Journal of Thermal Science Vol.28, No.4 (2019) 695−704

https://doi.org/10.1007/s11630-019-1124-6 **Article ID: 1003-2169(2019)04-0695-10**

Thermodynamic Analysis of a Modified Ejector-Expansion Refrigeration Cycle with Hot Vapor Bypass

LI Yunxiang, YU Jianlin*

Department of Refrigeration & Cryogenic Engineering, School of Energy and Power Engineering, Xi'an Jiaotong University, Xi'an 710049, China

Abstract: In this study, a modified ejector-expansion refrigeration cycle (MERC) is proposed for applications in small refrigeration units. A vapor bypass circuit is introduced into the standard ejector expansion refrigeration cycle (ERC) for increasing the ejector pressure lift ratio, thereby lowering the compressor pressure ratio in the MERC. A mathematical model has been established to evaluate the performances of MERC. Analysis results indicate that since a two phase vapor-liquid stream is used to drive the ejector in the MERC, a larger ejector pressure lift ratio can be achieved. Thus, the compressor pressure ratio decreases by 21.1% and the discharge temperature reduces from 93.6°C to 82.1°C at the evaporating temperature of -55°C when the vapor quality of two phase vapor-liquid stream increases from 0 to 0.2. In addition, the results show that the higher ejector component efficiencies are effective to reduce the compressor pressure ratio and the discharge temperature. Actually, the discharge temperature reduces from 91.4°C to 82.1°C with the ejector component efficiencies increasing from 0.75 to 0.85 at the two phase stream vapor quality of 0.2. Overall, the proposed cycle is found to be feasible in lower evaporating temperature cases.

Keywords: compressor pressure ratio, discharge temperature, ejector, refrigeration cycle, vapor bypass

1. Introduction

Recently, the use of two-phase ejector in single-stage vapor compression refrigeration and heat pump systems has drawn many attentions because the two-phase ejector can contribute to the system performance [1-4]. Principally, a two-phase ejector as expansion device applied in a single-stage vapor compression system can efficiently recover the loss of throttling process by generating isentropic expansion process to improve the system performance. Hence, a large number of theoretical and experimental studies have been conducted on the performance of the single-stage vapor compression cycle with an ejector.

Lawrence et al. [5] proposed three different two-phase ejector refrigeration cycles, and the result showed that these three cycles show the improvement in *COP* compared to the conventional refrigeration cycle. Ersoy and Sag [6] carried out preliminary experiments on an R134a refrigeration system with a two-phase ejector, and found that this refrigeration system exhibits higher *COP* than that of the system without ejector by 6.2%-14.5%. Li et al. [7] evaluated performance of the R1234yf ejector expansion refrigeration cycle, and confirmed that the cycle is superior to the standard cycle, especially at extreme operating conditions. Hassanain et al. [8]

Received: Dec 27, 2018 Associate editor: WANG Lin Corresponding author: YU Jianlin E-mail: yujl@mail.xjtu.edu.cn

Nomenclature

COP	cooling coefficient of performance	x	vapor quality
ERC	ejector expansion refrigeration cycle	***Greek letters***	
h	specific enthalpy/J·kg^{-1}	γ	pressure lift ratio
MERC	modified ejector expansion refrigeration cycle	η	efficiency
$\dot{m}$	mass flow rate/kg·s^{-1}	μ	entrainment ratio
p	pressure/Pa	ω	mass flow rate allocation ratio
Δp	pressure drop/Pa	***Subscripts***	
q_v	actual cooling capacity per unit volumetric displacement/J·m^{-3}	com	compressor
$\dot{Q}_c$	cooling capacity/W	d	diffuser
t_c	condensing temperature/°C	eje	ejector
t_e	evaporating temperature/°C	is	isentropic
t_d	compressor discharge temperature/°C	m	mixing chamber
v	refrigerant specific volume/m^3·kg^{-1}	n	nozzle
$\dot{V}$	volumetric displacement of the compressor/m^3·s^{-1}	p	primary fluid
w	velocity/m·s^{-1}	s	secondary fluid
$\dot{W}$	compressor input power/W	1-7, 2′-4′, 3″	refrigerant state points

conducted the ejector design and performance evaluation for an ejector expansion refrigeration system with refrigerant R134a. Also, Sag et al. [9] conducted an experimentally study on the ejector expansion refrigeration system in terms of energetic and exergetic aspects. This system applied R134a as refrigerant. The results showed that the total irreversibility of this ejector expansion refrigeration system is lower than that of the basic system. He et al. [10] carried out performance assessment of a transcritical CO_2 refrigeration system using an adjustable ejector, and revealed that the system has a better dynamic performance characteristic. To sum up, the ejector is an ideal expansion device, and it can be used in the vapor compression cycle system to improve the system performance.

Actually, the ejector is also a vapor compression device, i.e. the so-called thermo-compressor, which is widely used in heat-driven ejector refrigeration systems [11,12]. This means that an ejector can be used to mainly generate compression function similar to conventional compressors. Thus, ejectors could constitute an attractive alternative for conventional compressors in vapor compression refrigeration cycles. As well known, conventional single-stage compression cycles can only be operated in a limited temperature range. Their performances may degrade when they operate at low refrigeration temperatures, such as low coefficient of performance (*COP*), low cooling capacity, high discharge temperature and large pressure ratio of compressor. For this case, these drawbacks could be overcome by utilizing two-stage compression cycles [13-15]. Typical two-stage vapor compression cycle usually uses two individual compressors or a compound compressor with vapor injection. Compared with the compressor, the ejector has no moving parts, lower cost, simple structure and lower maintenance requirements. This fact makes the use of ejector that is an alternative to the commonly used compressor in a two-stage compression cycle may have some technical advantages, particularly in terms of low cost and high reliability. However, this idea has not been founded in the opening literature. Obviously, the idea of using an ejector as an additional compressor could be of interest for new ejector applications in the single-stage vapor compression cycle, which is favorable for reliable operation in a wide range of refrigeration temperature similar to a two-stage vapor compression cycle. Hence, it is meaningful to propose a modified ejector-expansion refrigeration cycle from this idea, and the investigations on performances of this cycle are much needed.

In this study, a modified ejector-expansion refrigeration cycle (MERC) is presented based on the standard ejector expansion refrigeration cycle (ERC). In the MERC, a compressor discharge vapor bypass circuit is configured for bypassing the condenser. In this case, the two phase vapor-liquid stream can be used to drive the ejector in the MERC, thus allowing the ejector pressure lift ratio to be increased, resulting in decreased compressor pressure ratio. Thus, the MERC may employ a compressor to operate at wide temperature and pressure ranges which could be attractive for small refrigeration apparatus applications. A mathematical model has been established to simulate the performances of MERC. In the simulations, the main parameters are analyzed and discussed including the compressor pressure ratio and discharge temperature, the performances of ejector and the cooling capacity per unit volumetric displacement. The study mainly concentrates on a theoretical investigation of the performance of this proposed cycle for developing the ejector expansion refrigeration technologies application in small refrigeration units.

2. Cycle Description and Modeling

The schematic diagram of the MERC system is shown in Fig. 1. The system components for MERC include a compressor, a bypass circuit with the flow regulating valve 1, a condenser circuit with the flow regulating valve 2, an ejector, a separator, a throttling device (expansion valve) and an evaporator. The schematic *P-h* diagram for the MERC is shown in Fig. 2. From Fig. 1, it is noticed that the MERC is very similar to the ERC. In the ERC, the ejector is mainly used as an expansion device to recover the energy loss in the throttling process, and its pressure lift is limited. However, the ejector in the MERC is applied to mainly provide the large compression effect in associate with expansion work recovery, resulting in the increase of the compressor suction pressure, for that a larger compression effect can be realized by the two phase flow driven ejector in the MERC. In this case, two flow regulating valves in both the bypass circuit and condenser circuit can be used to adjust relevant flow rates for obtaining the expected dryness quality of the mixed fluid (primary fluid of the ejector). The working process of MERC cycle is described as follows: a greater portion of the compressed refrigerant vapor from the compressor flows through the flow regulating valve 2 and enters the condenser. This refrigerant condenses into saturated or subcooled liquid at the condenser outlet by rejecting heat to the surroundings (process 2-2″-3′); the other portion of the vapor flows through the flow regulating valve 1 and mixes with the liquid refrigerant from the condenser (process 2-2′ and 2′, 3′-3); After that, this two-phase fluid flows into the nozzle of the ejector as the primary fluid to drive the ejector; Meanwhile, the secondary fluid, which is the saturated or superheated refrigerant vapor from the outlet of evaporator, is entrained to the ejector; In the ejector, the process 3-3″, (3″, 7)-4′ and 4′-4 are the expansion process of primary fluid, the mixing process between primary fluid and secondary fluid as well as compression process of mixed fluid, respectively; The two-phase refrigerant at the ejector outlet enters separator and separates into the saturated vapor (process 4-1) and the saturated liquid (process 4-5); At last, the saturated vapor refrigerant flows into the compressor where it is continually compressed to the condensing pressure (process 1-2); The saturated liquid refrigerant flows through the throttling device and finally enters the evaporator where the working fluid becomes saturated or superheated (process 5-6-7).

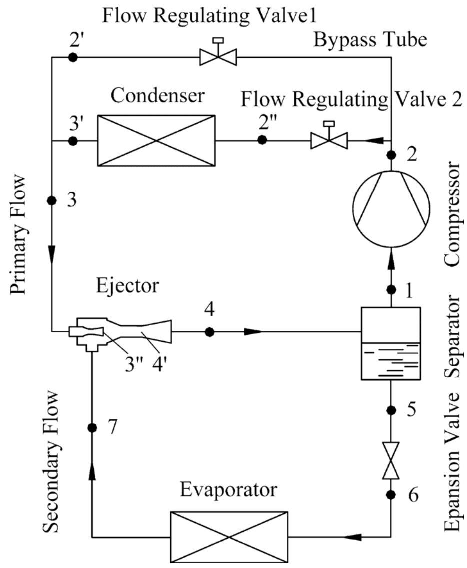

Fig. 1 The schematic diagram for the MERC system

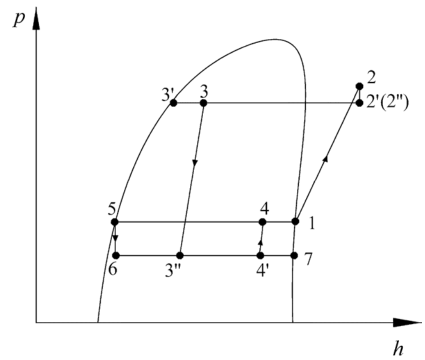

Fig. 2 The schematic *p-h* diagram for the MERC

The performances of MERC are theoretically assessed based on a mathematical model, and some basic assumptions made before establishing the model are as follows [16]:

(1) The steady-state and steady-flow process is carried out for all components;

(2) A variable isentropic efficiency is used to represent irreversible loss in the compressor;

(3) An isenthalpic process is used to represent throttling process in throttling device;

(4) The vapor and liquid discharged from the separator are saturated;

(5) The kinetic energies of the primary fluid and

secondary fluid inlets in ejector are neglected;

(6) The pressure drops and heat losses of refrigerant in the cycle are neglected;

(7) Ejector component efficiencies are introduced to represent friction losses in ejector.

On the basis of assumptions above, the mathematic model of MERC can be established in terms of the mass, momentum and energy conservations. Actually, the entrainment ratio μ and the pressure lift ratio γ_{eje} are important parameters, which can be used to evaluate the performance of ejector. The μ and the γ_{eje} are defined as,

$$\mu = \dot{m}_s / \dot{m}_p \tag{1}$$

$$\gamma_{eje} = p_4 / p_7 \tag{2}$$

where $\dot{m}_s$ is the mass flow rate of the ejector secondary fluid; $\dot{m}_p$ is the mass flow rate of the ejector primary fluid; p_4 is the ejector exit pressure; p_7 is the pressure of the ejector secondary fluid inlet.

In the nozzle, the primary fluid is accelerated, which creates a low pressure zone at the nozzle exit plane. Thus, the secondary fluid can be entrained to the low pressure zone of the ejector. On the basis of the energy balance of the primary fluid between the inlet and outlet of the nozzle, the velocity of primary fluid at the nozzle outlet is defined as,

$$w_{p2} = \sqrt{2\eta_n \left(h_{p1} - h_{p2s}\right)} \tag{3}$$

where h_{p1} is specific enthalpy of the primary fluid at the nozzle inlet; and h_{p2s} is the specific enthalpy of primary fluid at the nozzle outlet under an isentropic process; η_n is the nozzle efficiency, which is defined as,

$$\eta_n = \left(h_{p2} - h_{p1}\right) \big/ \left(h_{p2s} - h_{p1}\right) \tag{4}$$

where h_{p2} is the practical specific enthalpy of primary fluid at the nozzle outlet.

Keenan et al. [17] proposed two common mathematical models, the constant-area mixing (CAM) model and the constant-pressure mixing (CPM) model, to describe the mixing mechanism of fluids in the mixing chamber. The CPM is widely adopted in ejector design due to the superior performance in companion with the CAM [18-20]. And the CPM model is applied in the modeling i.e. the primary fluid mixes with the secondary fluid in the ejector at constant pressure. Then the momentum conservation equation for the mixing process of fluids in the mixing chamber can be obtained,

$$\left(\dot{m}_s + \dot{m}_p\right) w_{m2,is} = \dot{m}_p w_{p2} \tag{5}$$

where $w_{m2,is}$ is the ideal velocity of the mixed fluid at the mixing chamber outlet.

Furthermore, the mixing efficiency η_m is defined as,

$$\eta_m = w_{m2}^2 \big/ w_{m2,is}^2 \tag{6}$$

where w_{m2} is the practical velocity of the mixed fluid at the mixing chamber outlet.

And then the w_{m2} can be obtained by combining Eq. (1) with Eq. (5) and Eq. (6),

$$w_{m2} = w_{p2}\sqrt{\eta_m} \big/ (1+\mu) \tag{7}$$

The energy conservation equation of the mixing process can be expressed,

$$\left(\dot{m}_s + \dot{m}_p\right)\left(h_{m2} + w_{m2}^2/2\right) = \dot{m}_s h_{s1} + \dot{m}_p h_{p1} \tag{8}$$

where h_{s1} is the specific enthalpy of the secondary fluid inlet.

And then the specific enthalpy h_{m2} of the mixed fluid at the outlet of the mixing chamber is obtained as,

$$h_{m2} = \frac{h_{p1} + \mu h_{s1}}{1+\mu} - \frac{w_{m2}^2}{2} \tag{9}$$

In the diffuser chamber, the kinetic energy of mixed fluid is converted into pressure energy. At the diffuser exit, the velocity is reduced and the pressure is raised enough to cause discharge. Applying the energy conservation to the compression progress of the diffuser, the velocity of mixed fluid at the ejector outlet can be obtained as,

$$w_{d2} = \sqrt{w_{m2}^2 - 2\left(h_{d2} - h_{m2}\right)} \tag{10}$$

where h_{d2} is the specific enthalpy of mixed fluid at the ejector outlet, which can be obtained by,

$$h_{d2} = h_{m2} + \left(h_{d2,is} - h_{m2}\right) \big/ \eta_d \tag{11}$$

where $h_{d2,is}$ is the specific enthalpy of mixed fluid at the ejector outlet under an isentropic process for the same exit pressure; η_d is the diffuser efficiency.

As assumed that the kinetic energy of mixed fluid at the ejector out is neglected, the μ can be obtained by combining Eq. (1) with Eqs. (3-11),

$$\mu = \sqrt{\eta_n \eta_m \eta_d \frac{h_{p1} - h_{p2,is}}{h_{d2,is} - h_{m2}}} - 1 \tag{12}$$

The allocation of mass flow rate ω between the bypass pipe and condenser is written as,

$$\dot{m}_{2'} = \omega \dot{m}_2, \dot{m}_{3'} = (1-\omega)\dot{m}_2 \tag{13}$$

Mass balances for the ejector, flash tanks and condenser can be written as,

$$\dot{m}_3 + \dot{m}_7 = \dot{m}_4 = \dot{m}_1 + \dot{m}_5 \tag{14}$$

$$\dot{m}_2 = \dot{m}_{2'} + \dot{m}_{3'} = \dot{m}_3 \tag{15}$$

Using Eq. (1) and Eqs. (13-15), the fundamental equation between the entrainment ratio and refrigerant quality at the ejector outlet can be expressed as,

$$1 + \mu = 1/x_4 \tag{16}$$

For the compressor, the input power can be written as

$$\dot{W} = \dot{m}_p (h_2 - h_1) = \frac{\dot{m}_p (h_{2,is} - h_1)}{\eta_{is}} \tag{17}$$

$$\dot{m}_p = \frac{\eta_v \dot{V}}{v_1} \tag{18}$$

where h_1 is the refrigerant specific enthalpy at the compressor inlet; h_2 is the refrigerant specific enthalpy at the compressor outlet; $h_{2,is}$ is the refrigerant specific enthalpy at the compressor outlet under an isentropic process; $\dot{V}$ is the volumetric displacement of the compressor; v_1 is the refrigerant specific volume at the inlet of the compressor; η_{is} is the compressor isentropic efficiency; η_v is the compressor volumetric efficiency, which can be obtained [7,21],

$$\eta_{is} = 0.874 - 0.0135 \frac{p_2}{p_1} = 0.874 - 0.0135 \gamma_{com} \tag{19}$$

$$\eta_v = 0.959 - 0.00642 \frac{p_2}{p_1} \tag{20}$$

where γ_{com} is the compressor pressure ratio, p_1 is the refrigerant pressure at the compressor inlet; p_2 is the refrigerant pressure at the compressor outlet.

The cycle cooling capacity is

$$\dot{Q}_c = \dot{m}_s (h_7 - h_6) = \frac{\mu \eta_v \dot{V} (h_7 - h_6)}{v_1} \tag{21}$$

where h_6 is the refrigerant specific enthalpy at evaporator inlet; h_7 is the refrigerant specific enthalpy at the evaporatoroutlet. In addition, an actual cooling capacity per unit volumetric displacement q_v can be obtained by Eq. (21),

$$q_v = \mu \eta_v (h_7 - h_6) / v_1 \tag{22}$$

The cycle cooling coefficient of performance (*COP*) can be expressed by

$$COP = \dot{Q}_c / \dot{W} \tag{23}$$

Finally, the performance of the MERC is simulated using the above model. The simulation code is written in Fortran. The refrigerant properties are calculated by the NIST database and subroutines [22]. The flow diagram of the calculation process is shown in Fig. 3.

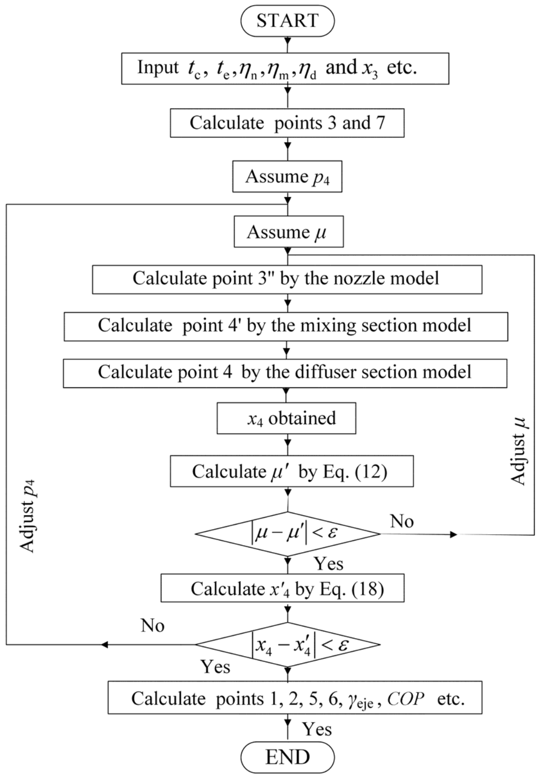

Fig. 3 The flow diagram of the calculation process for MERC

3. Results and Discussion

The R290 has excellent thermodynamic properties, the zero ODP and the low GWP, thus it is widely used in refrigeration systems [23]. Therefore, R290 is selected as the working fluid to evaluate the performance of MERC in this research. The correlative conditions are given as follows: the condensing temperature t_c changes from 30 to 40°C; the evaporating temperature t_e changes from -55 to -45°C [24]; the ejector component efficiencies η_n, η_m and η_d are assumed to be varied from 0.75 to 0.85 based on the study of literatures [25-28]. It should be noted that although in the MERC the two streams flowing through the regulating valves and condenser undergo a pressure drop Δp in practice, the effect of Δp on MERC

Table 1 The performances of MERC at the t_e of -55°C, t_c of 40°C and ejector nozzle isentropic efficiencies of 0.85

	$\Delta p = 0\times10^4$ Pa			$\Delta p = 1\times10^4$ Pa			$\Delta p = 2\times10^4$ Pa		
	γ_{com}	t_d/°C	q_v/J·m^{-3}	γ_{com}	t_d/°C	q_v/J·m^{-3}	γ_{com}	t_d/°C	q_v/J·m^{-3}
x_3=0	16.1	93.6	0.378	16.2	94.2	0.378	16.3	94.8	0.377
x_3=0.05	15.2	90.6	0.375	15.3	91.1	0.375	15.4	91.7	0.374
x_3=0.1	14.3	87.6	0.371	14.4	88.2	0.370	14.5	88.8	0.370
x_3=0.15	13.5	84.8	0.366	13.6	85.4	0.365	13.7	85.9	0.365
x_3=0.2	12.7	82.1	0.359	12.8	82.7	0.359	12.9	83.2	0.359

performances is relatively small at the conventional range of pressure drop, as shown in Table 1. Hence, the Δp is ignored when effects of the x_3 on the performance of MERC are studied in Section 3.1~3.3.

3.1 Effects of the x_3 on the performance of MERC at the different t_e

The mass flow rates of the bypass circuit and the condenser can be varied by controlling the regulating valves, resulting in variable vapor quality x_3 at the ejector nozzle inlet. Actually, the ejector performances (the pressure lift ratio γ_{eje} and the entrainment ratio μ) are varied with the x_3. Therefore, the compressor pressure ratio γ_{com} and the discharge temperature t_d will be affected by the x_3 due to the interaction of the γ_{com} and the $\gamma_{eje.}$ In this section, effects of the x_3 on the performance of MERC are investigated at different t_e. The t_c is assumed to be 40°C, the nozzle isentropic efficiency η_n, the mixing efficiency η_m and the diffuser isentropic efficiency η_d are fixed at 0.85, respectively.

Fig. 4 shows that the changes of the γ_{com} and t_d with x_3 at different t_e for MERC. It can be seen that both the γ_{com} and t_d are reduced with the increase of x_3, and as the reduction in t_e, the drops of γ_{com} and t_d with x_3 are raised. For instance, when the x_3 increases from 0 to 0.2, the γ_{com} decreases by 18.0% at the t_e of -45°C, while the γ_{com} decreases by 21.1% at the t_e of -55°C. Furthermore, the t_d reduces from 93.6 to 82.1°C and from 76.9 to 70.6°C at the t_e of -55°C and -45°C when the x_3 increase from 0 to 0.2. The reason is that the decrease of t_e could result in a serious rise of γ_{com} in a ERC, and the rise of γ_{com} can impair the η_{is}, thus the t_d could be increased hugely. However, the γ_{eje} increases with the rise of x_3 especially at a lower t_e in MERC, as shown in Fig. 5. Therefore, the γ_{com} reduces with the x_3 and the reduction rate rises with the decrease of t_e. Hence, the t_d is dropped sharply with the rise of x_3 especially at the lower t_e. The result indicates that a lower evaporating temperature can be achieved when the MERC is applied in a practical refrigeration unit, since the MERC is able to reduce the compressor pressure ratio and discharge temperature remarkably.

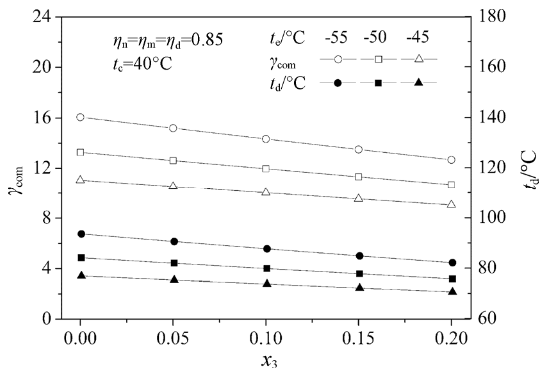

Fig. 4 The changes of the γ_{com} and t_d with x_3 at different t_e

Fig. 5 shows that the changes of the γ_{eje} and μ with x_3 at different t_e for MERC. It can be seen in Fig. 5 that when the x_3 increases from 0 to 0.2, the γ_{eje} increases by 21.9% ~ 26.7% at the t_e in the range of -45~ -55°C. This case may be explained as that compared with the liquid stream, the change in specific enthalpy of the primary fluid through the nozzle is larger when the two-phase vapor-liquid stream is used as primary stream. And thus the velocity of primary fluid at the nozzle outlet is higher for this two-phase vapor-liquid stream from energy conservation. This results in an increase of the kinetic energy of the mixed stream in the ejector mixing chamber because of the momentum conservation for the mixing process. And then more pressure energy of the exit fluid would be obtained because the higher kinetic energy can be converted to pressure energy in the diffuser. Hence the higher pressure lifting ratio of ejector γ_{eje} can be obtained when two-phase primary fluid is used. Furthermore, Fig. 5 shows that the entrainment ratio μ decreases by 22.7% ~ 23.5% at the t_e in the range of -45~ -55°C when the x_3 increases from 0 to 0.3. The reason is that as the rises of the x_3 and the γ_{eje}, the ejector outlet vapor quality x_4 increases. Hence, the reduction in μ is obtained because of the fundamental relation between the entrainment ratio and refrigerant quality at the ejector outlet, which is shown in Eq. (16).

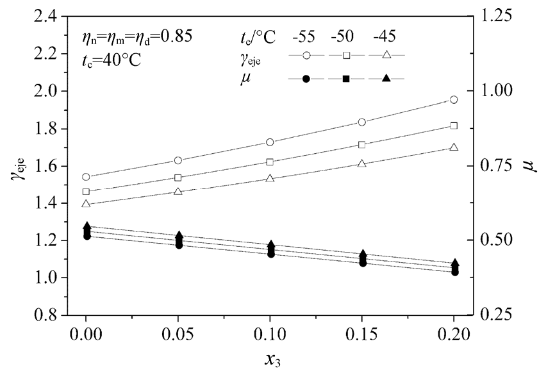

Fig. 5 The changes of the γ_{eje} and μ with x_3 at different t_e

Fig. 6 shows that the changes of the q_v and ω with x_3 at different t_e for MERC. It can be seen in Fig. 6 that the q_v decreases with the rise of x_3 when the t_e is fixed. The reason is that the μ is reduced with the increase of x_3, therefore the $\dot{m}_s$ through the evaporator decreases, thus the amount of q_v is reduced with the increase of x_3. However, the γ_{eje} rises with the increase of x_3, therefore the v_1 decreases. Furthermore, the η_v increases due to the reduction in γ_{com}, which results in the increase of the $\dot{m}_p$. On the basis of these reasons, the effect of x_3 on q_v is limited in the given range of x_3 actually. For instance, the q_v decreases by 5.0% ~ 8.2% at the t_e in the range of -55~ -45°C when the x_3 increases from 0 to 0.2. Besides, it can be seen in Fig. 6 that the allocation of mass flow rate ω increases with the rise of x_3, and it increases in a continuously accelerated speed as the t_e goes up, i.e. when the t_e is -55°C, the ω increases from 0 to 0.154; however, when the t_e is -45°C, the ω increases from 0 to 0.164 with the x_3 changing from 0 to 0.2. The result means that the mass flow rate of the refrigerant through the bypass circuit should be decreased at a lower t_e to maintain the same x_3. Note, the *COP* decreases from 1.027 to 0.897, i.e. decreases by 12.7%, at the t_e of -55°C when the x_3 changes from 0 to 0.2. It means that the reduction in *COP* is the cost of obtaining a lower t_e and keeping an applicable t_d.

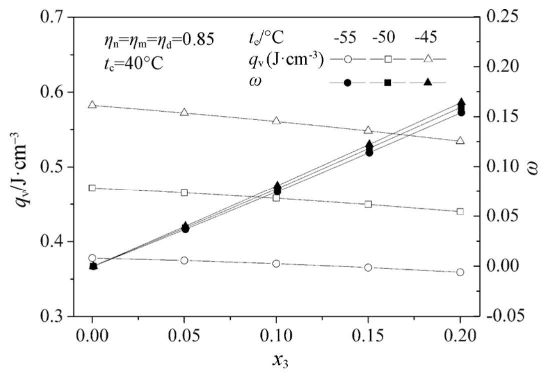

Fig. 6 The changes of the q_v and ω with x_3 at different t_e

3.2. Effects of the x_3 on the performance of MERC at the different t_c

The changes of the performances of the ejector and the MERC with the x_3 at different t_c will be discussed in this section. The t_e is assumed to be -55°C, the η_n, η_m and η_d are fixed at 0.85, respectively. It can be seen in Fig. 7 that when the x_3 changes from 0.1 to 0.2, the γ_{com} decreases by 19.8% ~ 21.1% at the t_c in the range of 30~ 40°C. Furthermore, when the t_c is 30, 35 and 40°C, the t_d is reduced to 70.2, 76.3 and 82.1°C with the x_3 increasing to 0.2. The results show the γ_{com} and the t_d decrease obviously as the x_3 rises, and decrease rates accelerate continuously as the t_c goes up. The reason is similar to Fig. 4, so that there is no more detailed description here. Fig. 8 shows that when the x_3 changes from 0.1 to 0.2, the γ_{eje} increases by 24.7% ~ 26.7% at the t_c in the range of 30~ 40°C. The explanation on the variation of γ_{eje} with the x_3 is the same as what is analyzed in Fig. 5. Furthermore, the simulation results show the γ_{eje} increases with the t_c at a constant of the x_3. This is due to the fact that the primary fluid in the motive nozzle with higher pressure energy would result in the mixed fluid with higher kinetic energy in the mixing chamber, so that the higher pressure lifting ratio can be obtained at the ejector exit. In addition, it can be seen in Fig. 8 that when the x_3 changes from 0.1 to 0.2, the μ decreases from 0.568 to 0.438, 0.540 to 0.415 and 0.513 to 0.392 for the t_c of 30, 35 and 40°C, respectively. It means that the $\dot{m}_s$ through the evaporator decreases with the rise of x_3 at a constant $\dot{m}_p$, which can lead to the change of the q_v. It can be seen in Fig. 9 that when the x_3 changes from 0.1 to 0.2, the q_v is decreased by 5.8% ~ 5.0% at the t_c in the range of 30~40°C. As same as the Fig. 6, the effect of x_3 on q_v is limited at different t_c for MERC. Furthermore, Fig. 9 shows when the x_3 changes from 0.1 to 0.2, the ω

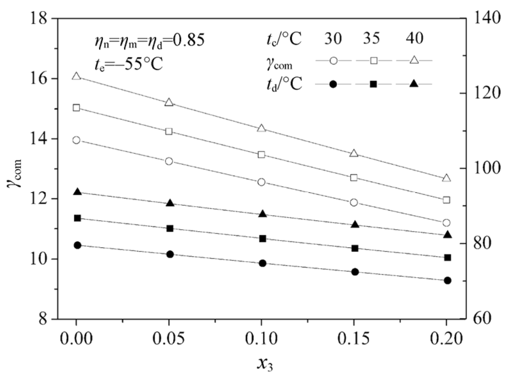

Fig. 7 The changes of the γ_{com} and t_d with x_3 at different t_c

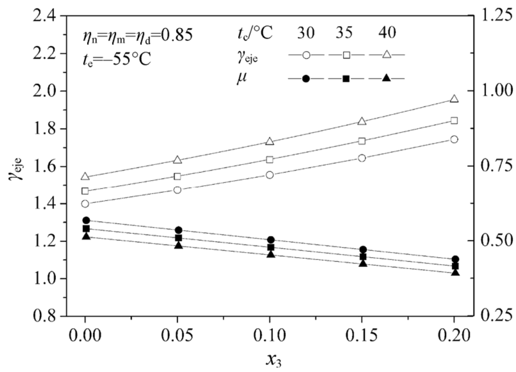

Fig. 8 The changes of the γ_{eje} and μ with x_3 at different t_c

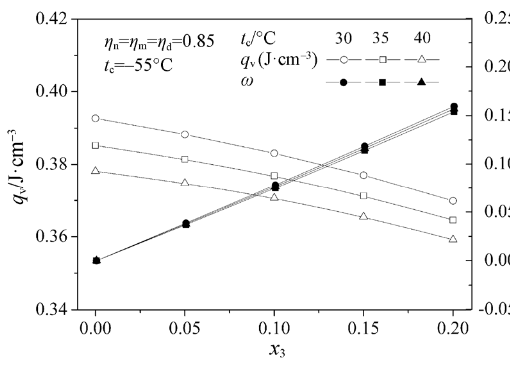

Fig. 9 The changes of the q_v and ω with x_3 at different t_c

increases from 0 to 0.160, 0 to 0.157 and 0 to 0.154 for the t_c of 30, 35 and 40°C, respectively. It means that the refrigerant vapor through the bypass circuit should be decreased to maintain the same x_3 when the MERC systems are applied in the higher ambient temperatures.

3.3 Effects of the x_3 on the performance of MERC at the different η_n, η_m and η_d

The working fluid flow in the ejector is very complicated, which causes large amount of the irreversibility losses, decreasing the ejector performance, and thus the irreversibility losses of the working process in ejector are usually considered by introducing the ejector component efficiencies [29]. In this section, three different ejector component efficiencies (0.75, 0.80 and 0.85) have been selected from ranges of 0.7~1.0 [25-28] to discuss the variations of the cycle performance and the ejector performance. The t_e and the t_c is assumed to be -55 and 40°C, respectively.

Fig. 10 shows the changes of the γ_{com} and t_d with x_3 at different ejector component efficiencies for MERC. It can be seen in Fig. 10 that when the x_3 changes from 0.1 to 0.2, the γ_{com} decreases by 15.7%, 18.3% and 21.1% for the ejector component efficiencies of 0.75, 0.85 and 0.95, respectively. The result is related to the fact that high ejector component efficiencies can result in the low irreversible loss in the nozzle, the mixing chamber and the diffuser, thus a higher pressure of refrigerant at the ejector outlet could be obtained. It results in the increase of the γ_{eje}, which can be shown in Fig. 11, therefore the γ_{com} decreases. Furthermore, when x_3 is 0.2, the t_d is reduced to 91.4, 86.7 and 82.1°C at the ejector component efficiencies of 0.75, 0.80 and 0.85. In a short conclusion, lifting the ejector component efficiencies is effective to reduce γ_{com} and t_d. Fig. 11 shows that when the x_3 changes from 0.1 to 0.2, the γ_{eje} increases by 18.6%, 22.4% and 26.7% for ejector component efficiencies of 0.75, 0.80 and 0.85, respectively, i.e. γ_{eje} is more sensitive to the variation of the x_3 when ejector component efficiencies are relatively high. In addition, the μ decreases with x_3, i.e. when x_3 changes from 0.1 to 0.2 the μ decreases from 0.497 to 0.370, 0.504 to 0.380 and 0.513 to 0.392 for ejector component efficiencies of 0.75, 0.80 and 0.85, respectively.

It can be seen in Fig. 12 that when the x_3 changes from 0.1 to 0.2, the q_v is decreased by 12.6%, 9.0% and 5.0% at the ejector component efficiencies of 0.75, 0.80 and 0.85. The result is related to the fact that the increase of ejector component efficiencies can observably reduce the irreversible loss in the ejector, thus the high μ could be obtained. Therefore, the reduction rate in q_v due to the deceasing μ can be decreased with the ejector component efficiencies increase. Furthermore, Fig. 12 shows when the x_3 changes from 0.1 to 0.2, the ω increases from 0 to 0.147, 0 to 0.150 and 0 to 0.154 for the ejector component efficiencies of 0.75, 0.80 and 0.85, respectively. It means that the allocation of mass flow rate between the bypass pipe and condenser needs to increase to maintain the same x_3 at the higher ejector component efficiencies.

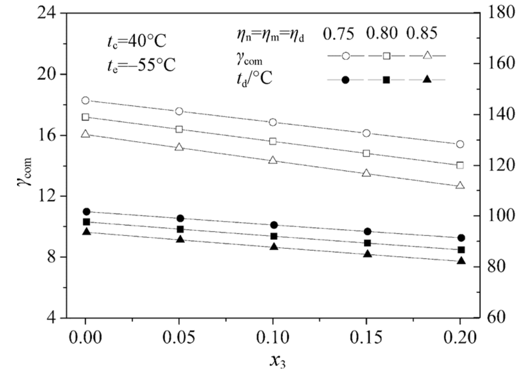

Fig. 10 The changes of the γ_{com} and t_d with x_3 at different η_n, η_m and η_d

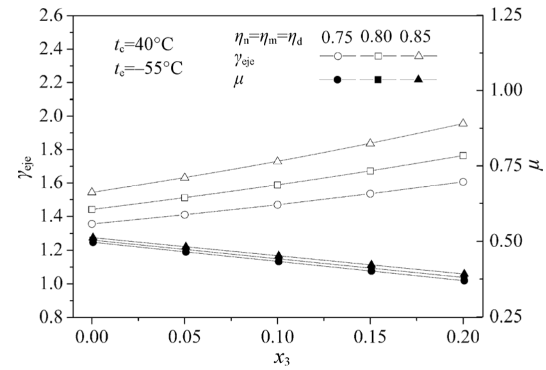

Fig. 11 The changes of the γ_{eje} and μ with x_3 at different η_n, η_m and η_d

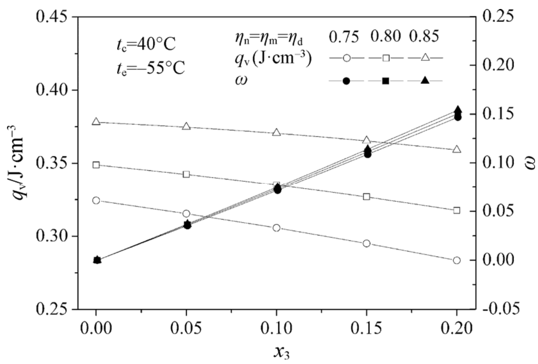

Fig. 12 The changes of the q_v and ω with x_3 at different η_n, η_m and η_d

4. Conclusions

A modified ejector-expansion refrigeration cycle (MERC) is presented in this paper. In the MERC, a bypass circuit with a flow regulating valve to regulate the x_3 is integrated in the ERC for increasing the γ_{eje} and lowering the γ_{com}. Furthermore, the theoretical investigation on the MERC performances (the γ_{com}, the t_d and the q_v etc.) is conducted. The main conclusions include:

1) The γ_{com} and t_d can be decreased when the x_3 is raised, i.e. the γ_{com} decreases by 18.0% ~ 21.1% with the x_3 increasing from 0 to 0.2 at the t_e in the range of -45~-55°C. The t_d reduces from 93.6 to 82.1°C at the t_e of -55°C when the x_3 increases from 0 to 0.2.

2) A small reduction in q_v is conducted with the rise of x_3, for instance, the q_v decreases by 5.0% ~ 8.2% at the t_e in the range of -55~-45°C when the x_3 increases from 0 to 0.2.

3) The higher ejector component efficiencies are effective to reduce γ_{com} and t_d, for instance, when the x_3 changes from 0.1 to 0.2, the γ_{com} decreases by 15.7%, 18.3% and 21.1% for the ejector component efficiencies of 0.75, 0.80 and 0.85, respectively.

In general, the key point of this article is lowering the γ_{com} and t_d by the MERC, aiming to obtain a lower t_e. The proposed MERC could broaden the ejector expansion refrigeration technologies, which could work in a wider temperature range than the ERC system. Certainly, further experimental studies on the MERC system operating performance are required in the next step to confirm the practicability of this cycle system.

Acknowledgements

This research is funded by the National Natural Science Foundation of China (NSFC) No.51776147. The authors thank NSFC for the support.

References

[1] Sarkar J., Ejector enhanced vapor compression refrigeration and heat pump systems—A review. Renewable & Sustainable Energy Reviews, 2012, 16(9): 6647–6659.

[2] Sumeru K., Nasution H., Ani F.N., A review on two-phase ejector as an expansion device in vapor compression refrigeration cycle. Renewable & Sustainable Energy Reviews, 2012, 16(7): 4927–4937.

[3] Kim H.D., Lee J.H., Setoguchi T., Matsuo S., Computational analysis of a variable ejector flow. Journal of Thermal Science, 2006, 15: 140–145.

[4] Kim H., Lee Y., Setoguchi T., Yu S., Numerical simulation of the supersonic flows in the second throat ejector-diffuser systems. Journal of Thermal Science, 1999, 8: 214–222.

[5] Lawrence N., Elbel S., Theoretical and practical comparison of two-phase ejector refrigeration cycles including First and Second Law analysis. International Journal of Refrigeration, 2013, 36(4): 1220–1232.

[6] Ersoy H.K., Sag N.B., Preliminary experimental results on the R134a refrigeration system using a two-phase ejector as an expander. International Journal of Refrigeration, 2014, 43: 97–110.

[7] Li H.S., Cao F., Bu X.B., Wang L.B., Wang X.L., Performance characteristics of R1234yf ejector-expansion refrigeration cycle. Applied Energy, 2014, 121(15): 96–103.

[8] Hassanain M., Elgendy E., Fatouh M., Ejector expansion refrigeration system: Ejector design and performance evaluation. International Journal of Refrigeration, 2015, 58: 1–13.

[9] Sag N.B., Ersoy H.K., Hepbasli A., Halkaci H.S., Energetic and exergetic comparison of basic and ejector expander refrigeration systems operating under the same external conditions and cooling capacities. Energy Conversion and Management, 2015, 90(15): 184–194.

[10] He Y., Deng J.Q., Zheng L.X., Zhang Z.X., Performance optimization of a transcritical CO_2 refrigeration system using a controlled ejector. International Journal of Refrigeration, 2017, 75: 250–261.

[11] Chen J.Y., Jarall S., Havtun H., Palm B., A review on versatile ejector applications in refrigeration systems. Renewable & Sustainable Energy Reviews, 2015, 49: 67–90.

[12] Besagni G., Mereu R., Inzoli F., Ejector refrigeration: A comprehensive review. Renewable & Sustainable Energy Reviews, 2016, 53: 373–407.

[13] Llopis R., Torrella E., Cabello R., Sánchez D., HCFC-22 replacement with drop-in and retrofit HFC refrigerants in a two-stage refrigeration plant for low temperature. International Journal of Refrigeration, 2012, 35(4): 810–816.

[14] Park C., Lee H., Hwang Y., Radermacher R., Recent advances in vapor compression cycle technologies. International Journal of Refrigeration, 2015, 60: 118–134.

[15] Jiang S., Wang S.G., Jin X., Yu Y., The role of optimum intermediate pressure in the design of two-stage vapor compression systems: A further investigation. International Journal of Refrigeration, 2016, 70: 57–70.

[16] Tan Y., Chen Y., Wang L., Thermodynamic analysis of a mixed refrigerant ejector refrigeration cycle operating with two vapor-liquid separators. Journal of Thermal Science, 2018, 27(3): 230–240.

[17] Keenan J.H., Neuman E.P., Lustwerk F., An investigation of ejector design by analysis and experiment. ASME Journal of Applied Mechanics, 1950, 72: 299–309.

[18] Manjili F.E., Yavari M.A., Performance of a new two-stage multi-intercooling transcritical CO_2 ejector refrigeration cycle. Applied Thermal Engineering, 2012,

40: 202–209.

[19] Wang X., Yu J.L., Zhou M.L., Lv X.L., Comparative studies of ejector-expansion vapor compression refrigeration cycles for applications in domestic refrigerator-freezers. Energy, 2014, 70(1): 635–642.

[20] Lucas C., Koehler J., Experimental investigation of the COP improvement of a refrigeration cycle by use of an ejector. International Journal of Refrigeration, 2012, 35(6): 1595–1603.

[21] Brunin O., Feidt M., Hivet B., Comparison of the working domains of some compression heat pumps and a compression-absorption heat pump. International Journal of Refrigeration, 1997, 20(5): 308–318.

[22] Lemmon E.W., Huber M.L., McLinden M.O., NIST Thermodynamic and Transport Properties of Refrigerants and Refrigerant Mixtures (REFPROP) Version 8.0. NIST, 2007.

[23] Navarro E., Urchueguía J.F., Gonzálvez J., Corberán J.M., Test results of performance and oil circulation rate of commercial reciprocating compressors of different capacities working with propane (R290) as refrigerant. International Journal of Refrigeration, 2005, 28(6): 881–888.

[24] Aroraa A., Kaushikb S.C., Theoretical analysis of a vapour compression refrigeration system with R502, R404A and R507A. International Journal of Refrigeration, 2008, 31(6): 998–1005.

[25] Alexis G.K., Rogdakis E.D., A verification study of steam-ejector refrigeration model. Applied Thermal Engineering, 2003, 23(1): 29–36.

[26] Sun D.W., Variable geometry ejectors and their applications in ejector refrigeration systems. Energy, 1996, 21(10): 919–929.

[27] Vereda C., Ventas R., Lecuona A., Venegas M., Study of an ejector-absorption refrigeration cycle with an adaptable ejector nozzle for different working conditions. Applied Energy, 2012, 97: 305–312.

[28] Elbel S., Hrnjak P., Experimental validation of a prototype ejector designed to reduce throttling losses encountered in transcritical R744 system operation. International Journal of Refrigeration, 2008, 31(3): 411–422.

[29] Zheng L.X., Deng J.Q., Research on CO_2 ejector component efficiencies by experiment measurement and distributed-parameter modeling. Energy Conversion and Management, 2017, 142(15): 244–256.