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A novel transcritical CO_2 refrigeration cycle with two ejectors

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ABSTRACT

In recent years, CO_2 is being revisited as a fully environmentally friendly and safe refrigerant. However, basic CO_2 transcritical refrigeration cycle suffers from large expansion loss due to high pressure difference between gas cooler and evaporator. Then, it is crucial to find effective and economic way to reduce the expansion loss. Here, a novel cycle with two ejectors is proposed for the first time. Compared with conventional ejector-expansion CO_2 cycle with only one ejector, this novel cycle with two ejectors is able to recover more expansion loss, thus improving the system performance further. A computational model is designed to simulate the double ejector CO_2 cycle. Simulation results show its high system COP. Effects of parameters, such as ejector nozzle efficiency, gas cooler pressure, entrainment ratios of the two ejectors, gas cooler outlet temperature, on the cycle performance are also analyzed by using the computational model.

Nouveau cycle frigorifique au CO_2 transcritique employant deux éjecteurs

Mots clés : Dioxyde de carbone ; R744 ; Cycle transcritique ; Éjecteurs ; Modélisation ; COP

1. Introduction

Carbon dioxide is a promising refrigerant due to its environment-benign nature (Lorentzen, and Pettersen, 1990; Kim et al., 2004). However, previous literature (Elbel and Hrnjak, 2008; Li and Groll, 2005; Nickl et al., 2005; Robinson and Groll, 1998; Sarkar et al., 2005, 2008, 2009; Yang et al., 2009; Yari, 2009) have reported that high pressure drop in basic transcritical CO_2 refrigeration cycle results in much larger thermodynamic expansion loss compared to conventional refrigeration cycles.

In order to recover throttling loss, several measures have been proposed. Replacement of the expansion valve by an expander (Robinson and Groll, 1998; Nickl et al., 2005; Yang

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Nomenclature

a	area per unit total ejector flow rate
COP	coefficient of performance
h	specific enthalpy
m	mass flow rate
P	pressure
Q	heat capacity
q	specific heat capacity
R	relative performance
s	specific entropy
t	temperature
u	velocity
v	specific volume
W	work load
x	quality

Greek symbols

η	isentropic efficiency
ω	entrainment ratio of the ejector

Subscripts and superscripts

b	receiving chamber or basic transcritical CO_2 cycle
comp	compressor
d	diffuser
drop	pressure drop in the receiving section of the ejector
e	evaporator
f	saturated liquid
g	saturated vapor
gc	gas cooler
is	isentropic process
m	motive nozzle
mb	motive flow at receiving chamber
mi	motive flow at nozzle inlet
mix	outlet of mixing section
n	ejector expansion transcritical CO_2 cycle
o	outlet
s	suction nozzle
sb	suction flow at receiving chamber
sh	superheat
si	suction flow at nozzle inlet
I	the first ejector
II	the second ejector

et al., 2009) is a direct measure. Although replacing the expansion valve with a turbine can significantly improve the performance of CO_2 transcritical cycle, such extensive hardware addition may not be economically feasible for many practical applications, especially for small capacity CO_2 cycle (Sarkar et al., 2005). Other measures include using ejector-expansion device (Sarkar et al., 2005; Li and Groll, 2005; Sarkar, 2008; Elbel and Hrnjak, 2008; Yari, 2009; Robinson and Groll, 1998; Yang et al., 2009; Sarkar, 2009) or vortex tube (Sarkar, 2009) to replace the expansion valve. This ejector-expansion device has advantages, such as low cost, no moving parts and ability to handle two-phase flow without damage, making it attractive for the development of high-performance CO_2 refrigeration system (Yari, 2009).

Li and Groll (2005) performed a thermodynamic analysis with respect to a transcritical CO_2 cycle of different expansion devices. It was found that the COP of the ejector-expansion transcritical CO_2 cycle can be improved by more than 16% over the basic transcritical CO_2 cycle for typical air conditioning operation conditions. Sarkar (2008) presented an optimization study along with optimum parameter correlations, using constant area mixing model (Li and Groll, 2005) for an ejector-expansion transcritical CO_2 heat pump cycle with either conventional or modified layout. He pointed out that the ejector may be the best alternative expansion device at least for low-capacity transcritical CO_2 heat pump systems. Elbel and Hrnjak (2008) conducted experimental validation of a prototype ejector designed to reduce throttling losses encountered in transcritical CO_2 system operation. Their experimental results showed that for the best conditions considered, the cooling capacity and COP were simultaneously improved by up to 8% and 7%, respectively.

Under typical air-conditioning operation conditions, the pressure difference across the throttling valve is reduced from 6 to 7 MPa for basic CO_2 cycle to about 3–4 MPa for the CO_2 cycle with one ejector. However, compared with R134a or R22 cycle, 3–4 MPa pressure difference across the throttling device is still quite large. That is to say, there is still a lot of expansion loss needs further recovering. Thus, to solve this problem we propose a double ejector-expansion CO_2 cycle in the present paper for the first time. The expansion loss will be recovered twice in this novel double ejector-expansion CO_2 cycle and COP may be further improved compared with conventional single-ejector cycle.

To understand the characteristics of the novel double ejector-expansion CO_2 cycle, we adopt the ejector model of Li and Groll (2005) for the thermodynamic simulation of the new cycle. Effects of some parameters on the performance of this new cycle are theoretically analyzed.

2. Double ejector-expansion CO_2 transcritical cycle layout

Based on the single-ejector cycle proposed by Li and Groll (2005), a double-ejector cycle, schematically shown in Fig. 1, is proposed. The p-h diagram of this cycle is depicted in Fig. 2. The cycle is composed of a compressor, a gas cooler, two ejectors (ejector I and ejector II), two separators (separator I and separator II), four throttling valves and an evaporator. The working process of the cycle is described in detail as follow:

One unit mass of compressed CO_2 stream in supercritical state is introduced into a gas cooler. The cooled-down stream then enters ejector I as the primary flow to eject ω_1 unit mass of low-pressure fluid from separator II. The $1 + \omega_1$ unit mass of fluid mixes and passes the diffuser of ejector I and flows into separator I. Due to the reason described by Li and Groll (2005), the quality (χ_1) of separator I must be larger than the mass flow rate of the stream sucked into the compressor to

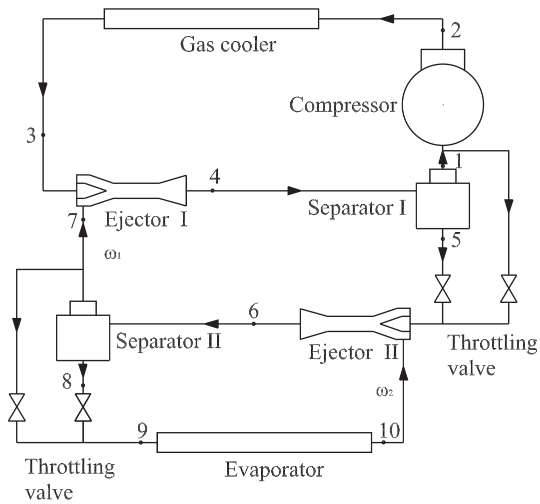

Fig. 1 – Schematic diagram of CO_2 cycle with double ejector-expansion device.

ensure mass conservation and maintain a steady-state operation, i.e., $(1+\omega_1)\chi_1 > 1$. Consequently, the excess vapor is combined with liquid flow through a bypass throttle valve and flows into ejector II. The liquid and part of the vapor from separator I serve as the primary flow of ejector II and mix with ω_2 unit mass of suction flow from the evaporator. The mixed flow passes the diffuser of ejector II and enters separator II. Likewise, the quality of separator II must satisfy the condition, $((1+\omega_1)\chi_1 - 1 + \omega_2)\chi_2 > \omega_1$. The ω_1 unit mass of the vapor is sucked into ejector I and the residual vapor and the liquid of separator II passes through the throttling valve and enters into the evaporator. The stream flowing out of the evaporator is sucked into ejector II. Then a closed loop is completed. It should be noted that one may introduce more ejectors to the cycle using the same way described above. However, too many ejectors make the refrigeration cycle more complicated to control and more expensive, thus not feasible in practice.

From Fig. 2, it is seen that the pressure at the compressor inlet 1 is much higher than that at the evaporator outlet 10.

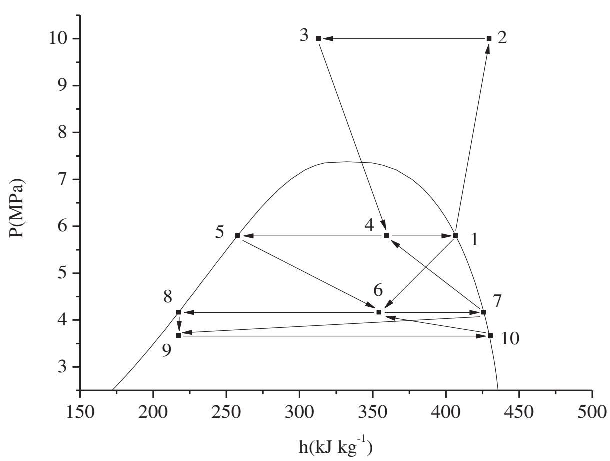

Fig. 2 – p-h diagram of CO_2 cycle with double ejector-expansion devices.

Therefore, the work consumption of the compressor is largely reduced; the pressure drop 8–9 across the throttling valve is also much smaller than the conventional cycles, including the basic cycle without ejector and the ejector-expansion cycle with one ejector.

3. Mathematical modeling and simulation

The ejectors are important components of the cycle. Firstly, an iterative calculation sub-routine of ejector modeling is developed. This ejector modeling module is inserted into a thermodynamic analysis model designed for the whole double-ejector cycle. To facilitate the modeling and theoretical analyses, the following assumptions are made:

(i) The cycle is operated at a stable condition. Pressure drops in the gas cooler, evaporator and the connection tubes are neglected.
(ii) The system is a closed loop, with no heat exchange with the environment.
(iii) The evaporator has a given outlet superheat and the gas cooler has a given outlet temperature.
(iv) Vapor stream from the separator and liquid stream from the separator are saturated fluids.
(v) The flow across the expansion valve or the throttle valves is isenthalpic.
(vi) The compressor has a given isentropic efficiency.

(vii) The flow in the ejector is considered one-dimensional homogeneous equilibrium flow.

(viii) The motive flow and suction flow share a same pressure at the inlet of the constant area mixing section of the ejector. There is no mixing between the two flows before reaching the inlet of the constant area mixing section.
(ix) The expansion efficiencies of the motive flow and suction flow are given constants. The diffuser of the ejector also has a given efficiency.
(x) Kinetic energies of the refrigerant at the ejector inlet and outlet are negligible.

3.1. *Ejector modeling*

Working process of an ejector is schematically shown in Fig. 3. The employed ejector model is similar to Li and Groll's (2005). Fig. 4 displays the iterative flowchart of the two-phase ejector calculation sub-routine together with governing equations involved. This model requires knowledge of the refrigerant states at inlets to the motive and suction nozzles, respectively. Firstly, the pressure at the inlet of the constant area mixing section of the ejector P_b and entrainment ratio of mass flow rate of motive flow to suction flow ω are set. The calculation of the specific enthalpy of the motive flow at the exit plane of the nozzle involves using an equation of sate and an assumption for the motive nozzle efficiency defined by Eq. (1). Then velocity of the motive flow at the exit plane of the nozzle can be calculated by using the energy conservation equation. Using the mass conservation equation, the area occupied by

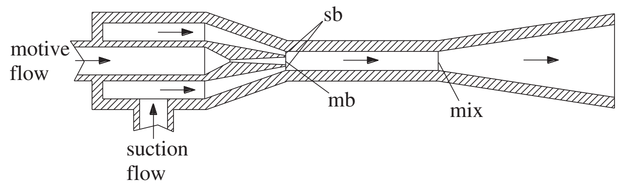

Fig. 3 – Schematic of ejector working process.

the motive stream at the inlet of constant area mixing section per unit total ejector flow rate a_{mb} is calculated. Similarly, the area occupied by the suction stream at the inlet of constant area mixing section per unit total ejector flow rate a_{sb} is calculated by assuming an efficiency (expressed by Eq. (2)) and by involving an equation of state, equations of mass and energy conservation.

To determine refrigerant state at the outlet of the mixing section, an iteration loop is applied. First, the outlet pressure P_m is guessed. By virtue of momentum and energy conversation, the velocity and the enthalpy of the mixing stream at the mixing section outlet can be found. The specific volume of the mixing stream can be found by using an equation of state. A subsequent check based on the conservation of mass for constant area mixing section determines if the initially guessed mixing section outlet pressure P_m needs to be updated for next iteration.

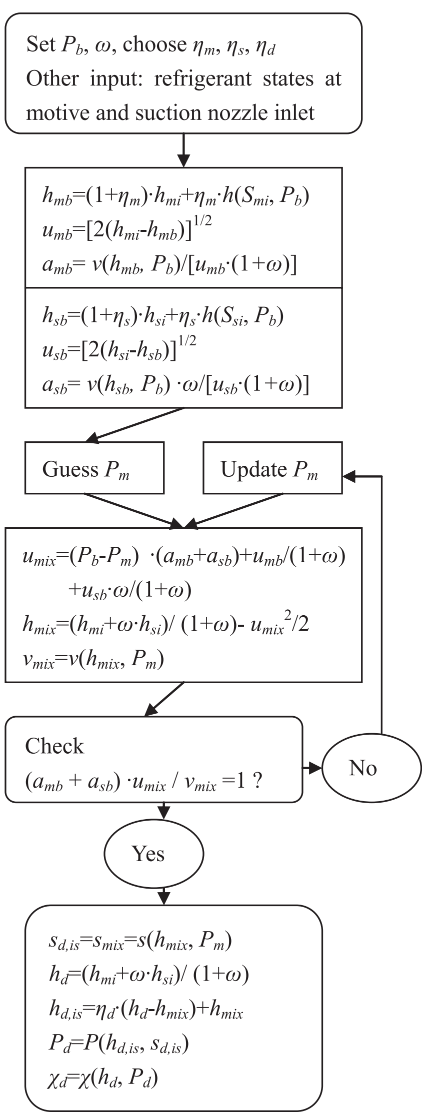

Fig. 4 – Flowchart of the ejector calculation sub-routine.

Once the iteration loop is terminated as mass conservation for constant area mixing section is satisfied, the calculation of the diffuser section of the ejector is followed. The entropy of the mixing stream at the outlet of the mixing section is obtained by using an equation of state. The stream enthalpy at the diffuser outlet can be calculated by using the energy conservation equation. The isentropic enthalpy at the diffuser outlet is calculated by assuming an efficiency (expressed by Eq. (3)). The diffuser outlet pressure and quality are then obtained using an equation of state.

$$\eta_m = \frac{h_{mi} - h_{mb}}{h_{mi} - h_{mb,is}} \tag{1}$$

$$\eta_{sb} = \frac{h_{si} - h_{sb}}{h_{si} - h_{sb,is}} \tag{2}$$

$$\eta_d = \frac{h_{d,is} - h_m}{h_d - h_{mix}} \tag{3}$$

It should be noted that entrainment ratio of an ejector and the ejector outlet quality must satisfy Eq. (4) in order to realize the cycle.

$$(1+\omega)\chi_d > 1 \tag{4}$$

The ejector subroutine aforementioned will be called by the main program twice. In addition, it should be noticed that ω in this subroutine denotes the mass flow rate ratio of suction flow to motive flow. When it applied to ejector II, the mass flow rate of the motive flow should not be equal to 1. And in this paper, ω_2 is the mass flow rate ratio of suction flow to the flow across the compressor, instead of motive flow.

3.2. *Double ejector transcritical cycle modeling*

Process diagram for modeling the whole double ejector transcritical cycle is depicted in Fig. 5. First, some basic parameters of the cycle such as T_{ev}, P_{gc}, $T_{gc,o}$, T_{sh}, P_{drop}, ω_1, ω_2, are input. Three efficiencies of the ejector mentioned above and the isentropic efficiency of the compressor η_c are preset. The pressure at the inlet of ejector I suction nozzle P_{si1} is guessed to start the iteration of this cycle calculation. Using an equation of state, refrigerant states at the inlets of the motive and suction nozzles of ejector I can be obtained. Then the ejector calculation sub-routine is called to calculate the pressure and quality at the diffuser outlet of ejector I. We then calculate the refrigerant states at the inlets of the motive and suction nozzles of ejector II using an equation of state and a given entrainment ratio for ejector I, ω_1. The ejector calculation subroutine is called again to calculate the pressure and quality at the diffuser outlet of ejector II. A subsequent check is performed to judge if the initially guessed pressure P_m at the outlet of the mixing section needs to be updated for the next iteration. The program checks three conditions, both ejector

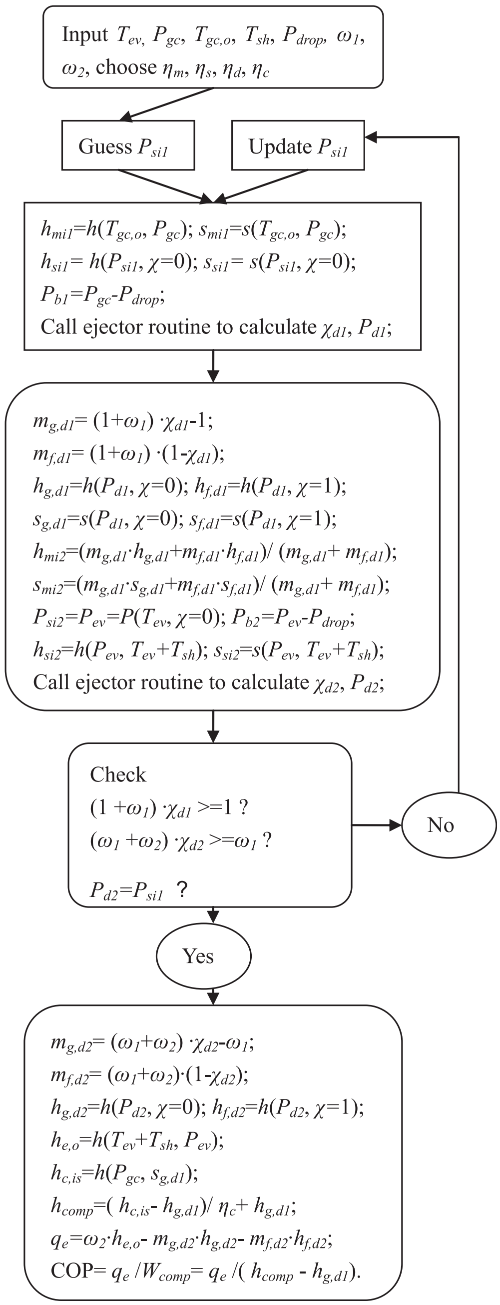

Fig. 5 – Flowchart of the calculation program designed for the double ejector refrigeration cycle.

outlets must satisfy Eq. (4) to realize the cycle and the calculated P_{d2} needs to be equal to the previous guessed P_{si1}. If the three conditions are all satisfied the refrigerant states at the inlet and outlet of the evaporator can be calculated in terms of an equation of state and by using the preset entrainment ratio ω_2. The compressor work is obtained once the specific enthalpy at the outlet of the compressor is determined in terms of an equation of state and the compressor isentropic efficiency defined by Eq. (5).

$$\eta_{comp} = \frac{h_{comp,is} - h_{g,d}}{h_{comp} - h_{g,d}} \tag{5}$$

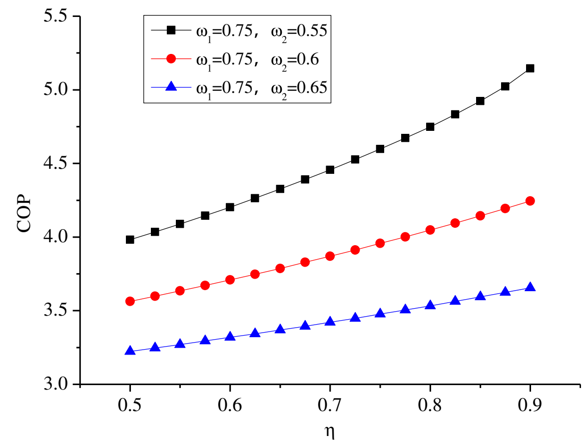

Fig. 6 – Variation of COP with nozzle efficiencies.

The final output of the computational program is the cycle COP.

4. Results and discussion

To investigate characteristics of the double ejector cycle, a base case is assumed: $P_{gc} = 10$ MPa, $T_{gc,o} = 40$ °C, $T_e = 5$ °C, $T_{sh} = 5$ °C, $\eta_m = \eta_s = 0.9$, $\eta_d = 0.8$, $\eta_{comp} = 0.75$, $P_{drop} = 0.03$ MPa. In the following case studies, we vary one of these operating parameters within a certain range to study the parameter-dependency of the system performance.

Ejectors are crucial components for a CO_2 double ejector refrigeration or heat pump cycle, and need appropriate and careful design. Low efficiency ejector will result in bad system performance, even worse than the basic CO_2 cycle system, and the double ejector cycle system may lose its advantages. Fig. 6 shows the effect of motive, suction and diffuser nozzle efficiencies on the system COP. Nozzle efficiencies are assumed to be equal. From Fig. 6, it is obvious that COP increases with increasing nozzle efficiency. We must confess that the ejector nozzle efficiencies are arbitrarily set due to lack of

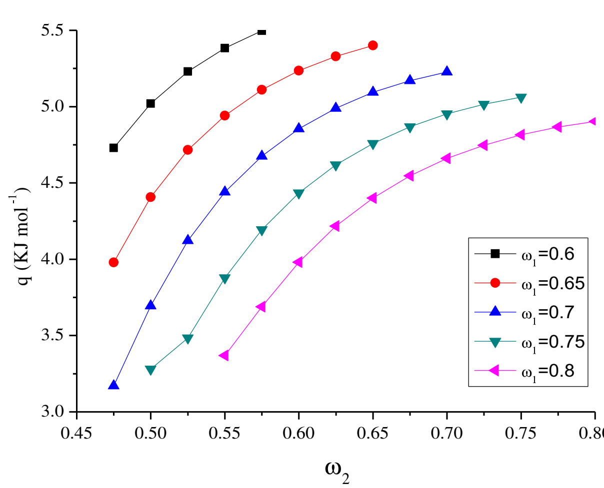

Fig. 7 – Variation of cooling capacity with entrainment ratios.

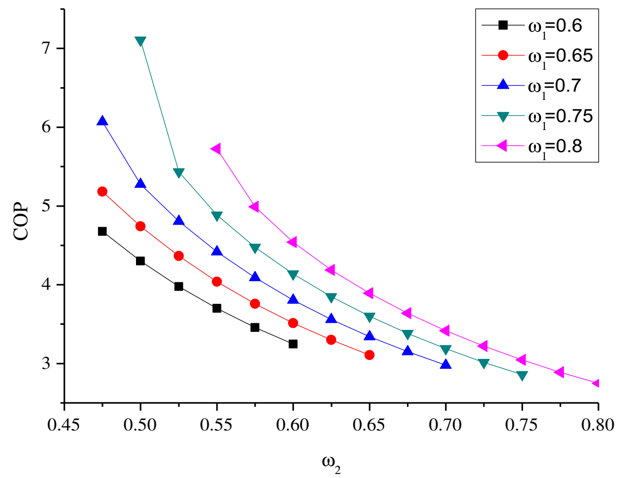

Fig. 8 – Variation of COP with entrainment ratios.

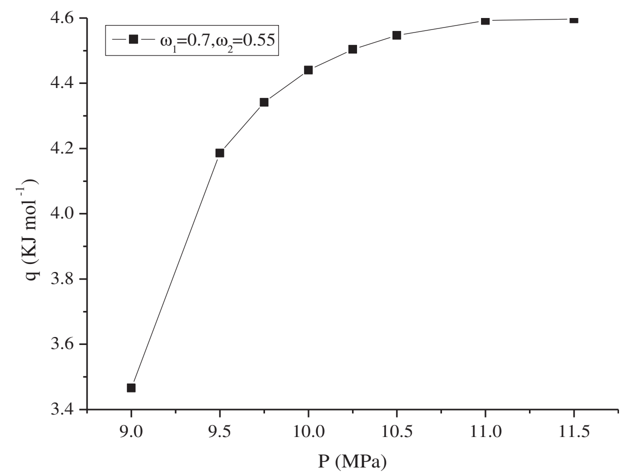

Fig. 10 – Variation of cooling capacity with gas cooler pressure.

experimental data. These efficiency values used in these figures are probably too high, which makes the calculated COP seemingly too high and may not be realized in practice. Therefore, it is worth pointing out that the present work concentrates only on analyzing the general trend about parameter-dependency of the system performance, instead of aim to provide exact values about the system performance or validate any experimental data.

Figs. 7 and 8 show effects of the two ejector entrainment ratios on cooling capacity and COP, respectively. With increase of ω_2 and decrease of ω_1, the cooling capacity increases and COP decreases. Both increase of ω_2 and decrease of ω_1 make the mass flow rate across the evaporator increase, and reduce the mass flow rate contributing to the recovery of expansion loss. Consequently, the cooling capacity increases and COP decreases.

From Fig. 9, it can be seen that the ejector outlet pressure increases with the decrease of ω_2. Larger ω_2 implies that mass flow rate through the evaporator is larger, the expansion loss before refrigerant stream enter the evaporator is also enlarged and the mass flow rate used for expansion loss recovery is reduced. A smaller suction pressure increases the energy consumption of the compressor and results in smaller COP.

Figs. 10 and 11 show effects of gas cooler pressure on the cooling capacity and COP, respectively. As displayed by Fig. 10, the cooling capacity increases, at a gradual decreasing rate, with the increase of the gas cooler pressure. The capacity curve almost levels off when gas cooler pressure exceeds about 10 MPa. It is seen from Fig. 11 that there exists an optimal gas cooler pressure, about 10 MPa, at which the system COP reaches its maximum value. Increase of gas cooler pressure may lead to increasing cooling capacity, more compressor power consumption due to high compressing ratio, higher expansion loss. The increasing cooling capacity has positive influence on the system performance, while the other effects bring negative influence. This is the reason why COP becomes decreasing when the gas cooler pressure is beyond about 10 MPa.

Fig. 12 shows that COP of the double ejector cycle increases with increasing evaporation temperature. The double ejector cycle is more suitable for working under conditions of high evaporation temperature. It is worth pointing out that at any

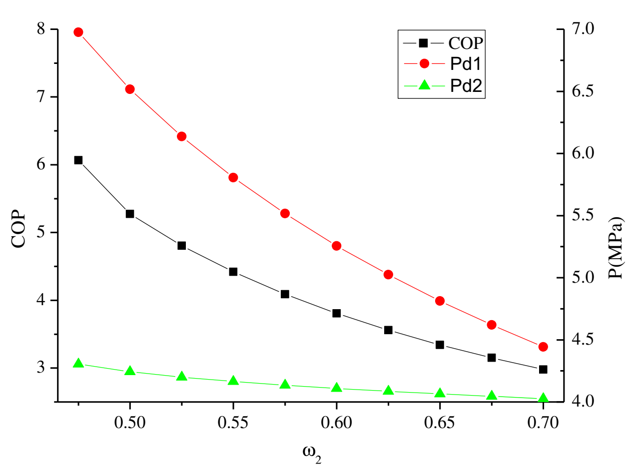

Fig. 9 – Variation of COP and two ejector outlet pressures with ω_2 (ω_1 = 0.7).

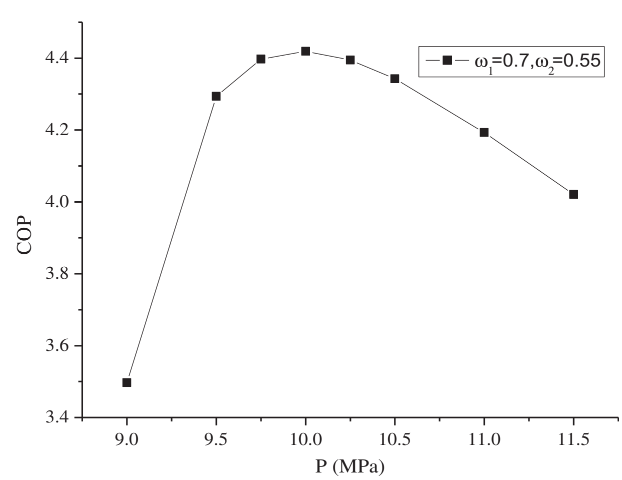

Fig. 11 – Variation of COP with gas cooler pressure.

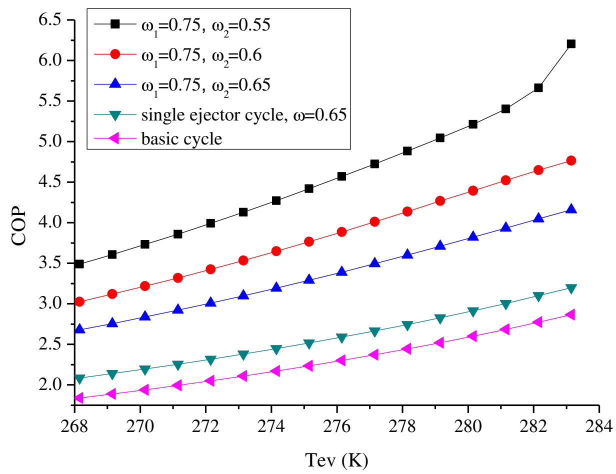

Fig. 12 – Variation of COP with evaporation temperature.

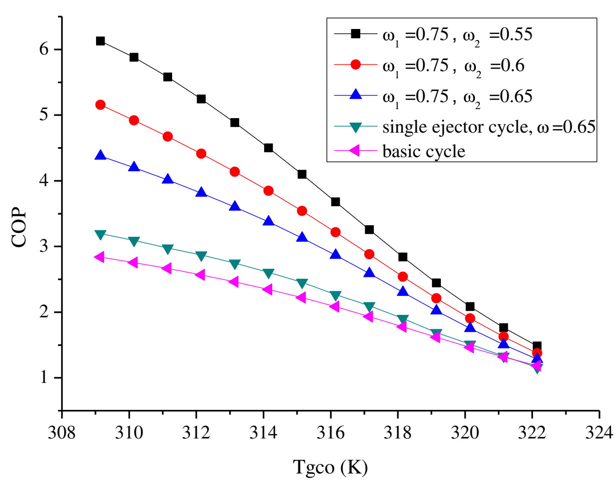

Fig. 13 – Variation of COP with gas cooler outlet temperature.

evaporation temperature, the system COP of double ejector cycle calculated is much higher than that of the basic CO_2 cycle and single ejector cycle. The modeling of single ejector cycle is the same as Li and Groll (2005).

Fig. 13 shows that COP decreases rapidly with increasing gas cooler outlet temperature. Therefore, having a double ejector cycle work with a sufficiently low temperature at the outlet of gas cooler can better realize its superiority. Again, it is worth pointing out that the COP of double ejector cycle is always higher than that of the basic CO_2 cycle and single ejector cycle.

5. Conclusion

A novel CO_2 transcritical refrigeration cycle with two ejectors is proposed in this work. The most advantage of this complicated cycle is that it can further recover the expansion loss of supercritical CO_2 compared with the single-ejector cycle. Computational model of the double ejector CO_2 cycle is developed to analyze the system characteristics. From the calculated results, conclusions can be drawn as follows.

Nozzle efficiencies of the ejectors have significant effect on the system COP. With the increase of ω_2 and decrease of ω_1, the cooling capacity increases and COP decreases. Similar to the basic CO_2 transcritical refrigeration cycle and single-ejector cycle, the cooling capacity increases with increasing gas cooler pressure and there exists an optimal gas cooler pressure, at which COP reaches its maximum value. COP of the double-ejector cycle increases with increasing evaporation temperature and decreases rapidly with increasing gas cooler outlet temperature. At any evaporation temperature and gas cooler outlet temperature, COP of the double-ejector cycle is higher than that of the basic CO_2 cycle.

Acknowledgments

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