Thermodynamic Analysis On A Novel Gas-gas Ejector Enhanced Autocascade Refrigeration Cycle

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Thermodynamic Analysis On a Novel Gas-gas Ejector Enhanced Autocascade Refrigeration Cycle

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ABSTRACT

A novel gas-gas ejector enhanced autocascade refrigeration cycle is proposed in this paper. The novel cycle uses an additional gas-gas ejector and a phase separator to accomplish a secondary composition separation for the zeotropic mixture used in the cycle. This proposed cycle can lift the evaporating pressure for a specific evaporating pressure and further improve the cycle performance. The performance comparisons between the novel cycle and a basic autocascade refrigeration cycle using the zeotropic mixtures of R23/R134a are conducted by theoretical method. In the evaporator inlet temperature range of -60°C to -40°C, the coefficient of performance (COP) and volumetric cooling capacity of the novel cycle can be improved by up to an average of 25.9% and 32.7% compared to that of the basic cycle, respectively. The effect of some key parameters on the performance of the novel cycle is further investigated. As the intermediate pressure of the novel cycle decreases in a certain range, the entrainment ratio and pressure lift ratio of the ejector increase and it further leads to the performance improvement of the novel cycle. In a given vapor quality range, the COP of the basic cycle show a monotonic increase tendency with the vapor quality at the condenser outlet, but the novel cycle has an optimal vapor quality to obtain the maximum COP. The simulation results shows that the novel cycle yields an improvement of 40.1-21.4% in the COP over the basic cycle as the vapor quality varies from 0.3 to 0.5. Besides, there exist optimal mixture compositions for the two cycles to obtain the optimum performance. The maximum COP of the two cycles can be obtained under the considered condition when the R134a mass fraction of the zeotropic mixture R23/R134a at the compressor inlet is fixed at about 0.6.

1. INTRODUCTION

With the development of the food, biology and space technology, the demands for low and ultralow temperature environment increase in recent years. However, the single-stage vapor compression refrigerators is not suitable for the refrigerating temperature range below −40 °C because the pressure ratio of the compressor will be too high to guarantee normal operation. Actually, different refrigerants can be mixed to possibly become the zeotropic mixtures, which have the features of temperature glide and composition shift during phase change (Missimer, 1997). Utilizing these two features, the autoascade refrigeration cycles (ARCs) with the zeotropic mixtures can achieve multilevel auto-cascade using one compressor and then greatly simplify the system. Thus, the ARC system has been an efficient method to obtain low and ultralow refrigerating temperature. Du et al. (2009) presented the experimental results of the cycle characteristics of a basic autocascade refrigeration system using zoetropic mixture of R23/R134a and constituted a useful source for the autocascade refrigeration system’s design and analysis. Zhao et
al. (2014) conducted a thermodynamic analysis of an autocascade heat pump cycle and indicated that the cycle could significantly improve the heating performance in cold climate compared with the single-stage air-source heat pumps. Moreover, there are also many researches focusing on the choice of the zerotropic refrigerants and the system configuration optimization. Kim and Kim (2002) proposed to replace the CFC refrigerants with R744 and carried out a simulation and experimental test of an autocascade refrigeration system using zeotropic mixtures of R744/R134a and R744/R290. Wang et al. (2013) simulated an ARC refrigerator operating with two vapor–liquid separators and study the effect of the main factors on its coefficient of performance (COP). Zhang et al. (2015) proposed a small-sized autocascade refrigeration cycle using CO2/propane as the refrigerant, in which a fractionation heat exchanger was introduced to improve the separation efficiency of the mixture.

As seen from the literature survey outlined above, the ARC system has been an effective alternative to obtain the relatively low temperature, and thus it's meaningful to design more efficient ARC cycles for improving the performance. For this purpose, a novel gas-gas ejector enhanced autocascade refrigeration cycle (NARC) is proposed in this paper, which uses an additional gas-gas ejector and a phase separator to accomplish a secondary composition separation for the zeotropic mixture. With the help of the additional gas-gas ejector and phase separator, the NARC can significantly lift the evaporating pressure for a specific evaporating temperature and further improve its performance. The performance comparisons between the NARC and ARC will be conducted by theoretical method, and the effect of main operation parameters on the two cycles’ performance will be also evaluated.

2. CYCLE SYSTEM DESCRIPTION

The schematic diagram of a basic autocascade refrigeration cycle system widely used in the low temperature refrigeration fields is illustrated in Fig. 1. The basic ARC cycle system is composed of a compressor, a condenser, a phase separator, a cascade condenser, two expansion valves and an evaporator. With the help of the phase separator, the ARC conducts a composition separation process for the zeotropic mixture and achieve better performance for low temperature application compared with the single-stage refrigeration cycle. Actually, the performance of the ARC can be further improved by conducting a secondary composition separation for the zeotropic mixture. For this purpose, a novel gas-gas ejector enhanced autocascade refrigeration cycle (NARC) is proposed, which uses an additional gas-gas ejector and a phase separator to accomplish a secondary composition separation for the zeotropic mixture. The schematic diagram of NARC is illustrated in the Fig. 2.

The NARC system operates in the following manner: the superheated refrigerant vapor (state 2) discharged from the compressor partially condenses to two-phase refrigerant fluid in the condenser; this two-phase refrigerant fluid (state 3) accomplishes the first composition separation process in the phase separator-1 and splits into the saturated liquid and saturated vapor. After leaving the phase separator-1, the saturated vapor (state 3v) enters the motive nozzle of the ejector as the primary fluid; this saturated liquid (state 3l) expands to two-phase fluid in the expansion valve-1; this two-phase fluid (state 4) accomplishes the secondary composition separation process in the phase separator-2 and splits into the saturated liquid and saturated vapor; after leaving the separator-2, the saturated liquid (state 4l) expands to two-phase fluid (state 5) in the expansion valve-2 and then evaporates to the saturated or superheated vapor (state 6) in the cascade condenser; the saturated vapor (state 4v) leaving the phase separator-2 enters the suction nozzle of the ejector as the secondary flow; the refrigerant fluid (state 7) leaving from the ejector exit condenses to saturated or subcooled liquid in the cascade condenser; this saturated or subcooled liquid (state 8) expands to two-phase fluid (state 9) through the expansion valve-3 and then evaporates to the saturated or
superheated vapor in the evaporator; this saturated or superheated vapor (state 10) mixes with the saturated or superheated vapor (state 6) leaving from the cascade condenser to state 1 and returns to the compressor at last.

**Figure 1:** Schematic diagram of the ARC system  
**Figure 2:** Schematic diagram of the NARC system

### 3. SIMULATION MODEL

As mentioned above, a gas-gas ejector is added in the novel autocascade refrigeration cycle, and thus it’s necessary to establish a recognized simulation model for the ejector. According to the studies of other researchers (Sarkar, 2012), it’s reasonable to give the assumptions as follow.

1. The refrigerant is mixed on the constant pressure in the ejector.
2. The isentropic efficiencies of the ejector can be assumed as constant.
3. The ejector is assumed to be at the steady state and adiabatic.
4. The pressure drop and heat losses in the ejector are neglected.
5. The kinetic energy of refrigerant at the inlet and outlet of the ejector can be negligible.

Based on the assumptions above and the conservation of energy, mass and momentum, the simulation model of the ejector is built. The entrainment ratio $\mu$ and the pressure lift ratio $r_p$ of the ejector are defined as

$$\mu = \frac{\dot{m}_s}{\dot{m}_p}$$  \hfill (1)

$$r_p = \frac{P_d}{P_s}$$  \hfill (2)

where $\dot{m}_p$ and $\dot{m}_s$ are the mass flow rates of the primary and secondary flow, respectively; $P_i$ and $P_o$ are the inlet pressure of the suction nozzle and exit pressure of the diffuser, respectively.

The kinetic energy of the refrigerant at the ejector inlets can be negligible, and thus the exit velocity of primary fluid leaving the nozzle can be obtained

$$w_{pi} = \sqrt{2\eta_s (h_{pi} - h_{p,i,o}) \times 1000}$$  \hfill (3)

where $\eta_s$ is the isentropic efficiency of the nozzle; $h_{pi}$ is the inlet specific enthalpy of the primary fluid; $h_{p,i,o}$ is the ideal outlet specific enthalpy of the primary fluid at the nozzle outlet under the isentropic expansion condition.

The momentum conservation equations and energy conservation equations for the mixing process in the mixing
The chamber can be written as follows:

\[(m_s + m_p)w_{m2} = m_p w_{p2} \sqrt{\eta_m}\]  

\[(m_p + m_s)(h_{m2} + \frac{w^2_{m2}}{2})/1000 = m_s h_{s1} + m_p h_{p1}\]  

And then the exit velocity \(w_{m2}\) and the specific enthalpy \(h_{m2}\) of the mixed fluid at the outlet of the mixing chamber are obtained as

\[w_{m2} = \frac{w_{m2}}{1 + \mu} \sqrt{\eta_m}\]  

\[h_{m2} = \frac{h_{p1} + \mu h_{s1}}{1 + \mu} - \frac{w^2_{m2}}{2000}\]

where \(\eta_m\) is the mixing efficiency in the mixing chamber; \(h_{s1}\) is the specific enthalpy of the secondary flow at the suction nozzle inlet.

In the diffuser section, the mixed flow converts the kinetic energy into the pressure energy. And then the specific enthalpy of the mixed fluid at the diffuser outlet can be obtained as

\[h_{d2} = h_{m2} + \frac{h_{d2, is} - h_{m2}}{\eta_d}\]

where \(h_{d2, is}\) is the ideal outlet specific enthalpy of the mixed fluid at the diffuser outlet under an isentropic compression condition; \(\eta_d\) is the diffuser isentropic efficiency.

As mentioned above, the kinetic energy of refrigerant at the outlet of the ejector is negligible, and thus the entrainment ratio for the ejector can also be determined as

\[\mu_e = \sqrt{\frac{\eta_s \eta_m \eta_d}{h_{d2, is} - h_{m2}} - 1}\]

For simplicity, the simulation model of the NARC cycle is built based on the common thermodynamic analysis method. The input specific work of compressor is

\[\hat{W}_{com} = \hat{m} \frac{h_i - h}{\eta_{com}}\]

\(h_i\) and \(h\) is the specific enthalpy of the refrigerant at the inlet and outlet of the compressor; \(\hat{m}\) is the refrigerant flow rate in the compressor; \(\eta_{com}\) is compressor isentropic efficiency, which can be obtained (Brunin et al., 1997)

\[\eta_{com} = 0.874 - 0.0135 \frac{P_2}{P_1}\]

where \(P_1\) and \(P_2\) is the refrigerant pressure at the inlet and outlet of the compressor.

The energy balance equation of the cascade condenser is

\[\hat{m} \left[\chi_{con} + \chi_{exp1} \left(1 - \chi_{con}\right)\right] (h_i - h_s) = \hat{m} (1 - \chi_{con}) (1 - \chi_{con}) (h_i - h)\]

where \(h_s\), \(h\), \(\chi_{con}\), \(\chi_{exp1}\) are the specific enthalpies of the mixture refrigerant; \(\chi_{con}\) is the vapor quality at the condenser outlet (point 3); \(\chi_{exp1}\) is the vapor quality at the expansion valve-1 outlet (point 4), which is dependent.
on the intermediate pressure $P_{im}$. The intermediate pressure ratio $\phi_{im}$ is defined as the ratio of the intermediate pressure to the condensing pressure $P_c$

$$\phi_{im} = \frac{P_{im}}{P_c}$$

(13)

The cooling capacity of the system can be written as

$$\dot{Q}_c = m\left[\chi_{con} + \chi_{exp} (1 - \chi_{con})\right](h_{c0} - h_c)$$

(14)

The volumetric cooling capacity of the system is

$$q_v = \left[\chi_{con} + \chi_{exp} (1 - \chi_{con})\right](h_{c0} - h_c)/\nu_i$$

(15)

where $h_c$ and $h_{c0}$ is the refrigerant specific enthalpy at the inlet and outlet of the evaporator, respectively; $\nu_i$ is the specific volume of the refrigerant at the compressor inlet.

The coefficient of performance (COP) of the system is given as

$$\text{COP} = \frac{\dot{Q}_c}{W_{com}}$$

(16)

It should be noted that the theoretical model of the ARC will be not detailed for simplicity, because it has been investigated in many literatures (Somasundaram et al., 2004). However, a basic ARC system will be still simulated to evaluate the performance improvement of the NARC over the ARC. The simulation program is written in the Fortran Language, and the refrigerant thermodynamic properties are obtained by invoking the property subroutines of REFPROP 8.0 (Lemmon et al., 2007).

4. RESULTS AND DISCUSSION

The thermodynamic properties of the mixture R134a/R23, i.e. the normal boiling points, -82.1°C for R23, -26.2°C for R134a, and the boiling point gap (59.9 °C), are satisfactory for the autocascade refrigeration cycle system (Bonekamp and Bier, 1997), so it is widely used as the mixture refrigerant for the auto-cascade refrigeration system (Hu et al., 2014). Thus, the mixture R23/R134a is selected as the typical refrigerant for the ARC and NARC systems in this paper. The performance comparison between the NARC and ARC systems is conducted under the following operation conditions. The condenser outlet temperature $t_c$ is constant of 35°C; the evaporator inlet temperature $t_e$ is chosen as the evaporator operation parameter, which varies from -60°C to -40°C; both the superheating degree at the evaporator and the subcooling degree at the cascade condenser are kept at 0°C; the vapor quality at the condenser outlet $\chi_{con}$ varies from 0.3 to 0.5; the R134a mass fraction of the refrigerant mixture at the compressor inlet $z$ ranges from 0.4 to 0.7; the intermediate pressure ratio $\phi_{im}$ varies from 0.6 to 0.9. As known, the ejector efficiencies vary with the ejector geometries and operating conditions, but the value variations are too difficult to be predicted by theoretical approach. Thus, the general method adopted by researchers is to assume the ejector efficiencies as constants when analyzing the cycle performance. Considering this case, the isentropic efficiencies of the ejector in the NARC are assumed to $\eta_n = 0.8$, $\eta_m = 0.95$ and $\eta_d = 0.8$ (Liu and Groll, 2013; Yari, 2009). The performance variations of the NARC and ARC cycles with respect to the evaporator inlet temperature $t_e$ are displayed in Fig. 3. The intermediate pressure ratio $\phi_{im}$ is assumed as 0.7 and 0.6, when $z$ and $\chi_{con}$ are constant of 0.6 and 0.4, respectively. It can be seen from Fig. 3 that, as $t_e$ increasing, the volumetric cooling capacity and COP of the two cycles are improved as expected. Although the variation tendencies of the COP and volumetric
cooling capacity of two cycles are similar, the NARC outperforms the ARC in both aspects for the entire $t_e$ range. According to the simulation results, the NARC can yield the improvements of 29.2-23.9% in the COP aspect and 36.0-29.3% in the volumetric cooling capacity aspect compared with the ARC, as $t_e$ is increased from -60°C to -40°C. The reason can be attributed to that, the existence of the secondary composition separation can reduce the vapor quality at the evaporator inlet and further offer an increase in the evaporating pressure for a specified evaporating temperature due to the temperature glide feature of the zeotropic mixture. Actually, the compression ratio of the NARC can be lowered by an average of 9.2% compared with that of the ARC for $\phi_m$ of 0.7. Moreover, with the help of the additional gas-gas ejector and phase separator, the refrigerant flow through the evaporator will be increased and then leads to the increase of the cooling capacity. And it will also contribute to the performance improvement of the NARC.

Figure 3: the performance variations of two cycles versus $t_e$

Figure 4 displays that the COP and volumetric cooling capacity of the NARC are increased with decreasing the intermediate pressure ratio $\phi_m$. The evaporator inlet temperature is assumed as -45°C and -55°C, while $z$ and $X_{coh}$ are constant of 0.6 and 0.4, respectively. Actually, as $\phi_m$ is decreased from 0.9 to 0.6, the NARC can yield the improvements of 7.8-37.37% in the COP aspect and of 9.6-48.3% in the volumetric cooling capacity aspect over the ARC for $t_e$ of -55°C. This means that, within a certain range, lowering the intermediate pressure benefits the performance of the NARC. It can be attributed to that lowering the intermediate pressure means that more refrigerant vapor is separated from the separator and enters the ejector, which will further lead to the increase of the cooling capacity. Simultaneously, as mentioned above, lowering intermediate pressure also help to lift the evaporating pressure and decrease the working pressure ratio of the compressor. According to the simulation results, when $\phi_m$ is decreased from 0.9 to 0.6, the working pressure ratio of the compressor of NARC can be lowered by 2.8-14.1% compared to that of the ARC for $t_e$ of -55°C. Additionally, it should be stressed that there is lower limit for the intermediate pressure ratio $\phi_m$, because the discharge temperature of the compressor will rise with decreasing the $\phi_m$. For example, as $\phi_m$ decreasing from 0.9 to 0.6, the discharge temperature of the compressor in the NARC system will be increased from 86°C to 106°C for $t_e$ of -55°C.

The variation tendencies of the entrainment ratio $\mu$ and the pressure lift ratio $r_p$ with decreasing the intermediate pressure ratio $\phi_m$ are illustrated in Fig.5. As shown, both the entrainment ratio and the pressure lift ratio of the
The ejector in the NARC are increased with decreasing the intermediate pressure ratio \( \phi_m \). Obviously, as \( \phi_m \) decreasing, more refrigerant vapor leaves from the phase separator -2 and enters the ejector as the secondary flow, which certainly leads to the increase of the entrainment ratio. Moreover, according to the simulation results, both the inlet pressure of suction nozzle and exit pressure of the diffuser are reduced with decreasing the intermediate pressure, while the inlet pressure of suction nozzle is reduced more rapidly. Therefore, the pressure lift ratio \( r_p \) of the ejector also increases with decreasing the intermediate pressure ratio.

Figure 4: the performance variations of two cycles versus \( \phi_m \)

Figure 5: the performance variations of the ejector versus \( \phi_m \)

Figure 6 shows the effect of the vapor quality at the condenser outlet \( \chi_{\text{con}} \) on the performance of the two cycles when the intermediate pressure ratio \( \phi_m \) is 0.7 and 0.6. The evaporator inlet temperature \( t_e \) is \(-55^\circ\text{C}\) and the R134a mass fraction \( z \) is 0.6. As shown, in the given range of the vapor quality, the NARC actually has an optimal vapor quality to obtain the maximum COP, although the change of the COP is minor. According to the simulation results, the COP of the NARC can reach the peak value of 0.819 at \( \chi_{\text{con}} \) of 0.375, while that of the ARC show a monotonic increase tendency with increasing the vapor quality at the condenser outlet. In addition, the volumetric cooling capacity and the COP shows the similar variations tendency with increasing the vapor quality. The reason can be attributed to that the increase of \( \chi_{\text{con}} \) can lead to the increase of the cooling capacity and the decrease of the evaporating pressure, and the performance variation is dependent on their synthetic action. For the ARC, within the given vapor quality range, the increase of the cooling capacity always plays a major role in the cycle performance variation, and thus the performance is improved monotonically. For the NARC, as the vapor quality increasing, the performance is improved at first because the effect of the increase of the cooling capacity is more dominant than that of the decrease of the evaporating pressure. And then, the effect of the decrease of the evaporating pressure becomes more dominant when the vapor quality beyond a limit. Thus, the performance of the NARC will be deteriorated after reaching the peak value. Therefore, the NARC has optimal vapor qualities to obtain the maximum COP and volumetric cooling capacity. In addition, it also can be seen that the fluctuation in the COP of the NARC with the vapor quality is really minor, which indicates that the NARC can always maintain high COP for the entire vapor quality range. It can be attributed to that, for the NARC, the values of the vapor quality are always near the optimal
point under the considered range, and thus the variation tendency of the COP with the vapor quality is relatively flat.

The pressure-temperature relationship of the refrigerant mixture adopted by the system is dependent on its mass fraction, and further influences the system performance. The effect of the R134a mass fraction of the refrigerant mixture at the compressor inlet \( z \) on the performance of two cycles is presented in Fig. 7, when the intermediate pressure ratio \( \phi_m \) is 0.7 and 0.6. The evaporator inlet temperature \( t_e \) is -55°C and the vapor quality \( \chi_{\text{con}} \) is 0.4. It can be seen that there exist optimal mass fractions for the two cycles to obtain the maximum COP, while their volumetric cooling capacities shows monotonous downtrend with increasing the mass fraction. As shown, although the NARC always outperforms the ARC in the COP aspect, the optimum mass fractions of the two cycles corresponding to the maximum COP are very close and the values are approximately 0.6. It also indicates that the use of the additional gas-gas ejector and phase separator can improve the system performance without affecting the optimum mass quality. In addition, the R134a mass fraction in the evaporators of two cycles certainly increases with increasing the R134a mass fraction at the compressor inlet \( z \), and further lead to the reduction of the evaporating pressure. According to the simulation results, as the R134a mass fraction \( z \) is increased from 0.3 to 0.7, the evaporating pressure of the NARC and ARC will be lowered by 59.9% and 61.7%, respectively. And thus, the volumetric cooling capacity of two cycles is monotonously decreased with increasing the R134a mass fraction \( z \).

5. CONCLUSION

As an attempt to improve the performance of the ARC systems, a novel gas-gas ejector enhanced autocascade refrigeration cycle is proposed, which adopted an additional gas-gas ejector and a phase separator to accomplish a secondary composition separation for the zeotropic mixture. The theoretical model of the novel cycle is built, and then the performance comparisons between the NARC and a basic ARC cycles are conducted adopting the typical mixture refrigerant of R23/R134a. Under the given operation conditions, the NARC cycle can improve the coefficient of performance and volumetric cooling capacity by up to an average of 25.9% and 32.7% compared to the ARC cycle, respectively. The effect of some key parameters on the cycle performance is investigated by the theoretical method, including the intermediate pressure, vapor quality at the condenser outlet and the mass fraction of the refrigerant mixture. The simulation results show that, within a certain range, lowering the intermediate
pressure benefits the performance of the NARC. In the given vapor quality range, there exist optimal vapor qualities for the NARC to obtain the maximum COP and volumetric cooling capacity, while the COP and volumetric cooling capacity of the ARC show monotonic growth tendencies with increasing the vapor quality. Meanwhile, the fluctuation in the COP of the NARC with the vapor quality is actually minor, indicating that the NARC can always maintain relatively high COP for the entire vapor quality range. In addition, although the NARC always outperforms the ARC in the COP aspect, the two cycles can obtain the maximum COP at the R134a mass fraction of about 0.6 under the given conditions. The volumetric cooling capacity of two cycles is monotonously reduced with increasing the R134a mass fraction. It’s hoped that this research can be a guide for the design and operation of the NARC cycle.

### NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
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<tr>
<td>COP</td>
<td>coefficient of performance</td>
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<tr>
<td>$h$</td>
<td>specific enthalpy</td>
<td>(kJ kg$^{-1}$)</td>
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<tr>
<td>$m$</td>
<td>mass flow rate</td>
<td>(kg s$^{-1}$)</td>
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<tr>
<td>$P$</td>
<td>pressure</td>
<td>(kPa)</td>
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<td>$q_{vc}$</td>
<td>volumetric cooling capacity</td>
<td>(kJ m$^{-1}$)</td>
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<td>$Q$</td>
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<td>(W)</td>
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<tr>
<td>$\chi$</td>
<td>vapor quality</td>
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**Subscripts**

- 1-10: state points
- c: cooling
- com: compressor
- d: diffuser
- e: evaporator
- i: inlet
- im: intermediate
- m: mixing process
- n: nozzle
- o: outlet
- p: primary flow
- s: secondary flow
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