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Transcritical CO_2 refrigeration cycle with ejector-expansion device

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Abstract

An ejector expansion transcritical CO_2 refrigeration cycle is proposed to improve the COP of the basic transcritical CO_2 cycle by reducing the expansion process losses. A constant pressure mixing model for the ejector was established to perform the thermodynamic analysis of the ejector expansion transcritical CO_2 cycle. The effect of the entrainment ratio and the pressure drop in the receiving section of the ejector on the relative performance of the ejector expansion transcritical CO_2 cycle was investigated for typical air conditioning operation conditions. The effect of different operating conditions on the relative performance of the ejector expansion transcritical CO_2 cycle was also investigated using assumed values for the entrainment ratio and pressure drop in the receiving section of the ejector. 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.

Keywords: Carbon dioxide; Modelling; Simulation; Transcritical cycle; Expansion device; Ejector; COP

Cycle transcritique au CO_2 muni d'un détendeur à éjecteur

Mots clés : Dioxyde de carbone ; Modélisation ; Simulation ; Cycle transcritique ; Détendeur ; Éjecteur ; COP

1. Introduction

Many researchers have analyzed the performance of the transcritical CO_2 refrigeration cycle in order to identify opportunities to improve the system's energy efficiency. By performing a second law analysis, Robinson and Groll (1998) [6] found that the isenthalpic expansion process in a transcritical CO_2 refrigeration cycle is a major contributor to the cycle irreversibility due to the fact that the expansion process takes a path from the supercritical region into the two-phase region. Brown et al. (2002) [1] presented an evaluation of carbon dioxide as an R-22 substitute for residential air conditioning applications. The performance of CO_2 and R-22 in residential air-conditioning applications was compared using semi-theoretical vapor compression and transcritical cycle models. It was found that the R-22 system had a significantly better COP than the CO_2 system when equivalent heat exchangers were used in the CO_2 and R-22 systems. An entropy generation analysis showed that the highest level of irreversibility occurred in the CO_2 expansion device, and together with the irreversibility in the gas cooler, were greatly responsible for the low COP of the CO_2 system. Therefore, the reduction of the expansion process losses is one of the key issues to improve the efficiency of the transcritical CO_2 refrigeration cycle.

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Nomenclature

a	area per unit total ejector flow rate	*Subscripts and superscripts*	
COP	coefficient of performance	b	receiving chamber or basic transcritical CO_2 cycle
h	specific enthalpy	comp	compressor
m	mass flow rate	d	outlet of diffuser
P	pressure	e	evaporator
Q	heat capacity	f	saturated liquid
q	specific heat capacity	g	saturated vapor
R	relative performance	is	isentropic process
s	specific entropy	mb	motive stream at receiving chamber
t	temperature	mi	motive stream at nozzle inlet
u	velocity	mix	outlet of mixing section
v	specific volume	n	ejector expansion transcritical CO_2 cycle
W	work load	o	outlet
w	entrainment ratio of the ejector	sb	suction stream at receiving chamber
x	quality	si	suction stream at nozzle inlet
η	isentropic efficiency		

A free piston expander–compressor unit was proposed by Heyl et al. (1998) [2] to recover the expansion process losses. However, implementation of the concept requires a two-stage refrigeration cycle and complicated flow control devices. Li et al. (2000) [3] performed a thermodynamic analysis of different expansion devices for the transcritical CO_2 cycle. A vortex tube expansion device and an expansion work output device were proposed to recover the expansion losses. The maximum increase in COP using a vortex tube or expansion work output device, assuming ideal expansion process, was about 37% compared to the one using an isenthalpic expansion process. The increase in COP reduced to about 20% when the efficiency for the expansion work output device was 0.5. In order to achieve the same improvement in COP using a vortex tube expansion device, the efficiency of the vortex tube had to be above 0.38. Liu et al. (2002) [4] performed a thermodynamic analysis of the transcritical CO_2 vapor-compression/ejection hybrid refrigeration cycle based on the idea proposed by Kornhauser (1990) [5]. In this cycle, an ejector is used instead of a throttling valve to recover some of the kinetic energy of the expansion process. Through the action of the ejector the compressor suction pressure is higher than it would be in a standard cycle, resulting in less compression work and improved system efficiency.

A transcritical CO_2 refrigeration cycle with an ejector expansion device is analyzed here using a constant pressure mixing model for the ejector expansion device. The system was simulated at typical air-conditioning operating conditions to investigate its performance improvement over a basic transcritical CO_2 refrigeration system.

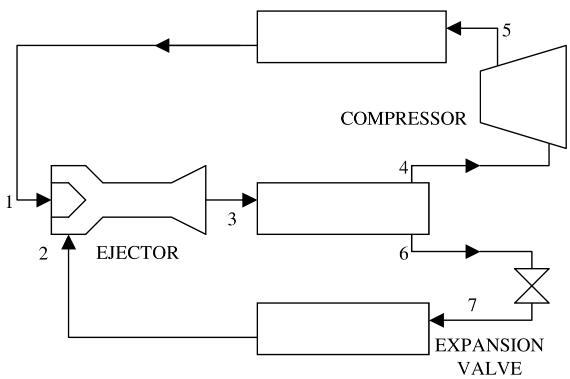

Fig. 1. Schematic of ejector expansion refrigeration cycle.

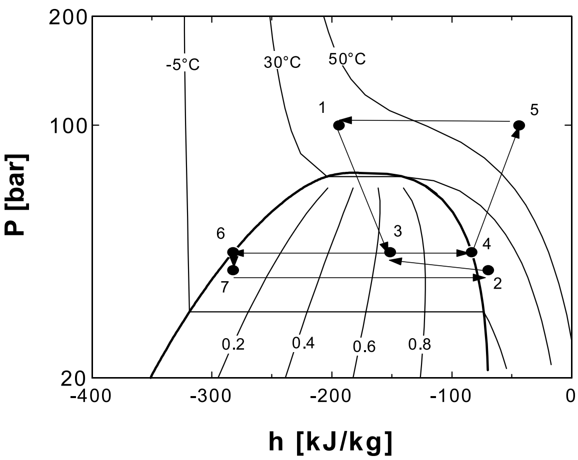

Fig. 2. Ejector expansion transcritical CO_2 refrigeration cycle in a *P–h* diagram.

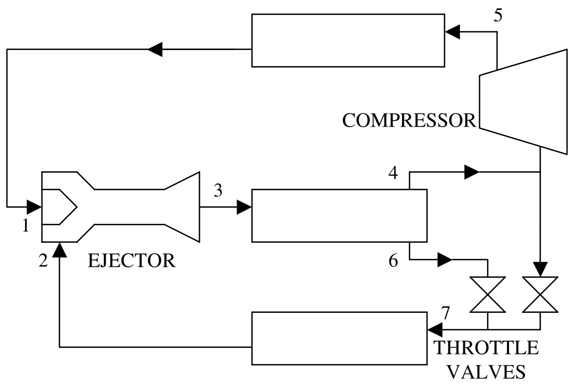

Fig. 3. Schematic of new ejector expansion refrigeration cycle.

2. Ejector-expansion transcritical CO_2 cycle

The ejector-expansion refrigeration cycle was first proposed by Kornhauser (1990) [5] as shown in the Fig. 1.

The cycle presented in Fig. 1 can be shown in a carbon dioxide *P–h* diagram as indicated in Fig. 2. It can be seen that the quality of state point four is fixed at one and the quality of state point six is fixed at zero. Thus, the entrainment ratio of the ejector, w, and the quality of the ejector outlet stream at state point three, x_3, has to satisfy $x_3 = 1/(1+w)$ to meet the mass conservation constraint for steady-state operation of the cycle. However, for a given ejector configuration, the entrainment ratio of the ejector is determined by the motive flow and suction flow and the ejector outlet pressure. This leads to a difficulty to control the operating conditions of a real system. To relax the constraints between the entrainment ratio of the ejector and the quality of the ejector outlet stream, a new ejector expansion transcritical CO_2 refrigeration cycle is proposed here as shown in Fig. 3. Part of the vapor in the separator is feed back to the evaporator inlet through a throttle valve, which regulates the quality at the evaporator inlet. The throttle valve can be controlled by the liquid level in the separator to ensure that the mass conservation is being satisfied to maintain a steady-state operation. The new ejector expansion cycle is also shown in a *P–h* diagram in Fig. 4.

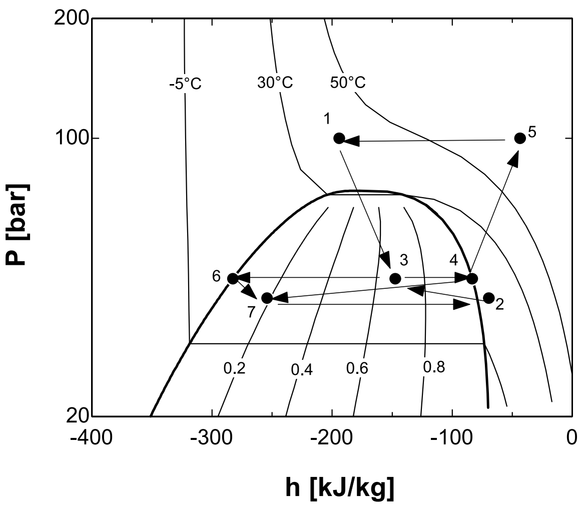

Fig. 4. New ejector expansion transcritical CO_2 refrigeration cycle in a *P–h* diagram.

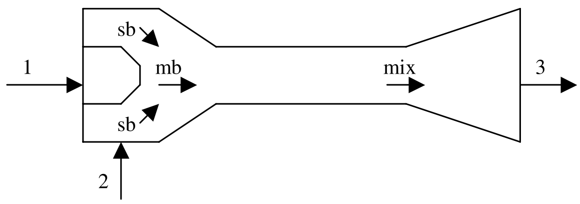

Fig. 5. Schematic of ejector working process.

The ejector working process is shown in detail as in Figs. 5 and 6. The motive stream expandes in motive nozzle from high pressure P_1 to receiving chamber pressure P_b. The enthalpy reduces from h_1 to h_{mb} and the velocity increases to u_{mb}. The suction stream expandes in suction nozzle from pressure P_2 to P_b. The enthalpy reduces from h_2 to h_{sb} and the velocity increases to u_{sb}. The two streams mix in the mixing section and become one stream with pressure P_m and velocity u_{mix}. This stream further increases its pressure to P_3 in the diffuser by converting its kinetic energy into internal energy.

3. Theoretical model

To simplify the theoretical model of the ejector expansion transcritical CO_2 refrigeration cycle, the following assumptions are made:

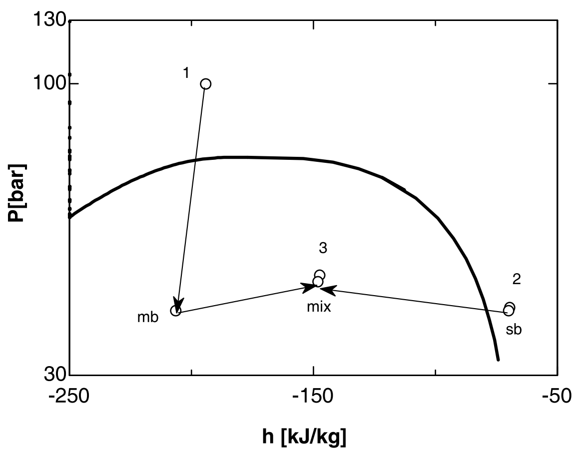

Fig. 6. Ejector working process in a *P–h* diagram of CO_2.

1. Neglect the pressure drop in the gas cooler and evaporator and the connection tubes.
2. No heat losses to the environment from the system, except the heat rejection in the gas cooler.
3. The vapor stream from the separator is saturated vapor and the liquid stream from the separator is saturated liquid.
4. The flow across the expansion valve or the throttle valves is isenthalpic.
5. The compressor has a given isentropic efficiency.
6. The evaporator has a given outlet superheat and the gas cooler has a given outlet temperature.
7. The flow in the ejector is considered a one-dimensional homogeneous equilibrium flow.
8. Both the motive stream and the suction stream reach the same pressure at the inlet of the constant area mixing section of the ejector. There is no mixing between the two streams before the inlet of the constant area mixing section.
9. The expansion efficiencies of the motive stream and suction stream are given constants. The diffuser of the ejector also has a given efficiency.

Using these assumptions, the equations for the ejector expansion transcritical CO_2 cycle were setup. Assuming that the pressure before the inlet of the constant area mixing section of the ejector is P_b and the entrainment ratio of the ejector is w, the following equations for the ejector section before the inlet of the constant area mixing section can be identified.

The motive stream is accelerated as it pressure drops from P_1 to P_b before it enters the mixing section. An isentropic expansion process is used to determine the actual exit state.

$$s_{mb,is} = s_{mi} \tag{1}$$

The corresponding enthalpy of the motive stream at the end of the isentropic expansion process can be determined from the property relationship, f, derived from the equation of state.

$$h_{mb,is} = f(s_{mb,is}, P_b) \tag{2}$$

Using the definition of expansion efficiency, the actual enthalpy of the motive stream at the inlet of the constant area mixing section of the ejector can be found.

$$\eta_m = \frac{h_{mi} - h_{mb}}{h_{mi} - h_{mb,is}} \tag{3}$$

Applying the conservation of energy across the expansion process, the velocity of the motive stream at the inlet of the constant area mixing section is given by Eq. (4).

$$u_{mb} = \sqrt{2(h_{mi} - h_{mb})} \tag{4}$$

The specific volume of the motive stream at the inlet of constant area mixing section can be found by a property relationship:

$$v_{mb} = f(h_{mb}, P_b) \tag{5}$$

Using the conservation of mass, the area occupied by the motive stream at the inlet of constant area mixing section per unit total ejector flow rate is given by:

$$a_{mb} = \frac{v_{mb}}{u_{mb}(1 + w)} \tag{6}$$

The calculation sequence for the suction stream is analogous to the one for the motive stream as shown in Eqs. (7)–(12).

$$s_{sb,is} = s_{si} \tag{7}$$

$$h_{sb,is} = f(s_{sb,is}, P_b) \tag{8}$$

$$\eta_{sb} = \frac{h_{si} - h_{sb}}{h_{si} - h_{sb,is}} \tag{9}$$

$$u_{sb} = \sqrt{2(h_{si} - h_{sb})} \tag{10}$$

$$v_{sb} = f(h_{sb}, P_b) \tag{11}$$

$$a_{sb} = \frac{v_{sb}}{u_{sb}} \frac{w}{1 + w} \tag{12}$$

To calculate the mixing section outlet conditions, an iteration loop is applied. First, a value of the outlet pressure P_m is guessed. By assuming that the momentum conservation is satisfied for the mixing process in the constant area mixing section, the velocity of the mixing stream at the mixing section outlet is calculated by using in Eq. (13).

$$P_b(a_{mb} + a_{sb}) + \frac{1}{1 + w} u_{mb} + \frac{w}{1 + w} u_{sb} = P_m(a_{mb} + a_{sb}) + u_{mix} \tag{13}$$

Using the conservation of energy, the enthalpy of the mixing stream at the mixing section outlet can be found.

$$h_{mi} + w h_{si} = (1 + w)\left(h_{mix} + \frac{1}{2} u_{mix}^2\right) \tag{14}$$

From a property relationship, the specific volume of the mixing stream can be found.

$$v_{mix} = f(h_{mix}, P_m) \tag{15}$$

In the last step, the conservation of mass for the constant area mixing section requires that Eq. (16) holds true.

$$\frac{(a_{mb} + a_{sb}) u_{mix}}{v_{mix}} = 1 \tag{16}$$

The mixing pressure is then iterated until Eq. (16) is satisfied.

In the next section, the calculations of the diffuser section of the ejector are presented. First, the entropy of the mixing stream at the outlet of the mixing section is found and set equal to the isentropic diffuser outlet entropy:

$$s_{mix} = f(h_{mix}, P_m) \tag{17}$$

$$s_{d,is} = s_{mix} \quad (18)$$

The stream enthalpy at the diffuser outlet can be found by applying the conservation of energy across the ejector:

$$(1+w)h_d = h_{mi} + wh_{si} \quad (19)$$

Given the efficiency of the diffuser, the isentropic enthalpy at the diffuser outlet can be found:

$$\eta_d = \frac{h_{d,is} - h_{mix}}{h_d - h_{mix}} \quad (20)$$

The diffuser outlet pressure and quality are then obtained from property relationships:

$$P_d = f(h_{d,is}, s_{d,is}) \quad (21)$$

$$x_d = f(h_{d,is}, P_d) \quad (22)$$

It should be noted that the entrainment ratio of the ejector and the ejector outlet quality must satisfy Eq. (23) in order to realize the cycle.

$$(1+w)x_d > 1 \quad (23)$$

Since it is assumed that the fluid streams leave the separator at saturated conditions, the gas and liquid enthalpies at the outlet of the separator can be found from property relationships:

$$h_{f,d} = f(P_d, x=0) \quad (24)$$

$$h_{g,d} = f(P_d, x=1) \quad (25)$$

Using a mass balance, the feed back vapor stream flow rate is given by:

$$m_{g,d} = (1+w)x_d - 1 \quad (26)$$

And the saturated liquid flow rate leaving the separator is given by:

$$m_{f,d} = (1+w)(1-x_d) \quad (27)$$

For a given superheat at the evaporator outlet, the enthalpy at the evaporator outlet can be found from a property relationship:

$$h_{e,o} = f(P_e, t_{e,o}) \quad (28)$$

The evaporator capacity can be calculated as:

$$Q_{o,n} = wh_{e,o} - m_{f,d}h_{f,d} - m_{g,d}h_{g,d} \quad (29)$$

The suction stream inlet enthalpy is equal to the evaporator outlet enthalpy:

$$h_{si} = h_{e,o} \quad (30)$$

To find the compression work of the compressor, first the isentropic compressor outlet conditions are evaluated:

$$s_{g,d} = f(P_d, x=1) \quad (31)$$

$$s_{comp,is} = s_{g,d} \quad (32)$$

$$h_{comp,is} = f(s_{comp,is}, P_c) \quad (33)$$

From the isentropic efficiency of the compressor, the actual enthalpy at the compressor outlet can be found:

$$\eta_{comp} = \frac{h_{comp,is} - h_{g,d}}{h_{comp} - h_{g,d}} \quad (34)$$

Then, the compression work done by the compressor can be found as:

$$W_{comp,n} = (h_{comp} - h_{g,d}) \quad (35)$$

The COP of the ejector expansion transcritical CO_2 cycle can be determined by:

$$COP_n = \frac{Q_{e,n}}{W_{comp,n}} \quad (36)$$

For the basic transcritical CO_2 cycle, the specific evaporator capacity is given by:

$$q_e = (h_{e,o} - h_{mi}) \quad (37)$$

The isentropic compressor outlet conditions for the basic transcritical CO_2 cycle can be found as follows:

$$s_{e,o} = f(P_e, t_{e,o}) \quad (38)$$

$$s^b_{comp,is} = s_{e,o} \quad (39)$$

$$h^b_{comp,is} = f(s^b_{comp,is}, P_c) \quad (40)$$

Using the definition of compressor isentropic efficiency, the actual enthalpy at the compressor outlet of the basic transcritical CO_2 cycle can be found by:

$$\eta_{comp} = \frac{h^b_{comp,is} - h_{e,o}}{h^b_{comp} - h_{e,o}} \quad (41)$$

Then, the specific compressor work of the basic transcritical CO_2 cycle is found by:

$$w_{comp} = h^b_{comp} - h_{e,o} \quad (42)$$

Finally, the performance of the basic transcritical CO_2 cycle is given by:

$$COP_b = \frac{q_e}{w_{comp}} \quad (43)$$

The relative performance of the ejector expansion transcritical CO_2 cycle to the basic transcritical CO_2 cycle is defined as:

$$R = \frac{COP_n}{COP_b} \quad (44)$$

Using the above theoretical model, the effects of the entrainment ratio w and the pressure drop in the receiving section of the ejector $P_e - P_b$ on the relative performance of the ejector expansion transcritical CO_2 cycle can be investigated. In addition, using a given entrainment ratio w and a given pressure drop in the receiving section of the ejector $P_e - P_b$, the effect of different operating conditions

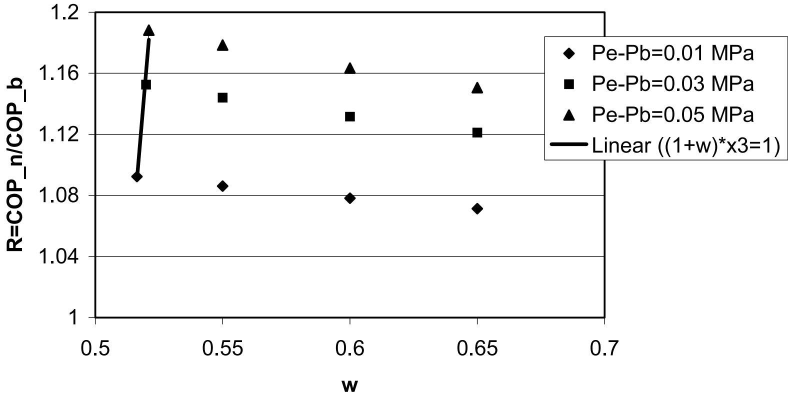

Fig. 7. Relative performance versus entrainment ratio ($P_{gc}=10$ MPa, $T_{gc,o}=40$ °C, $T_e=5$ °C, $T_{sh}=5$ °C).

on the relative performance of the ejector expansion transcritical CO_2 cycle can be investigated.

4. Results and discussion

To investigate the characteristics of the ejector expansion transcritical CO_2 cycle, the following standard operating conditions are assumed: $P_{gc}=10$ MPa, $T_{gc,o}=40$ °C, $T_e=5$ °C, $T_{sh}=5$ °C, $w=0.55$, $P_e-P_b=0.03$ MPa. The ejector is assumed to have the following efficiencies: $\eta_m=\eta_s=0.9$, $\eta_d=0.8$. The compressor is assumed to have an isentropic efficiency of 0.75.

The effect of the ejector entrainment ratio w for different pressure drops in the receiving section of the ejector P_e-P_b on the relative performance is shown in Fig. 7.

It can be seen that for the given conditions, the ejector expansion transcritical CO_2 cycle has a 7–18% improvement in COP over the basic transcritical CO_2 cycle. With a decrease in the entrainment ratio, the pressure elevation of the ejector increases, which, in turn decreases the pressure ratio across the compressor and thus, decreases the compression work and increases the improvement in COP. However, the decrease in the entrainment ratio is limited by the condition $(1+w)x_3=1$ as shown in Fig. 7. If $(1+w)x_3<1$, the cycle can not be realized since there is not enough vapor flow through the compressor that serves as the motive stream for the ejector. It can also be seen that the relative performance of the ejector expansion transcritical CO_2 cycle increases with an increase in the pressure drop in the receiving section of the ejector. However, the reachable pressure drop in the receiving section of the ejector is determined by the geometry of the ejector and the operating conditions of the ejector. Without having detailed experimental data for an ejector in a transcritical CO_2 cycle, the pressure drop in the receiving section, P_e-P_b, was assumed to be 0.03 MPa based on values found in the literature.

The effect of the gas cooler pressure on the relative performance of the ejector expansion transcritical CO_2 cycle is shown in Fig. 8. It can be seen that the relative performance of the ejector expansion transcritical CO_2 cycle increases with an increase in the gas cooler pressure

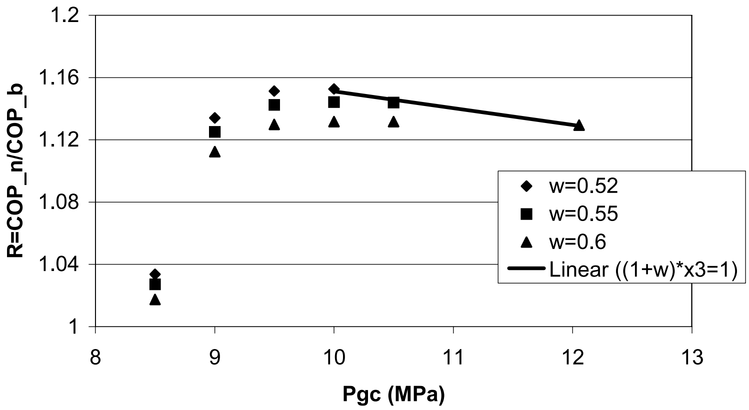

Fig. 8. Relative performance versus gas cooler pressure ($P_e-P_b=0.03$ MPa, $T_{gc,o}=40$ °C, $T_e=5$ °C, $T_{sh}=5$ °C).

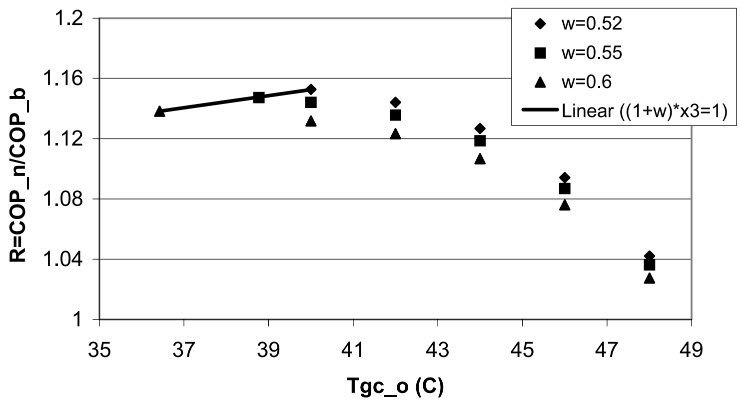

Fig. 9. Relative performance versus gas cooler outlet temperature ($P_e - P_b = 0.03$ MPa, $P_{gc} = 10$ MPa, $T_e = 5$ °C, $T_{sh} = 5$ °C).

until an optimum pressure is reached. As the gas cooler pressure increases, the expansion process losses of the basic transcritical CO_2 cycle increase as well. However, at the same time the ratio of the expansion process losses to the compression work is decreasing with an increase in the gas cooler pressure. Thus, there is an optimum gas cooler pressure at which the relative performance of the ejector expansion transcritical CO_2 cycle reaches a maximum value. It can also been seen that a high gas cooler pressure will lead to the limit in operating condition of $(1+w)x_3 = 1$.

The effect of the gas cooler outlet temperature on the relative performance of the ejector expansion transcritical CO_2 cycle is shown in Fig. 9. It can be seen that the relative performance decreases with an increase in the gas cooler outlet temperature. A high gas cooler outlet temperature affects the basic cycle performance through the loss of the evaporator capacity caused by a high quality of the refrigerant after the expansion process. These losses cannot be recovered by the ejector expansion device and the relative performance of the ejector expansion transcritical CO_2 cycle decrease with an increase in gas cooler outlet temperature. It can also been seen that a low gas cooler outlet temperature will lead to the limit in operating condition of $(1+w)x_3 = 1$.

The effect of the evaporation temperature on the relative performance of the ejector expansion transcritical CO_2 cycle is shown in Fig. 10. It can be seen that the relative performance increases with a decrease in evaporation temperature. With lower evaporation temperatures, the expansion process losses of the basic cycle increase because of the increase in pressure difference between the gas cooler pressure and the evaporation pressure. In this case, however, the ejector can recover some of the losses so that the relative performance of the ejector expansion transcritical CO_2 cycle will increase towards lower evaporation temperature applications. It can also been seen that a high evaporation

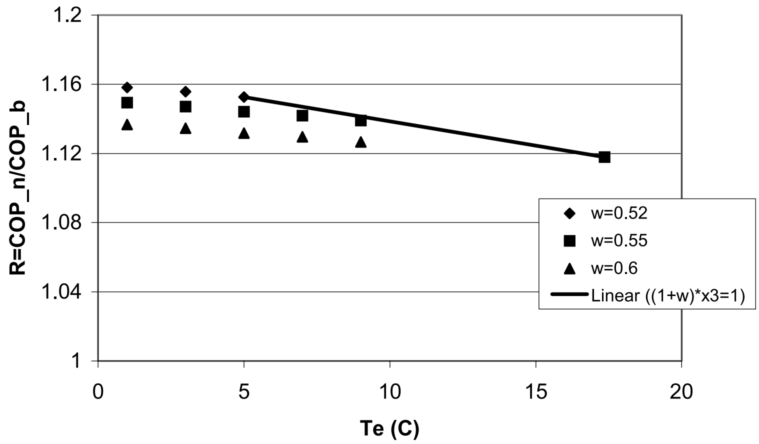

Fig. 10. Relative performance versus evaporation temperature ($P_e - P_b = 0.03$ MPa, $P_{gc} = 10$ MPa, $T_{gc,o} = 40$ °C, $T_{sh} = 5$ °C).

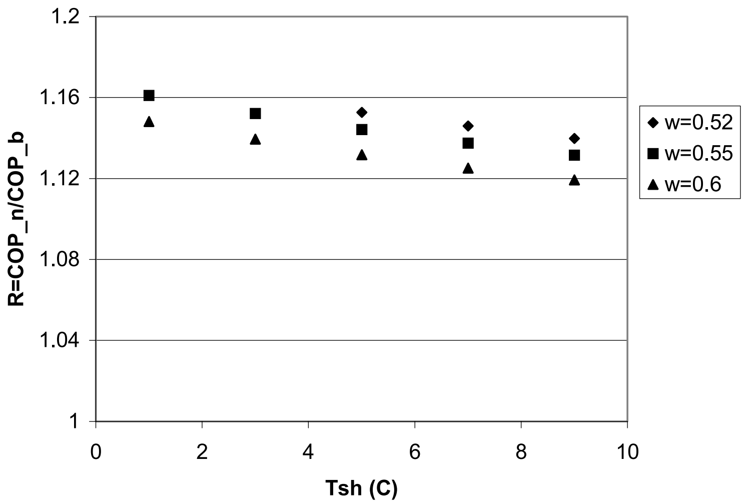

Fig. 11. Relative performance versus evaporator superheat ($P_e - P_b = 0.03$ MPa, $P_{gc} = 10$ MPa, $T_{gc,o} = 40$ °C, $T_e = 5$ °C).

temperature may lead to the limit in operating condition of $(1+w)x_3 = 1$.

The effect of the evaporator outlet superheat on the relative performance of ejector expansion transcritical cycle is shown in Fig. 11. It can be seen that a decrease in superheat increases the relative performance. Increasing the evaporator superheat will increase the specific evaporator capacity for both the basic cycle and the ejector expansion cycle. However, since there is less refrigerant flowing through the evaporator of the ejector expansion cycle than through the evaporator of the basic cycle per unit refrigerant flow through the compressor, the increase in evaporator capacity will be smaller for the ejector expansion cycle than for the basic cycle by increasing the evaporator superheat. That leads to a lower relative performance of the ejector expansion cycle with increasing the evaporator superheat. It should be noted that at low entrainment ratios, a small superheat may lead to the limit in operating condition of $(1+w)x_3 = 1$.

5. Conclusion

An ejector expansion transcritical CO_2 cycle is proposed to reduce the expansion process losses of the basic transcritical CO_2 cycle. A constant pressure-mixing model for the ejector was used to perform a thermodynamic cycle analysis of the ejector expansion transcritical CO_2 cycle. The effect of the entrainment ratio and the pressure drop in the receiving section of the ejector on the relative performance of the ejector expansion transcritical CO_2 cycle was investigated using the theoretical model. The performance improvement of the ejector expansion cycle over the basic transcritical CO_2 cycle for different operating conditions was investigated as well. It was found that the ejector expansion cycle improves the COP by more than 16% compared to the basic cycle for typical air conditioning applications.

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