



Thermodynamic performance of automobile air conditioners working with R430A as a drop-in substitute to R134a

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Abstract

The refrigerant R134a is to be phasing out soon in automobile air conditioning applications due to its high global warming potential of 1430. Hence, it is essential to identify a sustainable alternative refrigerant to phase out R134a in automobile air conditioners. This paper presents the experimental thermodynamic performance of R430A (composed of R152a and R600a, in the ratio of 76:24, by mass) as a drop-in substitute to replace R134a in automobile air conditioners. The experiments were carried out in an automobile air conditioner test setup equipped with a variable frequency drive electrical motor. During experimentation, the ambient temperature and ambient relative humidity were maintained at 35 ± 1 °C and $65 \pm 5\%$, respectively. The compressor speed was varied in the range between 1000 and 3000 rpm. The results showed that the coefficient of performance of an automobile air conditioner working with R430A was found to be 12–20% higher with 6–11% reduced compressor power consumption when compared to R134a. The R430A has 2–6 °C higher compressor discharge temperature when compared to R134a. The physical stability of the lubricant used in the compressor was retained while operating with R430A. The maximum exergy destruction occurs in the compressor (0.28 kW for R134a and 0.24 kW for R430A) followed by evaporator (0.16 kW for R134a and 0.14 kW for R430A), condenser (0.14 for R134a and 0.12 kW for R430A) and expansion valve (0.043 kW for R134a and 0.039 kW for R430A) at a compressor speed of 1000 ± 10 rpm. The exergy destruction of the system operating with R430A was found to be 12–28% lower when compared to R134a systems due to its favorable thermo-physical properties. The total equivalent warming impact of R430A was found to be lower when compared to R134a by about 47.3%, 35% and 32.4% for LPG, petrol and diesel vehicles, respectively. The results confirmed that R430A is a good drop-in substitute to replace R134a in existing automobile air conditioning systems.

Keywords Automobile air conditioners · Thermodynamic performance · R430A · R134a · Global warming

Abbreviations

COP	Coefficient of performance
GWP	Global warming potential
HC	Hydrocarbon
HFC	Hydrofluorocarbon
LPG	Liquefied petroleum gas

VFD Variable frequency drive

List of symbols

A	Area (m^2)
ex	Specific exergy ($kJ\ kg^{-1}\ K^{-1}$)
\dot{Ex}	Exergy rate ($kJ\ K^{-1}$)
E	Energy consumption per day (in $kW\ h\ year^{-1}$)
h	Specific enthalpy ($kJ\ kg^{-1}$)
L	Leakage rate in the system ($kg\ year^{-1}$)
\dot{m}	Mass flow rate ($kg\ s^{-1}$)
m	Mass of refrigerant (kg)
N	Life of the system (years)
n	Operating time per day
q	Quantity
R	Given function

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s	Specific entropy (kJ kg ⁻¹ K ⁻¹)
T	Temperature (°C)
w_r	Total uncertainty
W	Work (W)
w_1, w_2, \dots, w_n	Uncertainty in the independent variables
x_1, x_2, \dots, x_n	Independent variables

Greeks

α	Refrigerant recycling factor
β	Carbon-dioxide emission factor (assumed as 0.9)
ρ	Density (kg m ⁻³)
η	Efficiency (%)

Subscripts

0	Dead state
1	Compressor suction
2	Compressor discharge
3	Condenser outlet
4	Expansion valve outlet
cond	Condenser
comp	Compressor
dest,comp	Destruction in compressor
dest,cond	Destruction in condenser
dest,exp.val	Destruction in expansion valve
dest, evap	Destruction in evaporator
ele	Electrical
evap	Evaporator
ex	Exergetic
f	Fuel
gen	Generation
r	Refrigerant
vol	Volumetric

Introduction

More than 195 countries (including India) have signed under Kigali agreement to reduce the consumption of halogenated refrigerants with GWP more than 150. The automobile air conditioning manufacturers are looking for energy-efficient and environment-friendly alternative to phase out R134a. The automobile air conditioning systems have major share in HFC refrigerant emissions due to frequent refrigerant leakage caused by poor road conditions [1]. The consumption of R134a in automobile air conditioning systems can be reduced by phasing out during its servicing. Hence, it is essential to identify an energy-efficient and environment-friendly alternative to replace R134a in servicing sectors. The possible environment-friendly options for R134a reported in open literature are: HFC refrigerants with low global warming R152a, HFO

refrigerants (such as R1234yf and R1234ze) and hydrocarbon mixed refrigerants.

Ghodbane [2] investigated the performance of R152a and hydrocarbon refrigerants (such as R290, R600a and RC270). It was reported that R152a is a good option for replacing R134a in automobile air conditioners due to its higher energy efficiency and lower global warming potential. Similarly, Cabello et al. [3] reported the performance of a R134a refrigeration system retrofitted with R152a as drop-in replacement. It was reported that R152a has 13% improved COP with 10% loss in refrigeration capacity. Mohanraj et al. [4] reported that R134a has about 7% lower total equivalent warming impact due to its good energy efficiency and low global warming potential. However, the R152a is having 5 °C higher compressor discharge temperature when compared to R134a. Moreover, the pure hydrocarbon refrigerants are not possible to use as a drop-in substitute to R134a due to its mismatch in volumetric cooling capacity and operating pressure. The hydrocarbon refrigerant mixtures (composed of R290 and R600a) are energy efficient and environment friendly when compared to R134a [5, 6]. However, the hydrocarbon refrigerant mixtures are zeotropic in nature and it is not possible to be maintaining the composition of the refrigerant mixtures under leakage conditions [7]. Hence, hydrocarbon refrigerant mixtures are not preferred in automobile air conditioners due to its zeotropic behavior, which leads to composition shift [8].

During last decade, the HFO refrigerants such as R1234yf and R1234ze were identified as good drop-in substitutes to replace R134a in compression-based refrigeration systems [9, 10]. Zhao et al. [11] studied the performance of an automobile air conditioning system using micro-channel parallel flow condenser working with R134a and R1234yf. The mass-charge of R1234yf was reduced by about 5% when compared to R134a. The cooling capacity and COP of the system working with R1234yf were found to be about 12.4% and 9% lower when compared to R134a due to difference in thermo-physical properties. In another work, Zilio et al. [12] numerically investigated the performance of an automobile air conditioning system working with R134a and R1234yf. It was reported that R1234yf has lower cooling capacity and lower COP compared to R134a. However, the performance of the system was enhanced by modifying the area of the condenser and evaporator. Similarly, the performance of an automobile air conditioning system working with R134a and R1234yf was investigated by Lee and Jung [13]. It was reported that the COP and evaporator cooling capacity of an automobile air conditioning system using R1234yf were reduced by about 2.7% and 4.0% when compared to R134a. In another work, Qi [14] experimentally investigated the performance of an automobile air conditioning

system working with R1234yf as an alternative to R134a using micro-channel parallel flow evaporator. It was reported with 6.5% improved evaporator cooling capacity. In a similar investigation, the performance characteristics of an automobile air conditioning system working with R134a and R1234yf were compared by Cho et al. [15]. It was reported that R1234yf has 4% lower compressor power consumption and 7% lower cooling capacity. The performance of the system was improved by installing an internal heat exchanger. Recently, Meng et al. [16] investigated the performance of an automobile air conditioning system working with R134a and the mixture composed of R1234yf and R134a in the ratio of 89:11, by mass. It was reported that the new refrigerant mixture has lower COP by about 4–9% in cooling mode and 4–16% in heating mode.

However, the HFO refrigerants are unsaturated hydrofluorocarbons, which breaks down in the atmosphere and forms tri-fluoro acetic acid (TFA) due to the presence of fluorine atoms [17]. The environmental impacts of HFO refrigerants (R1234yf and R1234ze) are more significant than R134a. The accumulation of TFA in atmosphere will affect the aquatic system. Hence, HFO refrigerants will not sustain in the market due to its environmental issues. At present, R430A (the binary refrigerant mixture composed of R152a and R600a, in the ratio of 76:24, by mass) was reported as a good drop-in substitute for R134a with low global warming potential of about 107 [18–20]. Moreover, the R430A is compatible with both the lubricants (mineral oil and synthetic lubricants) used in automobile air conditioners. Hence, the lubricant replacement is not required for R430A retrofitting.

The major factors influencing the irreversibility in automobile air conditioners are thermal-physical properties (such as viscosity, density, thermal conductivity and specific heat) and flow properties (such as friction and pressure drop of refrigerants). It is essential to quantify the irreversibilities in each component of an automobile air conditioning system. Exergy analysis is proved to be a good tool to quantify the losses and identifying the inefficient components in automobile air conditioning systems [21–24]. In a related work, the performances of an automobile air conditioning system working with R134a and R1234yf as working fluids were investigated [21]. It was reported that the maximum exergy destruction was occurred in compressor followed by condenser, expansion valve and evaporator by contributing 53%, 21%, 15% and 11%, respectively. The exergy destruction of R134a was found to be about 35% lower when compared to R134a. Similarly, Cho and Park [22] investigated the exergy performance of an automotive air conditioning system using R134a and R1234yf at different compressor speeds. It was reported that R1234yf has lower exergy efficiency in the range between 3.4 and 4.6% at all the compressor speeds when

compared to R134a. In another work, Li et al. [23] investigated the thermodynamic performance of automobile air conditioning systems working with environment-friendly alternative refrigerants such as R152a and R290. It was reported that COP of R152a was found to be about 5% and 10% higher when compared to R134a in highway driving and idling conditions, respectively. Moreover, the COP of R290 was improved by about 8% in highway driving and dropped by about 15% in idling conditions. The exergy destruction of R152a and R290 was reduced by about 9.6% and 14.4% when compared to R134a in highway driving conditions. They also identified the compressor as the most inefficient component in the system, which needs more attention to optimize the operating parameters. Prabhakaran and Lal [24] also confirmed that the maximum exergy destruction occurs in the compressor of an automobile air conditioning system contributing about 60–70%. The enhancement of exergy efficiency of each component of the system is essential to achieve energy efficient and environment friendly. Further, the performance of automobile air conditioning system was investigated using micro-channel evaporator and integrated receiver-dryer condenser [25]. It was reported that the condensate retention of the modified system was reduced by about 17–31% when compared to the base line evaporator.

The cited literature review on environment-friendly alternative refrigerants for automobile air conditioners confirms that there is no specific work reported on the use of R430A for automobile air conditioners. Hence, a research attempt has been made in this work to explore the possibility of using R430A in automobile air conditioners at different compressor speeds by maintaining the ambient temperature and ambient relative humidity at 35 ± 1 °C and $65 \pm 5\%$, respectively.

Characteristics of R134a and R430A

In Fig. 1, the vapor pressures of R134a and R430A are compared. The vapor pressure of R430A was found to be 8–15% lower when compared to R134a. Hence, all the components of the system are physically stable, while retrofitting with R430A. The variation of latent heat of R134a and R430A is compared in Fig. 2. The latent heat of 430A was found to be about 36% higher when compared to 134a, which results in higher refrigeration capacity and reduced the compressor running time. The liquid density of R134a and R430A is compared in Fig. 3. It is observed that the liquid density of R430A was found to be about 37% lower by when compared to R134a. The lower liquid density of R430A has reduced the refrigerant mass-charge by about 50% (175 g) when compared to R134a. The liquid viscosity of R134a and R430A is illustrated in Fig. 4. It is

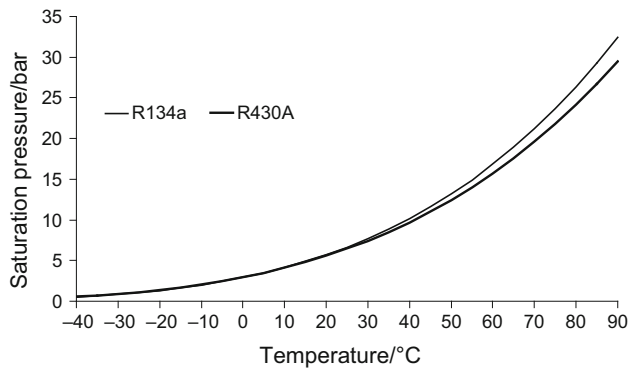


Fig. 1 Saturation pressure of R134a and R430A against temperature

observed that the liquid viscosity of R430A was found to be 20–30% lower when compared to R134a, which will reduce the friction resistance in the tubes of the condenser and increase the condenser heat transfer coefficient. The lower liquid density and liquid viscosity has reduced the irreversibility in the systems and improved the heat transfer coefficients. The thermodynamic properties of R134a and R430A are compared in Table 1. It is observed that R430A has better thermodynamic properties when compared to R134a. The R430A is a flammable mixture with ASHRAE flammable index of A3. The charge requirement of R430A was found to be about 175 grams, which will not form a flammable mixture inside the vehicle cabin in the case of leakage. However, the charge requirement of R1234yf (presently identified as a potential alternative to R134a in automobile air conditioners) is 95% of R134a, which has ASHRAE flammable index of A2L. The flammable risk of R430A was found to be lower than that of R1234yf in terms of refrigerant charge requirements. The GWP of R430A was found to be 107, which is significantly lower when compared to R134a. The presence of R600a in the mixture is compatible with mineral oil as well as synthetic oil. Hence, the same lubricant can be retained in the compressor, while retrofitting. The comparison of refrigerant properties confirmed that R430A has fulfilled the

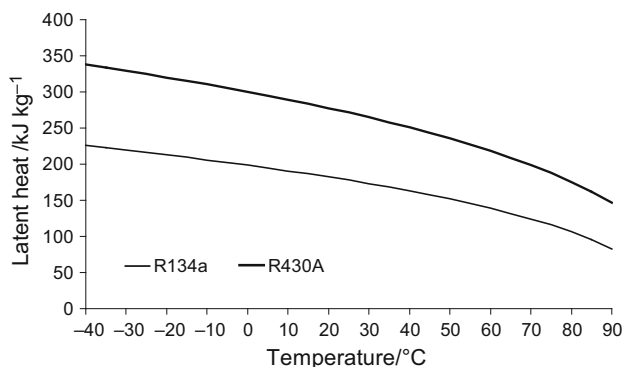


Fig. 2 Latent heat of R134a and R430A against temperature

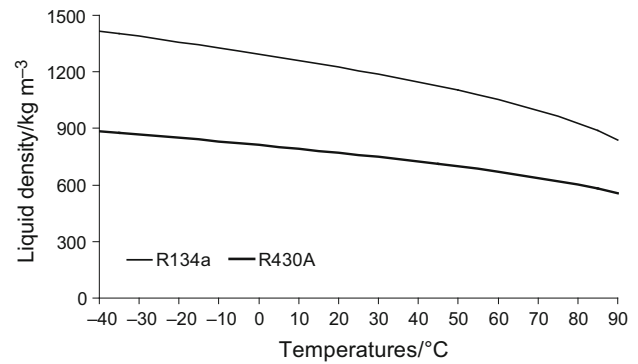


Fig. 3 Liquid density of R134a and R430A against temperature

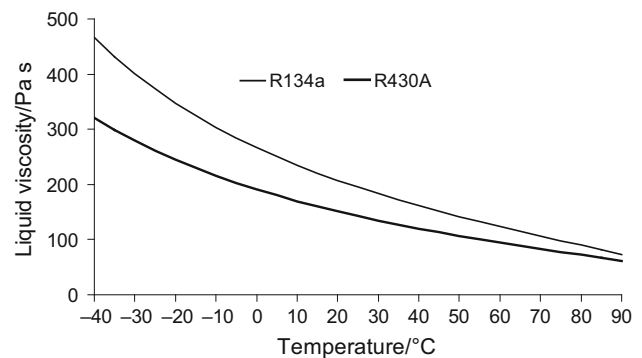


Fig. 4 Liquid viscosity of R134a and R430A against temperature

major requirements for phasing out the R134a in automobile air conditioners.

Experiments

The details of automobile air conditioner experimental setup, instrumentation and experimental procedure are described in this section.

Experimental

The schematic illustration of an experimental setup and its photographic views are illustrated in Fig. 5a, b, respectively. The experimental setup consists of a sedan-type passenger car compartment integrated with a compression-based air conditioning cycle. The air conditioning cycle consists of basic components such as compressor, condenser, expansion valve and an evaporator. In addition to the basic components, the accessories such as liquid receiver, sight glass and refrigerant liquid drier have been placed in the liquid line. The compressor was connected to an electrical motor of rated power input of 2.2 kW. The speed of the compressor was varied by a variable frequency drive. The air velocity approaching the condenser of an automobile air conditioner was varied using a 0.75 kW

Table 1 Properties of R134a and R430A

Refrigerant	Boiling point/°C	Molecular weight	Critical temperature/°C	Critical pressure bar	Flammable index	Ozone depletion potential	Global warming potential
R134a	− 26.4	102.03	101.1	4.06	A1	0	1430
R430A	− 21.4	64.14	118.43	4.3	A3	0	107

blower connected with variable frequency drive. An air heater with 2 kW rated input power was placed in the path of condenser air flow to vary the air temperature passing over the condenser. The occupancy load inside the car cabin was varied using a 2 kW resistance air heater. The humidity inside the vehicle cabin was controlled by a portable humidifier.

Instrumentation

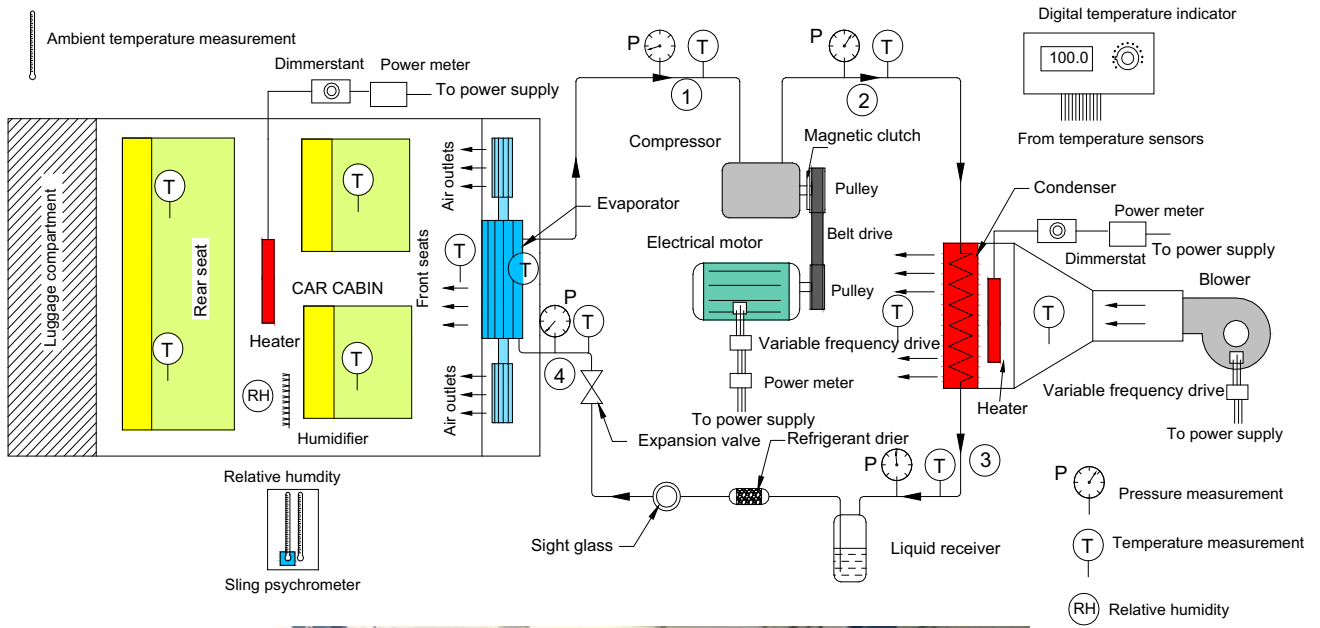
The experimental setup is equipped with four Bourdon tube pressure gauges with an accuracy of $\pm 2\%$ in the refrigerant circuit at compressor suction and discharge, condenser outlet and expansion valve outlet to measure the refrigerant pressure at respective points. In addition, twelve temperature sensors (RTD-Pt100) with an accuracy of $\pm 0.2\text{ }^\circ\text{C}$ were located in the refrigerant flow line at compressor suction and discharge, condenser outlet, expansion valve outlet, air temperature at condenser inlet and outlet, cabin temperatures at four different locations, air temperature at the evaporator inlet and outlet (as shown in Fig. 5a). The ambient temperature was measured using precision thermometer with $\pm 0.2\text{ }^\circ\text{C}$ accuracy. Ambient humidity was measured using sling psychrometer. All the temperature sensors were connected to the digital temperature indicator with a resolution of $0.1\text{ }^\circ\text{C}$. The compressor speed was measured using a digital tachometer with an accuracy of $\pm 5\text{ rpm}$. The air mass flow rate passing over the condenser was measured by a vane type anemometer with an accuracy of $\pm 0.1\text{ m s}^{-1}$. The power input to the motor was measured using a power meter. Similarly, the energy consumption by resistance air heaters located inside the car cabin and in the path of condenser air flow were measured using separate Wattmeter with an accuracy of $\pm 1\text{ W}$. The quantity of refrigerant charges in the system was measured by a digital weighing machine with an accuracy of $\pm 1\text{ g}$. The detailed specifications of measuring instruments and its range of usage are given in Table 2.

Experimental procedure

Initially, the air conditioning system was charged with nitrogen gas and maintained the pressure around 15 bar pressure for the period of 24 h to identify the leaks in the

system. After the leak test, the nitrogen gas was flushed out to remove the impurities inside the system. Then, the refrigerant circuit was evacuated using vacuum pump. About 350 g of R134a was charged in the system according to the manufactures specifications. The experimental observations were made for the test conditions shown in Table 3 with constant ambient temperature at $35 \pm 1\text{ }^\circ\text{C}$ with relative humidity $65 \pm 5\%$. The evaporator placed inside the car cabin was loaded for a single passenger. In addition, the ambient infiltration through the wind screens are quantified and simulated using resistance air heaters placed inside the car cabin. The air velocity approaching the condenser was maintained constantly at $15 \pm 1\text{ m s}^{-1}$. Before start experimental observations, the doors of the car cabin were kept opened for the period of 4 h to attain the steady-state conditions with ambient. Then, the system was switched ON and allowed to run for the period of 30 min to attain steady-state conditions at a compressor speed of 1000 rpm before experimental observations. The time taken to attain the steady-state cabin temperature of about $22\text{ }^\circ\text{C}$ from the initial cabin temperature of about $35\text{ }^\circ\text{C}$ was observed to study the pull-down characteristics of an automobile air conditioning system. During experimental observations, the pressures at typical locations in the refrigerant circuit, temperatures at all the typical locations in the refrigerant circuit, inside the car cabin, air temperatures at evaporator inlet and outlet and ambient temperature were measured. The experimental observations have been repeated for other compressor speeds such as 1500, 2000, 2500 and 3000 rpm using variable frequency drive attached with the electrical motor. After completing the base line tests with R134a, the refrigerant was recovered from the system and charged with required quantity of R430A. The equivalent refrigerant quantity of R430A was calculated using liquid densities of two refrigerants at ambient temperature. The equivalent refrigerant quantity was estimated to be about 175 g. Then, the complete test procedures were repeated for the automobile air conditioner charged with R430A. Three trial experiments were conducted to check the consistently of experimental observations. The variations in experimental observations were found to be $\pm 3\%$ deviations. Moreover, uncertainty analysis has been made to quantify the errors in experimental measurements.

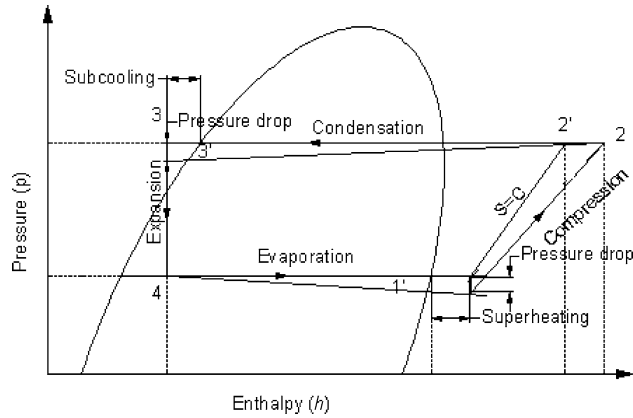
(a)



(b)



(c)



◀**Fig. 5 a** Schematic diagram of an automobile air conditioning system. **b** Photographic view of an experimental setup. **c** Pressure enthalpy diagram of compression cycle

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

The coefficient of performance (COP) of the automobile

Table 2 Instrumentation range and accuracy

Measured variable	Instrument	Range	Accuracy
Temperature	RTD-Pt100	0–199 °C	± 0.2 °C
Pressure	Bourdon Gauge	0–500 psi	± 5 psi
Air velocity	Anemometer	0.4/30 m s ⁻¹	± 3%
Compressor Speed	Photoelectric Tachometer	10–10,000 rpm	± 2%
Power Input	Wattmeter	Digital type	± 1 W

Table 3 Test conditions

Parameters	Unit	Range
Blower air velocity (condenser)	m s ⁻¹	15
Evaporator air velocity	m s ⁻¹	4.1
Solar irradiation	W m ⁻²	650
Compressor speed	rpm	1000, 1500, 2000, 2500 and 3000

Thermodynamic analysis of automobile air conditioning systems

The thermodynamic performance of automobile air conditioning systems were carried out to quantify the energy conversion of the system and also to quantify the losses in the system.

Energy analysis

The thermodynamic equations used for predicting the performance of an automobile air conditioner are presented in this section. The refrigerant mass flow rates through the system at various speeds were estimated using following equation [24]:

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

The volumetric efficiency and mechanical efficiency of the compressor were assumed as 0.85 and 0.8, respectively, based on the trial experiments. The mechanical and electrical power consumptions by the compressor are estimated by following relations [24]:

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

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

The efficiency of the motor and transmission is assumed as 0.95 and 0.9, respectively. The evaporator refrigeration capacity was estimated by following equation [24]

air conditioner was estimated using following equation [24]:

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

Amount of heat released in the condenser of an air conditioning system is given by following equation:

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

The energy and mass balances for steady-state process are given by:

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

and

$$\dot{m}_{r,1} = \dot{m}_{r,2} = \dot{m}_{r,3} = \dot{m}_{r,4} \tag{7b}$$

Exergy analysis

Exergy analysis is used to identify the inefficient components and to quantify the losses in each component of an automobile air conditioner. The pressure enthalpy diagram of a compression cycle is depicted in Fig. 5c. The following assumptions were made in the exergy analysis of an automobile air conditioner [27, 28]:

1. All the processes are assumed to be steady state.
2. The potential, kinetic and chemical effects are negligible.
3. Ambient conditions were considered as reference state.

The exergy performance of the system was quantified using a following equation [24]:

$$\dot{E}x_{dest} = T_0 s_{gen} \tag{8}$$

Exergy balance of the system is given by [24]:

$$\dot{E}x_{in} - \dot{E}x_{out} = \dot{E}x_{dest} \tag{9}$$

Specific exergy is given by:

$$ex = (h - h_0) - T_0(s - s_0) \tag{10}$$

The exergy rate is determined as [24]:

$$\dot{E}x = \dot{m}_r ex \tag{11}$$

The exergy destructions in compressor, condenser, expansion valve, evaporator and for the whole system are given by following equations [24]:

$$\dot{E}x_{dest,comp} = \dot{m}_r(h_1 - h_2) - \dot{m}_r T_0(s_1 - s_2) + W_{ele} \tag{12}$$

$$\dot{E}x_{dest,cond} = \dot{m}_r T_0(s_3 - s_2) + Q_{cond} \left(\frac{T_0}{T_{r,3}} \right) \tag{13}$$

$$\dot{E}x_{dest,exp.val} = \dot{m}_r T_0(s_4 - s_3) \tag{14}$$

$$\dot{E}x_{dest,evap} = \dot{m}_r T_0(s_1 - s_4) - Q_{evap} \left(\frac{T_0}{T_{r,1}} \right) \tag{15}$$

$$\dot{E}x_{dest,sys} = Ex_{dest,comp} + Ex_{dest,cond} + Ex_{dest,exp.val} + Ex_{dest,evap} \tag{16}$$

The relative exergy destruction is given by following relation [24]:

$$\dot{E}x_{relative} = \frac{Ex_{component}}{ED_{system}} \times 100 \tag{17}$$

The exergy efficiency of the system is given by following relation [29]:

$$\eta_{ex} = \frac{Q \left(1 - \frac{T_0}{T_{ref}} \right)}{W_{ele}} \tag{18}$$

Total equivalent warming impact

The environmental impacts caused by automobile air conditioning systems are due to direct emissions (refrigerant leakage during servicing, accidents and disposal of the equipment) and indirect emissions (during combustion of fuels to power the air conditioning equipment). The TEWI is an environmental impact indicator used to quantify the emissions (both direct and indirect) of air conditioning systems. It is given by the following equation [30]:

$$TEWI = GWP \times L \times N \times m + (GWP \times m \times (1 - \alpha) + \left(\frac{q_f}{10,000} \times \text{km/year} + \frac{q_f}{10,000 \times 100} \times m_{AC} \times \text{km/year} \right) \frac{COP_{reference}}{COP_{alternative}} \beta) \tag{19}$$

The TEWI of automobile air conditioning system was calculated using following assumptions:

1. Global warming potential of R134a and R430A are 1430 and 107, respectively.
2. Weight of automobile air conditioning systems and its accessories: 15 kg
3. Life of automobile air conditioning systems: 15 years
4. Mass of R134a and R430A charges in the system are 350 and 175 g, respectively.
5. Leakage rate is 20% every year (R134a: 70 g and R430A: 40 g).
6. Ambient temperature and relative humidity were maintained at 35 ± 1 °C and $65\% \pm 5\%$, respectively.
7. The COP of automobile air conditioning systems working with R134a (as reference working fluid) and R430A (as alternative working fluid) is evaluated using experimental test rig equipped with electrical motor.
8. The average fuel consumption observed in petrol, diesel and LPG fueled sedan model passenger car (Maruti Swift Dezire) operated with and without air conditioner is given in Table 4. In addition, the CO₂ emission factors for various fuels are given.

Uncertainty analysis

The uncertainties in experiments are estimated using following equation [26]:

$$w_r = \left[\left(\frac{\partial R}{\partial x_1} w_1 \right)^2 + \left(\frac{\partial R}{\partial x_2} w_2 \right)^2 + \dots + \left(\frac{\partial R}{\partial x_n} w_n \right)^2 \right]^{1/2} \tag{20}$$

Table 4 Fuel consumption by a sedan model passenger car (Maruti Swift Dezire)

Fuel	Petrol	Diesel	LPG
Without air conditioning (kmpl)	18.3	20.4	17.1
With air conditioning (kmpl)	16.4	18.1	15.8
Density (kg m ⁻³)	750	835	550
Carbon percentage	87%	86.2%	82.5%
CO ₂ emissions (kg of CO ₂ per liter)	2.31	2.68	1.64
CO ₂ emissions (kg of CO ₂ per km)	0.1408	0.1488	0.1037

In this experimentation, the compressor power consumption, speed of the compressor, temperature measurements at typical locations in refrigerant circuit and inside the vehicle cabin, refrigerant pressure at typical locations in refrigerant circuit were measured using Wattmeter, temperature sensors (RTD-PT100) and pressure gauges, respectively. The uncertainties in mass flow rate, refrigeration effect, compressor power consumption, COP, exergy destructions in compressor, condenser, expansion valve, evaporator are estimated to be $\pm 2.25\%$, $\pm 2.47\%$, $\pm 2.44\%$, $\pm 3.4\%$, $\pm 3.5\%$, $\pm 3.3\%$, $\pm 3.5\%$ and $\pm 3.5\%$, respectively.

Results and discussion

The thermodynamic performance of an automobile air conditioning system was experimentally investigated with R134a and R430A by maintaining the ambient temperature and ambient relative humidity at 35 ± 1 °C and $65 \pm 5\%$, respectively. The other parameters such as air velocity over the condenser face area, air velocity through the condenser and evaporator are maintained at 15 m s^{-1} and 4.1 m s^{-1} , respectively. The temperature and pressure at typical locations in the refrigerant circuit, temperature of air inside the vehicle cabin, air velocity passing through the condenser and compressor speed were measured experimentally. These measurements were used for evaluating the thermodynamic performance parameters such as COP, refrigeration effect, exergy destruction and relative exergy destruction of compressor, condenser, expansion valve and evaporator of the system. The thermodynamic performance results observed in the trial experiments are described in this section. Additionally, the behavior of R430A with lubricant is presented.

Experimental observations

The experimentally observed parameters such as compressor discharge temperature, compressor discharge pressure and refrigerant mass flow rate of R134a and R430A are compared in this section.

Compressor discharge temperature

The compressor discharge temperature is a major influencing factor affecting the compressor life. Hence, the compressor discharge temperature was measured experimentally for both the working fluids at different compressor speeds between 1000 rpm and 3000 rpm and compared in Fig. 6. It is observed that the compressor discharge temperature of R134a and R430A gets increased from

about 56 °C to 76 °C and from about 58 °C to 81 °C, respectively, with the increase in compressor speed from 1000 to 3000 rpm due to increase in refrigerant mass flow rate through the condenser. Moreover, it is observed that R430A has 2–6 °C higher compressor discharge temperature due to its higher specific heat ratio when compared to R134a. The compressor discharge temperature of R430A was found to be about 2–4 °C higher when compared to R134a in the case of domestic refrigerators [18–20]. The lubricant used in the compressor was physically stable, which can be ensured at elevated operating temperatures. Hence, the reliability of the compressor will not be affected while retrofitting with R430A.

Compressor discharge pressure

The compressor discharge pressures of R134a and R430A at different compressor speeds are compared in Fig. 7. It is observed that the compressor discharge pressures of R134a and R430A were increased by about 38% and 35%, respectively, with an increase in compressor speed from 1000 to 3000 rpm. The compressor discharge pressure of R430A was observed to be 7–10% lower when compared to R134a due to its lower vapor pressure and lower vapor density. The low operating pressure of R430A ensures that all the components of the system in high pressure side are under safe operating conditions.

Refrigerant mass flow rate

The refrigerant mass flow rates (using R134a and R430A) at different compressor speeds (range between 1000 and 3000 rpm) are compared Fig. 8. It is observed that the refrigerant mass flow rates of both the refrigerants get increased with the increase in compressor speed from 1000

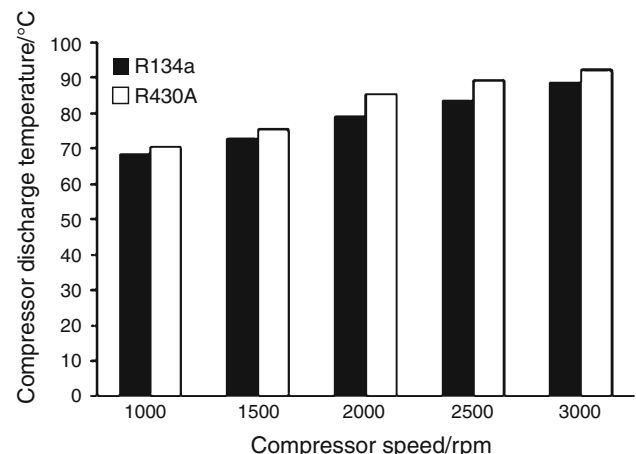


Fig. 6 Variation of compressor discharge temperature at different compressor speed

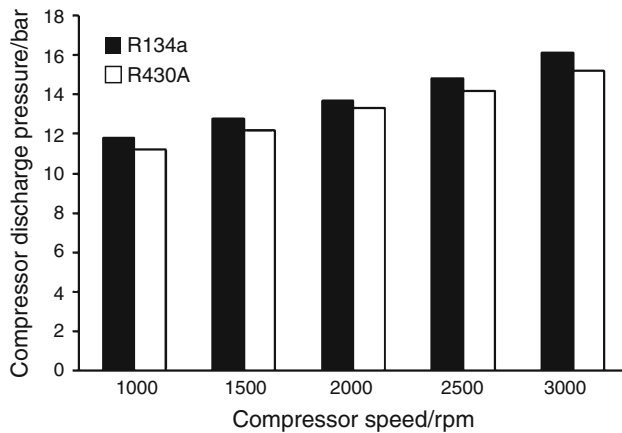


Fig. 7 Variation of compressor discharge pressure at different compressor speed

to 3000 rpm. The mass flow rates of R134a and R430A were increased by about 70% and 71%, respectively, with the increase in compressor speed from 1000 rpm to 3000 rpm. The mass flow rate of R430A was observed to be 37% to 41% lower when compared to R134a due to its lower charge requirement of R430A was 50% lower than the R134a. The mass flow rate of R134a observed in this work is similar to the earlier research work reported by Cho et al. [15] for R134a.

Energy performance

The energy performance parameters such as compressor power consumption, refrigeration capacity and COP of an automobile air conditioning system working with R134a and R430A as working fluids are described in this subsection.

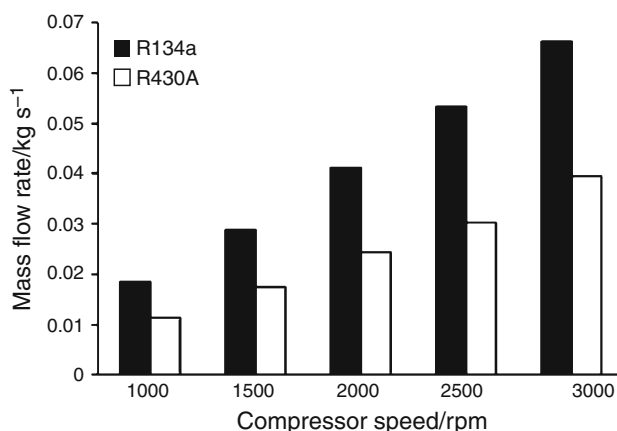


Fig. 8 Variation of refrigerant mass flow rate at different compressor speed

Pull-down time

The pull-down time characteristics of an automobile air conditioning system working with R134a and R430A to attain the final cabin temperature of about 22 °C from 35 °C are shown in Fig. 9. It is observed that the pull-down time required to attain the final steady-state cabin air temperature of about 22 °C using R134a and R430A was found to be about 630 s and 610 s, respectively, when the compressor is running at 3000 rpm. A reduction in pull-down time of about 3–6% was observed for R430A when compared to R134a due to its high latent heat of vaporization.

Compressor power consumption

The compressor power consumption of R134a and R430A at different compressor speeds in the range between 1000 and 3000 rpm is compared in Fig. 10. It is observed that the compressor power consumption of the system operating with R134a and R430A was increased by about 82% with the increase in compressor speed from 1000 to 3000 rpm. The compressor power consumption was increased from 0.4 to 2.2 kW and from 0.36 to 2 kW for R134a and R430A, respectively. In addition, it is observed that R430A has 6–11% lower compressor power consumption when compared to R134a due to its favorable thermo-physical properties such as lower refrigerant viscosity and lower density when compared to R134a. The presence of R600a in the mixture has enhanced the thermo-physical properties of the mixture [31]. Moreover, the operating pressure of R430A was found to be lower than that of R134a.

Refrigeration effect

The refrigeration effect (evaporator capacity) of an automobile air conditioning system using R134a and R430A is

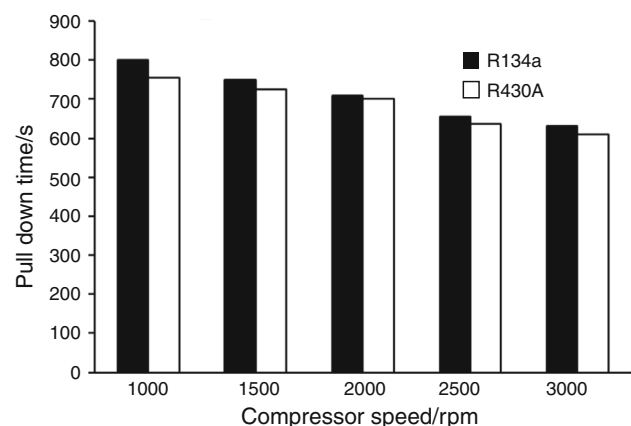


Fig. 9 Variation of pull-down time at different compressor speed

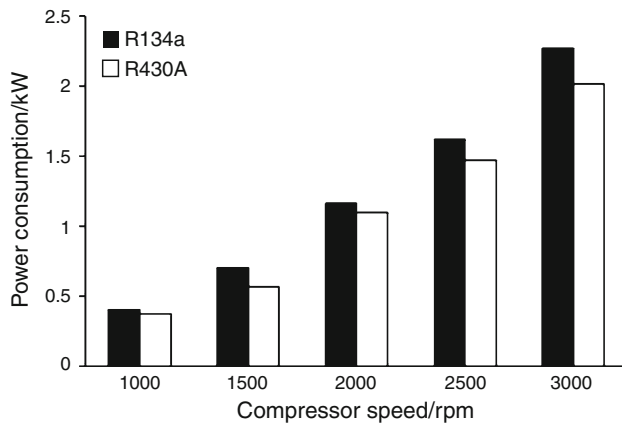


Fig. 10 Variation of power consumption at different compressor speed

compared in Fig. 11. It is observed that refrigeration effects (evaporator capacity) of R134a and R430A were increased by about 69% and 71%, respectively, with the increase in compressor speed from 1000 to 3000 rpm. The refrigeration effect of R430A was found to be 1–7% higher due to its favorable thermo-physical properties such as higher latent heat, higher thermal conductivity, lower viscosity and lower density when compared to R134a. The presence of hydrocarbon in the mixture has improved the thermo-physical properties of R430A when compared to R134a [31].

Coefficient of performance

The COP of an automobile air conditioner working with R430A as drop-in substitute to R134a is shown in Fig. 12. It is observed that COPs of the automobile air conditioning system using R134a and R430A were dropped by about 41% and 36%, respectively, with the increase in compressor speed from 1000 to 3000 rpm due to increase in compressor power consumption. The COP of the air

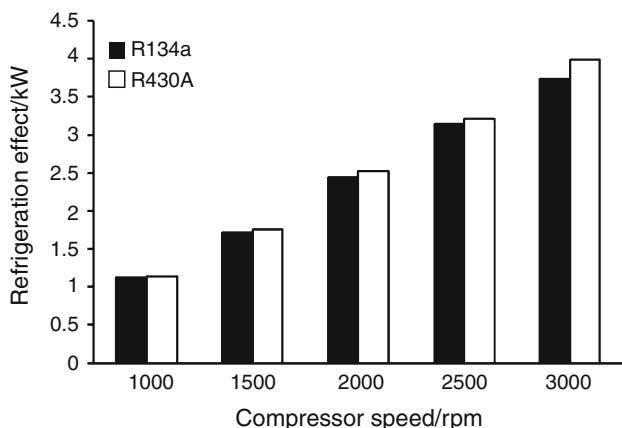


Fig. 11 Variation of refrigeration effect at different compressor speed

conditioner using R134a was dropped from 2.8 to 1.6, and the COP of the system using R430A was dropped from 3 to 1.9 with an increase in compressor speed from 1000 to 3000 rpm. The COP variations for R134a were found to be similar to the earlier work reported by Cho and Park [22]. Moreover, it is observed that R430A has 9–20% higher COP when compared to R134a due to its favorable thermo-physical properties such as lower density, lower viscosity, higher thermal conductivity and higher latent heat.

Exergy performance

The exergy performance of an automobile air conditioning system operating at different compressor speeds between 1000 and 3000 rpm is discussed in this subsection.

Exergy destruction in compressor

The exergy destructions of an automobile air conditioning system working with R134a and R430A at different compressor speeds between 1000 and 3000 rpm are shown in Fig. 13. The exergy destructions in the compressor were increased by about 43% with an increase in compressor speed from 1000 to 3000 rpm for both the refrigerants due to the increase in refrigerant operating pressures and compressor power consumption. The maximum exergy destructions of about 0.5 kW and 0.4 kW were observed for R134a and R430A, respectively. The exergy destructions in the compressor operating at different speeds ranging between 1000 and 3000 rpm while operating with 430A was found to be 12–30% lower when compared to R134a. The exergy performance variations observed in this work are found to be similar to the earlier reported investigations [21, 22]. The lower exergy destructions in compressor operating with R430A are due to its favorable

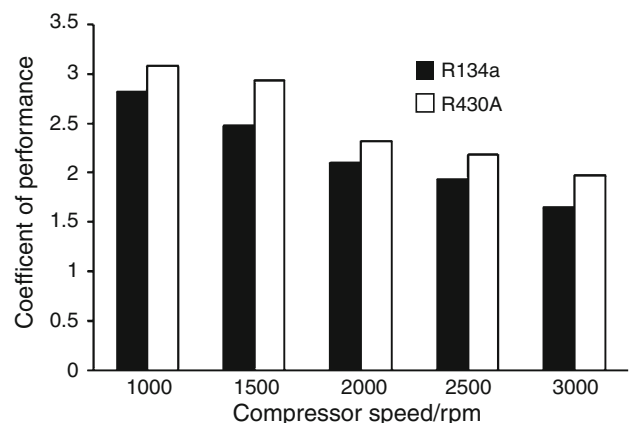


Fig. 12 Variation of coefficient of performance at different compressor speed

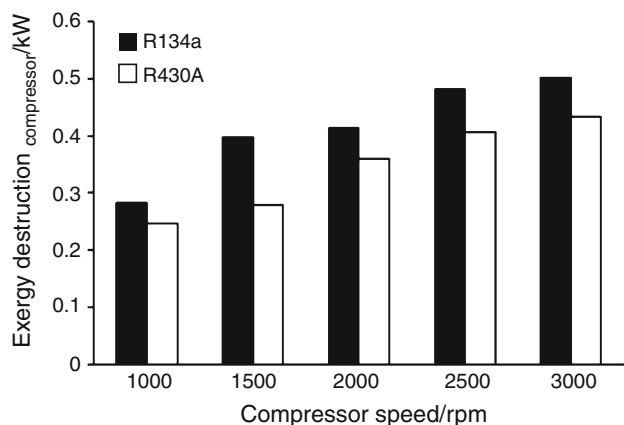


Fig. 13 Variation of exergy destruction of compressor at different compressor speed

thermo-physical properties such as lower density and lower viscosity.

Exergy destruction in condenser

Exergy destructions in the condenser operating between 1000 and 3000 rpm for both the refrigerants (R134a and R430A) are compared in Fig. 14. The maximum exergy destructions of about 0.42 kW and 0.4 kW were observed at the compressor speed of 3000 rpm for R134a and R430A, respectively. It is observed that the exergy destructions in the condenser of the system using R134a and R430A get increased by about 65% and 70%, respectively, with an increase in compressor speed from 1000 rpm to 3000 rpm. Moreover, the exergy destruction of R430A was found to be 3–16% lower when compared to R134a due to its higher thermal conductivity and lower liquid density and lower liquid viscosity. The exergy destructions observed in this research investigation are

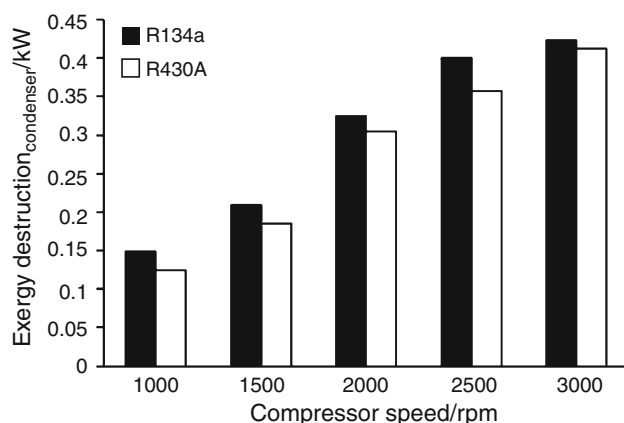


Fig. 14 Variation of exergy destruction of condenser at different compressor speed

similar to the earlier research work reported by Golzari et al. [21] and Cho and Park [22].

Exergy destruction in expansion valve

The exergy destructions in an expansion valve of automobile air conditioning systems working with R134a and R430A at different compressor speeds from 1000 to 3000 rpm are compared in Fig. 15. It is observed that the exergy destructions (both the refrigerants) in the expansion valve get increased by about 73% with the increase in compressor speed from 1000 to 3000 rpm. The maximum exergy destructions occur of about 0.16 kW and 0.14 kW were observed for R134a and R430A, respectively, at maximum compressor speed of 3000 rpm. The exergy destruction of R430A in an expansion valve was found to be 7–30% lower when compared to R134a due to its favorable thermo-physical properties such as lower operating pressure and lower density and lower viscosity as depicted in Figs. 1, 3 and 4, respectively.

Exergy destruction in evaporator

The exergy destructions in evaporator of an automobile air conditioning system operating at different compressor speeds in the range between 1000 and 3000 rpm are compared in Fig. 16. It is observed that the exergy destructions in the evaporator of the system using R134a and R430A get increased by about 50% and 53%, respectively, with the increase in compressor speed. The maximum exergy destructions of about 0.33 kW and 0.30 kW were observed for R134a and R430A at a peak compressor speed of 3000 rpm. Moreover, the exergy destruction of R430A was found to be 9–20% lower when compared to R134a due to its higher latent heat and higher thermal conductivity with lower viscosity and lower density. The

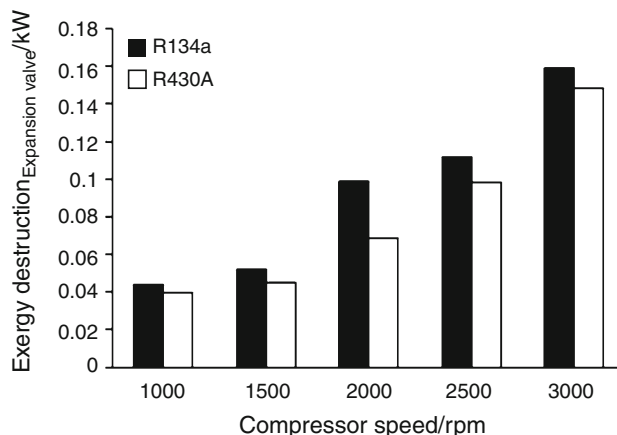


Fig. 15 Variation of exergy destruction of expansion valve at different compressor speed

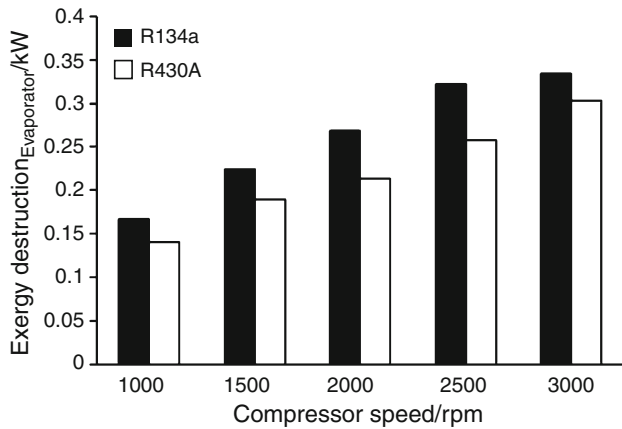


Fig. 16 Variation of exergy destruction of evaporator at different compressor speed

exergy destruction variations observed in an evaporator of the system working with R134a and R430A are found to be similar to the earlier reported investigations [21, 22].

Exergy destruction in the system

The exergy destructions of the system using R134a and R430A at different compressor operating speeds are compared in Fig. 17. It is observed that the system exergy destructions for using R134a and R430A get increased by about 55% and 57%, respectively, with the increase in compressor speed from 1000 to 3000 rpm. The maximum system exergy destruction of about 1.41 kW and 1.29 kW were observed for R134a and R430A, respectively, at compressor speed of 3000 rpm. The R430A has 9–21% lower system exergy destruction when compared to R134a for wide-range compressor speeds due to its favorable thermo-physical properties such as lower density, lower viscosity, higher specific heat, higher thermal conductivity and higher latent heat.

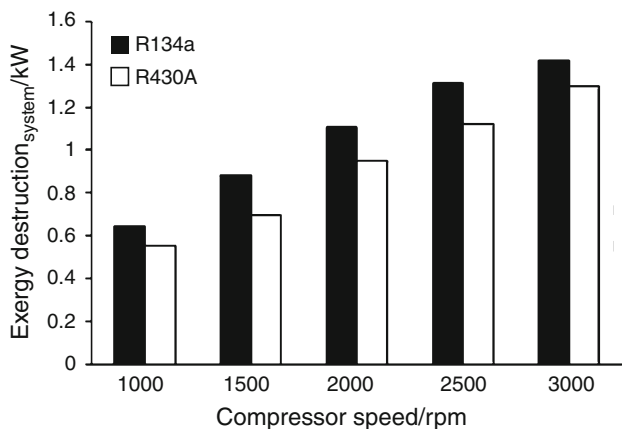


Fig. 17 Variation of exergy destruction of system at different compressor speed

Relative exergy destruction in the system

The influence of compressor speed on relative exergy destruction of each component of an automobile air conditioning system using R134a and R430A is depicted in Fig. 18. It is observed that the relative exergy destructions in the compressor using R134a and R430A were dropped from 44 to 35% and from 45 to 33%, respectively, with the increase in compressor speed from 1000 to 3000 rpm. Similarly, the exergy destructions in the condenser of the system operating with R134a and R430A were increased from 23 to 30% and from 23 to 31% with the increase in compressor speed from 1000 to 3000 rpm, respectively. The relative exergy destructions in the expansion valve of the system using R134a and R430A were increased from 7 to 11% with the increase in compressor speed from 1000 rpm to 3000 rpm. The relative exergy destructions in the evaporator of a system were dropped from 26 to 23.5% for R134a and from 25.4 to 23.3% for R430A. The maximum relative exergy destructions for R134a was observed in a compressor (by contributing about 44%), followed by an evaporator (by contributing 26%), condenser (by contributing 23%) and expansion valve (by contributing 7%) at a compressor speed of 1000 rpm. Similarly, the maximum exergy destructions of the system using R430A as working fluid were observed in the compressor (by contributing 45%), followed by an evaporator (by contributing 26%), condenser (by contributing 22%) and expansion valve (by contributing 7%). The relative exergy performance of an automobile air conditioning system is similar to the earlier reported investigations [21, 22]. The compressor operating parameters of the system need to be optimized to improve its overall performance.

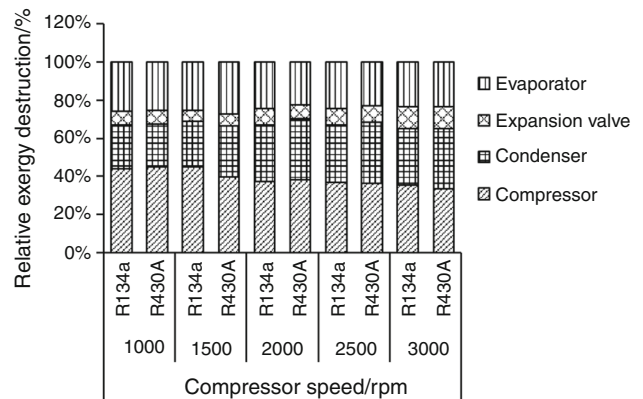


Fig. 18 Relative exergy destruction of each component of the system at different compressor speed

Exergy efficiency

The exergy efficiency variations of an automobile air conditioning systems working with R134a and R430A are compared in Fig. 19. The system exergy variations of R430A were observed to be about 15–24% higher when compared to R134a across the wide range of compressor operating speeds between 1000 and 3000 rpm. The exergy efficiency of the system using R134a and R430A drops by about 19.4% and 13.1%, respectively, with the increase in compressor speed from 1000 to 3000 rpm. The exergy efficiency of an automobile air conditioning system using R430A has 15–24% higher when compared to R134a due to its favorable thermo-physical properties such as higher specific heat, higher thermal conductivity, lower density, lower viscosity and higher latent heat when compared to R134a.

Mixture behavior

The R430A refrigerant mixture behaves like a near azeotropic mixture with a temperature glide of 0.2 °C. The presence of R600a in the mixture is miscible with mineral oil as well as with the polyol-ester. Hence, the lubricant replacement is not required in the case of R430A retrofitting. The composition of R430A was retained even in the case of 90% refrigerant leakage. The operating pressure of R430A observed at typical locations in the refrigerant circuit was observed to be lower when compared to R134a.

Total equivalent warming impact

The total equivalent warming impact (TEWI) of an automobile air conditioning systems (using LPG, petrol and diesel) working with R134a and R430A is compared in Fig. 20. It is observed that the vehicles using LPG, petrol and diesel fuel have TEWI of 5.45, 7.65, and 8.37 tons of

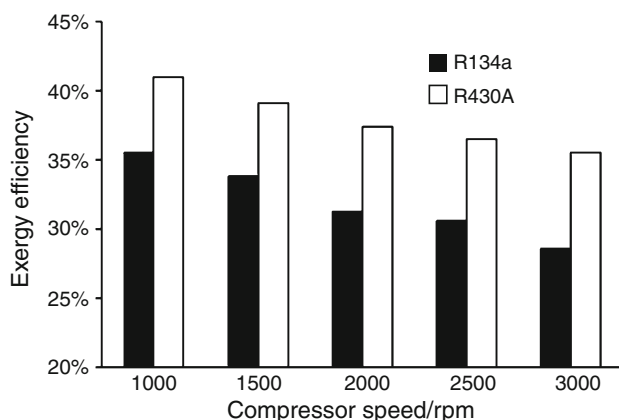


Fig. 19 Variation of exergy efficiency (η_{ex}) of system at different compressor speed

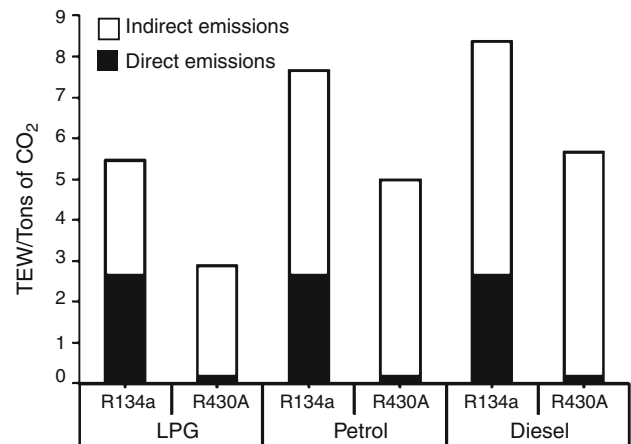


Fig. 20 Total equivalent warming impact of an automobile air conditioning system working with R134a and R430A for LPG, petrol and diesel

CO₂, respectively, for R134a air conditioners. Similarly, the LPG, petrol, and diesel fuel vehicles operating with R430A have TEWI of 2.87, 4.97 and 5.66, respectively. It is observed that the air conditioning systems operating with R430A has 47.3% 35% and 32.4% lower TEWI when compared to the system operating with R134a for LPG, petrol and diesel, respectively, due to its lower global warming potential.

Conclusions

The following major conclusions are arrived based on the experiments in R134a-based automobile air conditioning system retrofitted with R430A as drop-in substitute at ambient temperature and relative humidity of 35 ± 1 °C and $65 \pm 5\%$, respectively, with compressor speed variations between 1000 and 3000 rpm:

- The R430A refrigerant quantity was reduced by about 50% when compared to R134a.
- The R430A has 3–6% shorter pull-down time to attain the steady-state temperature of about 22 °C when compared to R134a.
- The R430A has higher compressor discharge temperature of about 2–6 °C due to its higher specific heat ratio when compared to R134a. The physical stability of lubricant used in compressor was retained at 6 °C elevated working temperature ensures the compressor reliability. Hence, the compressor life will not be affected.
- The R430A has 6–11% lower compressor power consumption when compared to R134a due to its favorable thermo-physical properties such as lower

viscosity and lower density in both liquid and vapor phases.

- The R430A has 9–20% higher COP due to its high latent heat, higher thermal conductivity and higher specific heat with lower viscosity and lower density when compared to R134a.
- The system exergy destruction for R430A was observed to be 9–21% lower when compared to R134a due to its favorable thermo-physical properties such as lower density and lower viscosity.
- The maximum exergy destruction for R134a and R430A was observed in the compressor, evaporator, condenser and expansion valve.
- The R430A has 47.3%, 35% and 32.4% lower TEWI when compared to the system operating with R134a for LPG, petrol and diesel vehicles, respectively, due to its lower global warming potential.

The results observed in this research confirmed that R430A is good drop-in substitute to phase out R134a in automobile air conditioners.

Acknowledgements The R430A sample provided by Hans Industrial Corporation is highly acknowledged.

Annexure

The uncertainty of calculated performance parameters are given by:

Mass flow rate

$$\dot{m}_r = \left[\left(\frac{\delta N}{N} \right)^2 + \left(\frac{\delta T_1}{T_1} \right)^2 + \left(\frac{\delta T_0}{T_0} \right)^2 + \left(\frac{\delta P_1}{P_1} \right)^2 \right]^{1/2}$$

Refrigeration effect

$$Q_{\text{evaporator}} = \left[\left(\frac{\delta \dot{m}_r}{\dot{m}_r} \right)^2 + \left(\frac{\delta T_1}{T_1} \right)^2 + \left(\frac{\delta P_1}{P_1} \right)^2 + \left(\frac{\delta T_4}{T_4} \right)^2 + \left(\frac{\delta P_4}{P_4} \right)^2 \right]^{1/2}$$

Compressor power consumption

$$W_{\text{compressor}} = \left[\left(\frac{\delta \dot{m}_r}{\dot{m}_r} \right)^2 + \left(\frac{\delta T_1}{T_1} \right)^2 + \left(\frac{\delta P_1}{P_1} \right)^2 + \left(\frac{\delta T_2}{T_2} \right)^2 + \left(\frac{\delta P_2}{P_2} \right)^2 \right]^{1/2}$$

Coefficient of performance

$$\text{COP} = \left[\left(\frac{\delta Q_{\text{evaporator}}}{Q_{\text{evaporator}}} \right)^2 + \left(\frac{\delta W_{\text{compressor}}}{W_{\text{compressor}}} \right)^2 \right]^{1/2}$$

Exergy destruction in compressor

$$\dot{E}x_{\text{dest,comp}} = \left[\left(\frac{\delta \dot{m}_r}{\dot{m}_r} \right)^2 + \left(\frac{\delta T_1}{T_1} \right)^2 + \left(\frac{\delta P_1}{P_1} \right)^2 + \left(\frac{\delta T_2}{T_2} \right)^2 + \left(\frac{\delta P_2}{P_2} \right)^2 + \left(\frac{\delta T_0}{T_0} \right)^2 + \left(\frac{\delta W_{\text{compressor}}}{W_{\text{compressor}}} \right)^2 \right]^{1/2}$$

Exergy destruction in condenser

$$\dot{E}x_{\text{dest,cond}} = \left[\left(\frac{\delta \dot{m}_r}{\dot{m}_r} \right)^2 + \left(\frac{\delta T_2}{T_2} \right)^2 + \left(\frac{\delta P_2}{P_2} \right)^2 + \left(\frac{\delta T_3}{T_3} \right)^2 + \left(\frac{\delta P_3}{P_3} \right)^2 + \left(\frac{\delta T_0}{T_0} \right)^2 + \left(\frac{\delta Q_{\text{condenser}}}{Q_{\text{condenser}}} \right)^2 \right]^{1/2}$$

Exergy destruction in an expansion device

$$\dot{E}x_{\text{dest,expansion}} = \left[\left(\frac{\delta \dot{m}_r}{\dot{m}_r} \right)^2 + \left(\frac{\delta T_3}{T_3} \right)^2 + \left(\frac{\delta P_3}{P_3} \right)^2 + \left(\frac{\delta T_4}{T_4} \right)^2 + \left(\frac{\delta P_4}{P_4} \right)^2 + \left(\frac{\delta T_0}{T_0} \right)^2 + \left(\frac{\delta Q_{\text{expansion}}}{Q_{\text{expansion}}} \right)^2 \right]^{1/2}$$

Exergy destruction in an evaporator

$$\dot{E}x_{\text{dest,evaporator}} = \left[\left(\frac{\delta \dot{m}_r}{\dot{m}_r} \right)^2 + \left(\frac{\delta T_4}{T_4} \right)^2 + \left(\frac{\delta P_4}{P_4} \right)^2 + \left(\frac{\delta T_1}{T_1} \right)^2 + \left(\frac{\delta P_1}{P_1} \right)^2 + \left(\frac{\delta T_0}{T_0} \right)^2 + \left(\frac{\delta Q_{\text{evaporator}}}{Q_{\text{evaporator}}} \right)^2 \right]^{1/2}$$

Exergy destruction in a system

$$\dot{E}x_{\text{dest,system}} = \left[\left(\frac{\delta W_{\text{compressor}}}{W_{\text{compressor}}} \right)^2 + \left(\frac{\delta Q_{\text{condenser}}}{Q_{\text{condenser}}} \right)^2 + \left(\frac{\delta Q_{\text{expansion}}}{Q_{\text{expansion}}} \right)^2 + \left(\frac{\delta Q_{\text{evaporator}}}{Q_{\text{evaporator}}} \right)^2 \right]^{1/2}$$

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