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Performance Analysis of Refrigerants R1234yf, R1234ze and R134a in Ejector-Based Refrigeration Cycle

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Performance enhancement of refrigeration and heat pump systems by cycle modification is an emerging research topic now-a-days to reduce the electricity consumption leading to mitigate the problems related to the environmental pollution by utility power plants. Due to no moving parts, low cost, simple structure and low maintenance requirements, the use of two-phase ejector has become a promising cycle modification recently. Use of ejector as an expansion device by replacing the throttle valve in the vapor compression refrigeration cycle seems to be one of the efficient ways to reduce the throttling losses or the expansion irreversibility in the refrigeration/heat pump cycle. Ejector also reduces the compressor work by raising the suction pressure to a level higher than that in the evaporator leading to the improvement of COP. The present work aims to evaluate the performance of an ejector based vapor compression refrigeration cycle under a wide range of operating conditions. Two newly proposed refrigerants i.e., R1234yf and R1234ze, and commonly used refrigerant R134a are considered for simulation and a comparative study has been carried out. A numerical model is developed and a parametric study of important parameters such as entrainment ratio, high side pressure (condenser pressure) and evaporator temperature are analyzed for the improvement of COP of the system. Results show that the COP of the R1234ze is highest compared to R1234yf and R134a for the given evaporating and condensing temperature.

Keywords: Ejector expansion refrigeration system; thermodynamic analysis, COP; refrigerants; R1234ze.

Nomenclature

- A : Cross sectional area (m^2)
- AR : Area ratio
- COP : Coefficient of performance
- C : Velocity (m/s)
- H : Enthalpy (Jkg^{-1})
- μ : Entrainment ratio
- P : Pressure (Pa)
- PR : Compression ratio/Pressure ratio
- q : Cooling capacity
- ρ : Density (kg/m^3)
- s : Entropy (kJ/kgK)
- T : Temperature (°C)
- v : Specific volume (m^3/kg)
- VCC : Volumetric cooling capacity (kJ/m^3)
- w : Specific work (kJ/kg)
- N : Efficiency
- dp : Pressure drop (kPa)
- PLR : Pressure lift radio

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Subscripts

c : Compressor
e : Evaporator
ms : Mixing section
mn : Motive nozzle
sn : Suction nozzle

1. Introduction

There are several ways of improving the performance of a vapor compression refrigeration cycle. Use of an ejector as expansion device is one of the alternative ways. The main advantage of the ejector may be found in the recovery of the expansion work normally wasted in throttling processes at a typical expansion valve. It also helps in reducing evaporator size by using flash gas bypass.

The use of ejector in vapor compression system for performance improvement was first proposed in 1990s by Kornhauser,[1] who analyzed the thermodynamic performance of the ejector expansion refrigeration cycle using R12 as a refrigerant. After his work, quite good number of research papers on modification of ejector geometry along with different working fluids (refrigerants) have been published in last two decades.[2,3]

Optimizing geometry of ejector for better performance is one of the challenging area in ejector based refrigeration system. Domanski[4] pointed out that the ejector efficiency significantly influences the cooling COP of the ejector expansion refrigeration cycle. Fan *et al.*[5,6] and Wu *et al.*[7] studied the modified ejector expansion refrigeration cycle with two heat sources. Nakagawa and Takeuchi[8] showed that a longer diverging section in the nozzle increased the nozzle efficiency. Disawas and Wongwises[9] experimentally investigated the performance of the ejector expansion refrigeration cycle without the expansion valve at upstream of the evaporator so that the evaporator is flooded with the refrigerant. Their tests showed an improved cooling COP at low heat sink temperatures.

Finding suitable working fluid for ejector expansion refrigeration cycle is one of the interesting areas of research. Performance of the cycle using different working fluids is also studied by various researchers.[10–15] Sarkar[11,12] and Ezaz *et al.*[13] worked on natural refrigerants based system whereas Li *et al.*[14] worked on a new refrigerant R1234yf and obtained the performance characteristics of the ejector-based system.

In the present work, performance analysis of ejector expansion refrigeration cycle using R1234ze (a new refrigerant) is studied. To compare the performance, a widely used refrigerant R134a and a new refrigerant R1234yf are taken into consideration. Results are obtained at various operating temperature and pressures of evaporator and condenser with different entrainment ratios.

2. Cycle Description

The schematic and p-h diagram of the EERC are shown in Figs. 1 and 2, respectively. The saturated refrigerant vapor leaving from the gas–liquid separator at state 1 enters the compressor in which its pressure is raised to superheated (state 2). The refrigerant vapor subsequently is discharged to the condenser where it gets condensed to a saturated

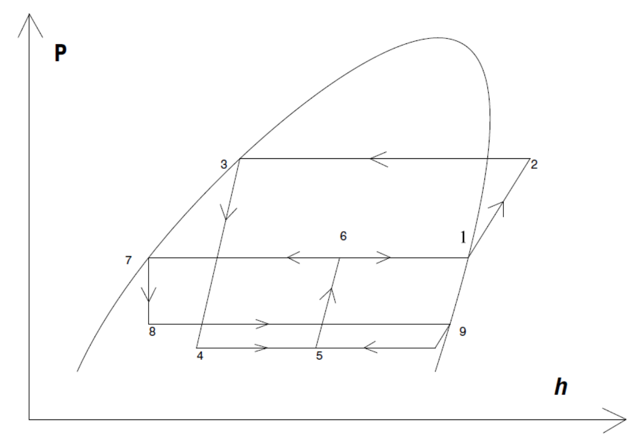

Fig. 1. p-h diagram of the R1234yf ejector-expansion refrigeration cycle.

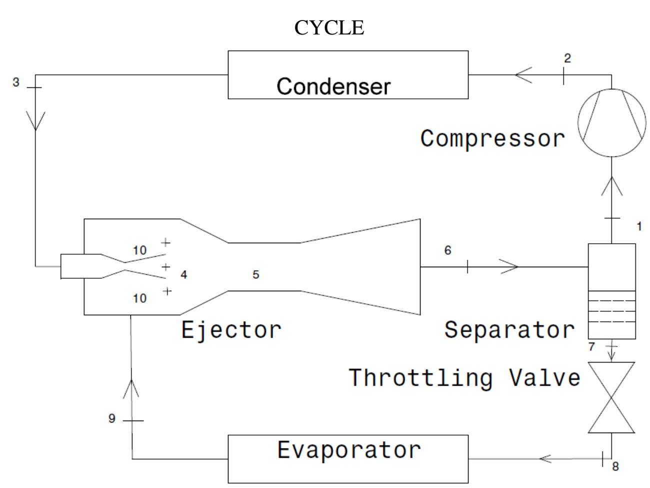

Fig. 2. Schematic of the ejector-expansion refrigeration cycle.

liquid (state 3). At the other side of the separator, saturated liquid (state 7) comes out and gets expanded through the throttle valve where its pressure and temperature drop to the evaporator condition (state 8). The refrigerant then enters the evaporator where it absorbs heat from the cooled medium and gets evaporated to saturated vapor (state 9). In the ejector, the motive flow from the condenser at state 3 expands through the motive nozzle, and its pressure decreases significantly and reaches the value below the evaporation pressure at the exit (state 4). Thus, the suction flow from the evaporator at state 9 is entrained into the ejector through the suction nozzle, and its pressure at the exit (state 10) equals to that of at state 4. Then, the two streams get mixed at constant pressure in the mixing section. The mixed flow at the end of the mixing section at state 5 enters the diffuser section where its velocity drops and pressure increases. Then the refrigerant leaves the ejector at state 6 and enters into the gas–liquid separator.

3. Thermodynamic Analysis of System

Following assumptions are considered for the analysis

(i) Steady state flow is considered.
(ii) Heat transfer through connecting pipes, ejector and separator is neglected.
(iii) Pressure drop in connecting pipe and heat exchanger is neglected.
(iv) The refrigerant leaving the condenser, evaporator and separator is saturated.
(v) The compressor has a given isentropic efficiency.
(vi) The velocities at the inlet and outlet of the ejector are neglected (as change in kinetic energy compared to change in enthalpy is negligible).
(vii) The friction losses inside the ejector are considered in terms of efficiencies of the nozzles, mixing section and diffuser.

With these assumptions, the mathematical model for the EERC can be established. Referring to Fig. 1 the analysis starts from determining the state parameters of the refrigerant flow at the outlets of the condenser and evaporator, i.e., the motive and suction streams of the ejector, respectively. At the condenser outlet,

$$P_3 = P_C = P \quad (\text{at } T = T_C, x = 0)\,, \tag{1}$$

$$h_3 = h \quad (\text{at } T = T_C, x = 0)\,, \tag{2}$$

$$s_3 = s \quad (\text{at } T = T_C, x = 0)\,. \tag{3}$$

At the evaporator outlet,

$$P_9 = Pe = P \quad (\text{at } T = T_e, x = 1)\,, \tag{4}$$

$$h_9 = h \quad (\text{at } T = T_e, x = 1)\,, \tag{5}$$

$$s_9 = s \quad (\text{at } T = T_e, x = 1)\,. \tag{6}$$

The ejector is a key component in the EERC, which can be classified into two types i.e., constant-pressure and constant-area ejectors. Based on the studies, researchers suggested that the constant-pressure mixing ejector has a better performance than that of the constant-area one.[2,3] So, in this study a constant pressure mixing ejector is used.

At motive nozzle outlet,

$$P_4 = P_9 - dp\,, \tag{7}$$

$$h_{4,s} = h \quad (\text{at } P = P_{4,s} = s_3)\,, \tag{8}$$

$$h_4 = h_3 - N_{\text{mn}}(h_3 - h_{4,s})\,, \tag{9}$$

$$c_4 = (2 \times (h_3 - h_4))^{0.5}\,, \tag{10}$$

$$A_4 = 1/((1+\mu) \times \rho_4 \times c_4)\,. \tag{11}$$

At the suction nozzle outlet,

$$P_{10} = P_9 - dp \tag{12}$$

$$h_{10,s} = h \ (\text{at } P = P_{10}, s = s_9)\,, \tag{13}$$

$$h_{10} = h_9 - N_{\text{sn}}(h_9 - h_{10,s})\,, \tag{14}$$

$$c_{10} = (2 \times (h_9 - h_{10}))^{0.5}\,, \tag{15}$$

$$A_{10} = \frac{\mu}{(1+\mu) \times \rho_{10} \times c_{10}}\,. \tag{16}$$

In the mixing section,

$$P_5 = P_4 = P_{10}\,, \tag{17}$$

$$c_5 = (N_{\text{ms}})^{0.5} \times \left(\frac{c_4}{1+\mu} + \frac{\mu \times c_{10}}{1+\mu}\right), \tag{18}$$

$$h_5 = \frac{1}{1+\mu} \times \left(h4 + \frac{c_4^2}{2}\right) + \frac{\mu}{1+\mu}\left(h_{10} + \frac{c_{10}^2}{2}\right) - \frac{c_5^2}{2}, \tag{19}$$

$$s_5 = s \ (\text{at } P = P_5, h = h_5)\,, \tag{20}$$

$$h_6 = h_5 + \frac{c_5^2}{2}\,, \tag{21}$$

$$h_{6,s} = h_5 - N_d(h_6 - h_5)\,, \tag{22}$$

$$P_6 = P \ (\text{at } P = P_{6,S}, S = S_5)\,, \tag{23}$$

In addition, the condition expressed by the following equation should be satisfied,

$$x_6 = \frac{1}{1+\mu}\,. \tag{24}$$

While T_3, T_9, dp, N_{mn}, N_{sn} and N_d are given, using Eqs. (1)–(24) the entrainment ratio μ of the ejector associated with state parameters can be calculated by iteration till Eq. (24) is true.

Then, another useful parameters for the ejector, i.e., the pressure lift ratio (PLR) is defined by Eq. (25).

$$\mathrm{PLR} = \frac{P_6}{P_9}\,. \tag{25}$$

Once the ejector performance is obtained, the performance of other components in the EERC can be calculated in ordinary way. At the gas–liquid separator outlets,

$$h_1 = h\ (\text{at } P = P_6, x = 1)\,, \tag{26}$$

$$h_7 = h\ (\text{at } P = P_6, x = 0)\,. \tag{27}$$

At the compressor outlet

$$h_{2,s} = h\ (\text{at } P = P_2, s = s_1)\,, \tag{28}$$

$$h_2 = h_1 + \frac{h_{2,s} - h_1}{N_c}\,. \tag{29}$$

The compressor isentropic efficiency is determined by an empirical relation proposed by Brunin *et al.*,[16]

$$N_c = 0.874 - 0.0135 \times \frac{P_c}{P_e}\,. \tag{30}$$

At the throttle valve outlet,

$$h_8 = h_7\,. \tag{31}$$

Then, the compressor work (W_c) and cooling capacity (q_e) of the EERC are defined by Eqs. (31) and (32), respectively.

$$W_c = \frac{h_2 - h_1}{1+\mu}\,, \tag{32}$$

$$q_e = \frac{\mu \times (h_9 - h_8)}{1+\mu}\,. \tag{33}$$

The COP of the cycle is determined by the following relation,

$$\mathrm{COP} = \frac{q_e}{W_c}\,. \tag{34}$$

The VCC of the cycle based on the specific suction volume of the compressor is given by,

$$\mathrm{VCC} = \frac{\mu \times (h_9 - h_8)}{v_1}\,. \tag{35}$$

Then the improvements in COP and VCC of the EERC over standard refrigeration cycle can be defined as,

$$\mathrm{COP}_{\mathrm{imp}} = \frac{\mathrm{COP} - \mathrm{COP}_s}{\mathrm{COP}_s}\,, \tag{36}$$

where COPs is the COP of the standard refrigeration cycle at same condensing and evaporation temperatures, and can be easily obtained through conventional calculation.

4. Results and Discussion

The performance characteristics of R1234yf, R1234ze and R134a based on EERC are investigated for various condensing temperatures (30–55°C) and evaporating temperatures (−10 to +10°C). In the analysis, the ejector is assumed to have the nozzle and diffuser efficiencies equal to 0.85, and mixing section efficiency equal to 0.95.[16]

The COP and pressure ratio (PR) of the R1234yf, R1234ze and R134a based EERC for different pressure drop (dP) in the ejector suction nozzle are presented in Figs. 3 and 4 for classical air-conditioning application with $T_e = 5$°C and $T_c = 40$°C. Results obtained for COP and pressure ratio are validated with the published data,[14] and found in good agreement. After validation of simulation model for R1234yf, the study is extended for obtaining performance of the system for different refrigerants at various conditions.

The comparison of COP of the refrigerants R134a, R1234ze and R1234yf is analyzed with the variation

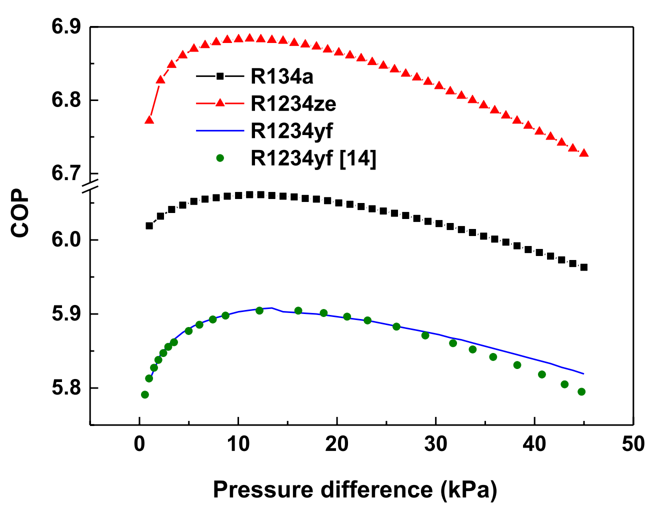

Fig. 3. COP variation of R1234ze, R134a and R1234yf versus pressure drop.

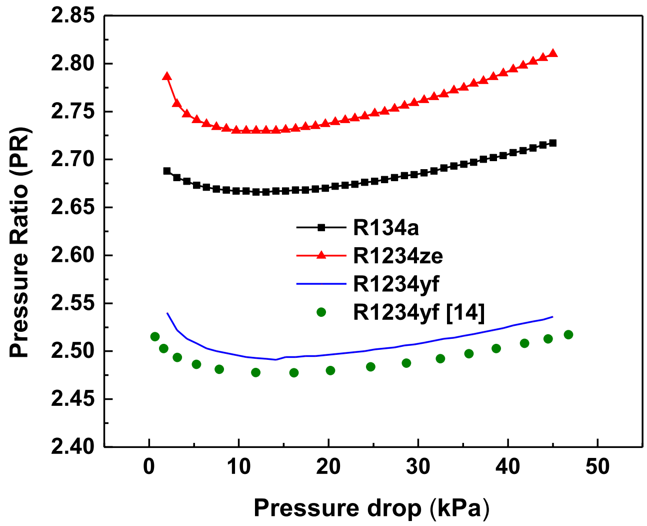

Fig. 4. Variation of pressure ratio of R1234ze, R134a and R1234yf versus pressure drop.

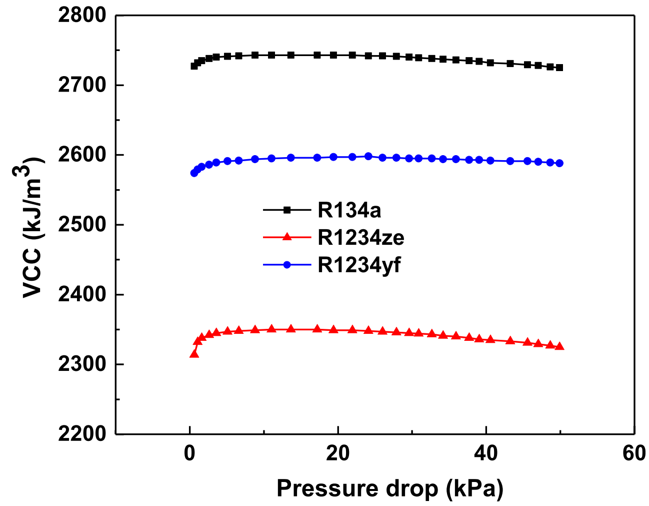

Fig. 5. Variation of VCC for R1234ze, R134a and R1234yf with pressure drop.

of pressure drop (dP). All three refrigerants have common trends and the maximum COP of all the three refrigerants obtained at the pressure drop between 11 kPa and 14 kPa as shown in Fig. 3. By comparing the COPs, it can be said that for the same pressure difference, R1234ze has highest COP, which occurs due to its better thermodynamic properties. The lowest COP is obtained for R1234yf and COP of R134a lies in between R1234ze and R1234yf. Beside COP, refrigerant 1234ze is also preferred over R134a due to very low global warming potential.

Figure 4 shows the variation of pressure ratio for R1234ze, R134a and R1234yf versus pressure drop in the ejector suction nozzle. The figure shows that R1234ze has maximum pressure ratio for given pressure drop, which occurs due to lower PLR of R1234ze compared to R134a and R1234yf as shown in Fig. 6. The lowest pressure ratio is obtained in the case of R1234yf.

Figure 5 shows variation of VCC at different pressure drop in the ejector suction nozzle. The maximum VCC of all three refrigerants occurs between the pressure drop of 5–12 kPa. It is observed that for the same pressure drop R134a has maximum VCC, which occurs due to high latent heat of vaporization of R134a. Results also show the lowest VCC in the case of R1234ze is attributed to the high specific volume of R1234ze at the suction of the compressor.

Figure 6 depicts the variation of PLR with different pressure drop in the ejector suction nozzle.

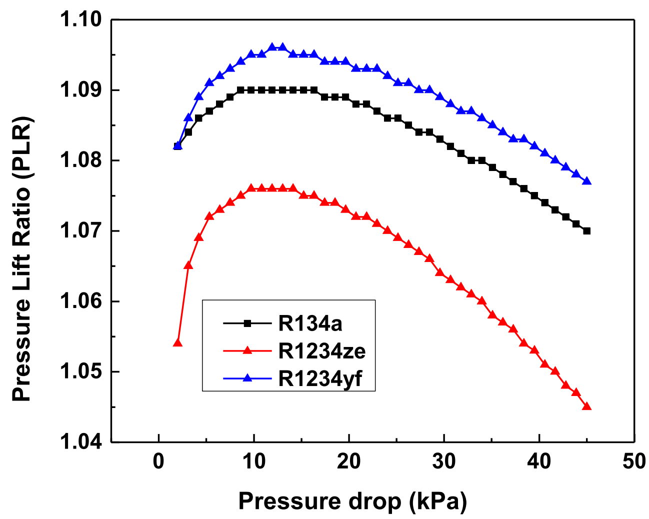

Fig. 6. Variation of PLR of R1234ze, R134a and R1234yf versus pressure drop.

The results show that PLR is maximum for the refrigerant R1234yf, and the minimum is for R1234ze.

Figures 7 and 8 represent variation of COP of the R1234ze, R134a and R1234yf EERC system versus different condensing temperatures and evaporating temperatures, respectively. From Fig. 7, it is seen that for a given condensing temperature as well as evaporating temperature, R1234ze has maximum COP. The trend of COP variation is the same in all three refrigerants. As condensing temperature increases, COP decreases (Fig. 7) due to increase in the difference between source and sink temperatures. As shown in Fig. 8, COP increases with increasing evaporating temperature, which occurs due

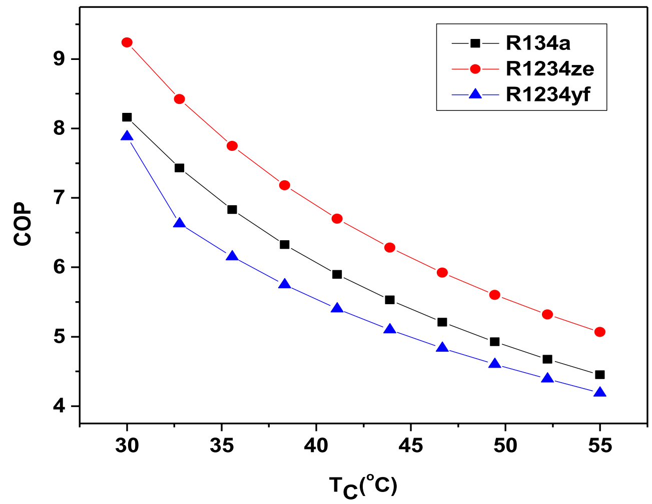

Fig. 7. Variation of COP of the R1234ze, R134a and R1234yf with condensing temperatures.

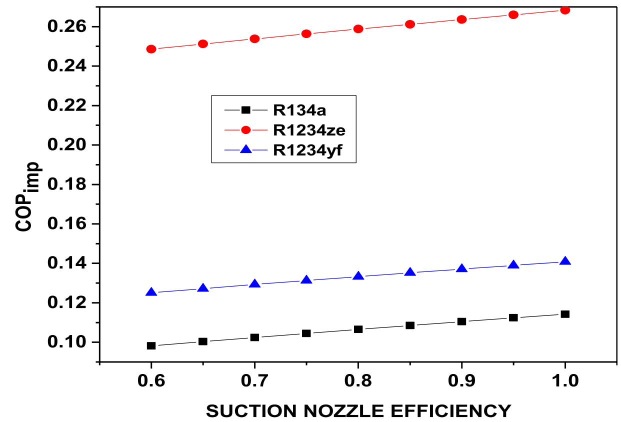

Fig. 10. COP improvement of the R1234ze, R134a and R1234yf versus suction nozzle efficiency.

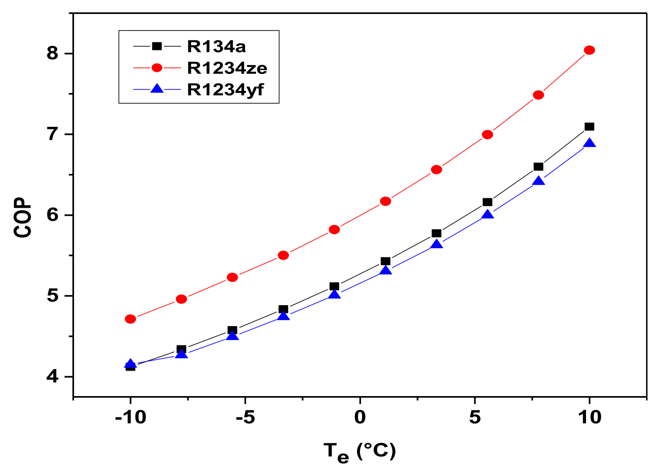

Fig. 8. Variation of COP of the R1234ze, R134a and R1234yf with evaporating temperatures.

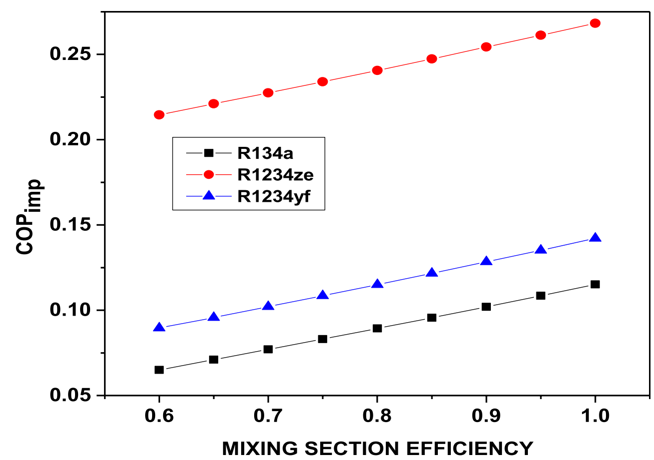

Fig. 11. COP improvement of the R1234ze, R134a and R1234yf for various mixing section efficiency.

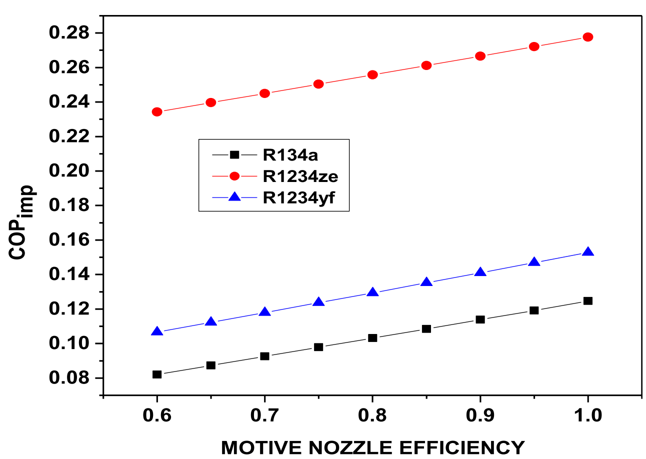

Fig. 9. COP improvement of the R1234ze, R134a and R1234yf versus motive nozzle efficiency.

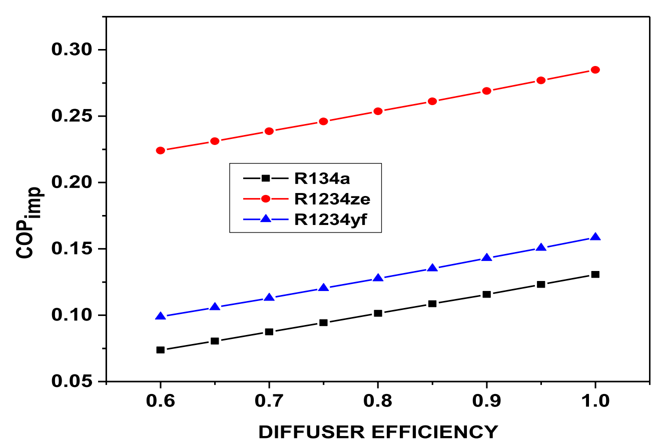

Fig. 12. COP improvement of the R1234ze, R134a and R1234yf for various diffuser efficiency.

to decrease in the difference between source and sink temperature.

Figures 9–12 show the effect of motive nozzle efficiency, suction nozzle efficiency, mixing section efficiency and diffuser efficiency on the improvement of COP for all three refrigerants considered in the present study. From all the cases, it can be seen that the improvement in COP increases with increase in efficiency for all refrigerants. By analyzing the performance among refrigerants, R1234ze shows much better improvement in COP compared to R1234yf and R134a.

5. Conclusions

Based on the above results of the three common refrigerants we can comment on the nature and performance of the refrigerants.

- For the given pressure drop between evaporator and ejector (dP), R1234ze offers maximum COP which occurs at a pressure drop of 11 kPa.
- For the given pressure drop between evaporator and ejector (dP), R134a has maximum VCC which occurs at a pressure difference of 10 kPa.
- The pressure ratio and PLR are maximum for R1234ze and R1234yf, respectively.
- For a particular condenser as well as evaporating temperature, R1234ze has a maximum COP.
- The COP_{imp} is higher for R1234ze compared to other refrigerants (R1234yf and R134a).
- By analyzing the performance of these three refrigerants, R1234ze yields better performance in most of the cases, which shows that this may also be a suitable candidate for refrigeration system.

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