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Performance characteristics of low global warming potential R134a alternative refrigerants in ejector-expansion refrigeration system

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Abstract Performance assessment of ejector-expansion vapor compression refrigeration system with eco-friendly R134a alternative refrigerants (R152a, R1234yf, R600a, R600, R290, R161, R32, and propylene) is presented for air-conditioning application. Ejector has been modeled by considering experimental data based correlations of component efficiencies to take care of all irreversibilities. Ejector area ratio has been optimized based on maximum coefficient of performance (COP) for typical air-conditioner operating temperatures. Selected refrigerants have been compared based on area ratio, pressure lift ratio, entrainment ratio, COP, COP improvement and volumetric cooling capacity. Effects of normal boiling point and critical point on the performances have been studied as well. Using ejector as an expansion device, maximum improvement in COP is noted in R1234yf (10.1%), which reduces the COP deviation with R134a (4.5% less in basic cycle and 2.5% less in ejector cycle). Hence, R1234yf seems to be best alternative for ejector expansion system due to its mild flammability and comparable volumetric capacity and cooling COP. refrigerant R161 is superior to R134a in terms of both COP and volumetric cooling capacity, although may be restricted for low capacity application due to its flammability.

Keywords: Eco-friendly refrigerant; Two-phase ejector; Optimization; COP; Area ratio; Critical temperature

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Nomenclature

a	–	cross-sectional area, m^2
C	–	volumetric refrigeration capacity, kJ/m^3
COP	–	coefficient of performance
h	–	specific enthalpy, kJ/kg
$\dot{m}$	–	mass flow rate, kg/s
p	–	pressure, kPa
PLR	–	pressure lift ratio
q	–	specific cooling effect, kJ/kg
s	–	specific entropy, kJ/kgK
t	–	temperature, °C
V	–	fluid velocity, m/s
v	–	fluid specific volume, m^3/kg
w	–	specific work, kJ/kg
x	–	two-phase quality

Greek symbols

η	–	component efficiency
μ	–	entrainment ratio
ϕ	–	ejector area ratio
ρ	–	fluid density, kg/m^3

Subscripts

b	–	nozle exit
c	–	condenser
co	–	compressor
d	–	diffuser
e	–	evaporator
m	–	mixing
mn	–	motive nozzle
sn	–	suction nozzle

1 Introduction

Today world face a lot of environmental problem such as global warming and ozone layer depletion due to use of high global warming potential (GWP) and ozone depletion potential (ODP) substances used as refrigerants. Hence, the conventional refrigerants are being phased out and need to be replaced by zero ODP and low GWP refrigerants. However, the performance level of many alternatives is slightly lower than conventional refrigerant and hence research and development on the performance improvement of vapor compression system by various cycle modifications has gained special interest recently. One of the very promising modifications is ejector expansion technology, which reduces the throttling losses and com-

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pressor work in refrigeration cycle. Nowadays, ejector expansion technology is more popular due to its less complex design, no moving parts, low cost and less maintenance cost [1,2]. Use of ejector in subcritical vapour compression cycle was introduced by Kornhauser [3] and in transcritical cycle by Liu *et al.* [4].

Refrigerant R134a is commonly used for both mobile and residential air conditioning applications. Due to high GWP (Tab. 1), R134a is going to be phased-out and various alternative refrigerants have been proposed [5-7]. Within last decades, many prototypes have been made by using some proposed R134a alternative refrigerants, such as hydrocarbons for residential air-conditioner and R134yf for automobile air-conditioner [8]. In the present study, R290, R600, R600a, R1270, R32, R152a, R161, and R1234yf have been selected based on environmental safety (zero ODP and negligible GWP) as R134a alternatives. Detailed physical, environmental and operational properties of selected refrigerant are given in Tab. 1. The research and development activities on ejector expansion vapor compression refrigeration system have achieved a milestone recently. Worldwide research activities on this area can be broadly classified in the following groups: (i) ejector-expansion transcritical refrigeration systems, (ii) energetic and exergetic performance improvements by using various refrigerants, (iii) CFD simulation and design optimization, (iv) experimental studies and ejector flow control, and (v) various ejector based cycle modifications [9–17]. Although, the ejector expansion refrigeration system with R134a has been studied extensively (both theoretical and experimental), similar studies for its alternatives are very limited. Within selected alternatives, only R290, R600a, R32, R152a, and R1234yf have been studied in ejector expansion refrigeration system by various authors [9,12,14] and most of these works have considered constant mixing pressure in ejector, which is not so realistic. With best of author's knowledge, no previous study has considered all these issues.

Present paper focuses on analyzing the performance of vapor compression system with ejector expansion technology by using eco-friendly refrigerants. Suitable R134a substitutes have been selected for air-conditioning application based on environmental criteria. The ejector has been modeled by taking care of pressure drop occur during mixing and it affect on the ejector performance. Comparison of optimum coefficient performance (COP), COP improvement, ejector area ratio, entrainment ratio, pressure lift ratio and volumetric cooling capacity with R134a, and its eight alternative

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Table 1: Physical and environmental properties of R134a and its alternatives.

Refrigerant	Normal boiling point (°C)	Critical temperature (°C)	Critical pressure (MPa)	GWP	Flammability	Safety class
R134a	-26.1	101	4.059	1300	inflammable	A1
R152a	-24.1	113.3	4.520	140	flammable	A2
R290	-42.1	96.7	4.247	20	flammable	A3
Propylene	-47.7	92.4	4.665	3	flammable	A3
R600	-0.53	152	3.796	20	flammable	A3
R600a	-11.7	134.7	3.640	3	flammable	A3
R32	-51.7	78.1	5.784	650	slightly	A2L
R1234yf	-29.5	99.7	3.382	4	slightly	A2L
R161	-37.6	102.1	5.010	12	flammable	A3

refrigerants is done by modeling and simulation considering irreversibility in all components. Effects of refrigerant critical temperature and normal boiling point on COP and volumetric refrigeration capacity are analyzed as well.

2 Mathematical modeling and simulation

The layout of the ejector expansion vapour compression cycle is shown in Fig. 1 and pressure-enthalpy diagram in Fig. 2. Motive stream from condenser and suction stream from evaporator are expanded through nozzles (1-1b and 2-2b), which are irreversible processes and may be affected by friction and shock. Refrigerant flows from primary nozzle and secondary nozzle then mix in the mixing chamber (1b-3m-2b). In the mixing chamber, two-phase flow streams mixes in a highly irreversible process which may include two-phase mixing shock wave, which is much thicker than a shock wave in gas-dynamics. The large velocity and temperature mismatch between the motive and suction flows as well as mismatch between liquid and vapor velocities may result in viscous losses and heat transfer, as well as several irreversible oblique shocks, which are commonly referred to as shock train. Hence, the process in the mixing chamber may be characterized by both pressure rise due to compression process (two-phase mixing shock wave or shock train) and the pressure drop due to fluid friction. Af-

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ter mixing, refrigerant is discharged through the diffuser (3m-3) by pressure rise above evaporator pressure and enters the separator. In the separator, liquid and vapour are separated; liquid is circulated through expansion valve (6-7) and then evaporator (7-2) to give useful cooling whereas vapour is the directed to the compressor (4-5) and then condenser (5-1).

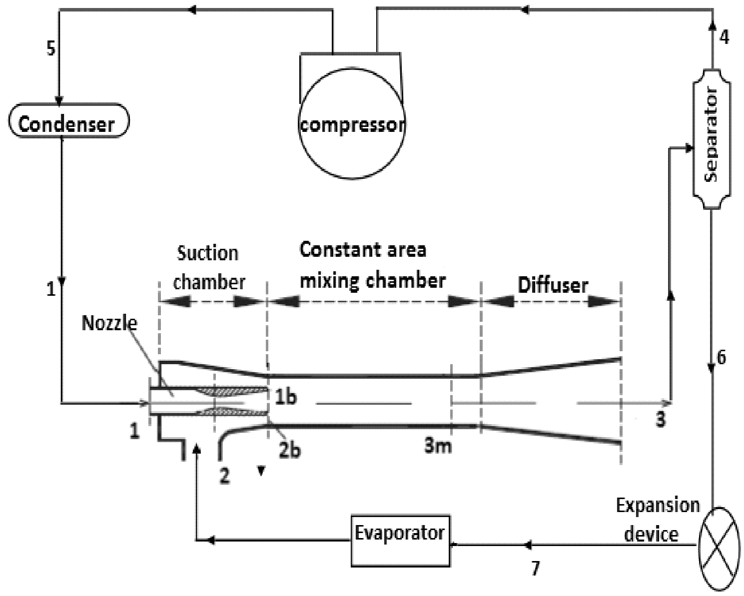

Figure 1: Schematic diagram of ejector expansion refrigeration cycle.

The ejector expansion vapour compression refrigeration cycle has been modeled based on mass, momentum and energy conservations. To simplify the theoretical model, the following assumptions have been made:

(i) pressure drops in heat exchangers and the connection tubes are negligible,

(ii) no heat transfer with the environment for the system except in the condenser,

(iii) refrigerant conditions at the evaporator and condenser exits are saturated,

(iv) both liquid and vapor streams separated from the separator are saturated,

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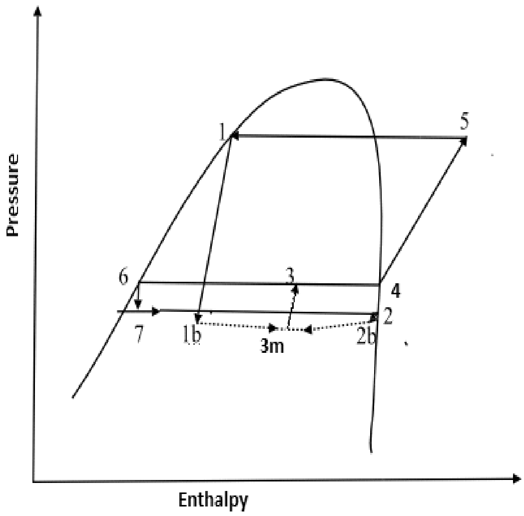

Figure 2: Pressure-enthalpy diagram of ejector expansion refrigeration cycle.

(v) flow across expansion valve or throttle valve is isenthalpic,

(vi) expansion efficiencies of the motive stream and suction stream are assumed as constant, diffuser of the ejector also has a given efficiency,

(vii) kinetic energies of the refrigerant at the ejector inlet and outlet are negligible,

(vii) both the motive stream and the suction stream reach the same pressure at the inlet of the constant area mixing section of the ejector without any premixing,

(ix) net mixing pressure drop is considered by using the mixing efficiency.

It may be noted that the assumptions for both nozzles and a diffuser are similar to the previous studies [12–14]. Hence, the outlet enthalpies and velocities of the motive stream and suction stream are given by [12]

$$h_{1b} = h_1 - \eta_{mn} \left(h_1 - h_{1s} \right) , \tag{1}$$

$$h_{2b} = h_2 - \eta_{sn} \left(h_2 - h_{2s} \right) , \tag{2}$$

$$V_{1b} = \sqrt{2 \left(h_1 - h_{1b} \right)} , \tag{3}$$

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$$V_{2b} = \sqrt{2\left(h_2 - h_{2b}\right)}\,. \tag{4}$$

Then, exit areas of motive and suction nozzles for given flow rates are as follows

$$a_{1b} = \frac{\dot{m}_{mn}}{\rho_{1b} V_{1b}}\,, \tag{5}$$

$$a_{2b} = \frac{\dot{m}_{sn}}{\rho_{2b} V_{2b}}\,. \tag{6}$$

Now, the ejector area ratio is defined as [12]

$$\phi = \frac{a_{1b} + a_{2b}}{a_{1b}}\,. \tag{7}$$

For modeling of the mixing chamber, the mixing efficiency has been used to account for the performance degradation due to both frictional losses as well as two-phase mixing shock wave in the mixing chamber. With assumed mixing efficiency, the mass, momentum and energy balance equations in ejector mixing section for unit total refrigerant mass flow rate are given by a set of equations to be solved simultaneously

$$\rho_{1b} a_{1b} V_{1b} + \rho_{2b} a_{2b} V_{2b} = \rho_{3m}(a_{1b} + a_{2b}) V_{3m}\,, \tag{8}$$

$$p_{2b}(a_{1b}+a_{2b}) + \eta_m\left[\dot{m}_{mn} V_{1b} + \dot{m}_{sn} V_{2b}\right] = p_{3m}(a_{1b}+a_{2b}) + (\dot{m}_{mn}+\dot{m}_{sn}) V_{3m}\,, \tag{9}$$

$$\frac{1}{1+\mu}\left(h_{1b} + \frac{V_{1b}^2}{2}\right) + \frac{\mu}{1+\mu}\left(h_{2b} + \frac{V_{2b}^2}{2}\right) = h_{3m} + \frac{V_{3m}^2}{2}\,. \tag{10}$$

At the exit of mixing section, four unknown quantities (density, pressure, enthalpy and velocity) can be found by simultaneous solving of above three equations and property function, $h_{3m} = f\left(p_{3m}, \rho_{3m}\right)$.
Then, by using energy balance and property function, the refrigerant enthalpy and pressure at the diffuser section exit of ejector can be found by

$$h_3 = h_{3m} + \frac{V_{3m}^2}{2}\,, \tag{11}$$

$$p_3 = p\big[h_{3m} + \eta_d\left(h_3 - h_{3m}\right), s_{3m}\big]\,. \tag{12}$$

Assuming thermal equilibrium, steady state and perfect separation (no liquid with saturated vapor stream and no vapor with saturated liquid stream)

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and applying mass balance in separator, the entrainment ratio for iteration should satisfy the following equation:

$$\mu = \frac{\dot{m}_{sn}}{\dot{m}_{mn}} = \frac{1 - x_3}{x_3} \,. \tag{13}$$

Due to negligible pressure drop in the suction nozzle, the process is much less irreversible [18] and hence the isentropic efficiency of suction nozzle has been taken as constant value of 85% [12]. Whereas flows in the motive nozzle, mixing section and diffuser are highly irreversible as discussed earlier [18] and the efficiencies are highly dependent on the fluid properties and ejector geometric parameters. Hence, the fluid pressure ratio and ejector area ratio dependent correlations, developed based on experimental data of R600a [18], have been used, which are given by

$$\eta_{mn} = -0.01615 \left(\frac{p_1}{p_2}\right)^2 + 0.06925 \left(\frac{p_1}{p_2}\right) - \frac{1.32811}{\phi} + \frac{1.40543}{\sqrt{\phi}} + 0.29577 \,, \tag{14}$$

$$\begin{aligned} \eta_m = {} & -0.50685 \left(\frac{p_1}{p_2}\right)^4 + 5.91466 \left(\frac{p_1}{p_2}\right)^3 - 25.43486 \left(\frac{p_1}{p_2}\right)^2 + 47.90428 \frac{p_1}{p_2} \\ & + \frac{12.50076}{\phi^2} - \frac{17.06804}{\phi^{1.5}} + \frac{7.52924}{\phi} - \frac{1.29759}{\sqrt{\phi}} - 32.60447 \,, \end{aligned} \tag{15}$$

$$\eta_d = 0.00482 \left(\frac{p_1}{p_2}\right)^2 - 0.06947 \frac{p_1}{p_2} + \frac{0.08833}{\phi} - \frac{0.04263}{\sqrt{\phi}} + 0.85588 \,. \tag{16}$$

Now, compressor exit enthalpy can be found by

$$h_5 = h_4 + \frac{h\left(p_c, s_4\right) - h_4}{\eta_{co}} \,, \tag{17}$$

where the compressor isentropic efficiency has been calculated by the following empirical relation, valid for twin screw compressors with pressure ratio up to 20 [19]

$$\eta_{co} = 0.874 - 0.0134 \frac{p_c}{p_e} \,. \tag{18}$$

Now, the pressure lift ratio can be calculated by the following equation

$$PLR = \frac{p_3}{p_2} \,. \tag{19}$$

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Volumetric capacity of refrigerant (to decide about the size of compressor) is given by

$$C = (h_2 - h_7)\frac{\mu}{v_4}\,. \tag{20}$$

Compressor work and refrigeration effects are given by, respectively,

$$w_{co} = \frac{(h_5 - h_4)}{1+\mu}\,, \tag{21}$$

and

$$q_e = (h_2 - h_7)\frac{\mu}{1+\mu}\,. \tag{22}$$

Coefficient of performance with ejector expansion technology is given by:

$$COP = \frac{q_e}{w_{co}}\,. \tag{23}$$

Simulation was done on the commercial Engineering Equation Solver (EES) software [20] based on the theoretical model of the ejector expansion vapor compression cycle presented above. Build-in property functions have been used for thermophysical properties of all refrigerants. For given evaporator and condenser temperatures, and secondary nozzle pressure drop, the similar algorithm presented by Sarkar [12] has been used, except the mixing section, where, proper iteration is used to solve mass, momentum, and energy relations simultaneously. Experimental data based correlations have been used for component efficiencies. Total mass flow rate of working fluid has been taken as 1 kg/s. It can be noted that the area ratio can be varied monotonically by changing the suction nozzle pressure drop. Coefficient of performance improvement with respect to basic cycle has been also calculated for all refrigerants.

3 Results and discussion

Previous studies (e.g. Sarkar [12]) showed that the COP of ejector expansion system increases initially and then decreases with the increase in ejector area ratio (decrease in secondary nozzle pressure drop) and gives some maximum value, which is due to the fact that the pressure lift ratio is maximum and yields minimum pressure ratio as well as minimum work done through the compressor. Hence, there exists an optimum area ratio yielding maximum cooling COP as well as COP improvement. The knowledge of optimum parameter will be useful to proper adjustment of primary and

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secondary nozzle areas in system operation, to get maximum performance. Hence, the performances of R134a and it alternatives in ejector-expansion refrigeration cycle are compared in this study based on the optimum ejector area ratio corresponding to the maximum cooling COP. Optimization has been done in EES [20]. For given component efficiencies, the performance parameters of ejector-expansion cycle at optimum condition are mainly dependent on evaporator and condenser temperatures.

The present numerical simulation code has been validated with previous results by Sarkar [12] available on open literature. However, the constant pressure mixing was assumed by Sarkar [12], which can be easily attained by assuming 100% mixing efficiency. For 40 °C condenser temperature and 5 °C evaporator temperature, the comparisons of various performance parameters (area ratio, entrainment ratio, pressure lift ratio, COP and COP improvement) using R290 and R600a are tabulated in Tab. 2 for mixing efficiency of 100%. As shown, the values are well matched with literature data with the maximum deviation of 5%. Another comparison for R1234yf

Table 2: Validation of present simulation code with Sarkar [12] results.

Refrigerant	Properties	Previous study	Present study	Deviation (%)
R290	ϕ	6.84	7.118	4
	μ	0.766	0.766	0
	PLR	1.0934	1.09	0.3
	COP	6.097	6.035	1
	ΔCOP (%)	12.48	11.81	5.3
R600a	ϕ	7.53	7.142	5.1
	μ	0.779	0.778	0.12
	PLR	1.0887	1.087	0.17
	COP	6.207	6.133	1.19
	ΔCOP (%)	10.19	9.843	3.4

with Li *et al.* [14] shows that for component efficiency $\eta_m = 95\%$ (others are same), the deviation maximum COP (5.979 in present study and 5.94 in previous study [14]) is about 1%. The code has been also validated with experimental data for R134a [21]. For the condenser temperature at 53.28 °C, and evaporator temperature at 8.67 °C with 12 °C superheat, and 79% compressor isentropic efficiency (calculated based on test data), the entrainment ratio (0.64 experimental and 0.671 predicted) and area ratio

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(5.35 experimental and 5.93 predicted) are deviated by 4.7% and 9.8%, respectively. Hence, the validity of the presented mathematical model is confirmed.

To investigate the characteristic of ejector expansion refrigeration technology with selected nine refrigerants (R134a and it alternatives: R152a, R1234yf, R600a, R600, R290, R161, R32, and propylene), the evaporation and condensation temperatures have been taken as 5 °C and 40 °C, respectively, typically used for air-conditioning application. Various performance parameters corresponding to the optimum condition of individual refrigerants have been compared and effects of normal boiling point and critical temperatures were studied to illustrate the various behavioral trends.

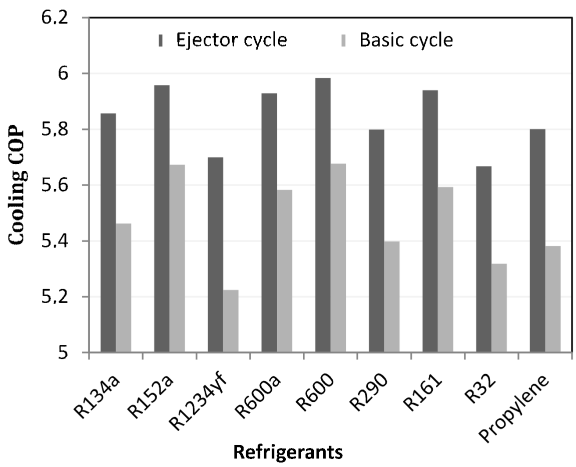

Figure 3: Comparison of maximum cooling COP with and without ejector.

Maximum cooling COP with and without ejector expansion technology with all nine refrigerants is shown in Fig. 3. Coefficient of performance is maximum with R600 followed by R152a and R161, but problem associated with these refrigerants is that, they are flammable in nature. The R1234yf could be good replacement of R134a however its COP is 4.3% lower in simple cycle. But difference in COP reduces after modification, and after modification COP of R1234yf is only 2.6% lower than R134a. Hence, the present study shows that R600 may be a best alternative to R134a (R600 yields about 2% higher COP than R134a) in air conditioning application for low charge system (flammability effect is less dangerous for low charge).

In Fig 4, it is clear that optimum area ratio is different for all differ-

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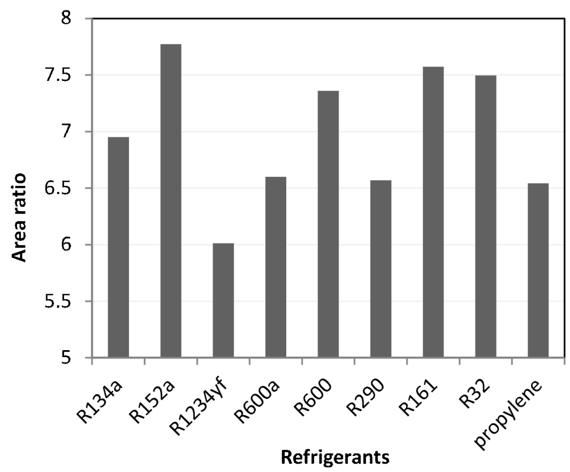

Figure 4: Comparison of optimum ejector area ratio.

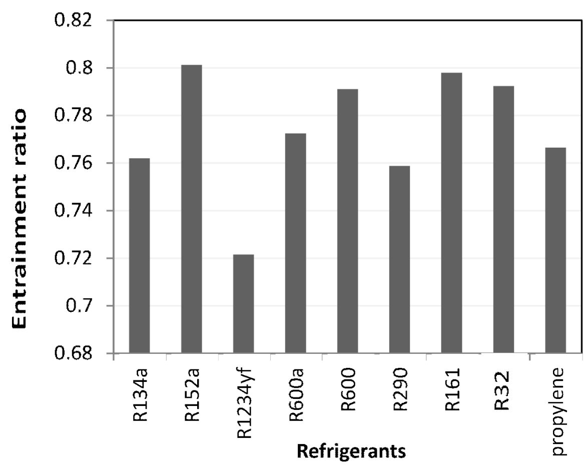

Figure 5: Comparison of optimum entrainment ratio of ejector.

ent refrigerants. It is due to unequal nozzle exit fluid velocity and other properties. Optimum area ratio is found maximum for R152a followed by R161 and R32 (7.773 for R152a, 7.573 for R161, and 7.497 for R32) whereas minimum is obtained for R1234yf (for R1234yf, it is 6.013). Figure 5 shows values of optimum entrainment ratio for all nine refrigerants. Entrainment ratio is the amount of refrigerant which is entrained and flows inside the evaporator by 1 kg of motive refrigerant flow. For each refrigerant for given operating condition and ejector efficiency optimum entrainment ratio occurs for which COP reaches maximum. Here it is important to note that if the entrainment ratio is less than optimum value, due to high vapor content most of the refrigerant from separator flows through compressor and only a small amount of refrigerant flows though evaporator. Hence COP decreases and compressor size increases. If entrainment ratio is more than an optimum value vapor content reduce and hence pressure lift ratio reduces. This increases compressor work and as a result COP reduces. It is clear from Fig. 5, that R152a has a maximum optimum entrainment ratio (entrainment ratio of R152a is 0.8013). Therefore compressor work is reduced in significant amount in comparison to R152a in ejector expansion technology. Refrigerant R1234yf has lowest optimum entrainment ratio (entrainment ratio of R1234yf is (0.7216) which leads to increase of the compressor size and work.

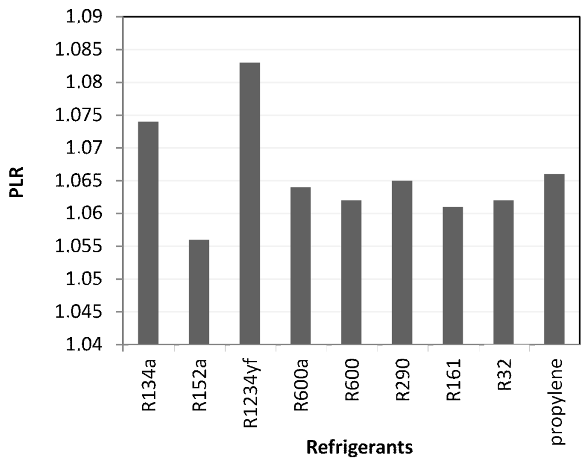

Figure 6: Comparison of pressure lift ratio of ejector.

Figure 6 shows the optimum value for pressure lift ratio (PLR). If pressure lift ratio is high then it increases the suction pressure of compressor and reduces the compressor work done. Hence selection of refrigerant should be done in such a way that its value of optimum pressure lift ratio is as high as possible. The maximum value of pressure lift ratio is noted in R1234yf followed by R134a (PLR for R1234yf is 1.083 and for R134a 1.074). R152a has a minimum pressure lift ratio followed by R161 (PLR for R152a is 1.056

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and for R161 is 1.061). COP improvement with all nine refrigerants is shown in Fig. 7. As discussed, higher pressure lift ratio reduces compressor work and increases COP; hence, maximum COP improvement is noted in R1234yf (10.1%) followed by propylene (8.745%) and R290 (8.417%).

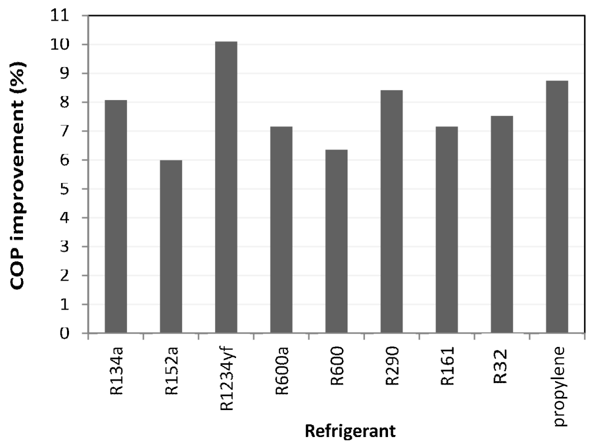

Figure 7: Comparison of COP improvement by using ejector expansion .

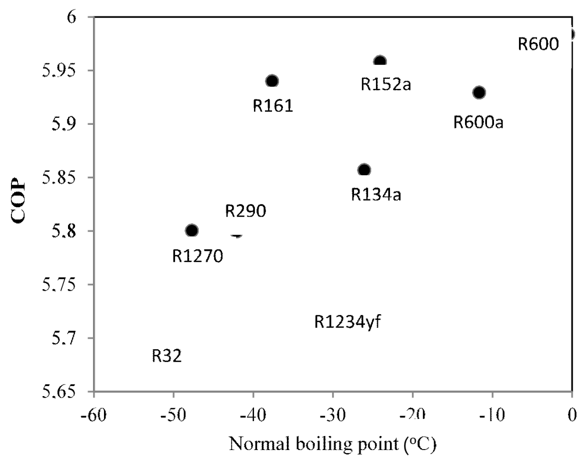

Figure 8: Variation of cooling COP with normal boiling point.

Figure 8 shows variation in COP with respect to normal boiling point of selected refrigerants. It is clear from the figure that there is no direct relation between normal boiling point and COP, but it affects the COP.

If normal boiling point increases the latent heat of vaporization also increases and a result the refrigerating effect increases with the increasing pressure ratio which subsequently increases the compressor work done. So depending upon dominant factor COP may increase or decrease. Figure 9 shows variation in volumetric refrigeration capacity with the normal boiling point. As the normal boiling point increases the volumetric capacity decreases which is quite obvious especially because high boiling point refrigerant has a low suction pressure and high specific volume at the inlet to the compressor.

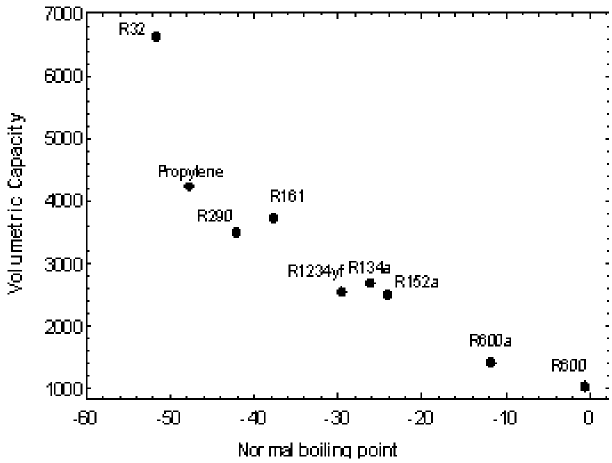

Figure 9: Variation of volumetric capacity with normal boiling point.

From Fig. 10 it is clear that critical temperature of the refrigerant should be high or reduced evaporation temperature should be low for high coefficient of performance. That is due to the fact that the refrigerant with higher critical temperature has a higher latent heat of evaporation and hence higher refrigerating effect. Figure 11 shows variation of volumetric refrigeration capacity with reduced evaporation temperature, which does not give any clear trend. However, as critical temperature increases, the volumetric capacity decreases. From Figs. 8 to 11, it is clear that R600 has a maximum COP but minimum volumetric capacity and R32 has a maximum volumetric capacity but minimum COP; both these condition are undesirable. Selection of refrigerant based of normal boiling point and critical point requires a trade-off between COP and volumetric capacity.

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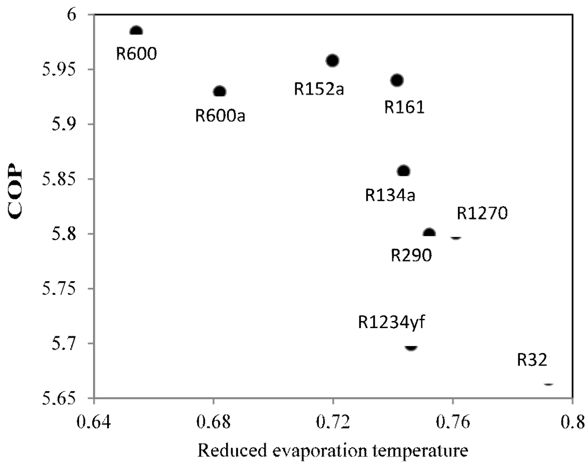

Figure 10: Variation of cooling COP with reduced evaporator temperature.

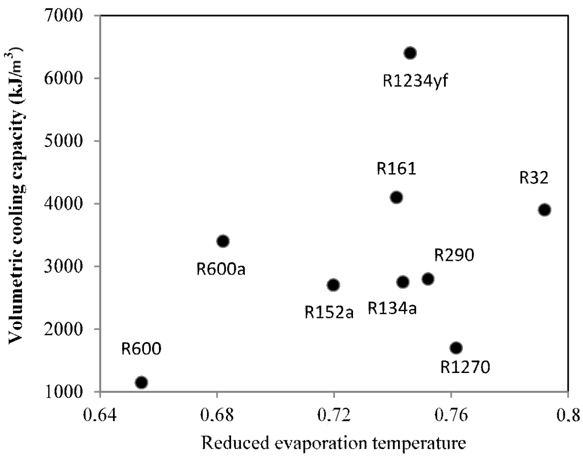

Figure 11: Variation of volumetric capacity with reduced evaporator temperature.

4 Conclusions

For air conditioning purpose (40 °C condensing temperature and 5 °C evaporating temperature), assessment of R134a and its possible alternatives (R152a, R1234yf, R600a, R600, R290, R161, R32, and propylene) in ejector expansion vapor compression system is presented in terms of COP, improvement in COP, volumetric refrigeration capacity, area ratio, entrain-

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ment ratio and pressure lift ratio. Variations of COP and volumetric capacity with normal boiling point and critical temperature are also presented in the study. Results show that R600 has a maximum COP followed by R152a; however, these refrigerants have low volumetric capacity compared to other refrigerants and another disadvantage with these refrigerants is they are flammable in nature. Refrigerant R32 has maximum volumetric capacity, but its COP is low compared to other refrigerants. R32 is also a high global warming potential refrigerant (GWP of R32 is 650) which is not suitable from the environmental point of view. R1234yf has noted maximum improvement in coefficient of performance among all other refrigerants with ejector expansion technology. Normal boiling point and critical temperature of R1234yf are very close to R134a. R1234yf has a very low GWP (GWP of R1234yf is 4 and for R134a, it is 1300) compared to R134a. Hence R1234yf could be the best replacement of R134a from environmental point of view with only a small performance compromise. Another advantage with R1234yf is that it is mildly flammable, which means it is safer to use compared to other alternative refrigerants. Performances of R290 and propylene are also comparable with R134a but they can be used only by taking a lot of safety measures because they are highly flammable (A3 category (Tab. 1)) in nature. R161 is superior to R134a in terms of both COP and volumetric cooling capacity although it is more flammable and hence can be recommended for a low charge system.

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