimum high-side pressure corresponding to the maximum COP decreasing as the CO_2 temperature approaches the heat sink temperature. For the same evaporator temperature of 5 °C, the COP at a gas cooler exit temperature of 36 °C is 23.0% more than at 40 °C.

Fig. 11 shows the variation of the COP with the ejector diffuser efficiency for various high-side pressures. The COP increases with increasing diffuser efficiency, but the influence of the high-side pressure on the COP is much more significant than the effect of the diffuser efficiency. Fig. 12 shows the variation of the COP with the ejector nozzle efficiency for various high-side pressures. As with the diffuser efficiency, the COP increases as the nozzle efficiency increases, but the effect is not significant.

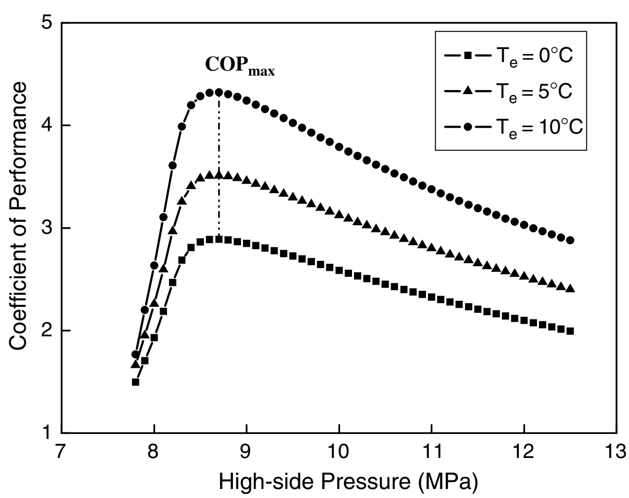

Fig. 9. Ejector expansion cycle cooling COP for various high-side pressures and evaporator temperatures.

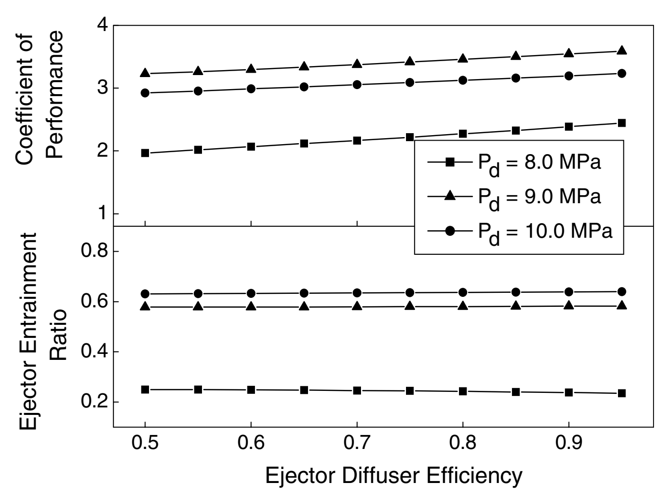

Fig. 11. Ejector expansion cycle cooling COP for various ejector diffuser efficiencies and high-side pressures.

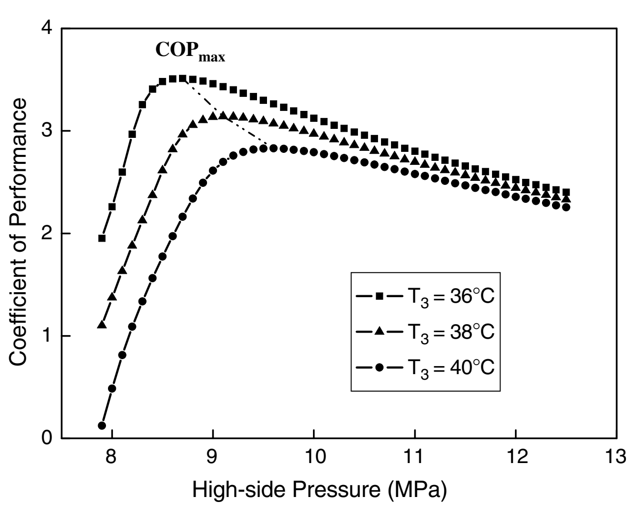

Fig. 10. Cooling COP for various high-side pressures and CO_2 temperatures at the gas cooler exit.

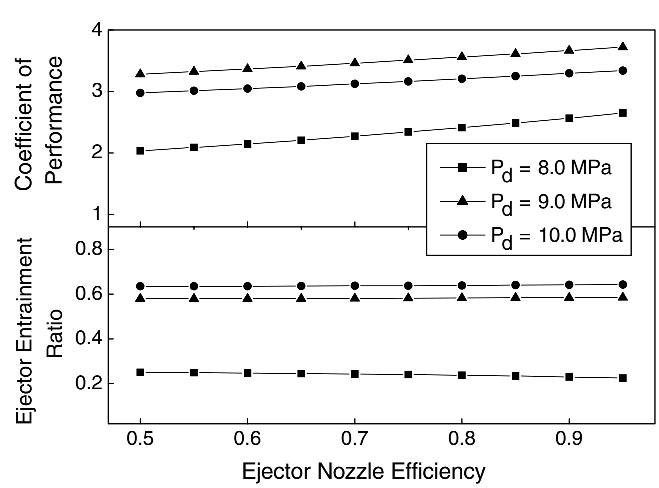

Fig. 12. Ejector expansion cycle cooling COP for various ejector nozzle efficiencies and high-side pressures.

Most prototypes of CO_2 systems have used small diameter or flat multiport tubes in the evaporators to handle the high pressure without adding heat exchanger weight. However, these evaporators face the challenging problem of how to distribute the developing two-phase flow from the header uniformly into many tubes. In a conventional cycle, the void fraction at the evaporator inlet exceeds 0.8 after about 6 MPa of throttling. But in the ejector expansion cycle, the pressure difference due to throttling is only about 2 MPa, so the mass flashed into vapor is greatly reduced and the two-phase flow can be more easily distributed. Therefore, the evaporator tube arrangement is not as critical in the ejector expansion cycle.

5. Conclusions

The COP and the refrigeration output of the transcritical CO_2 ejector expansion cycle were analyzed based on a detailed thermodynamic model of the system. The ejector expansion cycle performance was then compared with that of the internal heat exchanger system and a conventional vapor compression system. The analyses led to the following conclusions:

1. The ejector expansion cycle has a unique ejector entrainment ratio for a steady running system with a given set of working conditions. The system also has an optimum ejector entrainment ratio which gives the maximum system cooling COP.
2. In transcritical CO_2 refrigeration systems, the ejector expansion cycle cooling COP is higher than that of the internal heat exchanger cycle and the conventional cycle. For the working conditions described in this paper, the ejector improves the maximum COP by up to 18.6% compared to the internal heat exchanger system and by 22.0% compared to the conventional system.
3. In the transcritical CO_2 ejector expansion cycle, the throttling exergy loss is much less than in the internal heat exchanger cycle and in the conventional cycle. The exergy losses due to the compression and heat rejection processes are also reduced some. The evaporation exergy loss is not changed much.
4. The ejector expansion cycle refrigeration output is improved by up to 8.2% compared to the internal heat exchanger cycle refrigeration output and by 11.5% compared to the conventional cycle refrigeration output for the conditions used in this analysis.
5. The ejector expansion cycle performance is very sensitive to the operating conditions. This paper compares the influences of high-side pressure, evaporator temperature, CO_2 temperature at the gas cooler exit, ejector entrainment ratio, nozzle efficiency and diffuser efficiency on the cycle cooling COP.

The ejector expansion cycle has not been extensively studied even though the present study of the transcritical CO_2 ejector expansion cycle showed that the system performance can be improved by 25% relative to the conventional cycle which agrees with the experimental results given by the Denso Corporation [1]. However, many other experimental tests have obtained no more than 8% improvement perhaps because they did not use an optimum ejector design matched to the system operating conditions. Further study of the ejector expansion refrigeration cycle is needed to experimentally verify these results.

Acknowledgement

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