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System performance of R-1234yf refrigerant in air-conditioning and heat pump system – An overview of current status

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HIGHLIGHTS

- An overview of the system performance of R-1234yf in association with R-134a is carried out.
- The COP and heat capacity of R-134a system may suffer from direct drop-in via R-1234yf.
- The condenser performance for R-1234yf is appreciably lower than that of R-134a.
- The deterioration is around 0–27% depending on the operational conditions.
- By adding IHX, ejector, expander, or adjustment of the TXV, the deterioration can be relieved.

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ABSTRACT

In this study, an overview of the system performance of R-1234yf in association with R-134a is carried out. Based on the existing researches, it is found that the COP and heat capacity of R-134a system may suffer from direct drop-in replacement of R-1234yf. The deterioration is around 0–27% depending on the operational conditions. With the introduction of internal heat exchanger, ejector, expander, or adjustment of the thermal expansion valve, the deterioration can be relieved, and a comparable performance becomes likely. For the heat transfer performance in the evaporator, R-1234yf is almost comparable with that of R-134a. However, the performance in the condenser is inferior to R-134a. The phenomenon may be quite severe for a water cooled condenser since the dominant thermal resistance may fall in the refrigerant side. The volumetric efficiency of R-1234yf system is slightly lower than that of R-134a due to higher frictional drop of R-1234yf. For the same thermal expansion valve for controlling the suction superheat, it appears that higher suction superheat may occur for R-1234yf refrigerant. Hence further adjustment of spring in the valve is required for soft optimization.

1. Introduction

The European Union had initiated the F-gas Regulation to phase out gases with fluorine in 2006. In such regulation, the widely used HFC refrigerant, R-134a, are not recommended in new type vehicles by year 2011 until its total ban in all vehicles by 2017 [1]. Only refrigerants with a GWP of less than 150 will be permitted in the mobile air-conditioning market. Currently, only CO_2, R-152a, and R-1234yf meet the demand and are the potential candidates. However, the high system pressure and comparatively low efficiency of CO_2 and the flammability of R-152a make R-1234yf as the front runner of this race.

The ODP of R-1234yf is zero and its GWP is as low as 4 [2]. R-1234yf also features and has a very low toxicity compared with R-134a and has a mild flammability as listed in the "A2" classification [3]. Table 1 depicts the critical property and molecular weight for R-134a and R-1234yf. It shows that the critical pressure for R-134a is about 20% higher than that of R-1234yf. The thermophysical properties, cycle performance, and heat transfer performance of R-1234yf are the key parameters to assess the feasibility of using this new refrigerant in air conditioners. The thermophysical properties of refrigerant mixture were similar to those of R-134a (Arakawa et al., [2]), a more detailed comparisons of the relevant thermodynamic property and the transport property, in terms of ratios of R-1234/R-134a, are depicted in Fig. 1 based on the calculations of REFPROP 9.0 [4]. For the difference of thermodynamic properties depicted in Fig. 1(a), the specific heat for both phases and vapour pressure between R-134a and R-1234yf are normally within the 10%. The very similar vapour pressure of R-134a and R-1234yf suggests the refrigeration cycle can operate at similar conditions,

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Nomenclature

A	surface area (m^2)
d_o	outer tube diameter (m)
C_p	specific heat (J kg^{-1} K^{-1})
COP	coefficient of performance, dimensionless
f	compressor frequency, Hz
g	gravitational acceleration (m^2 s^{-1})
h	heat transfer coefficient (W m^{-2} K^{-1})
i	specific enthalpy (kJ kg^{-1})
IHX	internal heat exchanger
k	thermal conductivity (W m^{-1} K^{-1})
$\dot{m}$	mass flowrate (kg s^{-1})
P	pressure (Pa)
P_s	saturation pressure (Pa)
Pr	Prandtl number, dimensionless
P^*	reduced pressure
$\dot{Q}$	heat transfer rate (W)
T	temperature (°C)
T_o	evaporation temperature (°C)
T_s	saturation temperature (°C)
TXV	thermal expansion valve
x	vapour quality, dimensionless
Z	$= (1-x/x)^{0.8}(P^*)^{0.4}$

Greek letters

μ	viscosity (kg m^{-1} s^{-1})
σ	surface tension (N m^{-1})
ρ	density (kg m^{-3})
η_v	volumetric efficiency, dimensionless

Subscripts

air	inlet air at condenser
c	condensation
con	condenser
eva	evaporator
g	vapour phase
fg	difference between liquid phase and vapour phase
i	inside
f	liquid phase
o	outside
s	saturated
w	wall surface

Superscript

*	ratio (R-1234yf/R-134a)

thereby offering an opportunity as a drop-in solution for the current mobile air conditioners. Yet the liquid density for R-1234yf is about 10% lower than that of R-134a while the vapour density of R-1234yf is moderately about 15–20% higher than that of R-134a. The higher vapour density of R-1234yf implicates a larger mass flowrate provided by the compressor. However the corresponding latent heat of R-1234yf is about the same order of reduction as that of vapour density. Therefore it may imply a similar capacity of the two systems ($\dot{Q} \sim \dot{m} i_{fg}$).

In the meantime, as shown in Fig. 1(b), the difference in the transport properties for vapour phase (k_g, μ_g, Pr_g) and liquid Pr_f are generally within ±10%. The major difference occurs in liquid thermal conductivity (k_f) and liquid viscosity (μ_g). Both properties are approximately 20% lower than those of R-134a. The lower liquid thermal conductivity suggests a higher heat transfer barrier in the liquid phase. As a consequence, it may bring out a possible severe deterioration of heat transfer performance in condenser. This is because liquid film may cover on the heat transfer surface during condensation process and result in substantial thermal resistance. Yet it is applicable either for in-tube condensation or shell-side condensation. A recent review on the heat transfer performance by Wang [5] had clearly addressed the possible deficiency of condensation performance for R-1234yf. The major reason may be associated with its much lower liquid thermal conductivity. In this regard, re-design of the condenser may be one of the key issues in the R-1234yf system.

In addition to heat transfer performance, the system performance of R-1234yf relative to R-134a is the detrimental factor judging the suitability of R-1234yf as the possible drop-in solution. Therefore there had been some researches [6–15] on this topics for the past several years as tabulated in Table 2. Basically, almost all the studies reported a slight or moderate drop in cooling capacity and COP (coefficient of performance). The only exception is reference [6] who reported an almost identical system performance amid R-1234yf and R-134a. The system performance using R-1234yf and R-134a is contingent on many situations, including heat transfer performance in condenser and evaporator, refrigerant charge, expansion device, addition of expander/ejector/internal heat exchanger, compressor, and the like. Hence, it is the objective of this review to summarize recent efforts concerning the system performance between R-134a and R-1234yf.

Table 1
Fundamental constants of R-1234yf.

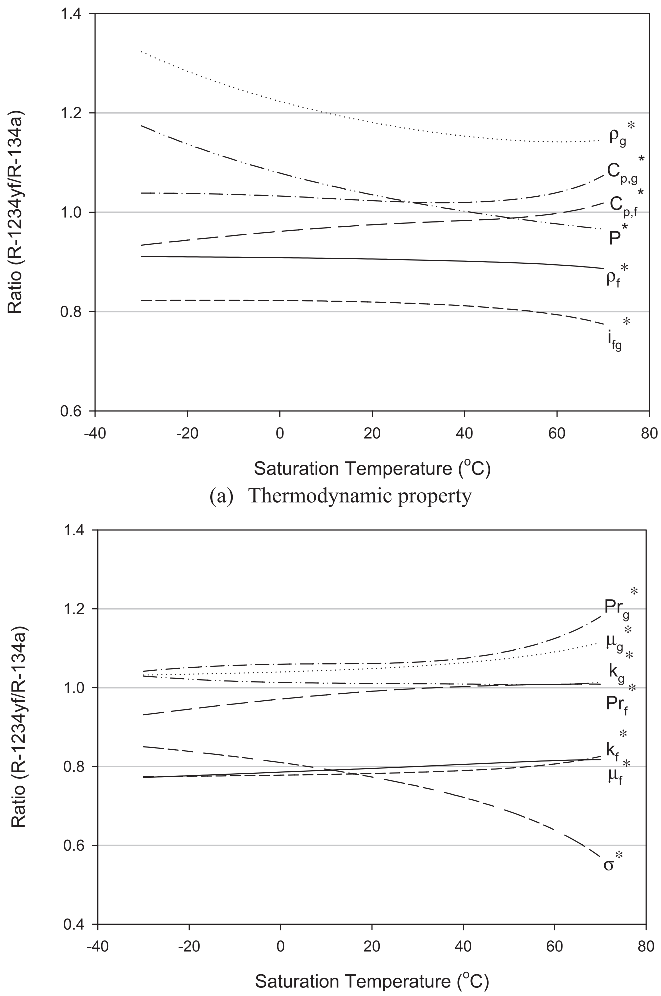
	Molecular weight	Critical temperature	Critical pressure
R-134a	102 g mol^{-1}	374.13 K	4.07 MPa
R-1234yf	114.042 g mol^{-1}	367.85 K	3.382 MPa

2. Heat transfer performance for R-1234yf

Wang [4] recently had provided a thorough overview about the recent efforts on the two-phase heat transfer characteristics of R-1234yf, including in-tube convective boiling, in-tube condensation, external condensation, nucleate boiling, and CHF. A short summary of his review and some latest results are presented subsequently. Further details and discussion can be found in his review.

2.1. Boiling heat transfer performance

The pool boiling heat transfer and convective boiling heat transfer performance of R-1234yf is approximately the same as that of R-134a. This can be made clear from the test results from Moreno et al. [16], Park and Jung [17], Saitoh et al. [18] and Lu et al. [19]. Thome [20] indicated that three mechanisms, namely bubble agitation, vapour–liquid change phenomenon, and evaporation characterized the mechanisms of the nucleate boiling heat transfer. From Table 1 the R-1234yf has a higher reduced pressure at the same saturation temperature. Note that the reduced pressure represents the ratio of the operational pressure of the associated refrigerant to its critical pressure. This is because its critical pressure is about 17% lower than of R-134a. For instance at a saturation temperature of 40 °C, the corresponding reduced pressure for R-1234yf is approximately 20% higher than that of R-134a, consequent resulting in a larger activation sites and improving the heat transfer performance accordingly. On the downside, the smaller



(a) Thermodynamic property

(b) Transport property

Fig. 1. Ratio of thermodynamic property and transport property between R-1234yf and R-134a.

bubble departure diameter ($\sim$ $(\sigma/g(\rho_f-\rho_g))^{0.5}$) of R-1234yf implies a lower bubble agitation, and a smaller vapour–liquid change contribution which offset the positive contribution from the higher reduced pressure. Hence similar nucleate boiling heat transfer prevails for R-134a and R-1234yf [16,17]. Moreover, the two-phase flow pattern for R-1234yf is about the same as that of R-134a (Padilla et al. [21]). In this regard, an almost identical two-phase convective boiling heat transfer coefficient amid R-134a and R-1234yf were reported on the published results [18,19].

2.2. Condensation heat transfer performance

From the well accepted Nusselt's equation applicable external condensation of a smooth tube:

$$h_c = 0.728\left[\frac{\rho_f\left(\rho_f-\rho_g\right)g i_{fg} k_f^3}{\mu_f\left(T_s-T_w\right)d_o}\right]^{1/4} \tag{1}$$

From Eq. (1) it appears that the liquid density, latent heat, liquid viscosity, and liquid thermal conductivity play essential role of the condensation. Note that the liquid thermal conductivity, is especially pivot due to its strong dependence ($h_c \sim k_f^{3/4}$). Unfortunately, three out of four of the foregoing properties of R-1234yf, ρ_f, i_{fg}, and k_f pose negative contribution to the heat transfer coefficient when compared to that of R-134a. Only liquid viscosity of R-1234yf casts positive influence of condensation heat transfer coefficient. In this sense, it is expected that the corresponding condensation heat transfer coefficient is much lower as compared to that of R-134a. In Wang's calculation [5], the decline of condensation heat transfer coefficient can be as large as 40%. Similar results for the in-tube condensation are also encountered. This can be made clear from the well-known Shah correlation [22] where:

$$h_c = h_f\left(1+\frac{3.8}{Z}\right) \tag{2}$$

where

$$Z = \left(\frac{1-x}{x}\right)^{0.8}\left(P^*\right)^{0.4} \tag{3}$$

$$h_f = \frac{k_f}{d_i}0.023\left(\frac{G(1-x)d_i}{\mu_f}\right)^{0.8} Pr_f^{0.4} \tag{4}$$

From Eq. (2), it appears that the in-tube condensation heat transfer coefficient is roughly proportional to $k_f^{0.6}$. Hence some published results like Col et al. [23] and Longo and Zilio [24] had pointed out an appreciable lower condensation heat transfer coefficient of R-1234yf. The reduction in heat transfer coefficient ranges from 12% to 35% depends on the operation conditions and heat exchanger. The results are also associated with the lower liquid density of R-1234yf. As depicted in Fig. 1(a), R-1234yf reveals approximately l0% lower liquid density. For the same mass flux and vapor quality, the condensate liquid film of R-1234yf is slightly thicker than R-134a and it would impair the condensation heat transfer performance. In summary of the foregoing discussion, a re-design of condenser are quite imperative especially for liquid cooled system where the dominant thermal resistance is usually on the refrigerant side. However, in a typical air-cooled system, the air-side resistance normally surpasses 80% or even 90% of the total thermal resistance (Wang et al. [25] and Wang [26]. Therefore the appreciably drop of in-tube condensation heat transfer performance may only jeopardize the performance of the condenser slightly.

3. System performance of R-1234yf vs. R-134a

Table 2 depicts related studies upon system performance amid R-1234yf and R-134a. These studies were mainly associated with drop-in replacement using R-1234yf as the substitute in the conventional R-134a system. Basically, the system performance, in terms of COP and capacity, are slightly lower than those of R-134a system. Hence, some soft optimizations, such as internal heat exchanger, adjustment of thermal expansion valve, introduction of expander or ejector are also considered. Further details are discussed accordingly.

3.1. Difference of refrigerant charge amount

The refrigerant charge amount was mainly affected by the system inner volume and refrigerant liquid density. For a drop-in replacement, the system inner volume was fixed for both R-134a and R-1234yf. Hence the refrigerant charge amount is mainly associated with the liquid density. In the meantime, the

Table 2
System performance in association with R-1234yf.

Authors	System	Working fluids	Test conditions	Conclusions
Motta et al. [6]	Compressor: reciprocating Condenser: fin-and-tube Evaporator: fin-and-tube Expansion device: needle valve System: vending machine	1.R-134a 2.R-1234yf 3.HFO-1234ze Cooling type: Air-cooled	Efficiency Test: 1.Outdoor: 32.2 °C, RH: 65% 2. Indoor: 2 °C . Capacity Test: 1. Outdoor: 40.5 °C, RH: 75% 2. indoor: 2 °C .	1. R-1234yf shows 1–2% lower COP but reveals 2–5% higher capacity as compared to R-134a. 2. R-1234yf shows 7–9% lower COP but reveals 11–13% higher capacity as compared to R-134a .
Lee and Jung [7]	Capacity: 3.5 kW Evaporator: tube-in-tube Condenser: tube-in-tube System: mobile air conditioner (MACs)	1. R-134a 2. R-1234yf Cooling type: Liquid-cooled	1. Evaporator/condenser: T_{sat} for HFC134a: 7 °C/45 °C and −7 °C/41 °C. 2. Test condition during summer and winter. The subcooling and superheat at the exits of the condenser and evaporator were fixed to be 5 °C	1. The COP of R-1234yf is 0.8–2.7% lower than R-134a. 2. The capacity of R-1234yf is up to 4.0% lower than that of R-134a. 3. The compressor discharge temperature of R-1234yf is 6.4°C–6.7 °C lower than R-134a. 4. The amount of refrigerant charge of R-1234yf is 10–11% lower than that of R-134a.
Zhao et al. [8]	Compressor: Variable displacement Condenser: microchannel parallel flow condenser Evaporator: laminated plate Expansion device: thermal expansion valve System: mobile air conditioner	1. R-134a 2. R-1234yf Cooling type: Air-cooled	Low load: Outdoor: 27 °C Indoor: 25 °C, RH = 40% Medium Load: Outdoor: 37 °C Indoor: 35 °C, RH = 40% High Load: Outdoor: 45 °C Indoor: 43 °C, RH = 40%	1. The optimum refrigerant charge amount of R-1234yf was approximately 95% of R-134a for the same system. 2. In all working conditions, the cooling capacity of the R-134a system was 12.4% larger than R-1234yf system at most, and the COP of the R-134a system was only 9% larger at most.
Jarall [9]	Refrigeration unit: 550 W Compressor: hermetic rotary type. Expansion device: thermal expansion valve System: Refrigeration unit	1. R-134a 2. R-1234yf Cooling type: Liquid-cooled	Condenser temperatures: 40 °C and 45 °C Evaporation temperature: −5~15 °C	1. The capacity and COP for R-1234yf relative to R-134a are decreased by 3.4–13.7% and 0.35–11.9%, respectively. 2. At similar conditions R-1234yf has lower pressure ratio than R134a which reduces the compressor power consumption.
Zilio et al. [10]	Compressor: a variable volume swash-plate type Condenser: minichannel parallel flow Evaporator: minichannel design Expansion device: thermal expansion valve System: automotive air conditioning system	1. R-134a. 2. R-1234yf Cooling type: Air-cooled	1. Evaporator air inlet: 35 °C & 40% RH, 25 °C & 80% RH, 15 °C & 80% RH 2. Evaporator air volumetric flowrate ($m^3 h^{-1}$): 400 ± 3%, 400 ± 3%, 400 ± 3% 3. Condenser air inlet: 35 °C, 25 °C, and 5 °C 4. Condenser air volumetric flowrate (m^3/h): 1580 ± 3%,1580 ± 3%, 1580 ± 3% 5. Compressor speed (rpm): 900–4000	1. For a given cooling capacity, R-1234yf systems present lower performance than the baseline R134a. 2. A numerical simulations were used to investigate the effects of" major" system modifications for improving condenser and/or evaporator. 2. The numerical simulations show that enhancing the face area of the condenser by 20%, evaporator by 10%, and using the overridden compressor, the R-1234yf system showed higher COP values than the baseline R-134a for equal cooling capacities
Chen et al. [11]	Compressor: (1). Scroll type (2). Capacity: 5000 W (3). Mass: 6–6.2/kg (4). Speed: 1000–7200 System: Mobile air-conditioning (MAC) in a hybrid electric vehicle	1. R-134a 2. R-1234yf Cooling type: Air-cooled	–	1. The "drop-in" of HFO-234yf in an R-134a MAC system can result in a slightly decreased system performance. 2. the cooling capacity and COP of the R-134a system could be 12.4% and 9% higher than those of the R-1234yf system, respectively,
Navarro-Esbri et al. [12]	Compressor: reciprocating type Condenser: shell-and-tube Evaporator: shell-and-tube Expansion device: thermal expansion valves System: Self-built test bench	1. R-134a 2. R-1234yf Cooling type: Liquid-cooled	T_{con}: 313.15–333.15 (K) T_{eva}: 265.65–280.15 (K) With and Without IHX Superheating degree: 5–10 (K) Compressor drive frequency: 30–50(HZ)	1. The cooling capacity of R-1234yf used as a drop-in replacement in a R134a refrigerant facility is about 9% lower than that presented by R134a in the test range. 2. The COP using R-1234yf are about 5%~30% lower than those obtained with R134a. 3. The volumetric efficiency of R-1234yf is about 5% lower in comparison with that obtained with R134a.
Cho et al. [13]	Compressor: Swash plate type $\dot{m}$ = 8.0 kg h^{-1} Expansion valve:(block type, 5.3 kW) System: automotive air conditioning	1. R-134a 2. R-1234yf Cooling type: air-cooled	1. Compressor speed (rpm): 200/1800/2500 2. Indoor temperature (DB/WB): 27/19.5 (°C) 3. Outdoor temperature (DB/WB): 35/24(°C)	1. The R-1234yf system showed a lower compressor power consumption and a lower cooling capacity by up to 4% and 7% in comparison with the R-134a system. 2. The cooling capacity and COP of the R-1234yf system without the internal heat exchanger decreased up to 7% and 4.5% compared to those of R-134a system. 3. Sufficient cooling capacity is ensured under various operation conditions and the system COP is improved by up to 4.6% by installing the internal heat exchanger into the R1234yf refrigeration system.
Navarro et al. [14]	compressor: open piston type System: automotive air conditioning	1. R-134a 2. R-1234yf 3. R-290	Evaporation temperature: −15 −15 °C Condensing temperature: 40 −65 °C	1. R-290 has shown a significant improvement in compressor and volumetric efficiencies and the heat losses are considerably lower than for the other two refrigerants. 2. R-1234yf shows a reduction of 10–15% in the system cooling capacity as compared to R-134a.

(continued on next page)

Table 2 (*continued*)

Authors	System	Working fluids	Test conditions	Conclusions
Qi et al. [15]	Evaporators types: (1). Laminated plate (2). Microchannel parallel flow (PF) type. System: Mobile air-conditioning	1. R-134a 2. R-1234yf Cooling type: air-cooled	1. Air T_{in}: 35/38/40 °C 2. RH: 40/60% 3. Pressure before expansion valve: 1.32–1.84 MPaG 4. Subcooling before expansion valve: 5–9 °C 5. P_{out} at evaporator: 0.2–0.25 MPaG 6. Superheat at evaporator outlet: 5 °C	1. R134a shows a better heat transfer and flow performance than that of R-1234yf in laminated plate evaporator. 2. Microchannel PF evaporator is a good option in R-134a drop-in replacement MAC system.

corresponding condensing pressure becomes higher whereas the liquid subcooling at the condenser outlet is decreased with the rise of refrigerant charge amount. Since the liquid density of R-1234yf was about 10% smaller; the optimum charge amount of R-1234yf was expected to be less than that of R134a. The determination criterion of refrigerant charge amount was that the refrigerant subcooling at the outlet of the condenser. The subcooling was evaluated based on the refrigerant temperature and pressure at condenser outlet. For a standard working condition, inlet air temperature at indoor unit were maintained at 27 °C, 50%, and 35 °C at the outdoor unit, Zhao et al. [8] reported that the optimum mass refrigerant charge of R-1234yf for maintaining an approximate subcooling is about 5% smaller than that of R-134a. With a relatively higher internal space of evaporator and condenser, Lee and Jung [7] reported a 10% lower R-1234yf refrigerant charge than R-134a.

3.2. Difference in compressor performance

Navarro-Esbrí et al. [13] used an open type reciprocating compressor, a shell-and-tube condenser, an internal tube-in-tube internal heat exchanger (IHX), a set of expansion valves, and a shell-and-tube evaporator to examine the drop-in performance of R-1234yf in association with R-134a. In Fig. 2 the influence of the compression ratio on the compressor volumetric efficiency using both refrigerants is presented. The volumetric efficiency, as expected, decreases with the rise of compression ratio. It also appeared that the compressor volumetric efficiency using R-1234yf is about 5% lower compared with that using R-134a. Furthermore, in this figure, one can observe that the dispersion obtained for the R-1234yf volumetric efficiency is larger than that of R-134a. From the foregoing discussion in Fig. 1, the required mass flowrate is generally higher for R-1234yf due to its much lower latent heat. This may lead to a rise of pressure drop and consequently Navarro-Esbrí et al. [13] attributed the lower volumetric efficiency to higher frictional pressure drops caused by R-1234yf. Fig. 3 presents the compressor power consumption using both refrigerants at different working conditions. In Fig. 3(a) the power consumption for R-1234yf is only marginal higher than that of R-134a for a condensing temperature of 333.15 K and is about 18–27% higher when the condensing temperature is raised to 313.15 K. The results suggests a minimum difference in the power consumption for higher condensing temperatures. This is because the refrigerant mass flowrate is low and the pressure drops are also low at elevated pressures. Notice that the critical temperature and critical pressure for R-1234yf is appreciably lower than that of R-134a. Hence the rise of condensing temperature cast even worse influence on the reduction of mass flowrate of R-1234yf. The results may also explain in part that the test data of Motta et al. [6] is the only one showing identical performance with R-134a during drop-in replacement of R-1234yf. Notice that the temperature difference from theirs is comparatively high and it results in a comparatively low mass flowrate. The effect of the superheating degree on the compressor power consumption is shown in Fig. 3(b). It appears that a decrease in the power consumption is encountered when the superheating increases from 5 K to 10 K. The influence of the compressor frequency on the power consumption subject to various evaporation temperature is seen in Fig. 3(c). Apparently, the difference between both refrigerants power consumption at 35 Hz is lower than that at 50 Hz. Again, this is due to the higher pressure drops using R-1234yf in comparison with those presented when using R-134a. Navarro et al. [14] used a similar test facility to examine the effect of lubricant between R-134a, R-1234yf, and R-290. The oil used for all tests was POE oil ISO 68. They reported that there are no significant differences between the refrigerants and in all cases the oil circulation ratio is lower than 3%. Yet the oil circulation ratio for low evaporation temperatures is higher than for the rest of the conditions.

3.3. Influence of evaporation temperature, internal heat exchangers

Navarro-Esbrí et al. [13] also investigated the variation of the COP with the operating parameters like evaporation temperature, internal heat exchanger, and condensing temperature as shown in Fig. 4. It is observed that the COP obtained using R-1234yf is about 5–27% lower than that observed when using R-134a with the operating pressures being changed as shown in Fig. 4(a). This difference in the COP using both refrigerants is lower for higher condensing temperatures, being about 8% for condensing

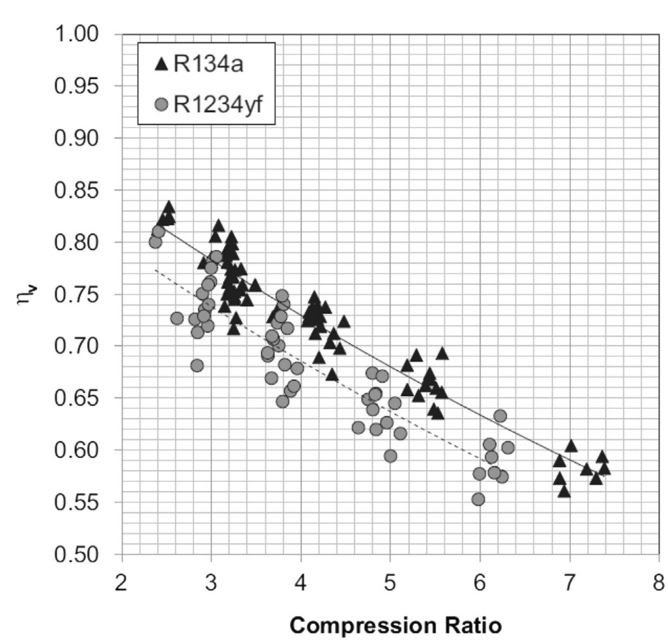


Fig. 2. Volumetric efficiency vs. compression ratio between R-1234yf and R-134a (from Navarro-Esbrí et al. [13]).

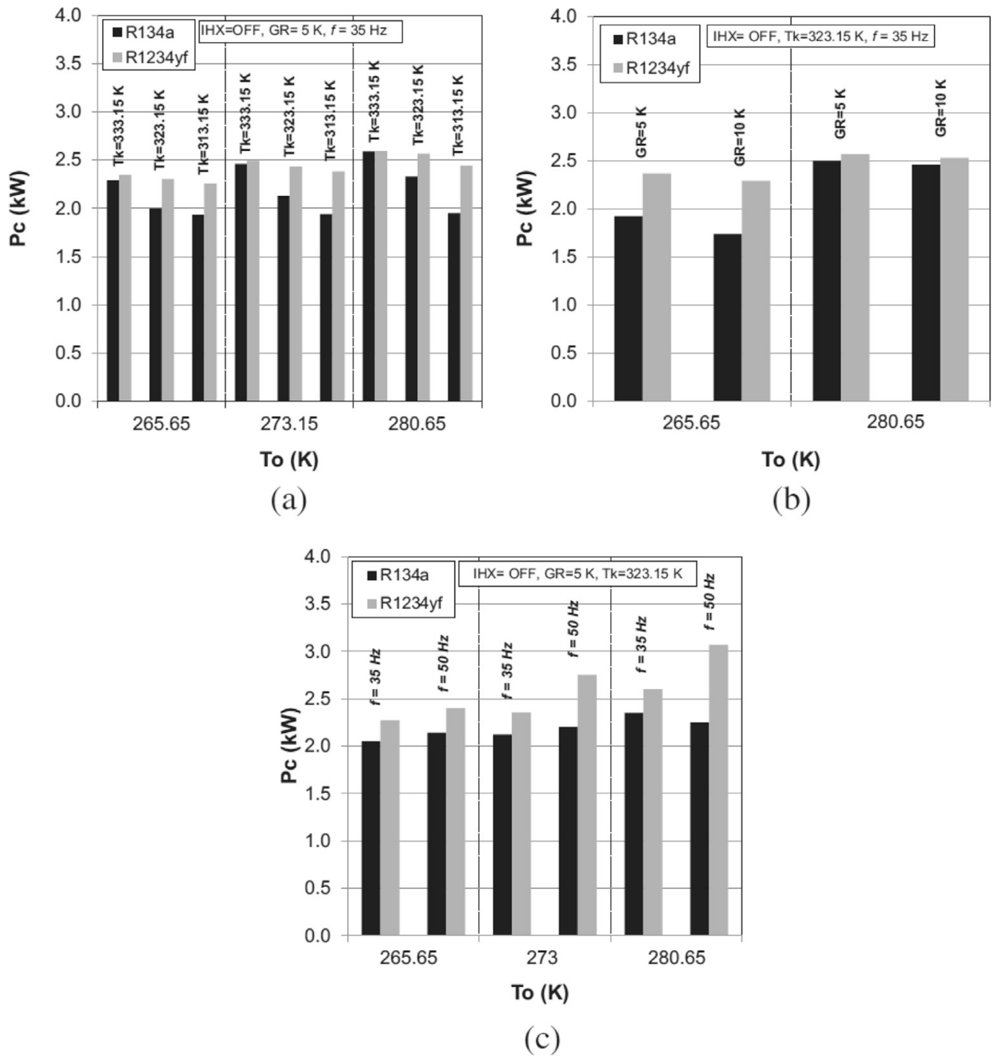


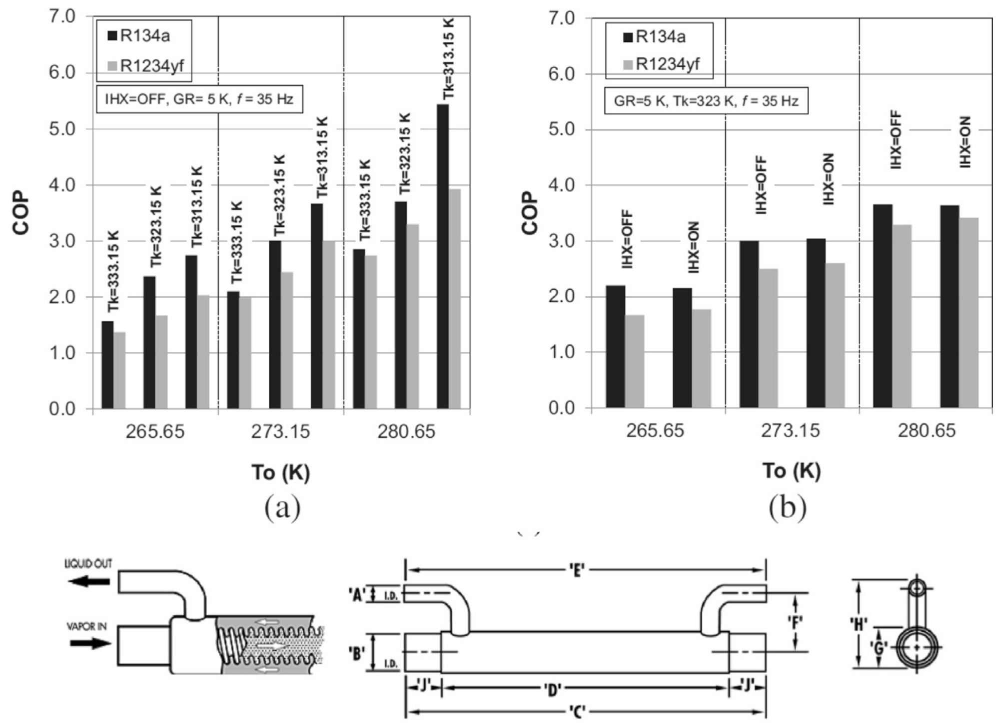
Fig. 3. Experimental COP variation vs. evaporation temperature To subject to influence of (a) condensing temperature; (b) superheating degree; and (c) compressor drive frequency (from Navarro-Esbrí et al. [13]).

temperatures of 333.15 K and about 25% for condensing temperatures of 313.15 K. The effect of condensing temperature is also associated with pressure drop (or the mass flowrate flowing along the system). For a compressor system, the mass flowrate is decreased with the rise of condensing temperature as aforementioned previously. On the other hand, it can be also seen that the IHX has a significant influence on the COP differences between both refrigerants (Fig. 4(b)). In fact, the COP for R-1234yf are 11–24% lower than those for R-134a when the IHX is not used but it becomes about 6–17% when the IHX is used. In a similar study also carried out by Navarro-Esbrí et al. [27] who had reported more details about the implementation of IHX between R-134a and R-1234yf. The schematic of the internal heat exchanger is shown in Fig. 4(c). They concluded that the introduction of the IHX for R-1234yf system produced an increase in the cooling capacity and COP, and it reveals comparable performance to the original R-134a system. The exploitation of an IHX would reduce the decrease in the cooling capacity and COP between 2 and 6%, almost compensating these reductions caused by using R-1234yf as a drop-in replacement for R-134a with an IHX for high compression ratios.

Cho et al. [12] conducted experiments concerning the influence of IHX in an automotive refrigeration systems with the refrigerants R-134a and R-1234yf. Performance test by using R-1234yf showed a lower power consumption (−4%) and a smaller cooling capacity (−7%) for using R-1234yf. In particular, the system performance between the R-1234yf and R-134a revealed a decline 7% of COP when IHX is not installed. However, with the introduction of IHX, the corresponding drop of COP is reduced to 2.9%. For optimization of the IHX configuration subject to R-1234yf, Seybold et al. [28] proposed several coaxial designs, and the optimized profiles of the IHX depends on the minimization of internal liquid volume, minimization of suction pressure drop, maximization of heat transfer, and limited liquid pressure drop.

3.4. Introduction of two-phase ejector and expander for performance improvement

Much of the recent work on two-phase ejectors has been focused on transcritical CO_2 cycles for its larger throttling loss and a lower cycle efficiency. Low-pressure working fluids like R-134a and R-1234yf had received less attention in the open literature since they are more difficult to successfully implement with the standard two-phase ejector cycle than CO_2 due to their lower work recovery potential. Lawrence and Elbel [29] developed an alternate two-phase ejector cycle, in which the pressure lift provided by the ejector was utilized in order to provide multiple evaporation



	A	B	C	D	E	F	G	H	I
L (mm)	15.9	34.9	361.9	279.4	361.9	60.3	41.3	90.5	41.3

(c)

Fig. 4. Experimental COP variation vs. To subject to (a) condensing temperature, (b) IHX, and (c) configuration of internal heat exchanger. ((a) & (b) are from Navarro-Esbrí et al. [13], and (c) is from Navarro et al. [27]).

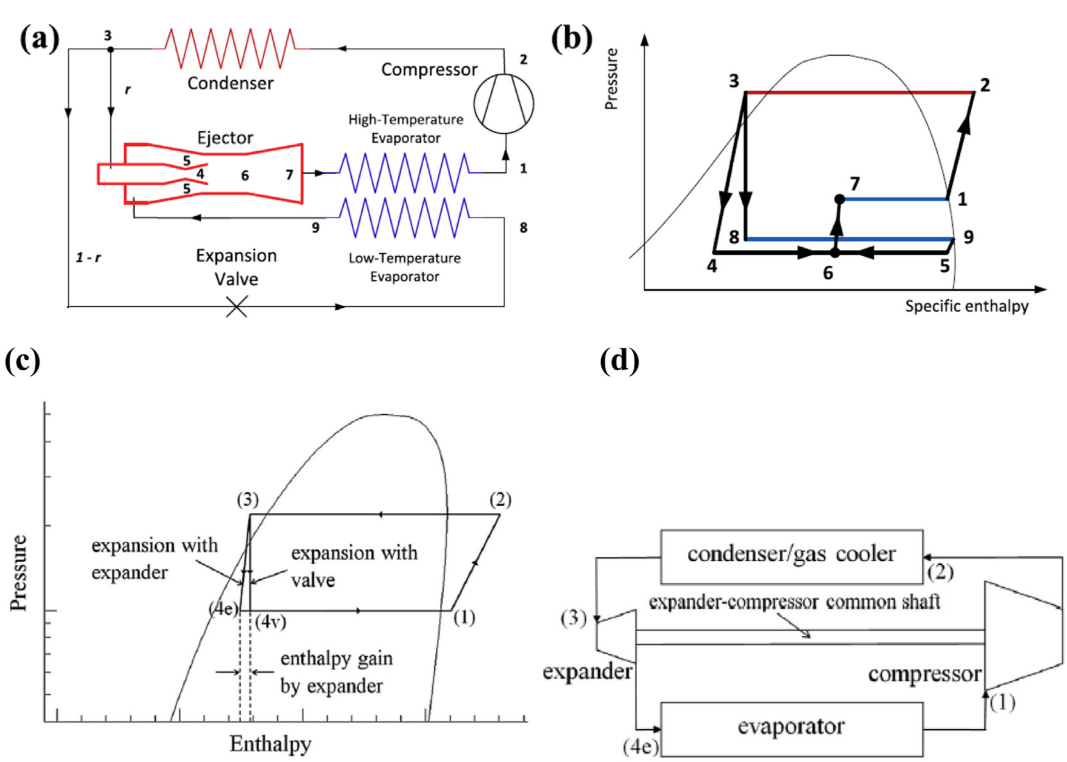


Fig. 5. (a) Layout diagram for ejector design and (b) pressure-specific enthalpy diagram of COS ejector cycle as proposed by Oshitani et al. [30]; and (c) pressure-specific enthalpy diagram for refrigeration system w/wo expander; and (d) schematic for refrigeration system with expander to recover work loss during expansion. (a) & (b) are from Lawrence and Elbel [29], and (c) and (d) are from Subiantoro and Ooi.[31].

temperatures. The base ejector cycle is based on the patent of Oshitani et al. [30] as shown in Fig. 5(a) and (b). The experimental results show that ejectors designed for low-pressure fluids were able to achieve similar but lower work recovery efficiencies compared to CO_2 ejectors. When compared to a two evaporation temperature expansion valve cycle, the ejector cycle showed maximum COP improvements of 12% with R-1234yf and 8% with R-134a.

Subiantoro and Ooi [31] performed an economic analysis of utilization of expanders in medium scale air-conditioners with several refrigerants (including R-1234yf). An expander improves the coefficient of performance (COP) of the system in two ways (Nickl et al., [32]): 1) by increasing the cooling capacity through performing a near-isentropic expansion, hence reducing the enthalpy of the refrigerant at the evaporator inlet (see Fig. 5(c)), and 2) by recovering the expansion energy, hence reducing the externally electrical power requirement of the compressor (see Fig. 5(d)). Their economic analysis showed that expanders are financially feasible to be installed into refrigeration systems, particularly for medium scale air conditioners. Expanders are especially attractive for CO_2 and R-404A systems with payback periods of less than 1 and 3 years, respectively. By increasing the expander efficiency from 30% to 60% in an R-1234yf system cuts the payback period from 5.3 to 3.4 years.

3.5. Performance difference in expansion valve

The superheat of an air-conditioning system is normally controlled by a thermal expansion valve. For a typical mobile air-conditioning, the superheat at the evaporator outlet is designed to fall between 5 °C and 10 °C. Zhao et al. [8] performed a drop-in test for R-1234yf in a typical mobile air-conditioning system originally filled with R-134a. Low ($T_{air} = 27$ °C), medium ($T_{air} = 37$ °C), and high load ($T_{air} = 45$ °C) test are conducted in an environmental chamber subject to R-134a and R-1234yf. Test results shown in Fig. 6(a) revealed that the superheat in the R-1234yf system could be 2.9–5.9 °C higher than that of the R-134a system under all working conditions. Basically, this result is mainly because the expansion valve was improper for the R-1234yf system. The opening of the expansion valve is determined by the location of the plug as shown in Fig. 6(b). The pressures in the bulb and the evaporator outlet simultaneously acted on the membrane. The fluid charged in the bulb can detect the refrigerant temperature at the outlet of the evaporator, thereby imposing pressure on the membrane. The thermal expansion valve was originally designed for R-134a, and the fluid charged in the bulb was R-134a. As a consequence, the initial setting (mainly the spring force) would be improper because the thermophysical properties for R-1234yf were somewhat different from R-134a. In this regard, the superheat of the R-1234yf system could not be controlled precisely to a proper value during a drop-in test, leading to a drop of system performance. Notice that excessively large superheat is a waste of the evaporator heat transfer area because the heat transfer in the superheat region is mainly sensible heat transfer which is not efficient as the latent heat transfer in the two-phase region. Therefore, the larger superheat of the R-1234yf system during the drop-in tests could decrease the system performance. In this regard, in order to obtain more reasonable superheat values for the R-1234yf systems, the TXV was modified by Zillo et al. [10] (referred to as “TXV tuned” in the figure) from its original setting of 0.20 MPa gauge at 0 °C to 0.29 MPa gauge at 0 °C. From this adjustment of spring, as shown in Fig. 6(c), it was found that the over superheat is improved. However, it is recommended that thermal expansion valve should be made based on R-1234yf to respond correctly all the operational conditions. Chen et al. [11] also commented that despite the R-1234yf displays moderate low performance during direct replacement of R-134a in the typical drop-in tests, the R-1234yf system can be optimized by adjusting the setting of the thermal expansion valve or adding an internal heat exchanger, and the system performance could be comparable with that of the R-134a system.

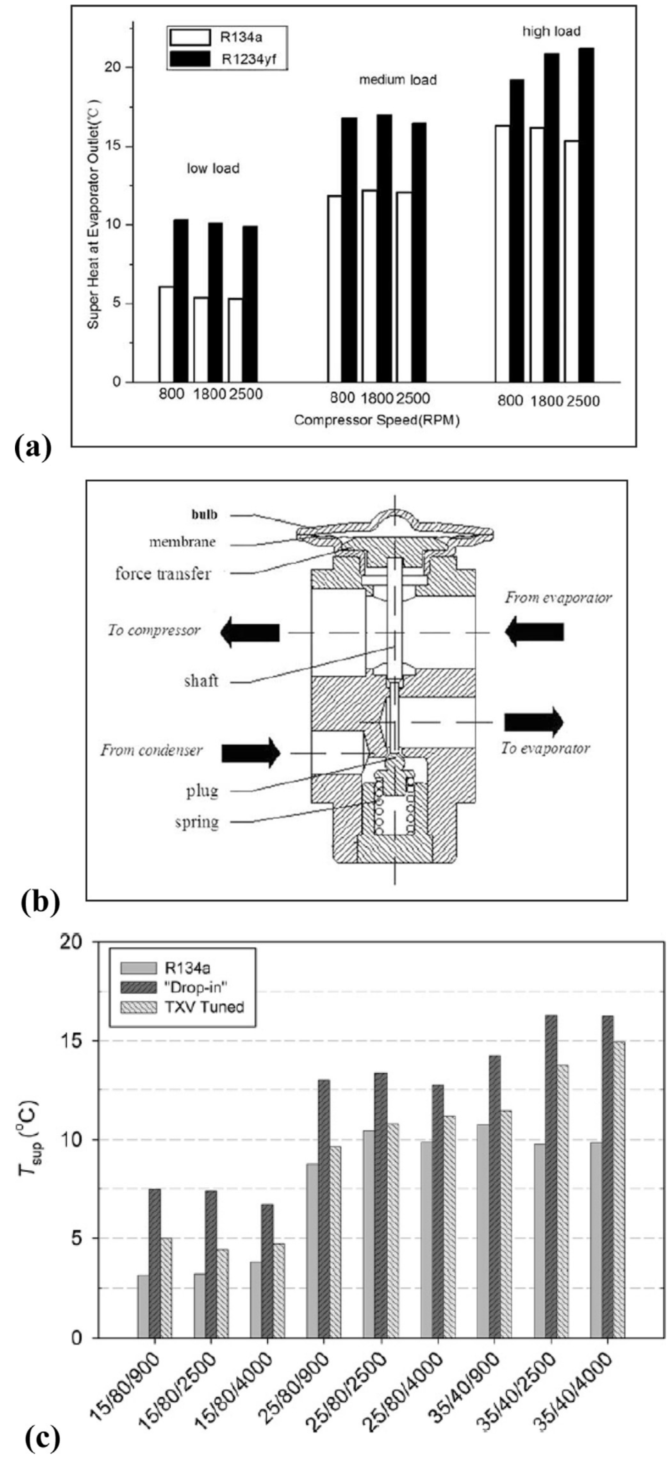


Fig. 6. Suction superheat subject to R-134a and R-1234yf (a) Test data from Zhao et al. [8]; (b) schematic for thermal expansion valve; (c) test data from Zillo et al. [10], Note: for a given label along the abscissa, the first number is the ambient temperature in °C, the second number is the relative humidity in percent, and the third number is the compressor speed in RPM.

4. Conclusions

The present review provides an overview of the system performance of R-1234yf in association with R-134a. The ODP of R-1234yf is zero and its GWP is as low as 4. R-1234yf also features and has a very low toxicity compared with R-134a and has a mild flammability. Currently, it is regarded as the most promising candidates to replace R-134a. Though the very similar vapor pressure of R-134a and R-1234yf suggests the possibility of a drop-in replacement of the current mobile air conditioners. The system performance, in terms of capacity and COP, of current R-1234yf system is normally lower than that of R-134a. This is because the system performance using R-1234yf and R-134a is contingent on many situations, including heat transfer performance in condenser and evaporator, refrigerant charge, expansion device, addition of expander/ejector/internal heat exchanger, compressor, and the like. Until now, there were tens of researches reporting system performance of R-134a and R-1234yf. The present overview had summarized previous efforts and concluded the following results:

(1) The COP and heat capacity of R-134a system may suffer from direct drop-in replacement of R-1234yf. The deterioration is around 0–27% depending on the operational conditions. With the introduction of internal heat exchanger, ejector, expander, or adjustment of thermal expansion valve, the deterioration can be relieved.
(2) R-1234yf gives a similar heat transfer performance with R-134a in evaporator but the performance in the condenser is inferior to R-134a. The phenomenon may be quite severe for a water cooled condenser since the dominant thermal resistance may fall in the refrigerant side. Therefore re-design of the condenser may become an important issues for future optimization of an R-1234yf system.
(3) The volumetric efficiency of R-1234yf system slightly lower than that of R-134a due to higher frictional drop of R-1234yf. There is no significant difference in oil circulation ratio for both refrigerants and the lubricant used in R-134a system can be used in R-1234yf system.
(4) For the same thermal expansion valve for controlling the suction superheat, it appears that higher suction superheat may occur for R-1234yf refrigerant. Hence further adjustment of spring in the valve is required for soft optimization.

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