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Research Article

Energy, exergy and entropy analysis with R1234yf as an alternate refrigerant to R134a of automobile air conditioning system

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ABSTRACT

A major portion of the worldwide emissions arise from mobile air-conditioning systems with hydrofluorocarbon refrigerant as working substance and which is one of major cause for the greenhouse effect. R134a refrigerant having GWP of 1400 has been extensively used in car air conditioning. To reduce greenhouse gas emissions, the current R134a refrigerant must be phase out as per Kigali Amendment. The present study deals with cooling load calculation of car model by heat balance method as per ASHRAE standard using local climate condition. Further, thermodynamic analysis of R1234yf as an alternate refrigerant to R134a has been carried out for automobile air conditioning system. The required properties of refrigerants are extracted from Engineering Equation Software. The thermodynamic analysis is carried out to study the effect of operating parameters viz. condensing temperature, evaporating temperature, degree of superheating and degree of subcooling on COP, EDR, exergy efficiency and entropy generation. The previous literature reports mainly focus on separate study of either cooling load calculation or energy analysis or exergy analysis of R1234yf and R134a for automobile air conditioning system, while this paper presents the comprehensive study of new low GWP R1234yf as an alternate refrigerant to R134a in automobile air conditioning system with cooling load calculation including the concept of energy, entropy and exergy analysis. The percentage difference in COP between R134a and R1234yf system varies from 2.44 % to 4.78 % while percentage difference in EDR varies from 6.79 % to 2.87 % when evaporating temperature varied from -10 °C to 10 °C. With 12 °C of superheating at compressor inlet, the COP of R134a is 3.9 whereas COP of R1234yf is 3.75, which makes 3.85 % lower than that of R134a. The R1234yf has 4.78 % lower value of exergy efficiency as compared to that of R134a at evaporating temperature of -10 °C and it is found that maximum exergy destruction takes place in compressor.

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INTRODUCTION

The impact of refrigerant on the environment became the most concern of scientific research. The emission of various gases from industries in the environment are comparable with carbon dioxide and reported by global warming potential (GWP). A high GWP gases significantly increases the level of earth's warmness. At present world uses the 3rd generation of HFC refrigerants having zero ODP and high GWP in the automobile industries. Hence, to overcome the problem of high GWP it is necessary to find the alternative of R134a as per Kyoto protocol and the Montreal protocol [1]. Regulation agencies are working on harmful refrigerants by phasing out and replacing them with environmentally friendly alternate refrigerants. It also reduces the emission of greenhouse gases and protects the ozone layer. The alternate refrigerant should also be safe and economical for the existing automobile air conditioning system. Some countries focus on reducing emissions by selecting suitable refrigerants, proper equipment design, maintenance and inspection. The Kigali Amendment [2] was approved at the 28th meeting of parties to the Montreal Protocol on October 15, 2016, in Kigali, Rwanda, seeks to phase out HFCs by decreasing their consumption and production. Before 2050, the global consumption of HFCs should be reduced by almost 85% under the Kigali Amendment. It has been found that the estimation of demand projection of passenger car population in India may increases up to 9% in 2017–2038 [3]. The average car life is considered to be 15 years, and the study proposes a limit of 200 passenger cars per 1,000 populations. However, the refrigerant demand scenario is studied based on different leakage rates from the system considering the technological development in the cooling system. According to India cooling action plan estimation the refrigerant demand would be 1900 to 24000 MT by 2038. One of the major factors for greenhouse gas emission is mobile air conditioning system. Study by Yang et al. [4]showed that the air conditioning system utilises up to 20% of total energy supplied to the vehicle from fuel energy. According to Shaikh et al. [5] R1234yf is a good alternative of R134a as both refrigerants having equal refrigerating effect, volumetric cooling capacity, COP and energy consumption. The comparison between the thermophysical properties of refrigerant R1234yf and R134a are summarized in Table 1.

The first law of thermodynamics provide only information about the loss of energy and system performance but degradation of energy can be determined by exergy balance [9]. The exergy loss is directly proportional to the suction and discharge temperature of the refrigerant in the system. It was found that the maximum exergy loss in vapour compression refrigeration system occurs in compressor device. According to the second law of thermodynamics the degradation of exergy and its efficiency is due to elevation of evaporator inlet air temperature of R1234yf and R134a automobile air conditioning system reported by Golzariet al. [10]. In compressor major exergy destruction of 53% occurred, while exergy destruction in thermostatic expansion valve, evaporator and condenser is found 15%, 11% and 21% respectively. Exergy efficiency of R1234yf was observed to be higher as compared to R134a. The comparative study of refrigerants R1234ze and R1234yf as replacements to R134a in a double evaporator vapour compressor refrigeration system was made by Yataganbaba [11]. The exergy efficiency significantly influenced by the condenser and evaporator temperature. Though performance of R1234yf is lower than that of R134a, but the performance gap is comparative small, so R1234yf is considered as suitable alternative to R134a due to its environment friendly properties [12]. Kamelet al. [13]suggests R1234yf and mixture of R1234yf/R134a as a promising long term eco-friendly solution for future with some minor modification in the system and also the refrigerant mixture is found non-flammable and less expensive. Alpaslan et al. [14] investigated effect of the range of temperature of air stream at condenser and evaporator inlet as well as compressor speed on energy and exergy analysis in automobile air conditioning system. The system consists of fixed capacity compressor, laminated plate evaporator, expansion valve and parallel flow micro channel condenser. The compressor operated by pulley belt mechanism with frequency inverter. The temperature variation is generated by 1.78 kW and 5.4 kW heater in condenser and evaporator duct respectively. It is also found that initial charge of R1234yf is 10% lower than R134a due to lower liquid density of R1234yf. Yumrutas et al. [15] made exergy analysis based computational model for the estimation of the effect of the condensing and evaporating temperatures on the COP, pressure losses, second law efficiency and the exergy losses of a vapor compression refrigeration system. It is concluded

Table 1. Refrigerant property comparison between R134a and R1234yf [6][7][8]

	Chemical composition	Molecular weight [kgkmol^{-1}]	Normal boiling point [°C]	Critical Temperature [°C]	Critical pressure [MPa]	ODP	GWP	Safety class (Security classification ASHRAE34)	Atmospheric lifetime (year)
R134a	CH_2FCF_3	102	-26.07	101.1	4.059	0	1400	A_1	14
R1234yf	$CF_3CF= CH_2$	114	-29.45	95	3.382	0	4	A_2L	<0.05 (11 days)

that the condensing and evaporating temperature strongly affect COP, second law efficiency and the exergy losses in the condenser and evaporator but has little effect on exergy losses of the other components. The exergy destruction in the compressor is maximum, while in the expansion valve and evaporator is lower for all the three refrigerants viz. R1234ze, R134a and R1234yf. A L Tarish [16]concluded that R134a and R1234yf has total exergy destruction of 2.7 kW and 2.48 kW respectively in complete vapour compression refrigeration system. The lowest exergy efficiency is observed for refrigerant R134a as compared to R1234yf and R1234ze. As subcooling temperature increased the irreversible loss increased hence the exergy efficiency decreased. Ozgur et al. [17] carried out energetic and exergetic analysis for R134a and R1234yf refrigerants in vapor compression refrigeration cycle. Difference in cycle efficiencies between R134a and R1234yf refrigerant is observed to be minimal. It was suggested that R1234yf is a better alternative to R134a based on similar thermodynamic properties aspect and it was concluded that problem of global warming originated due to R134a will be possibly resolved by system using R1234yf. Cho et al. [18] experimentally analyzed that the cooling capacity and COP of R1234yf is lower by 7% and 4.5% compared to R134a respectively in automobile air conditioning system. With addition of internal heat exchanger in system difference in performance reduced to 2.9% and 1.8% respectively. Zhaogang [19] performed experimental analysis using R1234yf and R134a on laminated plate and microchannel parallel flow type evaporator. The microchannel parallel flow type evaporator performed better and the pressure drop on air side is nearly close for both evaporators. Daviran et al. [20]simulated automobile air conditioning system with R134a and R1234yf as refrigerants using REFPROP 8.0. The discharge temperature, pressure ratio, pressure drop in condensation and evaporation and hence overall system price of R1234yf is found lower than R134a.The COP of R1234yf is 1.3–5% lower than that of R134a and identical in cooling capacity. Wantha [21] concluded that for similar operating conditions COP of R134a increased by 2.11% and that of R1234yf increased by 3.78% when a tube in tube internal heat exchanger is introduced in the existing refrigeration system**.** Gaurav and kumar [22] studied the effect of compression ratio on the material strength to reduce flammability. By lower charging amount of R1234yf refrigerant the compression ratio is lower so COP and cooling capacity of R1234yf can be improved. Ryo et al. [23] applied Patel-Teja (PT) equation and ECS (Extended corresponding state) to model and optimize various properties i.e density, enthalpy, entropy and specific heat of R1234yf. Atilla et al. [24] compared different refrigerants i.e. R513a, R1234ze, R134a, R450a and R1234yf within limiting the range of GWP, viscosity and liquid density and found that R1234yf can be used in place of R134a. Lee et al. [25] observed similarities in discharged temperature, COP and capacity of refrigerant R1234yf, R134a and the mixture of R1234yf/R134a on bench tester of heat pump in different weather condition. The initial charge required for R1234yf and mixture is 11% lower than R134a. Mixture become non-flammable at 10% and above composition of R134a. Aral et al. [26] developed and compared the empirical correlations for prediction of performance of R1234yf and R134a in an air conditioning system based on experimental results in the laboratory. The uncertainties of Exd, Wc, Qc, Qe and COP obtained during test i.e. 5.9%, 2.8%, 3%, 2.6% and 4.1% respectively. Direk et al. [27] experimentally investigated the influence of internal heat exchanger to enhance the performance. The maximum exergy destruction is found in compressor and total exergy destruction found reasonable with internal heat exchanger for refrigerant R1234yf and R134a. Alhendal et al. [28] concluded that refrigerant R1234yf thermal performance is close to R134a and exergy destruction is comparatively lower. Shin et al. [29] experimentally analyzed the energy and exergy of an automobile air conditioning system with R134a and R134/R1234yf mixture. The decrease in total exergy destruction of the system is observed conversely exergy efficiency observed to be increased.

The flow of analysis of present work consisting of first cooling load estimation small capacity car followed by energy analysis, exergy analysis and entropy analysis for low GWP refrigerant R1234yf as an alternate refrigerant to R134a for automobile air conditioning system presented in figure1 below.

The previous literature reports mainly focus on separate study of either cooling load calculation or energy analysis or exergy analysis of R1234yf and R134a for automobile air conditioning system, while this paper presents the comprehensive study of new low GWP R1234yf as an alternate

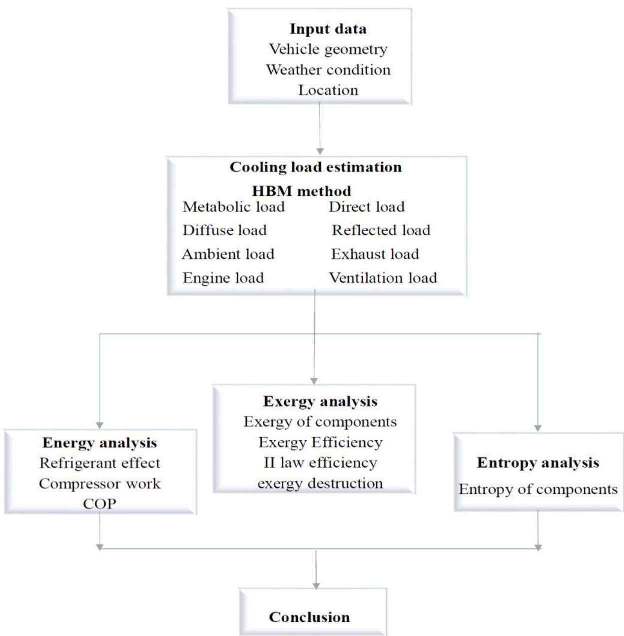

Figure 1. Flow diagram of the analytical thermal analysis.

refrigerant to R134a in automobile air conditioning system with cooling load calculation including the concept of energy, entropy and exergy analysis.

COOLING LOAD CALCULATION

Heat Balance Method (HBN) [30] is applicable to evaluate heating load and cooling load. To calculate the load of different types mathematical models of heat transfer is applied for the system. Mathematical models for load estimation are devised and summarized from various load approximations. In present work for this purpose Wagon R car is studied under random driving situations. Simplified car geometry and common material of vehicle body are considered as initial parameter for studies. For vehicle cabin a normal lumped model is considered. The net heat associated with vehicle body is classified in nine different terminologies. The instantaneous overall heat load gain by car body is summation of these different load. The mathematical representation of the heat load for the model [31] is given by equation (1),

$$Q_{total} = Q_{Met} + Q_{Dir} + Q_{Dif} + Q_{Ref} + Q_{Amb} + Q_{Exh} + Q_{Eng} + Q_{Ven} + Q_{AC} \quad (1)$$

In above equation Q is thermal energies with respect to time. Q_{total} is the effective thermal load come across the cabin. Q_{Met} is the metabolic load. Q_{Ref}, Q_{Dif} and Q_{Dir} and are the reflected, diffuse and direct radiation loads, respectively [32]. Q_{Amb} is ambient load. Q_{Eng} and Q_{Exh} and are engine load and the exhaust load due to high temperature in the engine and exhaust gases. The load due to ventilation is presented by the term Q_{Ven}, and the thermal load generated by air conditioning cycle is Q_{AC}. The equations (2) to (16) mentioned below determine the cooling load.

Metabolic load: The metabolic activities inside human body constantly create heat and humidity. This heat passes through the body tissues and is finally released to the cabin air. This load is considered as a heat gain by the cabin air and is called the metabolic load. The metabolic load can be calculated by

$$Q_{Met} = \sum M * 0.202 * \left(W^{0.425}\right)\left(H^{0.725}\right) \quad (2)$$

1. Direct load: The heat gain due to solar radiation is a significant part of the cooling loads encountered in vehicles. Direct radiation load is that part of the incident solar radiation which directly strikes a surface of the vehicle body, which is calculated from

$$Q_{Dir} = \sum S * \tau * cos\theta * \frac{A}{exp^{(B/Sin\beta)}} \quad (3)$$

2. Diffuse load: Diffuse radiation is the part of solar radiation which results from indirect radiation of daylight on the surface. During a cloudy day, most of solar radiation is received from this diffuse radiation. The diffuse radiation heat gain is found by,

$$Q_{Dif} = \sum S * \tau * C * I_{Dir} * \frac{1 + cos\sum}{2} \quad (4)$$

3. Reflected load: Reflected radiation refers to the part of radiation heat gain that is reflected from the ground and strikes the body surface of the vehicle. The reflected radiation is calculated by,

$$Q_{Ref} = \sum S * \tau * I_{Ref} \quad (5)$$

4. Ambient load: The ambient load [33] is the contribution of the thermal load transferred to the cabin air due to temperature difference between the ambient and cabin air. Exterior convection, conduction through body panels, and interior convection are involved in the total heat transfer between the ambient and cabin. The ambient load is calculated by,

$$Q_{Amb} = \sum S * U * \left(T_s - T_i\right) \quad (6)$$

where,

$$R = \frac{1}{h_o} + \frac{\lambda}{k} + \frac{1}{h_i}$$

$$h = 0.6 + 6.64\sqrt{v}$$

5. Exhaust load: Convectional and hybrid electric vehicle have an internal combustion engine that creates exhaust gases. The Exhaust gas temperature can reach as high as 1000 °C, because of the high temperature of the exhaust gas, some of the heat can be transferred to the cabin through the cabin floor. The Exhaust heat load calculated by,

$$Q_{Exh} = S_{Exh} * U * (T_{Exh} - T_i) \quad (7)$$

$$T_{Exh} = 0.138 * RPM - 1 \quad (8)$$

6. Engine load: Similar to the exhaust load, the high temperature engine of a conventional or hybrid car can also contribute to the thermal gain of the cabin. The engine load calculated by,

$$Q_{Eng} = S_{Eng} * U * \left(T_{Eng} - T_i\right) \quad (9)$$

Where,

$$T_{Eng} = \left(-2 * 10^{-6} * RPM^2\right) + 0.0355 * RPM + 77.5 \quad (10)$$

7. Ventilation load: Fresh air is allowed to enter the vehicle cabin to maintain the air quality for passenger. As the passengers breathe, the amount of CO_2 concentration linearly increases over the time. Thus, a minimum flow of fresh air should be supplied into the cabin to maintain the passenger comfort. For instant, minimum 13% fresh air is needed for a single passenger. The ventilation load [34] calculated by,

$$Q_{ven} = m_{Ven} * \left(e_0 - e_i\right) \tag{11}$$

Where,

$$e = (1000*T) + (2.051*10^6 + 1770*T)*X \tag{12}$$

$$X = 0.62198*\frac{(100*P)}{(100*P)-(\phi*Ps)} \tag{13}$$

8. Deep Thermal load: The thermal mass is a property of the substance of the selected volume that allows heat storage to provide inertia against temperature fluctuations, which is also called as the thermal flywheel effect. In automobiles, the vehicle cabin thermal inertia is known as deep thermal mass [31].

$$Q_{DTM} = \frac{\left(m_a * c_a + DTM\right)*\left(T_i - T_{Comf}\right)}{t_c} \tag{14}$$

$$\text{Where, } t_c = \frac{t_p}{ln\left(T_o - T_{Comf}\right)} \tag{15}$$

9. AC load: The duty of the air conditioning system is to compensate for other thermal loads so that the cabin temperature remains within the acceptable comfort range. In cold weather conditions, negative AC load is needed for maintaining the comfort conditions. The actual load creates by the AC system depends on the system parameter and working conditions.

$$Q_{AC} = -\left(Q_{Met} + Q_{Dir} + Q_{Dif} + Q_{Amb} + Q_{Ref} + Q_{Exh} + Q_{Eng} + Q_{Ven}\right) - Q_{DTM} \tag{16}$$

The data for cooling load calculations for small capacity car are as below.

- Total passenger with driver: - 04
- passenger average weight: - 75 kg
- average cabin temperature: - 25 °C
- average surface temperature: - 40 °C
- Engine speed: - 2500 RPM

The cooling load calculations are based ASHRE model on 17/05/2019 at location Surat, Gujarat, India (21.1663° N, 72.7833° E) and estimated by the empirical equations of HBM method. Furthermore, Automobile body surfaces consists of opaque part with insulation and glass surfaces, so that the direct radiation and diffuse radiation evaluated with consideration of the incident angle of sun radiation relative to the glass surfaces.

Table 2. Cooling load summary

Metabolic load (W)	321.48
Direct load (W)	1250.66
Diffuse load (W)	162.79
Reflected load (W)	11.64
Ambient load (W)	1268.11
Exhaust load (W)	20.57
Engine load (W)	81.53
Ventilation load (W)	-1232.35
Deep thermal mass load(W)	1582.08
Air conditioning load (W)	3466.51

THERMAL ANALYSIS

Following assumptions are considered for thermodynamic analysis to determine energy and exergy at various thermodynamic states in air conditioning system.

- Steady state condition.
- Process in expansion valve is adiabatic.
- Pressure losses in pipes are neglected.
- Heat transfer between system and the surroundings is neglected
- Pressure drop in condenser, evaporator and internal heat exchanger is neglected
- Potential energy and kinetic energy changes are insignificant.
- Cooling Capacity Q_L = 3.5 kW
- Isentropic Efficiency of compressor = 0.8
- Condensation temperature = 44 °C
- Evaporator temperature = 0 °C
- Degree pf Superheating = 5 °C

The cooling load calculated from data is 3.47 kW. For air conditioning design the cooling load value can be varying 5% to 10% higher than the design load [34] during different operating and weather conditions. For the numerical analysis the initial value of cooling load is considered as 3.5 kW, which is the product of the refrigerant flow rate and refrigerating effect. The thermodynamic system for automobile air conditioning system presented in Figure 2. The P-h diagrams for defined initial operating condition generated in EES for refrigerant R134a and R1234yf are shown in Figure 3 and Figure 4 respectively.

Energy Analysis

The refrigerant mass flow rate m is calculated as follow.

$$m = \frac{Q_L}{h_6 - h_5} \tag{17}$$

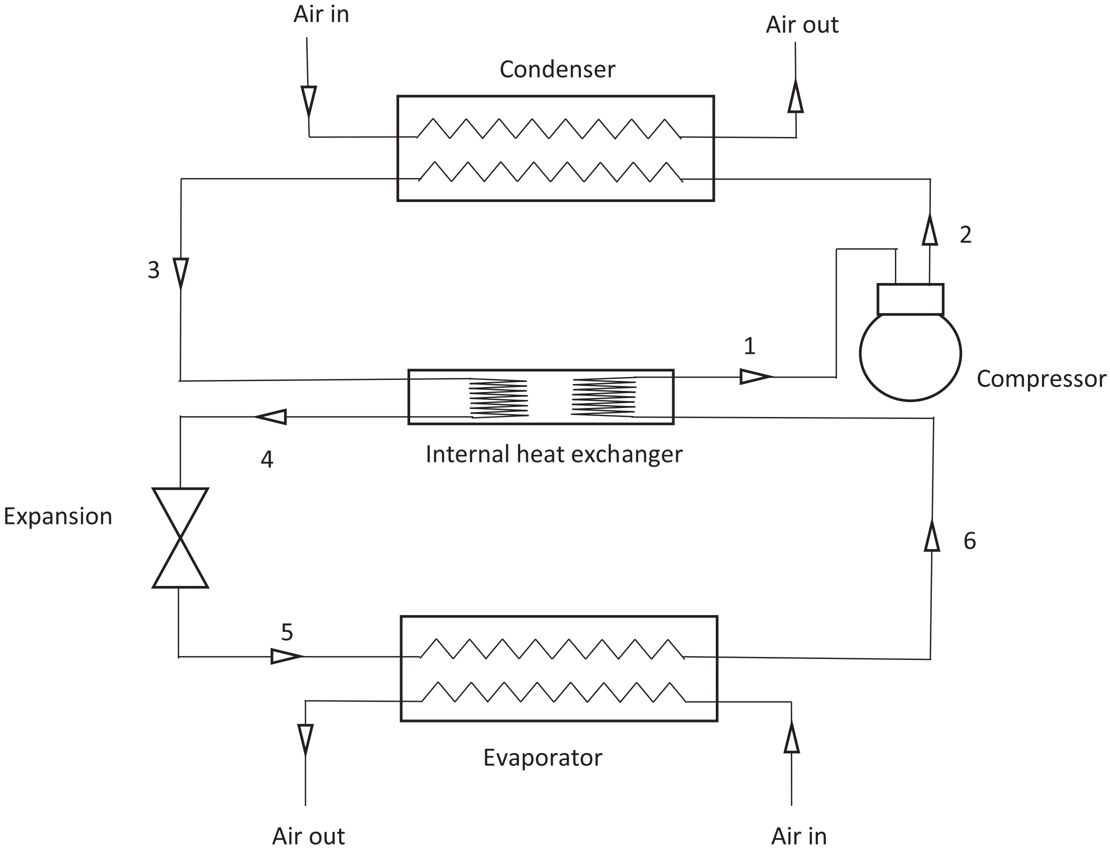

Figure 2. Schematic diagram of air conditioning system

The isentropic compressor work is determined by following equation.

$$Wc_{isen} = m*(h_{2s} - h_1) \tag{18}$$

The actual compressor work is the calculated as per following equation.

$$Wc = Wc_{isen} * \eta_s \tag{19}$$

The COP of the system is defined as ratio refrigerating effect produced by evaporator and compressor work.

$$COP = \frac{Q_L}{Wc} \tag{20}$$

This energy analysis is carried out to find effect of various operating parameters on COP of air conditioning

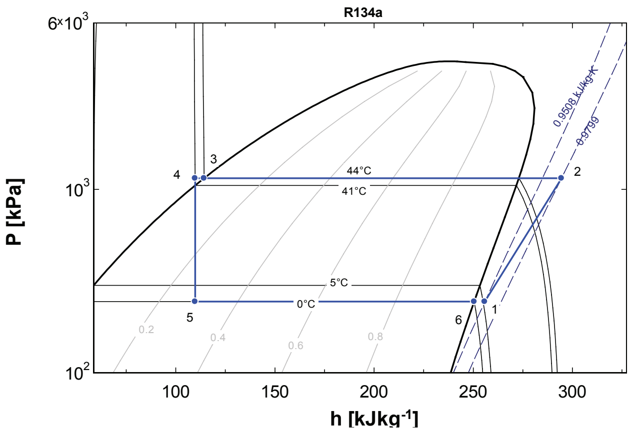

Figure 3. P-H diagram for R134a with IHX.

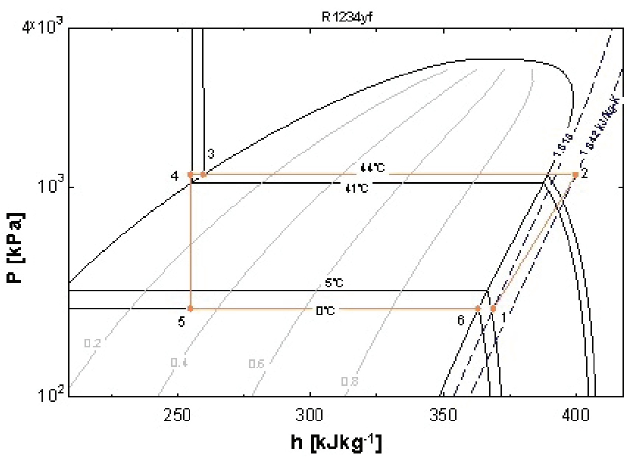

Figure 4. P-H diagram for R-1234yf with IHX.

system using Engineering Equation Software (EES) [35]. To carry out energy analysis condensing temperature varied from 30 °C to 60 °C, evaporating temperature varied from -10 °C to 10 °C, degree of superheating varied from 1 °C to 12 °C and degree of subcooling is varied from 1 °C to 12 °C. Results of energy analysis for two refrigerants i.e R134a and R1234yf is compared.

Exergy Analysis

The aim of exergy analysis is to evaluate the maximum performance of thermal system and point out the source of exergy destruction of the system. Exergy analysis of a thermal system can be evaluated by analysing all the components individually. The major exergy destruction in any component of the system is the point for major improvement. So, exergy analysis identifies and proposes the available scope of design higher efficient energy systems by minimising the inefficiencies. The exergy analysis leads to overcome the shortcomings. Exergy analysis identifying the source, magnitudes and causes of the process inefficiencies. Exergy analysis is carried out to find effect of various operating parameters on EDR, Second law efficiency, total exergy destruction and exergy destruction of each components of air conditioning system using Engineering Equation Software (EES) [35]. To carry out exergy analysis condensing temperature varied from 30 °C to 60 °C, evaporating temperature varied from -10 °C to 10 °C, degree of superheating varied from 1 °C to 10 °C and degree of subcooling is varied from 1 °C to 10 °C. Results of energy analysis for two refrigerants i.e R134a and R1234yf is compared.

The total exergy or maximum available energy of a system is determined by [36],

$$E_{total} = E_{Kinetic} + E_{Potential} + E_{Physical} + E_{Chemical} \tag{21}$$

Where, $E_{Kinetic}$, $E_{Potential}$, $E_{Physical}$, $E_{Chemical}$ are kinetic, potential, physical and chemical exergy respectively.

Because of elevation and speed changes for automobile are minimum, we have assumed the changes in potential energy and kinetic exergy are insignificant. Chemical exergy is assumed significant when a process involving combustion or chemical changes. But, in refrigeration process chemical substances does not get mixed or there is no chemical reaction. Hence, the value of chemical exergy is considered negligible and is not considered in exergy analysis. The physical exergy of the system is only considered in exergy analysis. The maximum work that can be developed by a flow system is exergy. In case of all processes in the environment and the system are reversible then exergy is conserved, while exergy is destroyed when the processes are irreversible. The exergy is due to entropy generation in the system and the exergy may be destructed. Exergy of flow stream of the mass is written as

$$\psi = (h - h_0) - T_0(s - s_0) \tag{22}$$

Where, s_o and h_o are specific entropy and enthalpy at dead state (T_o) [37].

Irreversibility of the system or Exergy destruction is the exergy difference between the flow entering to control volume and the leaving control volume by mass, heat and work transferred. It is represented by below exergy balance equation (23).

$$\Delta E = \sum (m * Ex)_{in} - \sum (m * Ex)_{out} + \sum W_{in} - \sum W_{out} + \sum Q_i * \left(1 - \frac{T_o}{T_s}\right)_{in} - \sum Q_{out} * \left(1 - \frac{T_o}{T_s}\right)_{out} \tag{23}$$

The exergy destruction equation for each component is obtained by writing exergy balance for each component individually. Total exergy destruction is algebraic summation of exergy destruction of individual components.

Exergy destruction in Compressor:

$$\Delta E_c = m\left[\left(h_1 - h_2\right) - T_o\left(s_1 - s_2\right)\right] + Wc \tag{24}$$

Exergy destruction in condenser:

$$\Delta E_{cond} = m * \left[(h_2 - h_3) - T_0(s_2 - s_3)\right] - Q_{cond} * (1 - \frac{T_0}{T_c}) \tag{25}$$

Exergy destruction in expansion valve:

$$\Delta E_{ex} = mT_o\left(s_5 - s_4\right) \tag{26}$$

Exergy destruction in evaporator:

$$\Delta E_{ev} = m\left(\left(s_5 - s_6\right) - \frac{RE}{T_e}\right) \tag{27}$$

Exergy efficiency:

$$Ex_{eff} = RE \left(\frac{T_o}{T_e} - 1\right) \tag{28}$$

Ratio of exergy destruction of the individual component to the system total exergy destruction is called as relative exergy destruction of that component. For the air conditioning system, the ratio of the actual COP to that of the corresponding reversible cycle or Carnot cycle is defined as second law efficiency,

$$\eta_{2^{nd}\,Law} = \frac{COP_{act}}{COP_{rev}} \tag{29}$$

Exergy destruction ratio (EDR) is ratio of the total exergy destruction to the exergy destruction of the product in the corresponding system [38]

$$EDR = \frac{\left(ED_{Total}\right)}{Q_{Evap} * \left(1 - \frac{T_0}{T_{eva}}\right)} \tag{30}$$

Entropy Generation

The entropy is randomness of molecules or measure of molecular disorder. At higher randomness of molecules the molecule position become less predictable which increase disorder and hence entropy of the system [39]. Entropy generation in compressor, condenser, expansion valve and evaporator are evaluated with following equation number 31 to 34.

Entropy generation in compressor

$$En_c = m(s_2 - s_1) \tag{31}$$

Entropy generation in condenser

$$En_{cond} = m\left((s_3 - s_2) - \frac{Q_c}{T_c}\right) \tag{32}$$

The expansion leads to generation of turbulence in the refrigerant, hence in expansion device increases the entropy [40].

Entropy generation in expansion devise

$$En_{ex} = m^* \ (s_5 - s_4) \tag{33}$$

Entropy generation in evaporator

$$En_{ev} = m^*\left((s_6 - s_5) - \frac{RE}{T_e}\right) \tag{34}$$

Total entropy generation is algebraic summation of entropy generation of individual components.

RESULTS AND DISCUSSION

The variation of COP and EDR with evaporating temperature for both R134a and R1234yf air conditioning system is shown in figure 5. Both COP and EDR increase with increase in evaporating temperature. COP of R1234yf is lower by 4.78% than COP of R134a and EDR of R1234yf is lower by 6.79% than EDR of R134a at evaporating temperature of -10 °C. The percentage difference in COP varies from 2.44% to 4.78% while percentage difference in EDR varies from 6.79% to 2.87% over given range of evaporating temperature.

The variation of exergy efficiency and second law efficiency with evaporating temperature for both R134a and R1234yf is shown in figure 6. The exergy efficiency decreases with increase in evaporating temperature. The R1234yf has 4.78% lower value of exergy efficiency as compared to that of R134a at evaporating temperature of -10 °C, the difference decreases up to 2.43% by when evaporating temperature has reached to 10 °C. The second law efficiency increases with evaporating temperature up to optimal value of 0 °C, but after this optimum value, the second law efficiency begins to decrease.

The variation of total exergy destruction and total entropy generation with evaporating temperature for both R134a and R1234yf air conditioning system is shown in

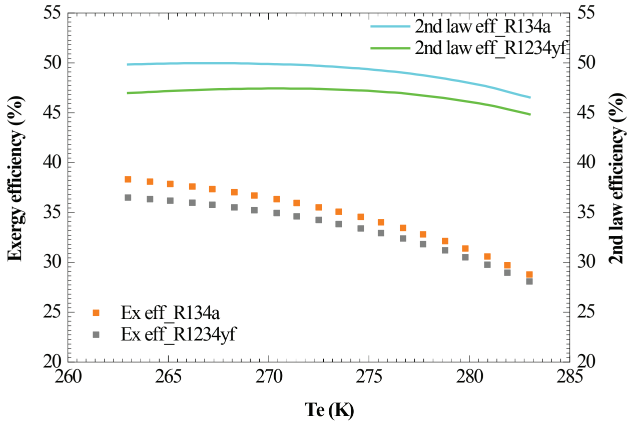

Figure 6. Effect of evaporating temperature on exergy efficiency and second law efficiency.

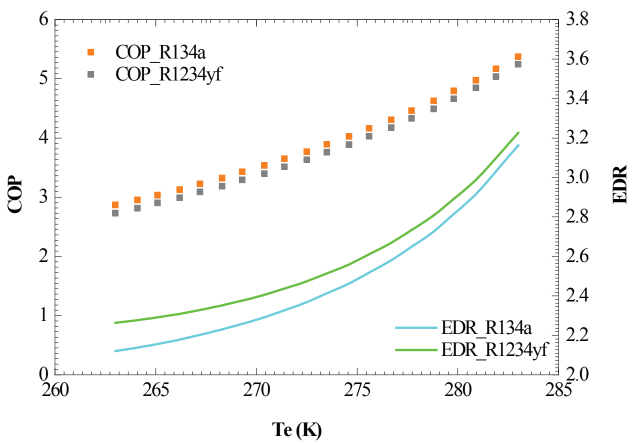

Figure 5. Effect of evaporating temperature on COP and EDR.

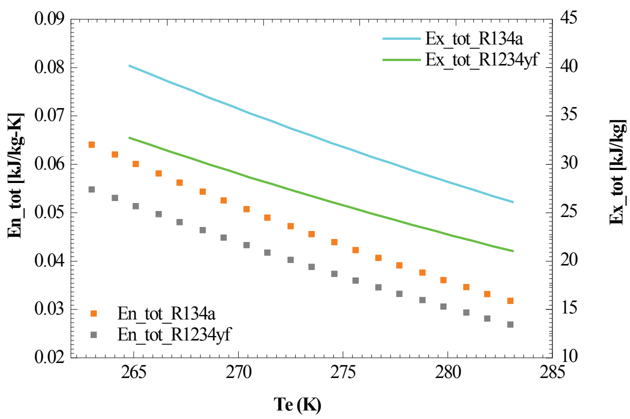

Figure 7. Effect of evaporating temperature on total exergy destruction and entropy generation.

Figure 7. Both total exergy destruction and entropy generation decreases with increase in evaporating temperature. The total entropy generation of R1234yf is lower by 14.42% than R134a at lower evaporating temperature of -10 °C and 15.39% at higher evaporating temperature of 10 °C. The total exergy destruction of R1234yf is lower by 18.55% than R134a at lower evaporating temperature of -10 °C and 19.4% at higher evaporating temperature of 10 °C.

Figure 8 shows exergy destruction of individual component with variation in evaporating temperature. The exergy destruction trend of individual component for both the R134a and R1234yf systems is identical. The maximum exergy destruction is found in compressor followed by condenser, expansion valve and evaporator. The exergy destruction of various components for R1234yf system is found lower than R134a system.

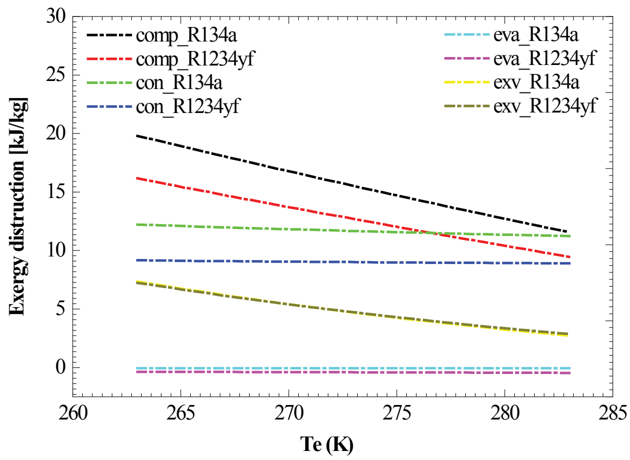

Figure 8. Effect of evaporating temperature on total exergy destruction of components.

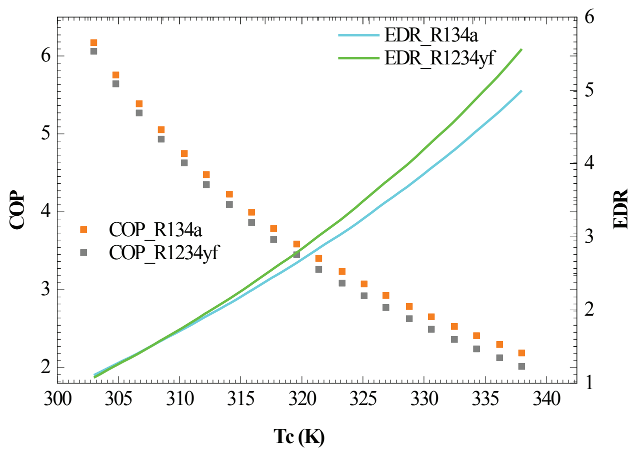

Figure 9. Effect of condenser temperature on COP and EDR.

The variation of COP and EDR with condensing temperature for both R134a and R1234yf air conditioning system is shown in Figure 9. COP decreases while EDR increases with increase in condensing temperature from 30 °C to 60 °C at fix evaporating temperature of 0 °C. Compressor work of R134a system is lower by 2.01% than compressor work of R1234yf system at condensing temperature of 30 °C and 10.18% when condensing temperature has reached to 60 °C. COP of R1234yf system is lower by 1.96% than R134a system at condensing temperature of 30 °C and 9.22% when condensing temperature reached to 60 °C.

The variation of exergy efficiency and second law efficiency with condensing temperature for both R134a and R1234yf air conditioning system is shown in Figure 10. Both total exergy efficiency and second law efficiency decreases with increase in condensing temperature. The second law efficiency of R1234yf is 1.99% lower as compared to R134a at condensing temperature of 30 °C and difference in value increases to 10.88% when condensing temperature has

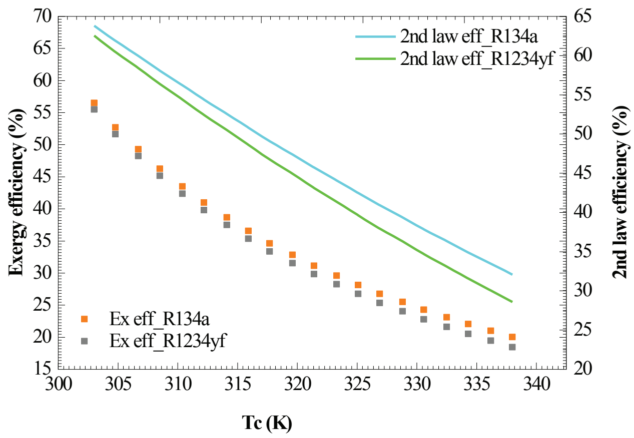

Figure 10. Effect of condenser temperature on exergy efficiency and second law efficiency.

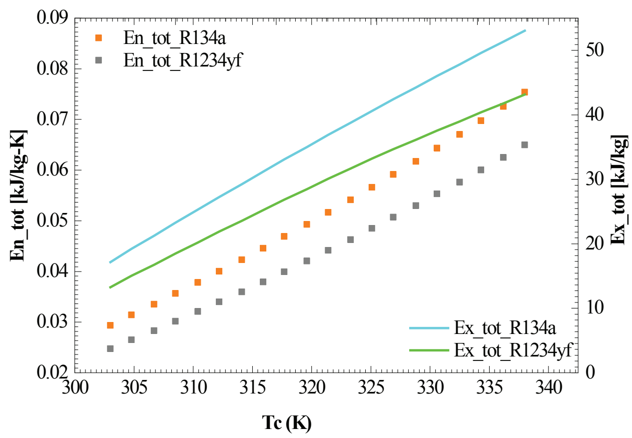

Figure 11. Effect of condenser temperature on total exergy destruction and entropy generation.

reached to 60 °C. The exergy efficiency of R1234yf is 1.79% lower as compared to R134a at condensing temperature of 30 °C and difference in value increases to 7.94% when condensing temperature has reached to 60 °C.

The variation of total exergy destruction and total entropy generation with condensing temperature for both R134a and R1234yf air conditioning system is shown in figure 11. Both total exergy destruction and entropy generation increases with increase in condensing temperature. The total entropy generation of R1234yf is lower by 15.74% than R134a at lower condensing temperature of 30 °C and decreases up to 13.85% at higher condensing temperature of 60 °C. The total exergy destruction of R1234yf is lower by 22.58% than R134a at lower condensing temperature of 30 °C and decreases up to 18.68% at higher condensing temperature of 60 °C.

Figure 12 shows exergy destruction of individual component with variation in condensing temperature

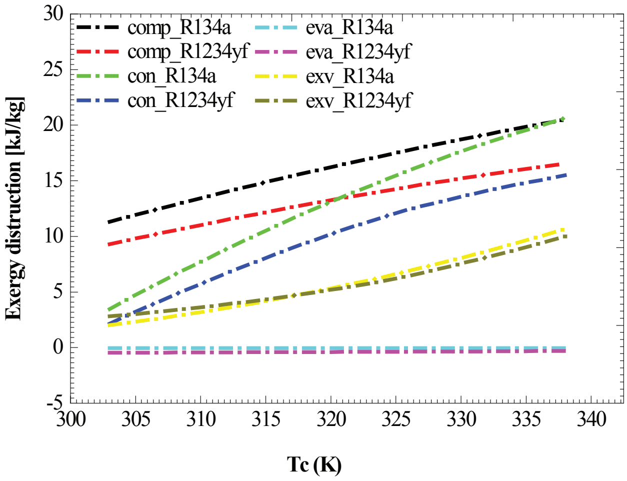

Figure 12. Effect of condensing temperature on total exergy destruction of components.

at constant evaporating temperature of 0 °C and cooling capacity of 3.5 kW. The exergy destruction of components increases with increase in condensing temperature for both the R134a and R1234yf systems. The maximum exergy destruction is found in compressor and minimum in the evaporator. The exergy destruction of various components for R1234yf system is found lower than R134a system.

The variation of COP and exergy efficiency with degree of superheating at the compressor inlet for R134a and R1234yf is shown in figure 13. As degree of superheating increase from 1 °C to 12 °C, COP and exergy efficiency increases for both R134a and R1234yf. COP of R1234yf system is lower by 3.95% than R134a system at 1 °C degree of superheating and 3.85% when degree of superheating reached to 12 °C. The exergy efficiency of R1234yf is 2.1% lower as compared to R134a at 1 °C degree of superheating and difference remains nearly same when degree of superheating reached to 12 °C.

The variation of COP and exergy efficiency with subcooling in the condenser for R134a and R1234yf is shown in figure 14. As degree of subcooling increase from 1 °C to 12 °C, COP and exergy efficiency increases for both R134a and R1234yf. COP of R1234yf system is lower by 17.74% than R134a system at 1 °C degree of subcooling and 2.5% when degree of subcooling reached to 12 °C. The exergy efficiency of R1234yf is 10.9% lower as compared to R134a at degree of subcooling 1 °C and difference in value decreases to 5.71% when degree of subcooling reached to 12 °C.

The exergy destruction diagram for R134a and R1234yf air conditioning system at 44°C condensing temperature and 0 °C evaporating temperature is shown in Figure 15 and Figure 16 respectively. Figure 15 shows that for R134a system out of total 100 % exergy input to motor, 31.26% exergy loss found in compressor, 17.92% exergy loss found in condenser, 13.04% exergy loss found in expansion valve and 0.15% exergy loss found in evaporator making finally exergy utilized to cooling space is 37.63%. It is concluded from figure 16 that for same operating condition exergy utilized to cooling space for R1234yf lower

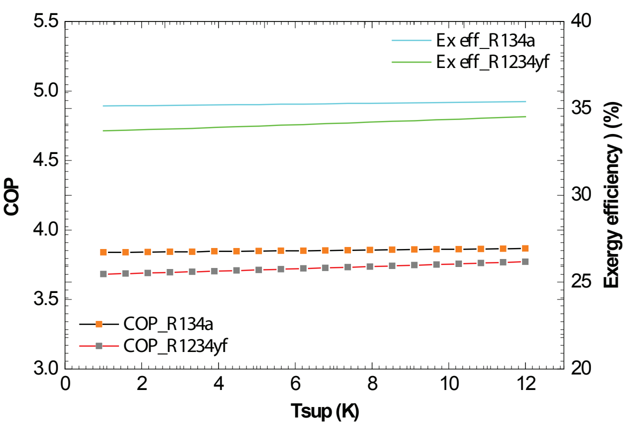

Figure 13. Variation of COP and exergy efficiency with degree of superheat.

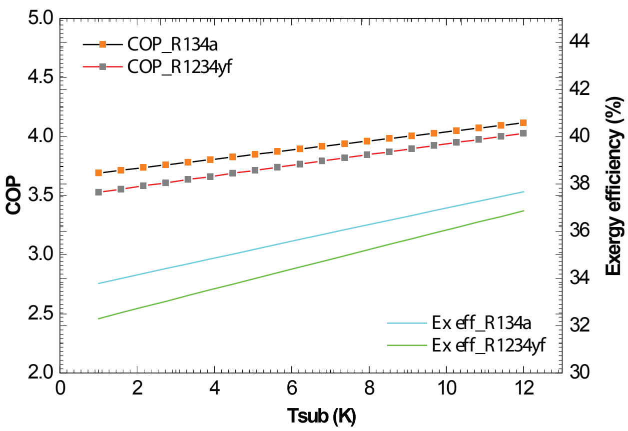

Figure 14. Variation of COP and exergy efficiency with degree of subcooling.

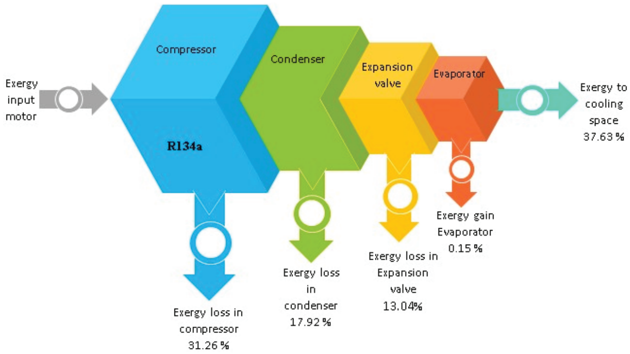

Figure 15. Exergy destruction diagram for R134a.

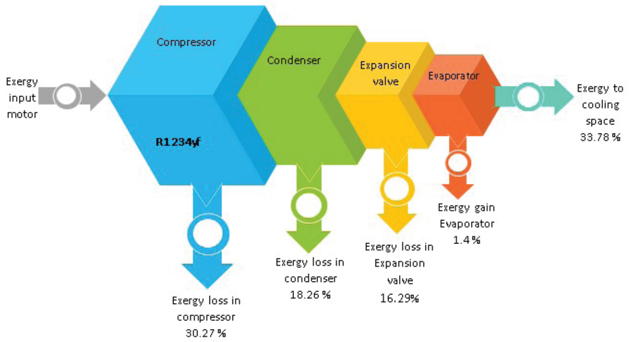

Figure 16. Exergy destruction diagram for R1234yf.

than R134a. In present study of automobile air conditioning system with operating conditions i.e., condensing temperature of 44°C and evaporating temperature of 0 °C, it is found that the compressor work of R1234yf is 27.36 kJ/kg and of R134a is 33.97 kJ/kg. Hence compressor work of R1234yf system is lower than the R134a system. On the contrary it is also found that the refrigerating effect of R1234yf is 115 kJ/kg is lower than that of R134a which is 148 kJ/kg. Hence COP of the system which is ratio refrigerating effect to compressor work is found 4.21 with R1234yf system and 4.36 with R134a system. This shows that COP of R1234yf is lower than R134a. The advantage of lower compressor work with R1234yf system vanishes due to lower refrigerating effect. Similar results have been observed by other researchers. The experimental study by Alpaslan et al. [14] concluded that for different operating conditions, the refrigerant effect of R1234yf is 0.4%-10.9% lower than that of R134a. Cho et al. [18] revealed that the refrigerant effect of R1234yf is 4% and compressor work is 7% lower than that of R134a. Daviran et al. [20] concluded that the refrigerant side over all heat transfer coefficient of R1234yf is 18% to 21% lower than that of R134a which results in lower refrigerating effect of R1234yf.

CONCLUSIONS

In present work the comprehensive study of new low GWP R1234yf as an alternate refrigerant to R134a in automobile air conditioning system is carried out with cooling load calculation including the concept of energy, entropy and exergy.

- Both COP and EDR increase with increase in evaporating temperature. The percentage difference in COP between R134a and R1234yf system varies from 2.44% to 4.78% while percentage difference in EDR varies from 6.79% to 2.87% when evaporating temperature varied from -10 °C to 10 °C.
- The exergy efficiency decreases with increase in evaporating temperature. The R1234yf has 4.78% lower value of exergy efficiency as compared to that of R134a at evaporating temperature of -10 °C, the difference decreases up to 2.43% by when evaporating temperature has reached to 10 °C.
- Exergy destruction of individual component shows that for both the R134a and R1234yf maximum exergy destruction is found in compressor followed by condenser, expansion valve and evaporator.
- COP decreases with increase in condensing temperature from 30 °C to 60 °C at fix evaporating temperature of 0 °C. COP of R1234yf system is lower by 1.96% than R134a system at condensing temperature of 30 °C and 9.22% when condensing temperature reached to 60 °C.
- Second law efficiency decreases with increase in condensing temperature. The second law efficiency of R1234yf is 1.99% lower as compared to R134a at condensing temperature of 30 °C and difference in value increases to 10.88% when condensing temperature has reached to 60 °C.
- As degree of superheating increase from 1 °C to 12 °C, COP and exergy efficiency increases marginally low for both R134a and R1234yf.
- As degree of subcooling increase from 1 °C to 12 °C, COP and exergy efficiency increases significantly for both R134a and R1234yf.

NOMENCLATURE

A	Apparent Solar Irradiation
A_2L	Mildly flammable refrigerants category
ADu	The Dubois area
B	Atmospheric Extinction Coefficient
C	Diffuse Radiation Factor
DTM	Deep Thermal Mass
E_n	Entropy generation
ΔE	Exergy destruction
e_o	The ambient enthalpy
e_i	The cabin enthalpy
h_i	Inner convection heat transfer coefficient
h_o	Outer convection heat transfer coefficient
H	The passenger height
HFO	Hydrofluoroolefin
HFC	Hydrofluorocarbon
I_{Dir}	The direct radiation heat gain per unit area
I_{Dif}	the diffuse radiation heat gain per unit area

I_{Ref}	the reflected radiation heat gain per unit area
M	The passenger metabolic heat production rate
m_{Ven}	The ventilation mass flow rate
ma	Cabin air mass
ODP	Ozone depleting potential
P_s	The water saturation pressure
Q_{Met}	Metabolic load
Q_{Dir}	Direct radiation load
Q_{Dif}	Diffuse radiation load
Q_{Ref}	Reflected radiation load
Q_{Amb}	Ambient load
Q_{Ex}	Exhaust load
Q_{Eng}	Engine load
Q_{Ven}	Ventilation load
R	The net thermal resistance
S	Surface area
T_s	The average surface temperature
T_i	The average cabin temperature
T_{Eng}	Engine temperature
T_{Exh}	Exhaust temperature
T_{comf}	Comfort temperature
t_c	Pull down constant
t_p	Pull down time
U	Overall heat transfer coefficient of surface element
v	The vehicle speed
W	The passenger weight
X	Humidity ratio in gram of water per gram of dry air

Symbols

τ	The surface element transmissivity
θ	The angle between the surface normal and the position of sun in the sky.
β	The altitude angle, calculated based on position and time
ϕ	Relative humidity
Ψ	specific exergy
Σ	The surface tilt angle measured from horizontal surface
η_{ex}	exergy efficiency
η_s	isentropic efficiency of compressor

Subscript

c	Compressor
cond	Condenser
ex	Expansion
ev	Evaporator
0	Dead state

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AUTHORSHIP CONTRIBUTIONS

Authors equally contributed to this work.

DATA AVAILABILITY STATEMENT

The authors confirm that the data that supports the findings of this study are available within the article. Raw data that support the finding of this study are available from the corresponding author, upon reasonable request.

CONFLICT OF INTEREST

The author declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

ETHICS

There are no ethical issues with the publication of this manuscript.

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