TECHNICAL PAPER

Experimental assessments on R430A as an environment-friendly replacement to R134a in vehicle air conditioners

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Abstract
The direct emission of refrigerants from air conditioners and indirect carbon dioxide emissions from its associated energy usage in vehicles contributes significantly toward global warming. The primary objectives of this investigation are to present the performance comparisons and highlight the environmental benefits of R430A as a replacement to R134a in vehicle air conditioners. The influences of seven critical parameters such as compressor angular velocity, ambient temperature, condenser air velocity, evaporator air velocity, passenger load capacity, relative humidity and solar radiation were considered for thermodynamic analysis of vehicle air conditioners. The experimental results revealed that the R430A has 4–10% higher refrigeration effect, 4–11% lower compressor power consumption, 7–12% higher coefficient of performance and 5–12% lower exergy destructions than R134a. The discharge temperature in the compressor of vehicle air conditioners using R430A was found to be 2–6 °C higher than R134a. The lubricants used in R134a systems were found physically stable at 6 °C elevated temperatures. The R430A has lesser total equivalent global warming potential by 32–40%, 30–35% and 44–50% than R134a in petrol, diesel and liquefied petroleum gas-fuelled vehicles.

List of symbols

A	Area (m^2)
E	Energy consumption per day (in kW h/year),
h	Specific enthalpy (kJ/kg)
L	Leakage rate in the system (kg/year)
$\dot{m}$	Mass flow rate (kg/s)
m	Mass of refrigerant (kg)
N	Life of the system (years)
n	Operating time per day
q	Quantity
R	Given function
s	Specific entropy (kJ/kg K)
T	Temperature (^{o}C)
w_r	Total uncertainty
W	Work (W)
$w_1, w_2 \ldots w_n$	Uncertainty in the independent variables
$x_1, x_2, \ldots x_n$	Independent variables

Greeks

ρ	Density (kg/m^3)

Subscripts

0	Dead state
1	Compressor suction
2	Compressor discharge
3	Condenser outlet
4	Expansion valve outlet
cond	Condenser
comp	Compressor
ele	Electrical
evap	Evaporator:
gen	Generation
r	Refrigerant

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vol   Volumetric
COP   Coefficient of performance
GWP   Global warming potential
LPG   Liquefied petroleum gas
HC   Hydrocarbon
HFC   Hydrofluorocarbon
HFO   Hydrfluoroolefins
TEGWI   Total equivalent global warming impact
VAC   Vehicle air conditioner
VFD   Variable frequency drive

Abbreviations

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1 Introduction

R134a is the extensively using refrigerant in vehicle air conditioners, which has global warming potential of 1430 [1]. The Paris protocol 2016 has restricted the consumption of halogenated refrigerants, which includes R134a. India has committed to the Paris protocol to minimize refrigerants' consumption with high global warming potential (GWP). As a result, a suitable low GWP alternative needs to be identified for R134a replacements [2]. The vehicle air conditioner (VAC) sector has substantial refrigerant emissions during its life cycle due to frequent leakages. Many research efforts have been made on identifying environment-friendly replacements for R134a in VACs during its servicing. The possible replacements are R152a, R1234yf and hydrocarbon refrigerant mixtures.

Mohmoud [3] assessed the performance of VAC systems using R152a as a possible replacement for R134a and reported with 11% improvement in coefficient of performance (COP) than R134a. Similarly, the refrigeration effect and COP of VAC systems using R152a were enhanced by 2% and 9%, respectively, compared to R134a [4]. Similarly, Li et al. [5] improved the COP of VAC systems using R152a by 5% in highway drive and 10% in idling compared to R134a. They also reported that the R152a mass was reduced by 28% than R134a. Cabello et al. [6] improved the COP of VAC systems using R152a by 13% with a 10% loss of cooling capacity than R134a. The refrigerants R152a and R1234yf have 22–32% and 17–29% lesser environmental impacts than

R134a [7]. However, R152a has a 5-9 °C higher discharge temperature in the compressor than R134a.

Zilio et al. [8] improved the performance of R134a-based VAC systems retrofitted with R1234yf by turning the thermostatic expansion valve. The R1234yf mass charge was reduced compared to R134a due to its 10% lesser liquid density. Similarly, Lee and Jung [9] reported that the VAC systems using R1234yf have 4% lesser cooling capacity and 2.7% lesser COP than R134a. The R1234yf has a 6 °C lesser discharge temperature than R134a. Zhao et al. [10] studied the performance of VAC systems using R134a and R1234yf and reported 9% improved cooling capacity and 12.4% improved COP than R134a by increasing the condenser and evaporator areas. The losses of cooling capacity and COP in VAC systems were reduced by adding internal heat exchangers [11]. Similarly, Qi [12] reported that the cooling capacity and COP observed in VAC systems using R1234yf were 7.7–10.6% and 4.8–7% lesser than R134a. Cho and Park [13] reported that the exergy efficiency of VAC systems using R1234yf was 3.4–4.6% lesser than R134a. Golzari et al. [14] have identified the compressor as the main ineffective component in VAC systems due to more exergy destruction compared to other components. They also concluded that the exergy destructions in VAC systems using R1234yf was lower than R134a. Prabhakar et al. [15] reported that the COP and exergy efficiency of VAC systems using R1234yf were improved by 4.3–8.6% and 3.7–5.1%, respectively, by turning the expansion valve. Meng et al. [16] reported that the VAC systems using R1234yf/R134a mixture (mass fraction: 0.88:0.12) has 4–9% lower COP in cooling and 4–16% in heating modes compared to R134a. Nevertheless, R1234yf decomposes rapidly in the environment and forms tri-fluoro acidic acid that is unsafe for aquatic systems [17]. Hence, R1234yf is not considered a sustainable replacement.

Maclaine-cross [18] reported that the hydrocarbon refrigerants are energy efficient and safe to use in VAC systems. The pure hydrocarbons such as R290, R600 and R600a cannot retrofit R134a due to a mismatch in working pressures and volumetric cooling capacities. Hence, the mixtures of R290 and R600a are preferred. The R290/R600a mixture has improved energy performance with reduced discharge temperature than R12 and R134a [19, 20]. Wongwises et al. [21] reported that the ternary mixture composed of R290, R600a and R600 (with a mass fraction of 0.5:0.1:0.4) has 21% higher compressor power consumption with 41% higher refrigeration capacity than R134a. The mixture has lower compressor discharge temperatures than R134a. However, the hydrocarbon mixtures are zeotropic in nature with a temperature glide of 5–7 °C. Therefore, significant performance degradation was observed in VAC systems working under high ambient temperatures. Furthermore, the mass fractions of hydrocarbon mixtures may vary during leakage conditions [22].

The R430A is a binary near an azeotropic mixture of R152a and R600a (0.76:0.24; mass fraction) with a temperature glide less than 0.2 °C was reported as a possible R134a replacement [23–25]. The discharge temperature of R152a was reduced by adding R600a. The performance of a refrigerator was tested using R134a and R430A as working fluids and reported with 3.9% reduced power consumption and 3.8–6.4% improved COP with 2–4 °C higher discharge temperature in the compressor [23]. He also reported that the R430A has 4–5% lower environmental impact than R134a. The VAC systems using R430A have a 12–20% higher COP than R134a [24]. The environmental impacts of VAC systems using R430A were reduced by 32.4%, 35% and 47.3% compared to R134a for the vehicles powered by diesel, petrol and liquefied petroleum gas (LPG), respectively [25].

The cited literature reviews confirmed that many research attempts had been reported on the performance of environment-friendly refrigerants in VAC systems. However, there is no comprehensive research on thermodynamic performance comparisons of VAC systems using R134a and R430A under the influence of seven critical parameters such as compressor angular velocity, ambient temperature and condenser air velocity, evaporator air velocity, passenger load capacity, relative humidity and solar radiation. Hence, the detailed research investigations have been made on the usage of R430A as a possible replacement to R134a in VAC systems.

2 Details of experimentation

The descriptions of experimental setup, measuring instruments, characteristics of refrigerants and the experimental procedures are presented in this section.

2.1 Description of experimental setup

The layout and photograph of a VAC experimental arrangement are shown in Fig. 1. The experimental arrangement consists of a compression refrigeration cycle using R134a is attached with a sedan passenger vehicle compartment. The compression refrigeration cycle consists of key components like the compressor, condenser, expansion device, evaporator and accessories such as sight glass, receiver and filter drier. The compressor angular velocity was varied using a variable frequency drive (VFD) electrical motor of 2.25 kW rated power. The condenser air velocity was varied using a blower connected with a VFD electrical motor of 0.75 kW rated power. The evaporator air velocity was changed using fan control. The resistance air heater (2 kW rated input power, connected with the dimmerstat) was kept in the airflow path toward the condenser to vary the ambient temperatures. The heat

gains inside the cabin are due to passenger load capacity; the temperature difference between cabin and ambient and solar heat gain were simulated using two electric resistance air heaters of rated input power of 2 kW. The electrical resistance heaters were connected with dimmerstat to control the heat outputs. The relative humidity (RH) inside the cabin was maintained between 55 and 85% using a portable humidifier.

2.2 Measuring instruments

The refrigerant temperatures (at four salient points in the refrigerant circuit as shown in Fig. 1a), cabin temperatures (at four locations inside the vehicle cabin as illustrated in Fig. 1a), and air temperatures (at inlets and outlets of the condenser and evaporator) were measured using twelve RTD sensors (Pt-100) having an accuracy of ± 0.2 °C. The RTD sensors are connected to the digital display with 0.1 °C resolution. The refrigerant pressures at four salient points (shown in Fig. 1a) in the refrigerant circuit were measured using four Bourdon tube pressure gauges with $\pm 2\%$ accuracy. The ambient temperature was observed by a calibrated thermometer with ± 0.2 °C accuracy. The ambient and cabin RH were measured using a sling psychrometer with $\pm 2\%$ accuracy. The digital tachometer (with ± 5 rpm accuracy) was used for measuring the compressor angular velocity. The air velocities at the condenser inlet and evaporator outlet were measured using a standard vane-type anemometer with ± 0.1 m/s accuracy. The instantaneous motor power consumption was monitored using a digital power meter with an accuracy of ± 3 W. The heat generated inside the vehicle cabin was quantified based on the electricity consumption of resistance heaters. The electricity consumptions of resistance heaters installed in the vehicle cabin and the condenser airflow path were noted using separate power meters with ± 3 W accuracy. The refrigerant mass charged in the system was quantified using a digital weighing machine with ± 5 g accuracy.

2.3 Characteristics of refrigerants

The thermophysical characteristics of R134a and its possible replacements are revealed in Fig. 2. It is identified that the R430A has 8–11% lower vapor pressure than R134a and R1234yf. Furthermore, it is noticed that with the R430A has 33–44% higher latent heat with 33–38% lesser liquid density and 19–29% lesser viscosity than R134a and R1234yf. The thermodynamic characteristics of R134a and the possible replacements are given in Table 1. Except for critical pressure, all the thermodynamic characteristics of R430A are found to be good compared to R134a and its other alternatives [26].
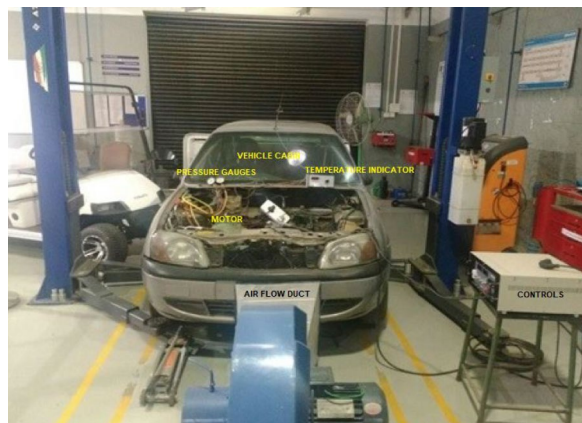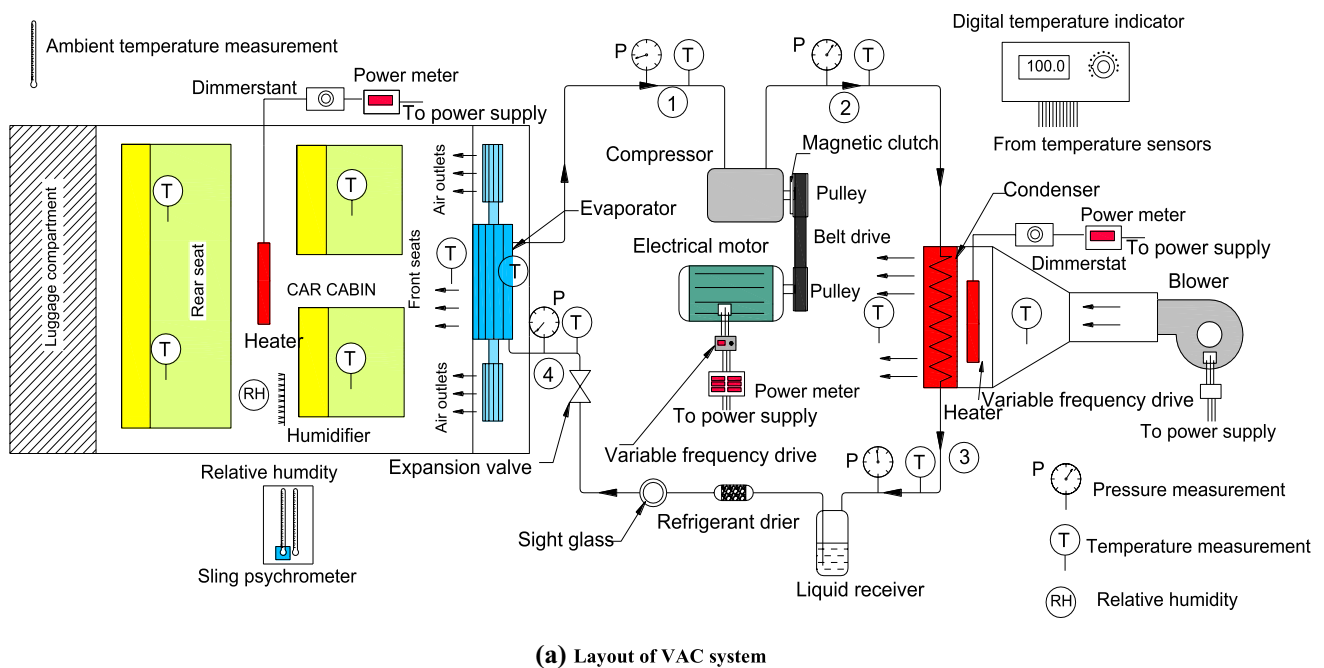
(a) Layout of VAC system

(b) Photographic view of VAC system

**Fig. 1** Schematic and photographic views of VAC systems

2.4 Experimental procedure

The refrigerant circuit of VAC experimental setup was flushed using nitrogen gas to purge the air and other non-condensable gases. Then, the system was evacuated using a vacuum pump. As per the manufacturer's recommendation, 350 ± 10 g of refrigerant R134a was charged. The experimental trials were made in a VAC experimental setup for the initial test conditions mentioned in Table 2. The compressor angular velocity was maintained between 1000 and 3000 rpm using a VFD electrical motor. The ambient temperature was simulated between 30 and 45 °C using electrical resistance heaters. The condenser and evaporator air

velocities varied from 2 to 15 m/s and from 1 to 4 m/s. The condenser and evaporator's air mass flow rates were calculated using dimensions of condenser and evaporator outlet dimensions. The heat gain inside the vehicle cabin due to passenger load capacity was varied for the seating capacities between 1 and 5. Moreover, the heat gains due to temperature differences between ambient and vehicle cabin and solar radiation (between 250 and 1000 W/m^2) infiltrated through the windscreen glasses were estimated and simulated by resistance heaters. The RH inside the VAC cabin was varied between 55 and 85% using a portable humidifier. Initially, the vehicle doors were kept open to reach steady-state with the surroundings. The VAC system was then switched ON
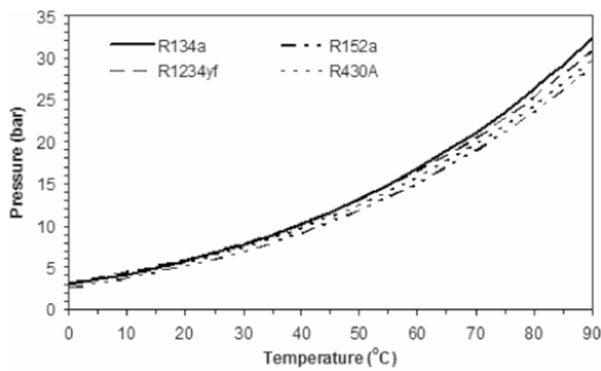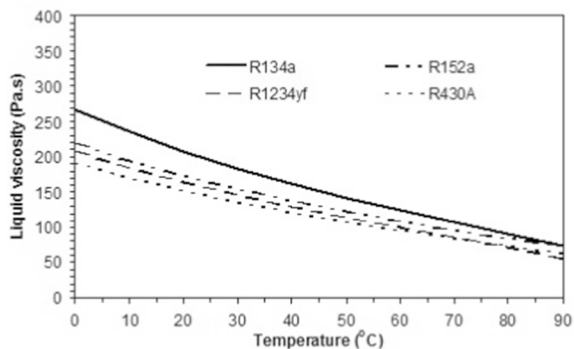
(a) Variation of pressure with temperature
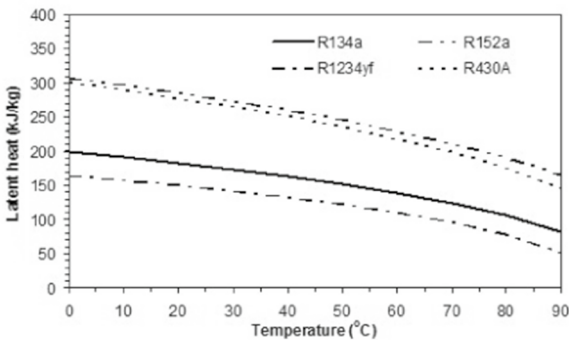
(b) Variation of latent heat with temperature
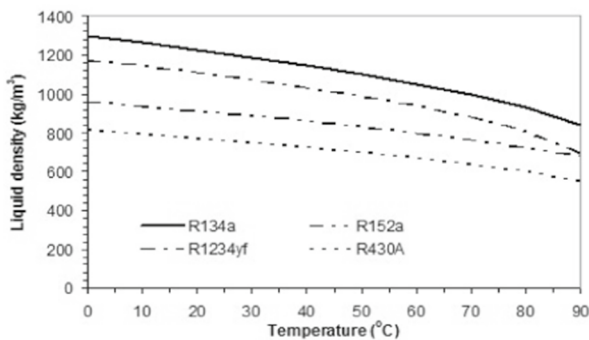
(c) Variation of liquid density with temperature

(d) Variation of liquid viscosity with temperature

Fig. 2 Thermophysical characteristics of R134a and its alternatives. **a** Pressure, **b** liquid density, **c** liquid density, **d** liquid viscosity

and allowed to reach the steady-state cabin temperature of 22 ± 0.5 °C. Initially, the compressor angular velocity was maintained at 1000 rpm. The time duration to reach 22 ± 0.5 °C from its initial cabin temperature was measured to investigate the pull-down behavior. During experimentations, the pressure and temperature of refrigerants at salient points in the refrigerant circuit, air temperatures inside the vehicle cabin, air temperatures at inlets and outlets of condenser and evaporator, compressor angular velocity, compressor power consumption and air velocities at condenser inlet and evaporator outlet were measured after the system has reached the stable operation. Further, the experiments have been done by varying the compressor angular velocities to 1500, 2000, 2500 and 3000 rpm using a VFD electrical motor. Following the reference test, the R134a was taken out and charged with an equal mass of R430A. The equal mass of R430A was estimated to be 175 ± 10 g based on its liquid density compared with R134a. Then, test procedures were repeated in a VAC system using R430A. The uncertainty analysis was carried out to ascertain the experimental errors.

3 Thermodynamic analysis of vehicle air conditioners

The observed pressure and temperature values at the typical locations in the refrigerant cycle of VAC and air temperatures at different locations inside the vehicle cabin were used for assessing the thermodynamic performance. The refrigerant characteristics were predicted using REFPROP. The equations used in the thermodynamic analysis of VAC systems are presented in this section.

3.1 Energy analysis

The mass flow rates of refrigerant in a VAC at different compressor angular velocities are estimated by:

$$\dot{m}_r = \frac{\text{RPM}}{60} \times V_{\text{dis,comp}} \times \rho_1 \times \eta_{\text{vol}} \tag{1}$$

Based on the experimental trials, the compressor volumetric efficiency was assumed as 0.85. The $V_{dis,comp}$ is assumed as 155 cm^3/revolution based on the compressor specifications. Further, the compressor power consumption is estimated by [25]:

$$W_{\text{comp}} = \dot{m}_r (h_2 - h_1) \tag{2}$$

$$W_{\text{ele}} = \frac{\dot{m}_r (h_2 - h_1)}{\eta_{motor} \times \eta_{transmission}} \tag{3}$$

Table 1 Thermodynamic characteristics of R134a and its alternatives

Refrigerants	Molecular weight (°C)	Boiling point (°C)	Critical temperature (°C)	Critical pressure (MPa)	Flammable index	ODP (R12=1)	GWP (CO$_2$=1)
R134a	101	−26	101	4.05	A1	0	1430
R152a	66	−24	113.3	4.52	A3	0	140
R1234yf	114	−29.5	94.7	3.38	A2L	0	<4
R430A	64.14	−21	118	4.30	A3	0	104

Table 2 Experimental conditions

Parameters	Conditions					
	Case -I	Case -II	Case -III	Case -IV	Case -V	Case -VI
Compressor angular velocity (rpm)	1000, 1500, 2000, 2500, 3000					
Ambient Temperature (°C)	30, 35, 40, 45	35	35	35	35	35
Condenser air velocity (m/s)	15	2, 5, 10, 15	15	15	15	15
Evaporator air velocity (m/s)	4	4	1, 2, 3, 4, 5	4	4	4
Passenger Load	2	2	2	1, 2, 3, 5	2	2
Relative humidity (%)	65	65	65	65	55, 65, 75, 85	65
Solar radiation (W/m^2)	500	500	500	500	500	250, 500, 750, 1000

Here, the efficiencies in belt power transmission and electrical motor are assumed to be 0.9 and 0.95, respectively. The refrigeration capacity of an evaporator is estimated by:

$$Q_{evap} = \dot{m}_r \times (h_1 - h_4) \tag{4}$$

$$Q_{evap} = \dot{m}_a c_{pa}(T_{evap,in} - T_{evap,out}) \tag{4a}$$

The COP of a VAC is estimated by:

$$COP = \frac{Q_{evap}}{w_{ele}} \tag{5}$$

The condenser heat rejection in a VAC is given by:

$$Q_{cond} = \dot{m}_r(h_2 - h_3) \tag{6}$$

$$Q_{cond} = \dot{m}_a c_{pa}(T_{evap,in} - T_{evap,out}) \tag{6a}$$

3.2 Exergy analysis

The inefficient component in VAC systems using R134a and R430A was identified using exergy analysis. The thermodynamic processes involved in the compression cycle of the VAC system are shown in Fig. 3. The assumptions made in exergy analysis are by [27]:
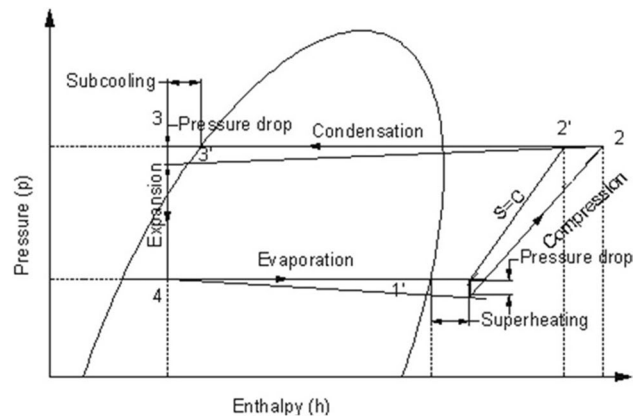

Fig. 3 Pressure–enthalpy chart of compression cycle

(i) The thermodynamic processes in VAC systems are assumed as steady state.

(ii) The effects of chemical, kinetic and potential in VAC systems are neglected.

(iii) The ambient condition is assumed as dead state conditions.

The energy and mass balances are given by:

$$E_{in} = E_{out} \tag{7}$$

$$\dot{m}_{r,1} = \dot{m}_{r,2} = \dot{m}_{r,3} = \dot{m}_{r,4} \tag{8}$$

The exergy performance is given by:

$$\dot{Ex}_{dest} = T_0 s_{gen} \tag{9}$$

The exergy balance of a VAC is given by:

$$\dot{Ex}_{dest} = \dot{Ex}_{in} - \dot{Ex}_{out} \tag{10}$$

The specific exergy at typical locations is estimated using the following relation:

$$ex = (h - h_0) - T_0(s - s_0) \tag{11}$$

$$\eta_{ex} = \frac{Q\left(1 - \frac{T_0}{T_{ref}}\right)}{W_{ele}} \tag{19}$$

3.3 Total equivalent global warming impact

Fischer [28] quantified the environmental impacts caused in VAC systems (due to direct refrigerant emissions and indirect carbon dioxide emissions associated with the energy consumption) in terms of total equivalent global warming impact (TEGWI). TEGWI is given by:

$$TEGWI = m \times L \times N \times GWP + (m \times GWP \times (1 - \alpha) +$$
$$\left(\frac{q_f}{10,000} \times km/year + \frac{q_f}{10,000 \times 100} \times m_{AC} \times km/year\right) \frac{COP_{reference}}{COP_{alternative}} \beta \tag{20}$$

The exergy rate is determined as:

$$\dot{Ex} = \dot{m}_r ex \tag{12}$$

The exergy destructions in various components of VAC systems are given by:

$$\dot{Ex}_{dest,comp} = \dot{m}_r(h_1 - h_2) - \dot{m}_r T_0(s_1 - s_2) + W_{ele} \tag{13}$$

$$\dot{Ex}_{dest,cond} = \dot{m}_r T_0(s_3 - s_2) + Q_{cond}\left(\frac{T_0}{T_{r,3}}\right) \tag{14}$$

$$\dot{Ex}_{dest,exp.val} = \dot{m}_r T_0(s_4 - s_3) \tag{15}$$

$$\dot{Ex}_{dest,evap} = \dot{m}_r T_0(s_1 - s_4) - Q_{evap}\left(\frac{T_0}{T_{r,1}}\right) \tag{16}$$

$$\dot{Ex}_{dest,sys} = \dot{Ex}_{dest,comp} + \dot{Ex}_{dest,cond} + \dot{Ex}_{dest,exp.val} + \dot{Ex}_{dest,evap} \tag{17}$$

The relative exergy destruction in various components of VAC systems is given by:

$$\dot{Ex}_{relative} = \frac{Ex_{component}}{Ex_{dest,sys}} \times 100 \tag{18}$$

The exergy efficiency of VAC systems is given by:

The assumptions used for estimating the TEGWI are:

- The GWP of refrigerants used in VAC systems is listed in Table 1.
- The VAC systems have significant refrigerant emissions due to more vibrations and poor road conditions [29].
- The weight of VAC and its accessories are assumed as 15 kg.
- Life of a VAC is assumed as 15 years.
- The refrigerant mass charge of R134a is 350 ± 10 g (manufacturer recommendation). The equivalent quantity of R430A is estimated as 175 ± 10 g.
- The annual refrigerant leakage rate is 20% of charge quantity.
- The COP of a VAC using R134a and R430A is assessed using an experimental setup powered with VFD electrical motor.
- The fuel consumption and carbon dioxide (CO_2) emissions observed in Indian make vehicles fuelled by LPG, petrol and diesel with air conditioners are given in Table 3 were used for environment impact assessments.

3.4 Uncertainty analysis

The errors in experimental results are ascertained by the following relation [30]:

Table 3 Fuel consumption by sedan passenger vehicle

Fuel	Petrol	Diesel	LPG
Fuel consumption with air conditioning (kmpl)	16.4	18.1	15.8
Vehicle carbon dioxide emissions (kg of CO_2 per liter)	2.31	2.68	1.64
Vehicle carbon dioxide emissions (kg of CO_2 per km)	0.1408	0.1488	0.1037

$$w_r = \left[\left(\frac{\partial R}{\partial x_1} w_1 \right)^2 + \left(\frac{\partial R}{\partial x_2} w_2 \right)^2 + \cdots + \left(\frac{\partial R}{\partial x_n} w_n \right)^2 \right]^{1/2}.$$
(21)

The errors in compressor exit temperature, compressor power consumption, refrigeration effect, COP, exergy destruction and exergy efficiency were estimated as $\pm 1.4\%$, $\pm 2.44\%$, $\pm 2.47\%$, $\pm 3.4\%$, $\pm 6.9\%$ and $\pm 5.8\%$, respectively.

4 Result and discussions

The thermodynamic performances of VAC systems were experimentally studied using R134a and R430A. The influences of seven critical parameters on the thermodynamic performance of VAC systems are described.

4.1 Discharge temperature

The variations of discharge temperatures of a compressor used in VAC systems using R134a and R430A are depicted in Fig. 4. The increase in compressor angular velocity has significantly increased the discharge temperatures in the compressors. The influences of ambient temperature on discharge temperatures are depicted in Fig. 4a (Case-I). It is noticed that the discharge temperatures of compressors using both the refrigerants were increased by 3–6 °C with a rise in ambient temperatures from 30 to 45 °C. The highest discharge temperatures of 90.5 and 95.6 °C were observed in the compressor using R134a and R430A refrigerants, respectively, at the compressor angular velocity of 3000 rpm and an ambient temperature of 45 °C. The discharge temperatures of R430A were observed to be 4–5 °C higher than R134a. Further, the discharge temperatures of R134a and R430A refrigerants under the influence of condenser air velocity are illustrated in Fig. 4b (Case-II). The discharge temperatures were dropped by 2–7 °C with increasing condenser air velocity from 2 to 15 m/s. The maximum discharge temperatures of 88.9 and 93.6 °C were observed in compressors of VAC systems using R134a and R430A, respectively, at the compressor angular velocity of 3000 rpm with a condenser air velocity of 2 m/s.

The influences of evaporator air velocity on discharge temperature are illustrated in Fig. 4c (Case-III). It is noticed that the discharge temperatures in the compressor using R134a and R430A were reduced by 2–5 °C with a rise in evaporator air velocity from 1 to 4 m/s. The maximum discharge temperatures of 89.6 and 93.7 °C were noticed in compressors of VAC systems using R134a and R430A systems, respectively, at the compressor angular velocity of 3000 rpm and evaporator air velocity of 1 m/s. The effects of increasing passenger load capacity on discharge temperature

are illustrated in Fig. 4d (Case-IV). It is noticed that the discharge temperatures were increased by 4–6 °C and 5–7 °C in R134a and R430A systems, respectively, with a rise in passenger load capacity from 1 to 5. The maximum discharge temperatures of 91.2 and 96.3 °C were noticed in compressors using R134a and R430A, respectively, at compressor angular velocity of 3000 rpm and maximum passenger seating load capacity of 5. The discharge temperature of R430A is 2–6 °C higher than R134a.

The effects of cabin RH on discharge temperature are illustrated in Fig. 4e (Case-V). It is observed that the discharge temperatures were dropped by 2–3 °C and 3–5 °C for R134a and R430A systems, respectively, with a rise in cabin RH. The highest discharge temperatures of 89.1 and 93.3 °C were noticed in compressors using R134a and R430A, respectively, at the compressor angular velocity of 3000 rpm and 55% cabin RH. The discharge temperature noted in the compressor using R430A was found to be 2–4 °C higher than R134a. The effects of solar radiation on discharge temperature are depicted in Fig. 4f (Case-VI). The discharge temperatures of R134a and R430A were raised by 5–6 °C under the influence of increased solar radiation from 250 to 1000 W/m^2. The maximum discharge temperatures of 95.3 and 100.4 °C were noted in VAC systems using R134a and R430A, respectively, at the compressor angular velocity of 3000 rpm and solar radiation of 1000 W/m^2. The discharge temperatures in compressors using R430A were 2–6 °C higher due to its high critical pressure than R134a. The existence of R600a in R430A has lowered the discharge temperature compared to R152a. The lubricants used in R134a compressors are stable at high working temperatures, ensuring the reliability and durability of R430A retrofitting [23–26].

4.2 Compressor power consumption

In Fig. 5, the variations of compressor power consumptions using R134a and R430A are illustrated. The rise in compressor angular velocity has significantly increased the power consumption of the compressor. The influences of ambient temperature on compressor power consumption are shown in Fig. 5a (Case-I). The rise in ambient temperature from 30 to 45 °C has increased the compressor's power consumption by 17–26% and 7–18% in the VAC systems using R134a and R430A, respectively. The maximum compressor power consumptions of 2.20 kW in R134a and 2.01 kW in R430A were observed in VAC systems at the compressor angular velocity and an ambient temperature of 3000 rpm and 45 °C, respectively. In Fig. 5b (Case-II), the variations of compressor power consumptions under the influence of condenser air velocity are depicted. The compressor power consumption was dropped by 8–21% and 6–16% in R134a and R430A systems, respectively, with an increase in condenser air
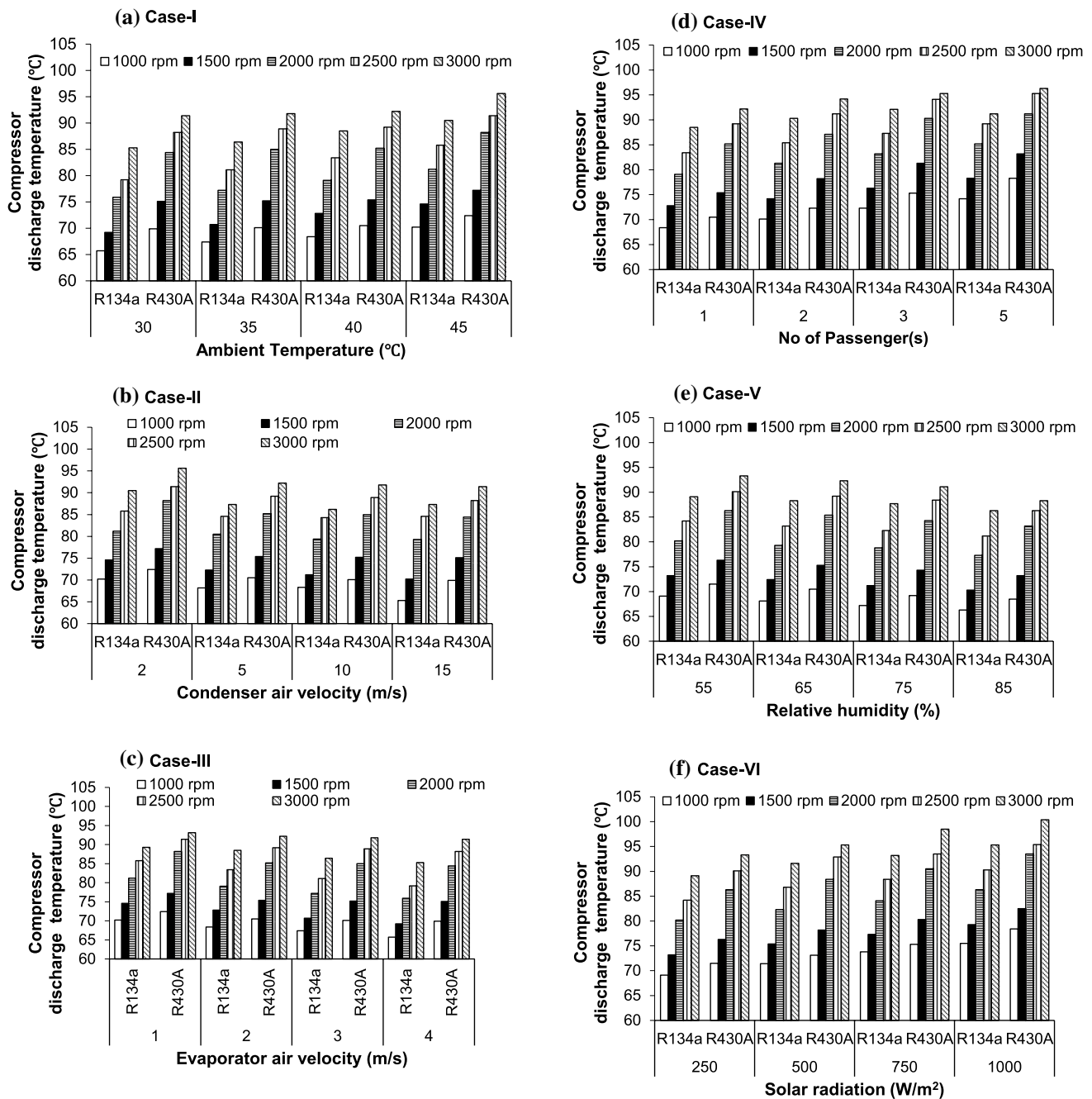
Fig. 4 Variation of Compressor discharge temperature. **a** Influence of ambient temperature; **b** influence of condenser air velocity; **c** influence of evaporator air velocity; **d** influence of passenger load; **e** influence of cabin RH; and **f** influence of solar radiation

velocity from 2 to 15 m/s. The maximum compressor power consumption of 2.21 kW in R134a and 2.01 kW in R430A was observed in VAC systems at the compressor angular velocity and condenser air velocity of 3000 rpm and 2 m/s, respectively.

The variations in compressor power consumptions due to the effects of evaporator air velocity are shown in Fig. 5c (Case-III). It is observed that an increase in evaporator air

velocity from 1 to 4 m/s has reduced the compressor power consumptions by 5–8% and 3–4% in R134a and R430A VAC systems, respectively. The highest compressor power consumptions of about 2.12 kW and 2.03 kW were observed in VAC systems using R134a and R430A, respectively, at the compressor angular velocity of 3000 rpm and evaporator air velocity of 1 m/s. The effect of passenger load capacity on compressor power consumption is illustrated in Fig. 5d
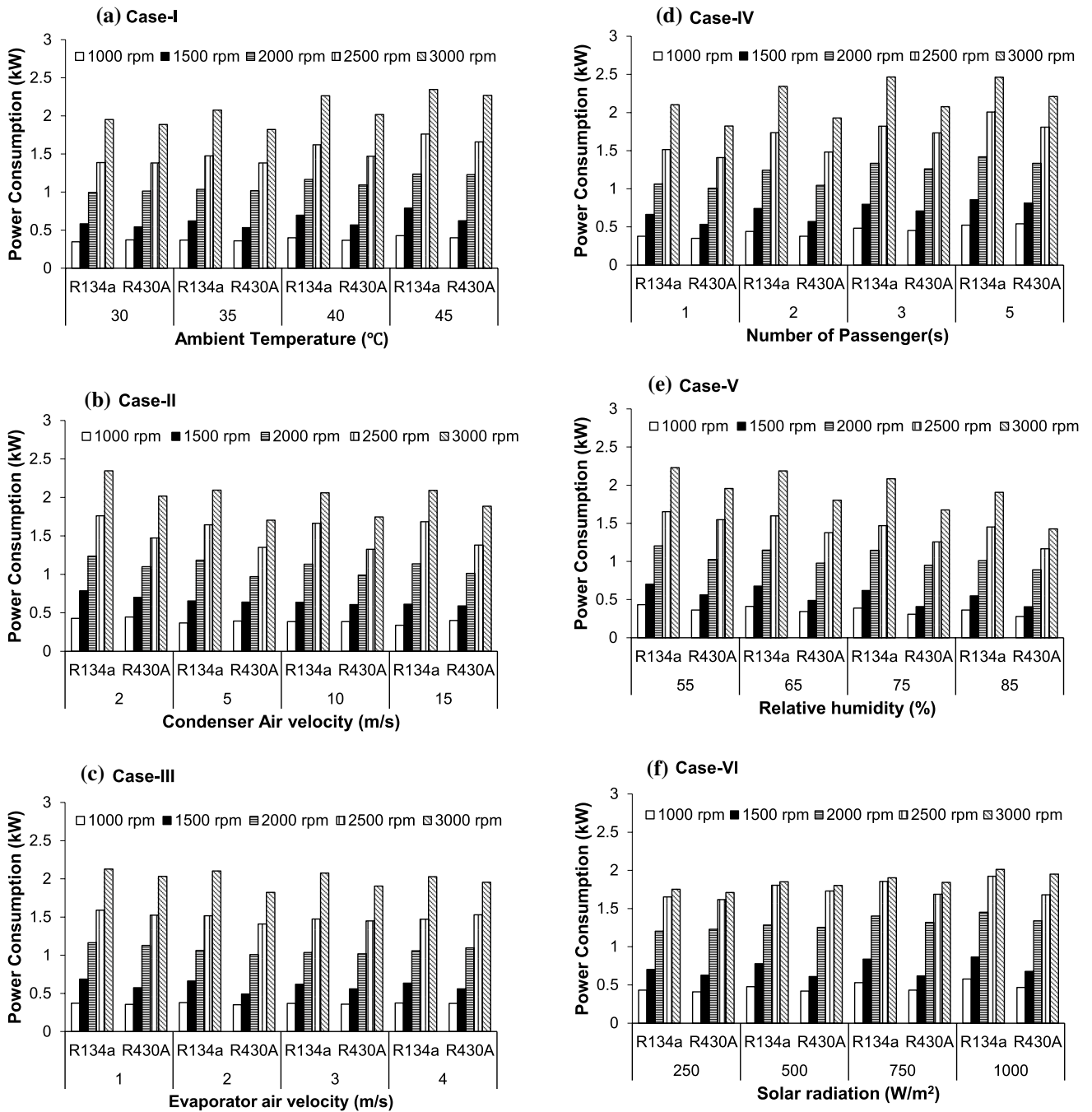
Fig. 5 Variation of compressor power consumption. **a** Influence of ambient temperature; **b** influence of condenser air velocity; **c** influence of evaporator air velocity; **d** influence of passenger load; **e** influence of cabin RH; and **f** influence of solar radiation

(Case-IV). It is noticed that the increase in passenger load capacity from 1 to 5, the compressor power consumption was increased by 28–54% and 17–38% in VAC systems using R134a and R430A, respectively. The highest compressor power consumptions of 2.21 and 2.12 kW were noticed in VAC systems using R134a and R430A, respectively, for the compressor angular velocity of 3000 rpm and passenger load capacity of 5.

The compressor power consumption variations under cabin RH influence are depicted in Fig. 5e (Case-V). It is noticed that the compressor power consumptions were dropped by 12–22% and 13–128% in R134a and R430A systems, respectively, with an increase in cabin RH from 55 to 85%. The R134a and R430A have maximum compressor power consumptions of 2.20 and 1.95 kW, respectively, in VAC systems at the compressor angular velocity

of 3000 rpm and cabin RH of 55%. The effects of solar radiation on compressor power consumptions are compared in Fig. 5f (Case-VI). The compressor power consumptions were raised by 6–8% and 3–4% in R134a and R430A systems, respectively, under the effect of increasing solar radiation from 250 to 1000 W/m^2. The maximum compressor power consumptions of 2.02 kW in R134a and 1.95 kW in R430A systems were noticed at maximum solar radiation of 1000 W/m^2 and compressor angular velocity 3000 rpm. The results confirmed that the R430A has 4–11% lesser power consumption in the compressor owing to its lesser liquid density and lesser liquid viscosity than R134a [24–26].

4.3 Refrigeration effect

In Fig. 6, the variations of refrigeration effects observed in VAC systems using R134a and R430A are compared. The refrigeration effects were increased under the effect of increasing compressor angular velocity. In Fig. 6a (Case-I), the influences of ambient temperature on refrigeration effect of a VAC system's refrigeration effect are depicted. The refrigeration effects of VAC systems were dropped by 16–22% in R134a and 12–20% in R430A systems under the influence of rising ambient temperatures from 30 to 45 °C. The maximum refrigeration effects of 3.9 and 4.1 kW were noticed in VAC systems using R134a and R430A, respectively, for the compressor angular velocity maintained at 3000 rpm and ambient temperature maintained at 30 °C. The refrigeration effects under the effect of condenser air velocity are compared in Fig. 6b (Case-II). The refrigeration effects were increased by 4–22% and 5–25% in VAC systems using R134a and R430A, respectively, under the effects of increasing condenser air velocity from 2 to 15 m/s. The highest refrigeration effects of 4.47 and 4.77 kW were observed in VAC systems using R134a and R430A, respectively, at 15 m/s air velocity approaching the condenser and compressor angular velocity of 3000 rpm. The influences of evaporator air velocity on refrigeration effect are depicted in Fig. 6c (Case-III). The results revealed that an increase in evaporator air velocity from 1 to 4 m/s has increased refrigeration effects by 5–14% and 7–20% in R134a and R430A systems, respectively. The maximum refrigeration effects of 3.93 kW in R134a and 4.19 kW in R430A were noticed in VAC systems at evaporator air velocity of 4 m/s and compressor angular velocity of 3000 rpm. In Fig. 6d (Case-IV), the influences of passenger load capacity on refrigeration effect are depicted. It is observed that the refrigeration effects were reduced by 8–13% and 7–8% in R134a and R430A systems, respectively, under the influence of increasing passenger load capacity from 1 to 5. The maximum refrigeration effects 3.8 kW in a VAC system using R134a and 4.2 kW in a VAC system using R430A were

noticed at the compressor angular velocity of 3000 rpm and single passenger load capacity.

The influence of cabin RH on the refrigeration effect is illustrated in Fig. 6e (Case-V). It is noted that the refrigeration effects of VAC systems using R134a and R430A were raised by 11–17% and 11–28%, respectively. The maximum refrigeration effects of about 4.1 and 4.2 kW were noticed in VAC systems using R134a and R430A, respectively, at 85% cabin RH and maximum compressor angular velocity of 3000 rpm. The influences of solar radiation on refrigeration effects of VAC systems using R134a and R430A are compared in Fig. 6f (Case-VI). The results revealed that the refrigeration effects observed in VAC systems using R134a and R430A were dropped by 8–11% and 4–12%, respectively. The highest refrigeration effect of 3.6 and 4.1 kW was noticed in VAC systems using R134a and R430A, respectively, at solar radiation of 250 W/m^2 and compressor angular velocity of 3000 rpm. The R430A has 4–10% higher refrigeration effect and shorter pull-down time because of its higher latent heat, lesser viscosity and lesser density than R134a [25]

4.4 Coefficient of performance

The variations of COPs of R134a and R430A in a VAC system are shown in Fig. 7. The results revealed that the COPs of VAC systems were dropped with a raise in compressor angular velocity. In Fig. 7a (Case-I), the variations of COP under the influence of ambient temperatures are depicted. An increase in ambient temperature from 30 to 45 °C has dropped the COPs by 28–40% and 27–38% in R134a and R430A VAC systems, respectively. The maximum COPs of 3.9 and 4.2 were noticed in VAC systems using R134a and R430A, respectively, at the compressor angular velocity of 1000 rpm and an ambient temperature of 30 °C. The effects of condenser air velocity on COP variations are compared in Fig. 7b (Case-II). It is observed that the COPs were increased by 7–25% and 20–22% in R134a and R430A systems, respectively, with an increase in condenser air velocity from 2 to 15 m/s. The maximum COPs of about 3.4 and 3.8 were noted in VAC systems using R134a and R430A, respectively, for the condenser air velocity of 15 m/s and compressor angular velocity of 1000 rpm [25].

The variations of COPs under the effect of evaporator air velocity are shown in Fig. 7c (Case-III). It is observed that the COPs of a VAC using R134a and R430A were increased by 8–21% and 7–21%, respectively, with rising in evaporator air velocity from 1 to 4 m/s. The maximum COPs of 3.6 and 4.0 were observed in a VAC system using R134a and R430A, respectively, at the evaporator air velocity of 4 m/s and the compressor angular velocity of 1000 rpm. The effects of passenger load capacity on COP variations of the VAC system are compared in Fig. 7d (Case-IV). It is noticed
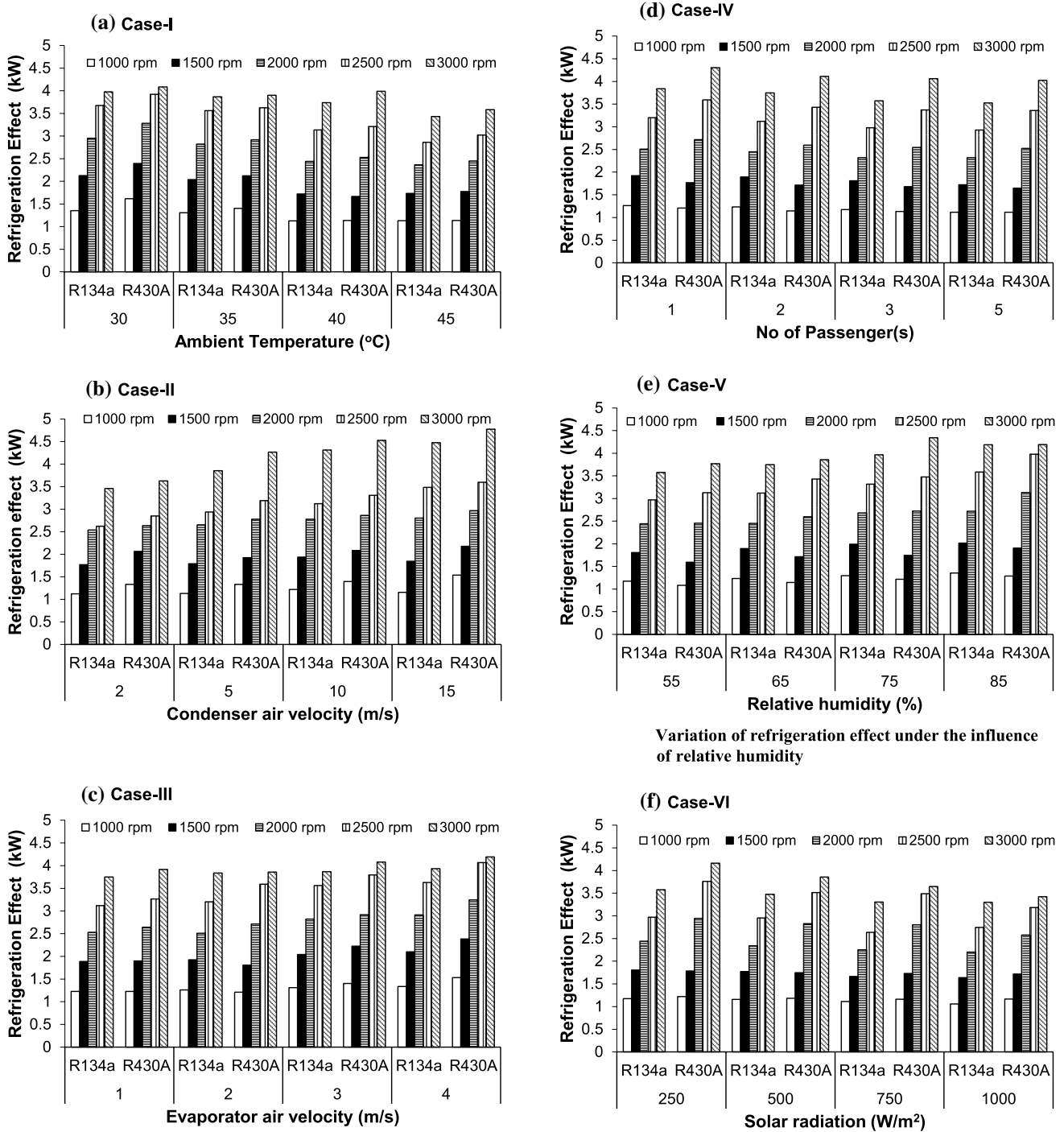
Fig. 6 Variation of refrigeration effect **a** influence of ambient temperature; **b** influence of condenser air velocity; **c** influence of evaporator air velocity; **d** influence of passenger load; **e** influence of cabin RH; and **f** influence of solar radiation

that the COPs were dropped by 20–32% and 20–30% in VAC systems using R134a and R430A systems, respectively, under the influence of increasing passenger load capacity from 1 to 5. The maximum system COPs of 3.3 and 3.4 were observed in VAC systems using R134a and R430A,

respectively, at the compressor angular velocity of 1000 rpm with single passenger seating capacity.

The effects of cabin RH on COP variations of VAC systems are illustrated in Fig. 7e (Case-V). It is noted that the COPs of VAC systems using R134a and R430A were raised
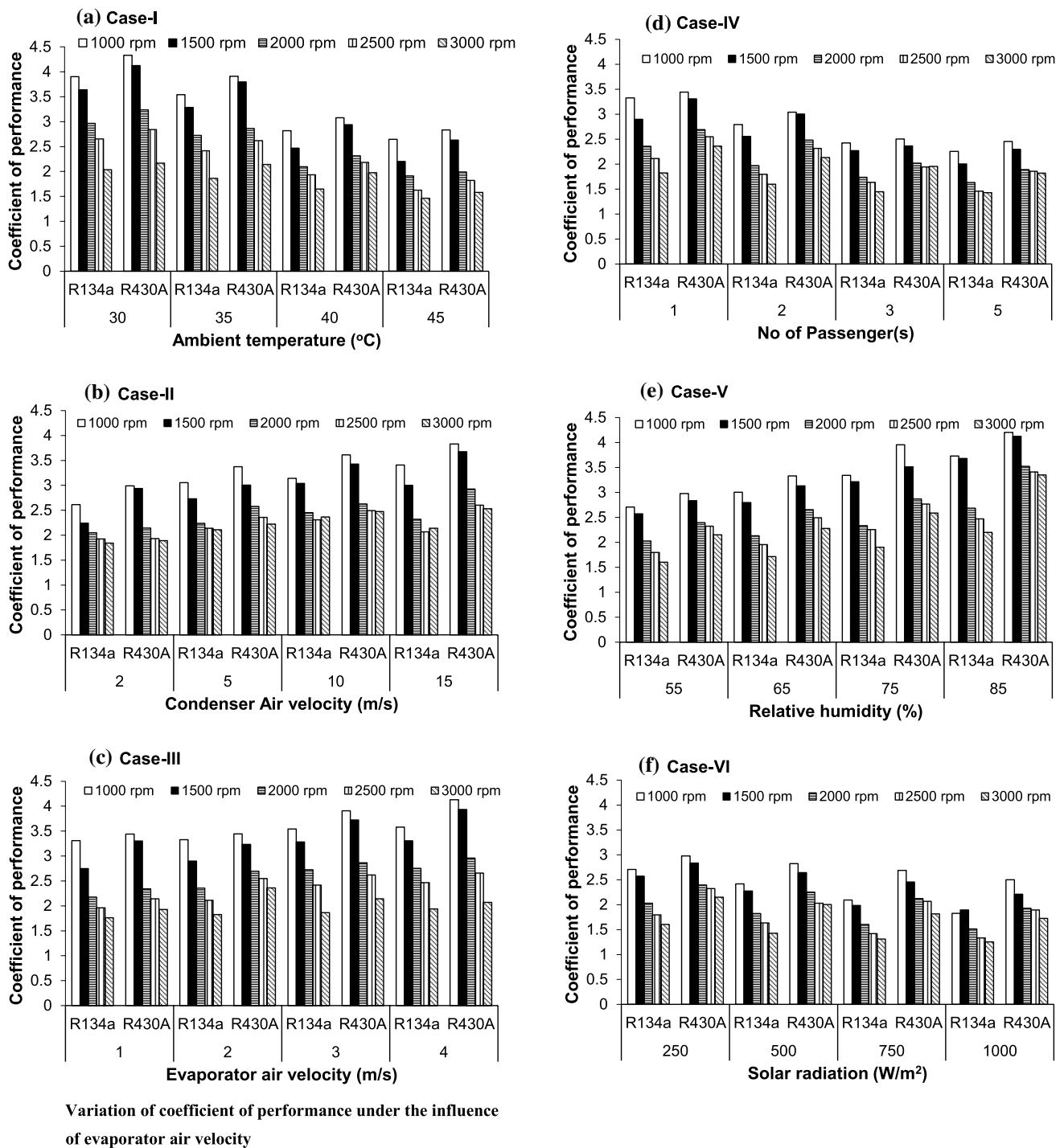
Fig. 7 Variation of coefficient of performance **a** influence of ambient temperature; **b** influence of condenser air velocity; **c** influence of evaporator air velocity; **d** influence of passenger load; **e** influence of cabin RH; and **f** influence of solar radiation

by 25–27% and 23–26%, respectively, under the influence of an increase in cabin RH from 55 to 85%. The maximum COP of about 3.7 and 4.0 was observed in VAC systems using R134a and R430A, respectively, at the maximum cabin RH of 85% and maximum compressor angular velocity of 1000 rpm. The COP variations under the effects of an

increase in solar radiation are depicted in Fig. 7f (Case-VI). It is noticed that the COPs of VAC systems using R134a and R430A refrigerants were reduced by 22–33% and 11–29%, respectively, under the effect of rising in solar radiation from 250 to 1000 W/m^2. The maximum COPs of 2.7 and 3.0 were noted in VAC systems using R134a and R430A, respectively,

at lean solar radiation of 250 W/m^2 and compressor angular velocity of 1000 rpm. The R430A has 7–12% improved COP due to its good thermodynamic and thermophysical characteristics compared to R134a [24]. The R430A has improved COP in VAC systems without major modifications. However, the R1234yf needs an internal heat exchanger to improve VAC systems' performance using R134a [11].

4.5 Exergy destructions

The variations of exergy destructions observed in VAC systems using R134a and R430A are compared in Fig. 8. The exergy destructions were increased significantly with an increase in compressor angular velocity due to an increase in associated irreversibility in VAC systems. As illustrated in Fig. 8a (Case-I), the exergy destructions in VAC systems
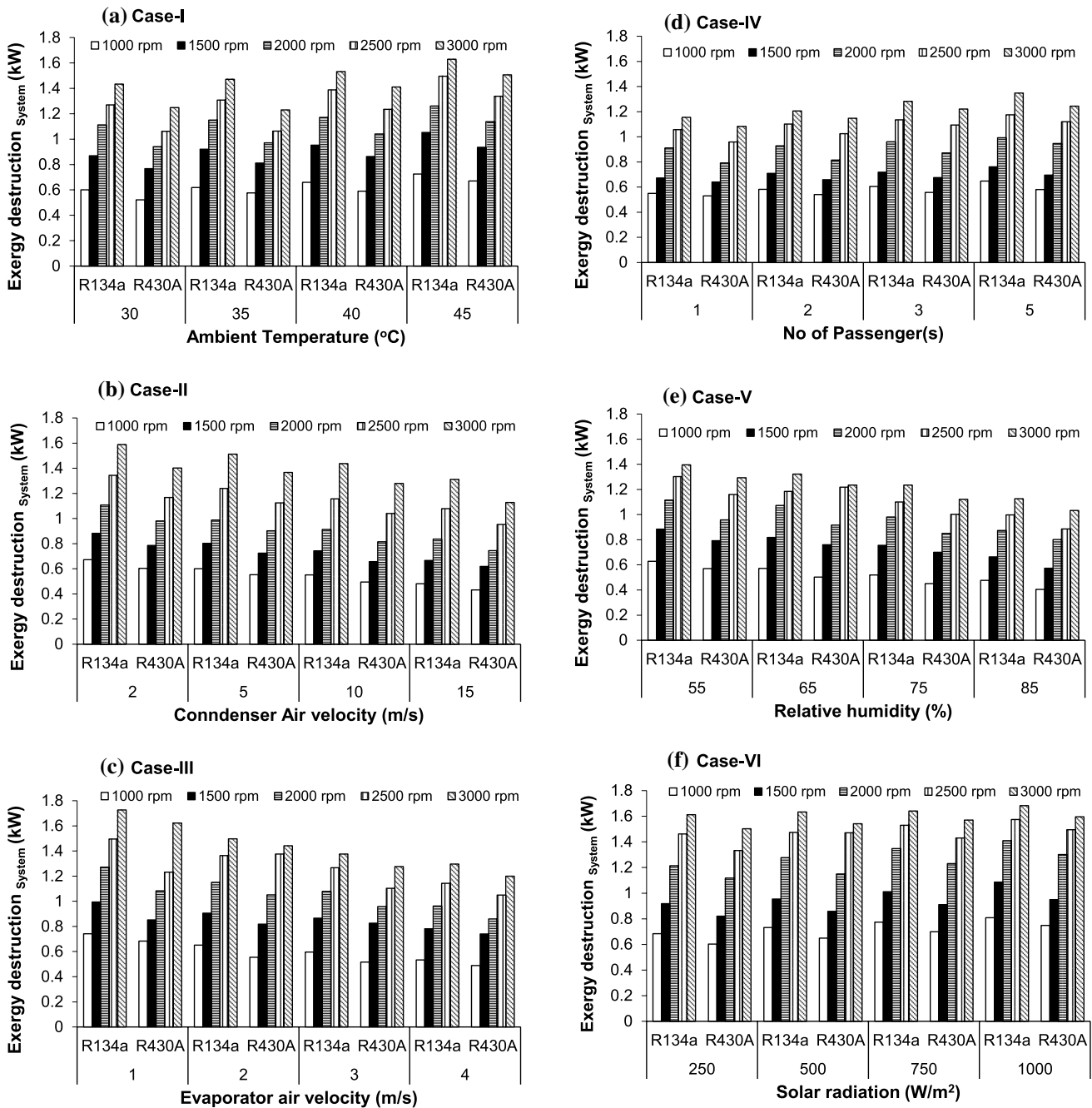

Fig. 8 Variation of exergy destruction **a** influence of ambient temperature; **b** influence of condenser air velocity; **c** influence of evaporator air velocity; **d** influence of passenger load; **e** influence of cabin RH; and **f** influence of solar radiation

using R134a and R430A were increased by 21–28% and 13–21%, with a rise in ambient temperatures from 30 to 45 °C. The maximum exergy destructions in VAC systems using R134a and R430A were observed as 1.62 and 1.50 kW, respectively, at a compressor angular velocity of 3000 rpm and an ambient temperature of 45 °C. In Fig. 8b (Case-II), the system exergy destructions under condenser air velocity influence are illustrated. The results revealed that the system exergy destructions were reduced by 21–40% and 23–40% in R134a and R430A VAC systems, respectively, under the influence of an increase in air velocity approaching the condenser from 2 to 15 m/s. An increase in condenser air velocity has improved condenser heat rejection. The highest exergy destructions of about 1.58 and 1.40 kW were observed in VAC systems using R134a and R430A, respectively, at the compressor angular velocity of 3000 rpm and a condenser air velocity of 2 m/s. The highest exergy destructions of 1.58 kW in R134a and 1.40 kW in R430A were observed at compressor angular velocity and condenser air velocity of 3000 rpm and 2 m/s, respectively.

The effects of evaporator air velocity on system exergy destructions are compared in Fig. 8c (Case-III). The exergy destructions observed in VAC systems using R134a and R430A were reduced by 27–39% and 15–40% with an increase in evaporator air velocity from 1 to 4 m/s. The highest exergy destructions of 1.72 kW and 1.62 kW were noticed in VAC systems using R134a and R430A at evaporator air velocity and compressor angular velocity of 1 m/s and 3000 rpm, respectively. The influences of passenger load capacity on system exergy destructions are compared in Fig. 8d (Case-IV). The results revealed that the exergy destructions were increased by 8–15% and 8–16% in VAC systems using R134a and R430A, respectively, increasing the passenger load capacity from 1 to 5. The VAC systems using R134a and R430A have the maximum system exergy destructions of 1.34 and 1.24 kW, respectively, for the passenger load capacity of 5 and compressor angular velocity of 3000 rpm [25].

The exergy destructions observed in the VAC system under the influence of cabin RH are illustrated in Fig. 8e (Case-V). The exergy destructions in VAC systems using R134a and R430A were reduced by 24–33% and 19–34%, respectively, under increasing cabin RH from 55 to 85%. The highest exergy destructions of 1.39 and 1.29 kW were observed in VAC systems using R134a and R430A, respectively, at compressor angular velocity of 3000 rpm and cabin RH of 55%. The variations of system exergy destructions under the influence of solar radiation are shown in Fig. 8f (Case-VI). The rise in solar radiation from 250 to 1000 W/m^2 has increased the exergy destructions by 4–15% and 6–19% in VAC systems using R134a and R430A, respectively. The highest exergy destructions of 1.68 and 1.59 kW were noticed in VAC systems using R134a and R430A,

respectively, at the maximum solar radiation of 1000 W/m^2 and maximum compressor angular velocity of 3000 rpm. The VAC systems using R430A have 5–12% lower exergy destructions because of its good thermophysical and thermodynamic characteristics than R134a [25]. The compressor of VAC systems using R134a and R430A was identified as the most ineffective component due to its more exergy destruction. Therefore, the operating parameters of the compressor are to be optimized. The expansion valve of VAC systems has minimum exergy destructions due to the absence of work and heat interactions.

4.6 Exergy efficiency

The variations of system exergy efficiencies of a VAC are depicted in Fig. 9. The exergy efficiencies of VAC systems were dropped under the influence of increasing compressor angular velocity. The exergy efficiency variations under the influence of ambient temperatures are depicted in Fig. 9a (Case-I). The rising ambient temperatures from 30 to 45 °C have dropped the exergy efficiencies by 26–35% in R134a and 23–34% in R430A VAC systems. The maximum exergy efficiencies in VAC systems using R134a and R430A were observed to be about 47% and 58%, respectively, at the compressor angular velocity of 1000 rpm and the ambient temperature at 30 °C. The effects of condenser air velocity on exergy efficiency variations are depicted in Fig. 9b (Case-II). The raising condenser air velocity from 2 to 15 m/s has dropped the exergy efficiencies of VAC systems using R134a and R430A by 5–15% and 20–30%, respectively. The highest exergy efficiencies were noticed in VAC systems using R134a and R430A by 42% and 51%, respectively at compressor angular velocity of 1000 rpm and condenser air velocity of 15 m/s.

In Fig. 9c (Case-III), the influence of evaporator air velocity on exergy efficiencies of VAC systems is compared. The results revealed that the exergy efficiencies of VAC systems using R134a and R430A were increased by 16–27% and 14–29%, respectively, under the influence of increasing evaporator air velocity from 1 to 4 m/s. The maximum exergy efficiencies of about 46% and 57% were observed for R134a and R430A at evaporator air velocity and compressor angular velocity of 4 m/s and 3000 rpm. The influences of passenger load capacity on exergy efficiency of a VAC are depicted in Fig. 9d (Case-IV). It is observed that the exergy efficiency was dropped by 11–19% and 9–25% in R134a and R430A systems, respectively, with an increase in passenger load capacity from 1 to 5. The maximum exergy efficiency of 38% and 43% was noticed in VAC systems using R134a and R430A, respectively, when the compressor angular velocity was maintained at 1000 rpm with the single passenger load capacity.
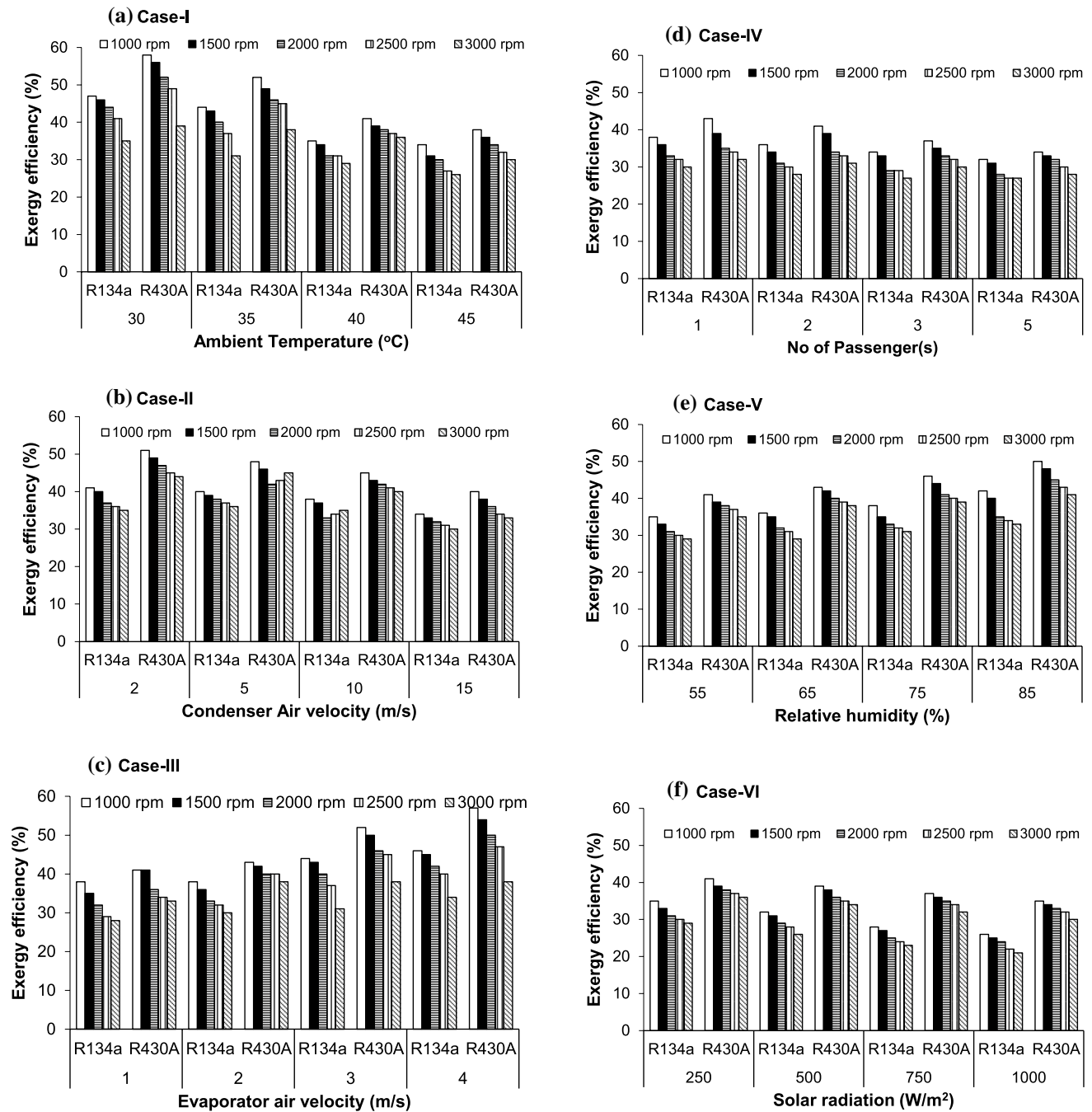
Fig. 9 Variation of exergy efficiency **a** influence of ambient temperature; **b** influence of condenser air velocity; **c** influence of evaporator air velocity; **d** influence of passenger load; **e** influence of cabin RH; and **f** influence of solar radiation

The exergy efficiency variations of VAC systems under the effect of cabin RH are shown in Fig. 9e (Case-V). The exergy efficiencies were raised by 12–18% and 14–19% in VAC systems using R134a and R430A, respectively, under increasing cabin RH from 55 to 85%. The maximum exergy efficiency of about 42% and 50% were noticed in VAC systems using R134a and R430A, respectively, at the compressor angular velocity of 1000 rpm and cabin

RH of 85%. The variations of exergy efficiency under the effects of solar radiation are depicted in Fig. 9f (Case-VI). The exergy efficiencies of VAC systems using R134a and R430A were reduced by 28–39% and 13–21%, respectively, with a rise in solar radiation from 250 to 1000 W/m^2. The maximum exergy efficiencies of about 35% and 41% were noticed in VAC systems using R134a and R430A at the compressor angular velocity of 1000 rpm

and solar radiation of 250 W/m^2, respectively. The exergy efficiency of VAC systems using R430A was 5–10% higher owing to good thermodynamic and thermophysical characteristics compared to R134a [25].

4.7 Total equivalent global warming impact

Ambient temperature has the major influence on TEGWI of VAC systems. The TEGWI of VAC systems using petrol, diesel and LPG as fuels under the influences of ambient temperatures and compressor angular velocities is depicted in Fig. 10. It is noted that the R430A has 30–35%, 32–40% and 44–50% lower TEGWI compared to R134a for the vehicles using diesel, petrol and LPG fuels, respectively. The lower TEGWI was observed with R430A due to lower GWP with improved energy performance than R134a [25].

5 Conclusion

The thermodynamic performances of VAC systems using R134a and R430A were experimentally assessed, and the following major conclusions are drawn:

- The thermophysical and thermodynamic characteristics of R430A were observed to be better than R134a. The quantity of R430A was reduced by about 50% because of its lesser liquid density than R134a.
- The VAC systems using R430A have a higher discharge temperature in the compressor than R134a by 2–6 °C. The lubricants used in R134a compressors are stable at 6 °C higher operating temperatures, making sure the compressor reliability.
- The R430A has a 5–10% higher refrigeration effect with 4–11% lesser compressor power consumption due to its positive thermophysical characteristics than R134a.
- The R430A has a 7–12% higher COP than R134a for the considered range of test conditions.
- The exergy destructions in VAC systems using R430A are 5–12% lesser because of its good thermophysical characteristics than R134a. The compressor was identified as the most ineffective component in the system due to more irreversibility.
- The TEGWI of a VAC system using R430A was found to be 30–35%, 32–40% and 44–50% lower than R134a because of its improved COP and lower GWP for the vehicles using diesel, petrol and LPG as fuels, respectively.
- The compressor speed is the major influencing parameter affecting the performance of a VAC, followed by condenser air velocity, ambient temperature, solar irradiation and passenger seating capacity. The relative humidity and
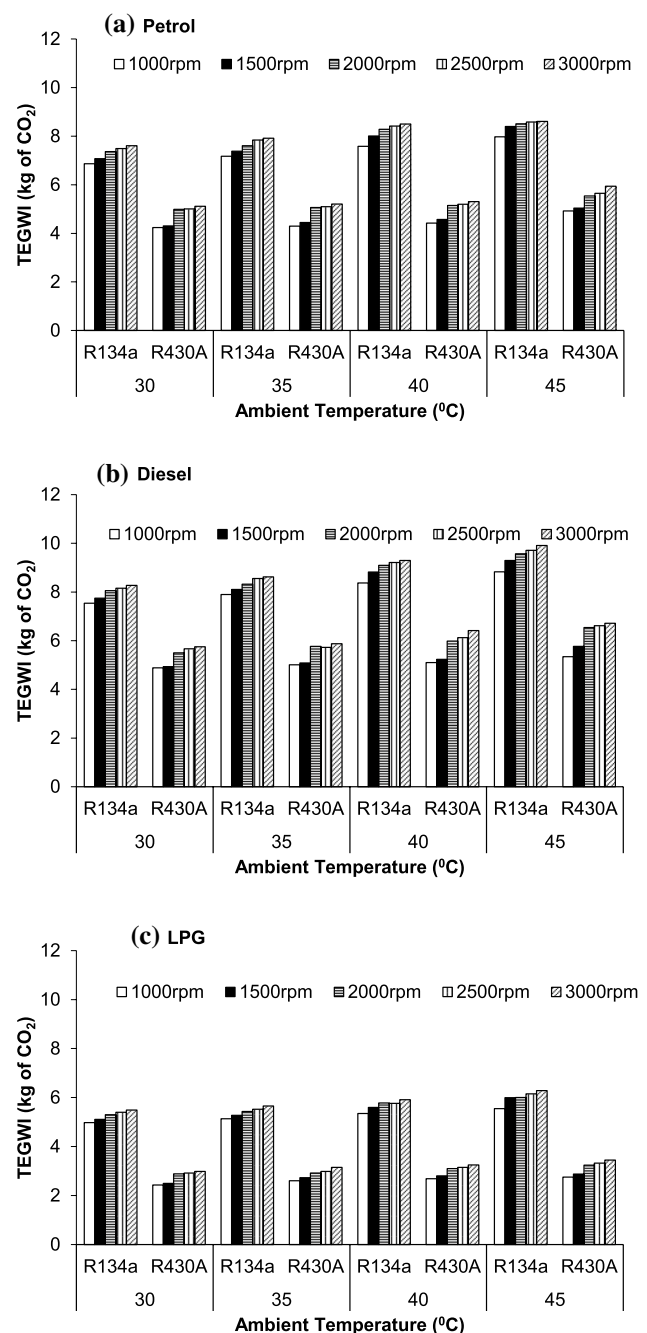

Fig. 10 Total equivalent warming impact (TEWI) of VAC using R134a and R430A under the influence of ambient temperature

evaporator air velocity have less influence on the thermodynamic performance of a VAC.

The outcome of this paper confirmed that R430A could be used as an environment-friendly replacement to phase-out R134a in VAC systems.

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