Khatoon and Karimi *Int. J. Air-Cond. Ref.* (2023) 31:2
https://doi.org/10.1007/s44189-022-00017-1

International Journal of
Air-Conditioning and Refrigeration

RESEARCH Open Access

Thermodynamic analysis of two evaporator vapor compression refrigeration system with low GWP refrigerants in automobiles

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Abstract

Due to the growing demand for refrigerators and air conditioners in automobiles, there is a need for an innovative and efficient design to achieve both refrigeration and air conditioning. To address this, the present work has evaluated a two-evaporator vapor compression system to eliminate the requirement for separate refrigeration and air conditioning units. Theoretical energy and exergy performance assessment of the same is carried out along with the variation in the condenser and evaporators temperature. Various low GWP refrigerants such as R1234yf, HFO1336mzz(Z), R513A, and R450A are compared against high GWP R134a and R452A. The results reveal that maximum exergy efficiency and COP and lowest compressor power of 31.50%, 2.47, and 6.304 kW, respectively, are obtained with HFO1336mzz(Z). After HFO1336mzz(Z), R134a shows the highest exergy efficiency and COP of 30.56% and 2.41, respectively, and the lowest compressor power of 5.61kW. HFO1336mzz(Z) exhibits the optimum performance, whereas R452A shows the worst thermodynamic performance in the system. It is also found that the performance of R450, R513A, R450A, and R1234yf is approximately equivalent to each other. Moreover, component-wise exergy destruction analyses indicate that the efficiency of the compressor needs to be improved as the maximum destruction of 61.84–56.00% occurs in the compressor while the minimum exergy destruction of 0.42–0.54% occurs in the expansion valve. This study proposes the two-evaporator system for both refrigeration and air conditioning in automobiles. It is also found that R450A, R1234yf, R513A, and HFO1336mzz(Z) can be the potential alternative to R134a.

Keywords Two evaporators, Refrigeration and air conditioning, Automobiles, Energy, Exergy, Low GWP refrigerant

1 Introduction

Air conditioning was once considered a luxury, but now it has become a necessity in automobiles as it provides thermal comfort to the passengers and the driver. It also reduces the drivers' fatigue and, hence, the chances of accidents [25, 30]. Generally, 1- to 1.5-ton air conditioning is used in automobiles. Many studies are performed on the different refrigerants in AAC system with cooling capacity of 3.5 kW [8, 32]. As of now, refrigerators are used in high-end car models; however, they are now gaining popularity in economical automobiles as well as they help to keep the food, drinks, and other consumable items at low temperatures, preventing these items from getting warmed while in the car. This increase in popularity can also be attributed to the rising purchasing power of consumers for innovative and technologically advanced products. With most people looking for comfort in cars, the global market for portable car refrigerators is expected to grow at a compound annual growth rate (CAGR) of 11.5% from 2021 to 2027 [1]. This growth can add to the existing pressure on fossil fuels to meet the energy demands leading to environmental degradation. The strategy should be to find innovative ways to

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optimize the performance reducing the energy requirements and impact on the environments. Even a slight increase in efficiency can tremendously increase profits and also benefits the environment.

In automobiles, the air conditioners and refrigerators can be installed as individual systems or can be coupled as a two-evaporator vapor compression refrigeration system (VCRS). But a two-evaporator vapor compression refrigeration (VCR) system, with the low-temperature evaporator used for refrigeration and the high-temperature evaporator used for air conditioning, can provide a better prospect of energy-saving and efficiency increase. This solution can fulfill the need of both refrigeration and air conditioning, providing high facilities of comfort, and can eliminate the need for two separate systems in automobiles. This arrangement is efficient because it can solve the space constraints in automobiles and the high initial cost of using individual systems. Moreover, a separate individual refrigerator in the cabin rejects heat indoor, increasing the heat load on air conditioning to pump out extra heat.

Two-evaporator VCR systems are already being used to fulfill the need for different refrigerating loads at different temperatures, such as in supermarkets and hotels, and are cost-effective and efficient. The two evaporator vapor compression systems save up to 20% power compared to a single evaporator system [16]. Two evaporators in a series-connected VCR system make the system more efficient and, hence, enhance the system performance compared to a single evaporator [7]. These systems also help to reduce compressor work as two-evaporator system has a large surface area for heat recovery [16, 17]. Therefore, the performance of two or more evaporator systems is better than a single evaporator system.

Improving the thermodynamic performance of automobile air conditioning (AAC) systems has been the core agenda, and to this effect, numerous studies have been done by modifying the design of VCRS systems. An experimental study performed by optimizing the expansion valve to replace R134a with R1234yf in AAC system investigated that the performance of R1234yf gets improved by adjusting the expansion valve under varied conditions as COP and cooling capacity both increased averagely by 8% and 11.3%, respectively [29]. The use of an internal heat exchanger (IHX) can also improve the performance of the system. An experimental study found improvement in the exergy efficiency and COP by 1.5 to 4.6% and 0.9%, respectively [6], while others found enhancement of 4.1% and 7.9%, respectively, in COP and cooling capacity of the R1234yf system with IHX [9]. Another experimental investigation for refrigerant R1234ze (Z) in R134a-based mobile air conditioning (MAC) determined that the performance of the MAC system gets enhanced by utilizing the IHX in the R1234ze system because of an average reduction of 19% in compressor power and an increment of 4% in COP. Furthermore, for the R1234ze system with IHX, there is a decrease in the value of exergy destruction of about 50% compared to R134a [10]. MAC system performance with R1234yf can also be improved by redesigning the expansion valve [36]. Apart from IHX, higher compressor efficiency can also enhance the performance of the R1234yf based MAC system [24]. The experimental analysis carried out to replace R134a with R1234yf in a VCRS found that the energy performances such as COP and cooling capacity of R1234yf are 9% and 19%, respectively, lower than R134a. And, with the introduction of an IHX, the differences in energy performances of R1234yf get reduced [22]. It has also been reported that though COP, cooling capacity, and volumetric efficiency of R1234yf and R1234ze are lower than R134a, but the introduction of IHX in VCRS can improve the performance of low GWP refrigerants such as R1234yf and R1234ze [20]. An experimental comparison of R513A and R134a in the MAC system showed that R513A could be a substitute to R134a with adjustment of thermostatic expansion, leading to remarkable performance and higher cooling capacity of R513A [19].

As suggested by many studies, the exergy-based analysis is a useful tool to determine the actual losses occurring in the system due to irreversibility. A theoretical study incorporating this analysis revealed that the maximum exergy destruction and entropy generation occurred in the compressor and the cycle components of the R1234yf system show lower exergy destruction and entropy generation but higher exergy efficiency than R134a in AAC [13]. Another study also found through exergy-based charge optimization technique in MAC that the maximum exergy destruction of 59.88 to 69.9% is obtained in the compressor, and minimum exergy destruction of 3.05 to 15.73% is obtained in the expansion valve [23]. This analysis can also reveal better alternative refrigerants considering their exergetic performance. A theoretical model of exergy analysis showed that the exergy efficiency of R1234ze is higher than that of R134a and R1234yf [34]. An exergy analysis to replace R134a with a mixture of R290/R600a in a domestic refrigerator determined that the hydrocarbon mixture R290/R600a shows maximum exergy efficiency of 42.1% and improvement in COP of 28.5% compared to R134a [27]. Experimental exergy analyses are also performed to replace R134a with R430a (mixture of R152a & R600a) in an existing AAC system [4]. They found that compressor power reduced from 6 to 11% with 12 to 20% enhancement in COP of R430a compared to R134a. Also, R430a has total exergy destruction of 12 to 28% lower than R134a. R1234ze has been found to have better energetic and exergetic performance than R152a, R134a, and R1234yf. This study reported that the total exergy destruction of R1234yf and R152a is 15% and 45%, respectively, lower than R134a, while total exergy destruction of R1234ze is 5.4% higher than R134a [3]. An exergetic comparative analysis of R1234yf and R134a in AAC systems and heat pumps found that the system exergetic efficiency of R1234yf in heating and cooling modes on average is 14.7% and 17.6%, respectively, lower than R134a [5]. While in comparison to natural refrigerant isobutane, R134a shows lower exergetic performance with energy and exergy efficiencies of isobutane being higher than R134a by 175% and 50%, respectively [2]. Another refrigerant R450A in VCRS reveals total irreversibility of 15.25 to 27.32% lower and exergy efficiency of 10.07 to 130.93% higher than R134a [12], while R513A in a small capacity refrigeration system shows 0.4% higher global exergy efficiency than R134a with maximum exergy destruction and minimum exergy efficiency obtained in the compressor [18].

From the above literature survey, it is evident that there are no studies either on the refrigerators in automobiles or on the two evaporator VCRS fulfilling the need for both air conditioning and refrigeration in automobiles. Most of the studies are focused on the optimization of conventional single-evaporator AAC systems. There is a need to explore innovative, compact, economical, efficient, and environmentally friendly alternatives such as two evaporator VCRS to meet the increasing demand for air conditioners and refrigerators in automobiles. Furthermore, it is observed that the studies related to refrigerants R513A, R450A, R452A, and HFO1336mzz(Z) are limited, and hence more extensive study on these refrigerants is required.

This study has four main objectives—the first is to propose the two-evaporator system as an alternative to two separate systems for refrigeration and air conditioning in automobiles. The second aim is to examine the effect of evaporator and condenser temperature on the thermodynamic performance of the system. The third aim is to present the comparative analysis of R452A, R134a, and the low GWP refrigerants such as HFO1336mzz(Z), R1234yf, R513A, and R450A so as to suggest alternative refrigerants to R134. The fourth aim is to find the component having the maximum losses.

2 Qualitative assessment of refrigerants

Refrigerant selection should consider thermophysical characteristics, environmental factors, economic aspects, and technological issues [15]. Environmental properties such as ODP and GWP are crucial to reduce environmental impact.

The Kigali amendment to the Montreal Protocol planned to restrict the manufacturing and usage of high GWP HFC refrigerants and terminate their usage by 2028 [31]. The refrigerant R134a has been used in automobile air conditioning systems for a long time; however, it has a very high GWP of 1300. Since the beginning of 2017 in European countries, the manufacturers have stopped the utilization of R134a in a newly manufactured automobile as the fluorinated gas mandate (F-gas) has directed to limit the usage of refrigerants having GWP more than 150 [11]. Hence, the alternative refrigerants should be more environmentally friendly and safe and also improve the energy efficiency of the system. The possible alternatives to R134a are natural refrigerants (R744, HFC152a), hydrocarbon refrigerants (R290, R600a), and hydro-fluoro olefins (R1234yf, R1234ze, HFO1336mzz(Z)). The natural refrigerant, R744, has a GWP of 1, but it is not suitable to replace R134a due to its high operating pressure and critical temperature. The complication associated with hydrocarbon refrigerants (R290, R600a) is their flammability risk. Many authors have reported that the potential refrigerant to replace R134a is R1234yf because the thermophysical properties of R1234yf are almost similar to R134a, and it has a very low GWP of 4 and mild flammability with an A2L safety rating as shown in Table 1. The flammability risk of R1234yf is lower than R152a and hydrocarbon refrigerants.

Table 1 Thermo-physical and environmental properties of selected refrigerants [18, 21, 26, 35]

Sr. No.	Refrigerants	Molecular wt. (kg kmol^{-1})	NBP (°C)	Critical temperature (°C)	Critical Pressure (MPa)	ODP	GWP	Safety group
1	R1234yf	114.04	−29.45	94.7	3.281	0	4	A2L
2	HFO1336mzz(Z)	164.10	33.4	171.3	2.90	0	2	A1
3	R134a	102.03	−26.01	101.1	4.059	0	1300	A1
4	R513A	108.4	−29.60	96.50	3.76	0	573	A1
5	R450A	109	−23.5	105.87	3.814	0	547	A1
6	R452A	103.5	−47.2	75.05	4.014	0	1945	A1

The disadvantage of R1234yf is that it does not improve the energy performance of the air conditioner [37]. To overcome such drawbacks, HFO and HFC refrigerants have been mixed in varying ratios to develop several refrigerants such as R450A, R452A, and R513A. R450A, mixture of R134a and R1234ze (42/58); R452A—mixture of R125, R32, R1234yf (59/11/30); R513A—mixture of R134a and R1234yf (44/56) [26]. These refrigerants have low toxicity and flammability risk (Table 1) and are safer to use. The detailed properties of refrigerants are listed in Table 1.

3 Description of two-evaporator system

This study couples both refrigerator and air conditioner in a two-evaporator system to assess the exergy and energy performance to achieve high efficiency and energy saving. Figures 1 and 2 show the schematic and P-h diagram of

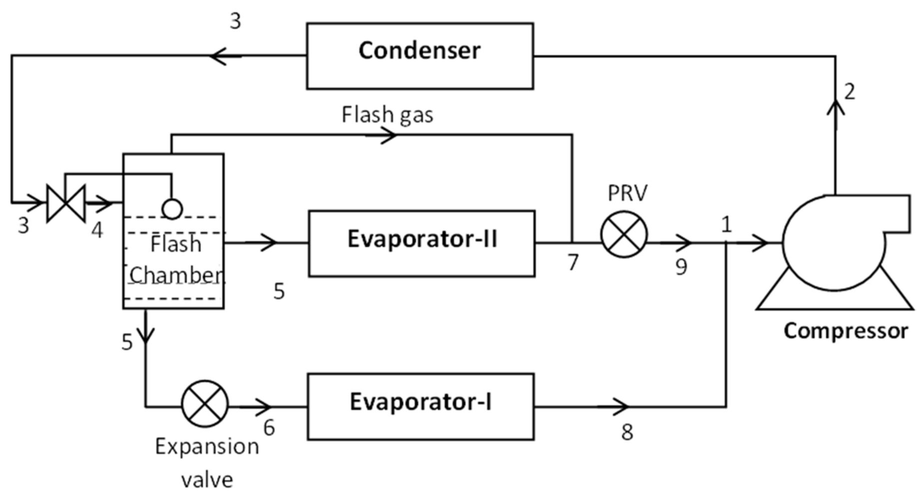

Fig. 1 Schematic diagram of two evaporator vapor compression refrigeration systems (VCRS)

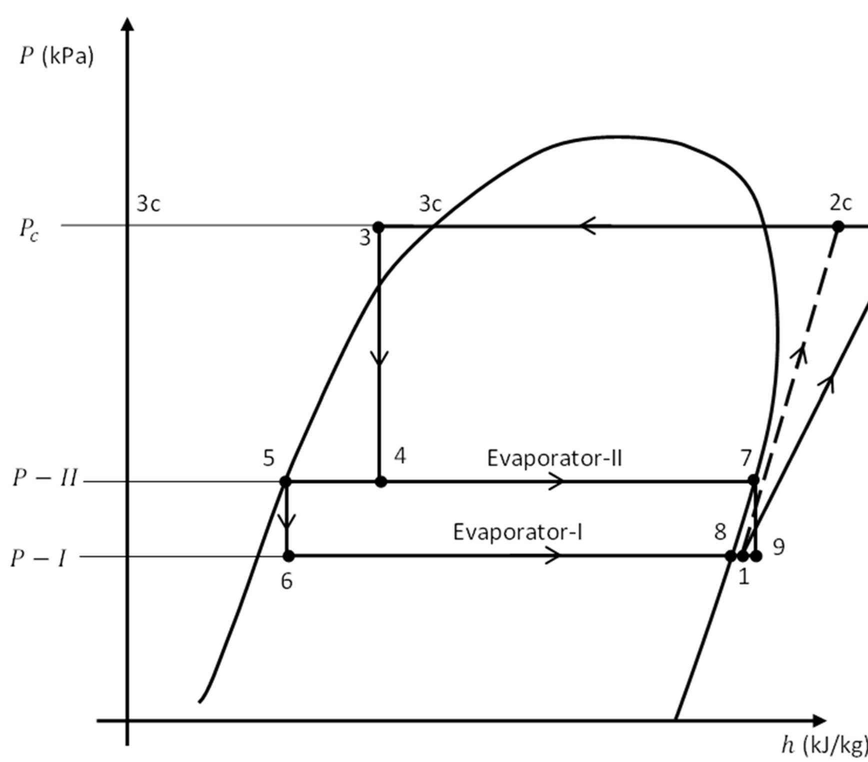

Fig. 2 Pressure enthalpy (P-h) state diagram of two evaporator VCR systems

two evaporator VCRS which consists of one compressor, one condenser, expansion valve-I for the whole cycle, flash chamber, two evaporators, an individual expansion valve for evaporator-I, and pressure regulating valve (PRV).

Figure 1 shows the two parallelly connected evaporators: evaporator-I works as a refrigerator and evaporator-II works as the air conditioner. This system can be used for the same or different temperatures with different cooling loads as per the requirement. Refrigeration and air conditioning system is a continuous process in which the refrigerant changes its phase from liquid to gas-liquid to liquid at lower temperature absorbing heat, gets its pressure and temperature increased, and then rejects heat at a higher temperature, converting it into liquid again. The refrigerant in the superheated region at state 1 enters the compressor, and after compression, temperature and pressure increase. The superheated refrigerant condenses in the condenser from state 2 to 3c and changes in liquid form by rejecting heat in the environment. From state 3c to 3, subcooling is there which reduces the temperature. After state 3, refrigerant is expanded from 3 to 4 and then saturated liquid (state 5) is separated from saturated vapor (state 7) in the flash chamber. The flash gas is passed directly into the PRV (7 to 9). One part of the pure liquid refrigerant, which is at state 5 (saturated liquid), directly enters evaporator-II and the other part (also at state 5) first expands (5 to 6) in the expansion valve and then enters evaporator-I. In evaporator-II, pure liquid refrigerant absorbs heat and converts into saturated vapor, producing air conditioning effect. The other part, which is at state 6 after expansion, is at lower pressure and can provide refrigeration in evaporator-I. PRV reduces the pressure at the exit of evaporator II to the pressure of evaporator I, so that the suction pressure at the inlet of the compressor is the same.

3.1 Thermodynamic and mathematical modeling for simulation

The mathematical model based on the equations is developed for thermodynamic analysis of two-evaporator vapor compression systems. The input parameters used for the simulation are listed in Table 2. For ease of analysis, the following assumptions are made:

1) All the components are assumed to be steady-state and steady-flow processes.
2) During the flow of refrigerants, the variation of kinetic and potential energies is negligible throughout the system.
3) The compression and expansion processes are adiabatic.
4) Negligible heat losses and pressure drops in linking pipes and system components.
5) The pressure is constant when refrigerant passes through the evaporator and condenser.
6) The dead state of the refrigerants is at pressure Po = 1.013 bar and temperature $T_o = 30°C$.

Table 2 Input parameters adopted for thermodynamic analysis

Parameters	Values
Cooling capacity for evaporator-II ($Q_{ev,II}$)	3.5kW [30]
Cooling capacity for evaporator-I ($Q_{ev,I}$)	10kW [34]
Sources temperature for evaporators (T_s)	15°C and −10°C
Sink temperature for the condenser (T_i)	30°C
Subcooling temperature, T_{sc}	5°C
Ambient temperature, T_a	30°C
Ambient pressure, P_o	101.325 kPa
Isentropic efficiency, η_{is}	0.75
Mechanical efficiency, η_{mech}	0.80
Electrical efficiency, η_{ele}	0.95

The cooling capacity of the evaporator-I working as refrigerator is assumed 10 kW for the study purpose. The cooling capacity can be varied as per the requirement

Exergy analysis is a powerful tool to determine the thermodynamic losses in each component and the whole system. This analysis also indicates the feasibility of the system and helps in the improvement of components. Exergy analysis also identifies the location, causes, and sources of deviation from the reference or ideal state. Exergy is defined as the maximum amount of work that can be produced by a process or system as it attains its equilibrium or reference state. The total energy at the inlet of the system is greater than the total energy at the outlet of the system, and this difference in the amount of energy is generally termed as " Internal energy losses" or "Exergy destruction".

The exergy balance in a general form can be expressed in the following Eq. (1) [33]:

$$\dot{E}_{x_{in}} - \dot{E}_{x_{out}} = \dot{E}_{x_D} \tag{1}$$

where $\left(\dot{E}_{x_{in}} - \dot{E}_{x_{out}}\right)$ is the rate of total exergy transferred due to heat, work, and mass interaction and $\dot{E}_{x_D}$ is the rate of exergy destruction due to irreversibility.

Generally, exergy balance for control volume in the steady process is expressed as [28]

$$\dot{E_{x_D}} = \sum \dot{E}_{x_{in}} - \sum \dot{E}_{x_{out}} + \sum \left[\dot{Q}\left(1 - \frac{T_a}{T}\right)\right]_{in} - \sum \left[\dot{Q}\left(1 - \frac{T_a}{T}\right)\right]_{out} + \sum \dot{W}_{in} - \sum \dot{W}_{out} \tag{2}$$

where $\dot{E}_{x_D}$ is exergy destruction flow and on the right side, the first two-term are stream exergy flow, the second two-term is exergy flow due to heat transfer, and the last two terms are exergy flow due to work transfer.

The specific exergy at any state point is written as

$$E_x = (h - T_a s) - (h_o - T_a s_o) \tag{3}$$

where h_o and s_o are the enthalpy and entropy at ambient temperature T_a, h *and* s are enthalpy and entropy at any point.

Exergy efficiency is a very beneficial tool to determine the effectiveness of a thermal system in respect to its performance in reversible conditions. In other words, exergy efficiency, also known as "second law efficiency", is the ratio of actual COP to the maximum COP. For reversible processes, the exergy efficiency equals 1 and for other cases, it is less than 1. The exergy efficiency of the system is calculated by below:

$$\eta_{ex} = \frac{(COP)_{act}}{(COP)_{max}} = 1 - \frac{\left(\dot{E}_{xD}\right)_{total}}{\dot{W}_{comp}} \tag{4}$$

where $\left(\dot{E}_{xD}\right)_{total}$ is total exergy destruction of the system, $\dot{W}_{comp}$ is work input to compressor, and η_{ex} is exergy efficiency.

$$\dot{W}_{comp} = \frac{\dot{m}(h_2 - h_1)}{\eta_{mech}\eta_{ele}} = \frac{\dot{m}(h_{2c} - h_1)}{\eta_{mech}\eta_{is}\eta_{ele}} \tag{5}$$

The performance of the compressor also depends on isentropic efficiency.

$$\eta_{is} = \frac{h_{2c} - h_1}{h_2 - h_1} \tag{6}$$

The above exergy equations are performed for each component of the vapor compression refrigeration system of two evaporator systems (Fig. 1). The exergy destruction of each component is described below.

Compressor:

$$\left(\dot{E}_{xD}\right)_{comp} = \dot{E}_{x_1} - \dot{E}_{x_2} + \dot{W}_{comp} = \dot{m}(T_a(s_2 - s_1)) \tag{7}$$

Condenser:

$$\begin{aligned}\left(\dot{E}_{xD}\right)_{cond} &= \dot{E}_{x_2} - \dot{E}_{x_3}\\ \left(\dot{E}_{xD}\right)_{cond} &= (h_2 - T_a s_2) - (h_3 - T_a s_3)\end{aligned} \tag{8}$$

Flash chamber:

$$\begin{aligned}\left(\dot{E}_{x_D}\right)_{FC} &= \dot{E}_{x_3} - \dot{E}_{x_5}\\ \left(\dot{E}_{x_D}\right)_{FC} &= \dot{m}(h_3 - T_a s_3) - \left[(\dot{m}_2 + \dot{m}_1)(h_5 - T_a s_5) + \dot{m}_3(h_7 - T_a s_7)\right]\end{aligned} \tag{9}$$

Expansion valve:

$$\begin{aligned}\left(\dot{E}_{xD}\right)_{exp} &= \dot{E}_{x_5} - \dot{E}_{x_6}\\ \left(\dot{E}_{xD}\right)_{exp} &= \dot{m}_1(T_a(s_6 - s_5))\end{aligned} \tag{10}$$

Evaporator-II:

$$\begin{aligned}\left(\dot{E}_{x_D}\right)_{ev,II} &= \dot{E}_{x_5} - \dot{E}_{x_7} + \dot{Q}_{ev,II}\left(1 - \frac{T_a}{T_s}\right)\\ \left(\dot{E}_{x_D}\right)_{ev,II} &= \dot{m}_2(h_5 - T_a s_5) - \dot{m}_2(h_7 - T_a s_7) + \dot{Q}_{ev,II}\left(1 - \frac{T_a}{T_s}\right)\end{aligned} \tag{11}$$

Evaporator-I:

$$\begin{aligned}\left(\dot{E}_{x_D}\right)_{evap,I} &= \dot{E}_{x_6} - \dot{E}_{x_8} + \dot{Q}_{ev,I}\left(1 - \frac{T_a}{T_s}\right)\\ \left(\dot{E}_{x_D}\right)_{evap,I} &= \dot{m}_1(h_6 - T_a s_6) - \dot{m}_1(h_8 - T_a s_8) + \dot{Q}_{ev,I}\left(1 - \frac{T_a}{T_s}\right)\end{aligned} \tag{12}$$

Pressure regulating valve (PRV)

$$\left(\dot{E}_{xD}\right)_{PRV} = \dot{E}_{x_7} - \dot{E}_{x_9}$$

PRV assumed to be isenthalpic, so $h_7 = h_9$
$\dot{m}_4 = \dot{m}_3 + \dot{m}_2$ where $\dot{m}_3$ is vapor mass.

$$\left(\dot{E}_{xD}\right)_{PRV} = \dot{m}_4(T_a(s_9 - s_7)) \tag{13}$$

Total exergy destruction:

$$\left(\dot{E}_{xD}\right)_{total} = \left(\dot{E}_{xD}\right)_{comp} + \left(\dot{E}_{xD}\right)_{cond} + \left(\dot{E}_{xD}\right)_{FC} + \left(\dot{E}_{xD}\right)_{exp} + \left(\dot{E}_{xD}\right)_{PRV} + \left(\dot{E}_{xD}\right)_{evap,I} + \left(\dot{E}_{xD}\right)_{evap,II} \tag{14}$$

Coefficient of performance (COP) of two evaporator systems:

$$(COP)_{act} = \frac{\dot{Q}_{ev,I} + \dot{Q}_{ev,II}}{\dot{W}_{comp}} \tag{15}$$

4 Result and discussion

4.1 Simulation validation

The model developed for the simulation of the present work has been validated with the studies done by Kabul et al. [14] and Yataganbaba et al. [34] by using the same operating parameters as used by the authors. Table 3 shows the deviation of the present model from the model of Kabul et al. [14] for R600a. The input parameters and values for the validation are $T_a = 20°C$, $Q_E = 1kW$, $T_e = -10°C$, $T_c = 40°C$, $T_{cc} = 0°C$, degree of superheating = 5°C, $\eta_{is} = 80\%$, $\eta_{el} = 90\%$, $\eta_{mech} = 85\%$.

Table 3 Comparison of the present study with a reference model [14]

R600a	Reference model	Present model	Deviation
Total exergy destruction	0.3228	0.3214	−0.43%
Exergy efficiency	0.1848	0.1856	0.43%
EER	11.26	11.3	0.36%
COP	3.3	3.313	0.39%
ER (energy efficiency)	0.627	0.6295	0.39%

Table 4 Comparison of the present study with a reference model of [34]

Exergy destruction of component	Reference model	Present model	Deviation
Compressor	2.41	2.406	−0.17%
Condenser	1.78	1.785	0.28%
Expansion valve-1	0.45	0.4529	0.64%
Expansion valve-2	1.24	1.245	0.40%
Evaporator-1	0.37	0.3685	−0.41%
Evaporator-2	0.49	0.4912	0.24%
Evaporator pressure regulator	0.87	0.8689	−0.13%
Mixing chamber	0.01	0.01036	3.60%

The deviation is between 0.36 and 0.43% and the results are reasonably matching with Kabul et al. [14].

As seen from the Table 4, the present model is validated with another theoretical study of Yataganbaba et al. [34] for refrigerant R1234yf. The input parameters and values considered are $Q_{ev,I} = 10kW$, $Q_{ev,II} = 15$ kW, $T_{ev\_I}= -5°C$, $T_{ev\_II} = -18°C$, $T_{con} = 40°C$, $T_L = 5°C$ and $-10°C$, $T_H = 25°C$, $T_o = 25°C$, isentropic efficiency $= 0.75$, degree of subcooling $= 5°C$. The results of the present model satisfactorily match with that of the reference model of Yataganbaba et al. [34] with deviation lying between 0.13 and 3.6%

From the above results, it is concluded that the Engineering Equation solver (EES) software used for the present study is reliable. Engineering equation solver is a powerful software for determining the thermophysical properties of refrigerants and for solving simultaneous equations.

4.2 Performance analysis

In the present study, the thermodynamic performance is evaluated to analyze two-evaporator vapor compression system for automobile. The thermodynamic properties at each state point of the two-evaporator vapor compression system are calculated with Engineering Equation Solver. In this context, the thermophysical properties such as temperature, pressure, enthalpy, and entropy of the cycle are shown in Table 5 for refrigerant R134a in terms of being an example among the six refrigerants used for thermodynamic analysis. The values is calculated at Tev-I= 258.15K, Tev-II=268.15K, and Tcon=308.15K.

The effect of evaporator-I, evaporator-II, and condenser temperature on compressor power for all selected refrigerants is shown in Figs. 3, 4, and 5. The constant temperatures of −15°C, −5°C, and 35°C are used for evaporator-I, evaporator-II, and condensing temperature, respectively. The compressor power consumption increases with a decrease in evaporator-I and with an increase in condenser temperature. There is a slight decrease in compressor power with increase in evaporator-II temperature; however, the effect is negligible as evident from Fig. 4. This phenomenon can be attributed to the fact that with the increase in evaporator-II temperature, there is a slight reduction in the overall mass flow rate of the system. It can be inferred that the decrease in evaporator-I and increase in condensing temperature cause an enhancement in pressure ratio, leading to an increase in

Table 5 Thermophysical properties at each stage point of the two evaporators system for R134a

Stage No.	*T(K)*	*P (kPa)*	$h\left(\frac{kJ}{Kg}\right)$	$s\left(\frac{kJ}{kg*K}\right)$	*x (Quality)*	*Phase*	$\dot{m}\left(\frac{kg}{s}\right)$
1	261.3	164	244.1	0.9514	-	Superheated steam	0.0897
2	318.2	887.5	279.8	0.9514	-	Superheated steam	0.0897
3	308	887.5	100.9	0.3714	0	Saturated liquid	0.0897
4	268	243.5	93.57	0.3603	0.2392	Liquid-vapor	0.0897
5	268	243.5	45.17	0.1798	0	Saturated liquid	0.0682
6	258.2	164	45.17	0.181	0.0629	Liquid-vapor	0.0173
7	268	243.5	247.5	0.9343	1	Saturated vapor	0.0388
8	258	164	241.5	0.9414	1	Saturated vapor	0.0173
9	265.4	164	247.5	0.9644	-	Superheated steam	0.0388

Similarly, the thermophysical properties of other refrigerants are calculated for thermodynamic analysis of the two-evaporator system

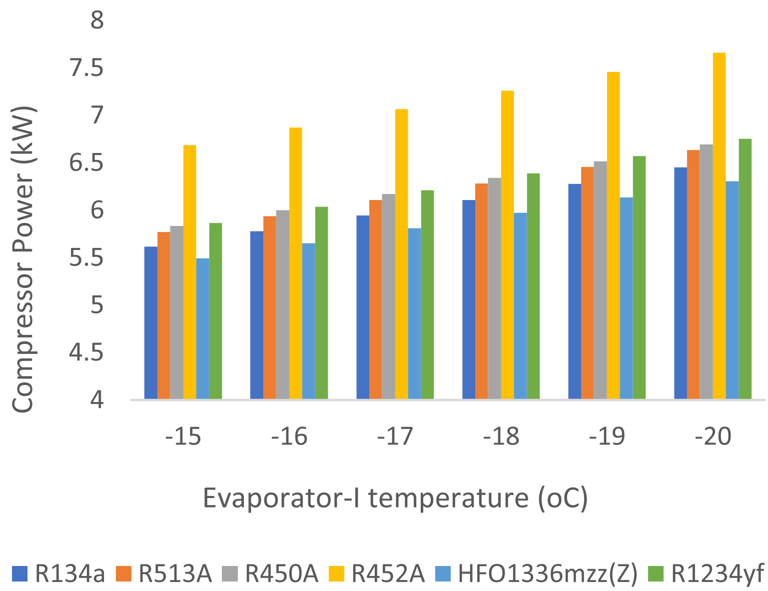

Fig. 3 Variation of compressor power as a function of evaporator-I temperature

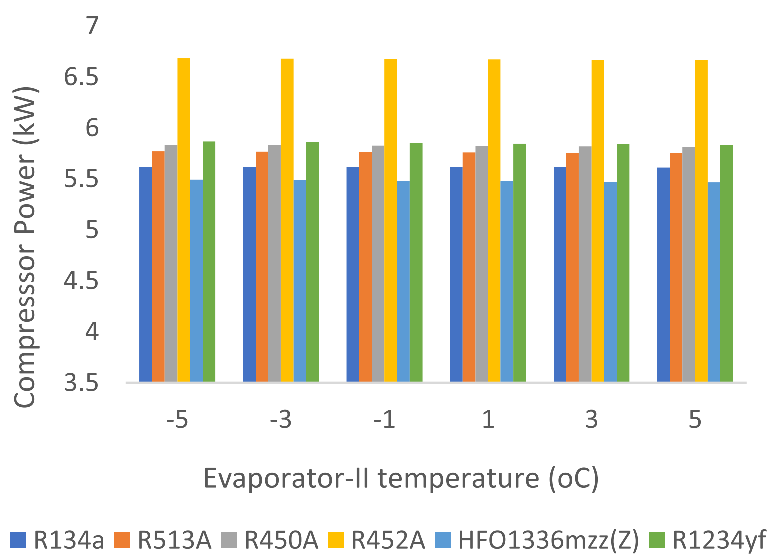

Fig. 4 Variation of compressor power as a function of evaporator-II temperature

compressor power. It can be concluded that the lower the evaporator-I and evaporator-II temperatures and the higher the condenser temperatures, the higher is the auxiliary load on the engine of the automobile. The highest compressor power is obtained with R452A and followed by R1234yf, R450A, R513A, R134a, and HFO1336mzz(z) throughout temperature variation. The power consumption of R513A, R450A, R1234yf, and R452A is 2.71–2.87%, 3.83–3.78%, 4.41–4.68%, and 18.97–18.73%, respectively, higher than R134a at evaporator-I; 2.71–2.48%, 3.83–3.62%, 4.41–3.97%, and 18.97–18.75%, respectively higher than R134a at evaporator-II; 2.71–3.33%, 3.83–3.91%, 4.41–5.49%, and 18.97–20.07%, respectively, higher than R134a at condensing temperature whereas power consumption of HFO1336mzz(Z) is lower than R134a by 2.22–2.26%, 2.22–2.60%, and 2.22–2.60% in comparison to R134a at evaporator-I, evaporator-II, and condensing temperature respectively. As per the compressor power analysis, the suitable candidate to replace with R134a are R513A, R450A, and HFO1336mzz(Z) although the compressor power of R513A and R450A is higher than R134a but their GWP is lower.

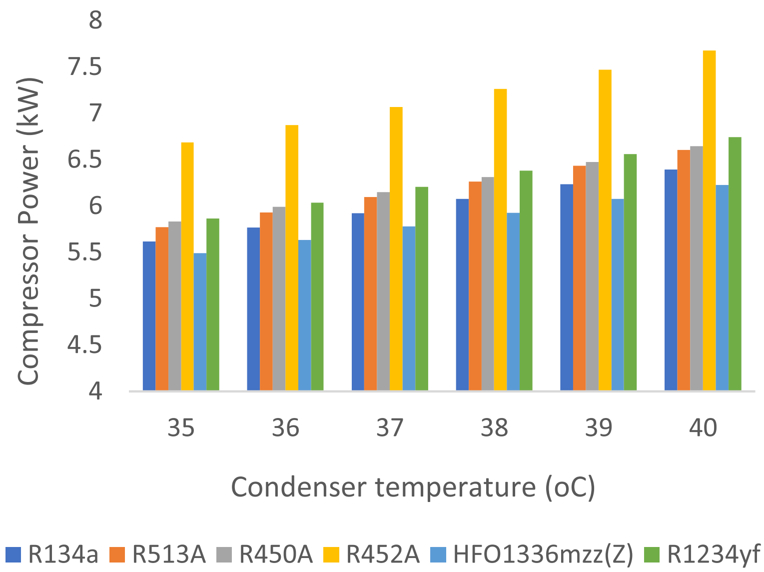

Fig. 5 Variation of compressor power as a function of condenser temperature

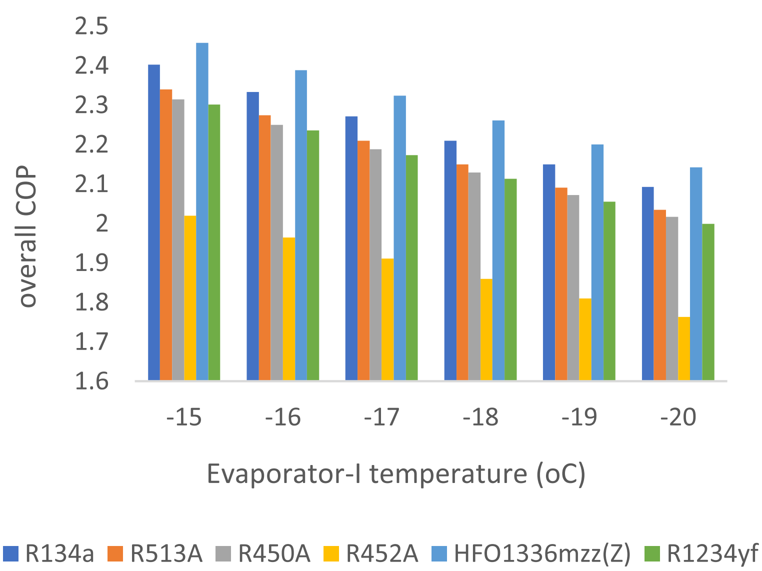

Fig. 6 Variation of overall COP as a function of evaporator-I temperature

The effects of evaporator-I, evaporator-II, and condenser temperatures on the overall COP of the system are shown in Figs. 6, 7, and 8. The result shows that the overall COP decreases with a decrease in evaporator-I and an increase in condenser temperature. There is a slight increase in overall COP of the system with evaporator-II temperature but the increase is almost negligible in comparison to evaporator-I. This is because of the fact that the variation of power consumption with evaporator-II temperature is very low. However, the overall COP is influenced by evaporator-I and condenser temperature as they affect the power consumption and the evaporation latent heat for all the refrigerants. Therefore, higher COP is obtained at lower condenser temperature and higher evaporator-I temperature. The highest and lowest COP is obtained for HFO1336mzz(Z) and R452A throughout the temperature variation. The COP of R513A, R450A, R1234yf, and R452A are 2.62–2.77%, 3.66–3.63%, 4.20–4.49%, and 15.94–15.77%, respectively, lower than R134a for evaporator-I; 2.62–2.45%, 3.66–3.53%, 4.20–3.82%, and 15.94–15.79%, respectively, lower than R134a for evaporator-II; 2.62–3.22%, 3.66–3.79%,

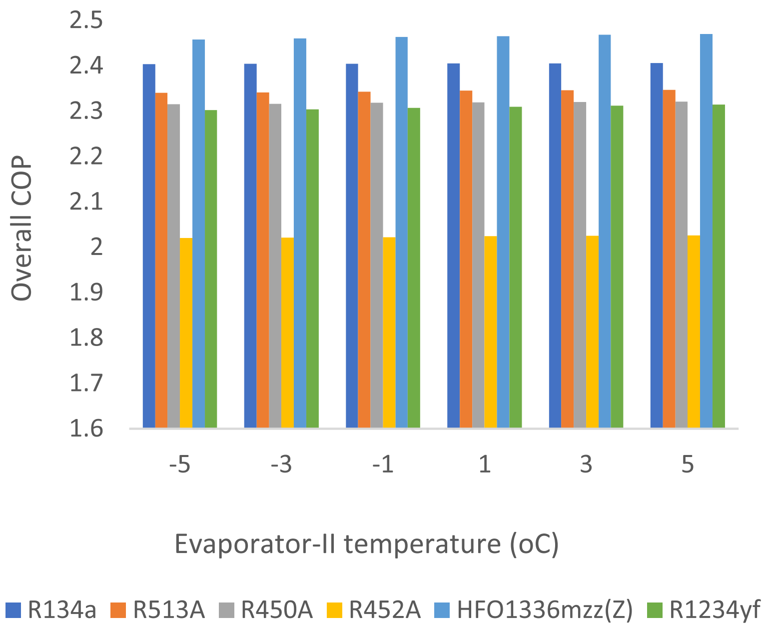

Fig. 7 Variation of overall COP as a function of evaporator-II temperature

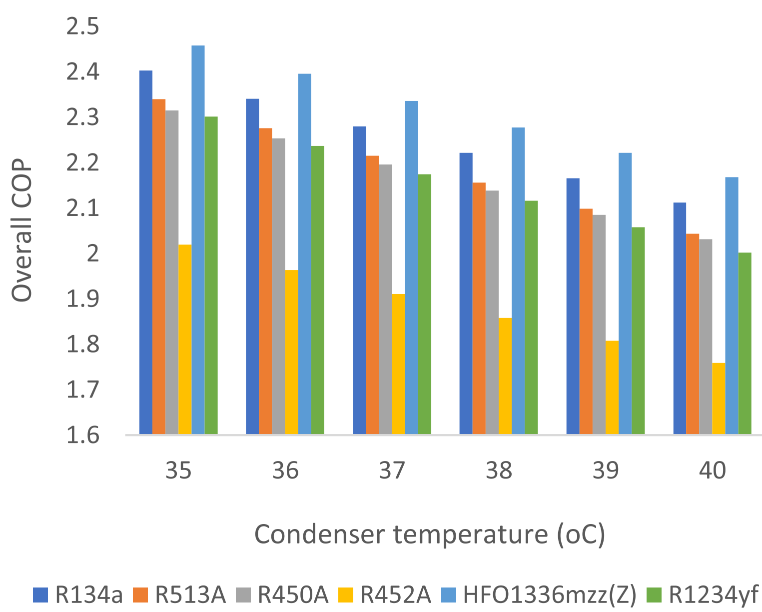

Fig. 8 Variation of overall COP as a function of condenser temperature

4.20–5.21%, and 15.94–16.71%, respectively, lower than R134a for condenser. HFO1336mzz(Z) indicates higher COP than R134a by 2.29–2.34%, 2.29–2.66%, and 2.29–2.65%, respectively, at evaporator-I, evaporator-II, and condenser temperature. R1234yf, R513A, and R450A have comparable COP to each other at all the points.

Figures 9, 10, and 11 depict the effect of evaporator-I, evaporator-II, and condenser temperature on total exergy destruction of the system. The result shows that the total exergy destruction increases with a decrease in evaporator-I temperature as the entropy flow across the evaporators increases with a decrease in evaporator-I temperature. The effect of evaporator-II temperature on the total exergy destruction can be ignored in comparison to evaporator-I as the variation of mass flow rate through evaporator-II is almost negligible with

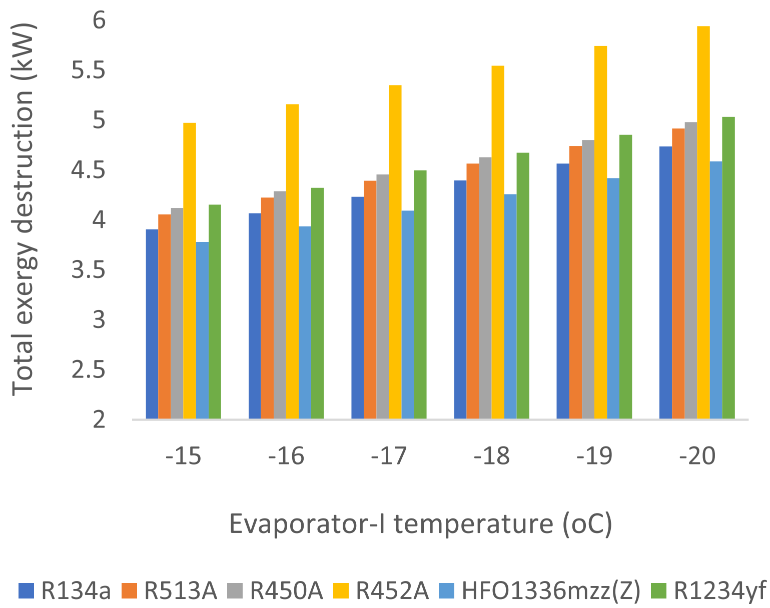

Fig. 9 Variation of total exergy destruction as a function of evaporator-I temperature

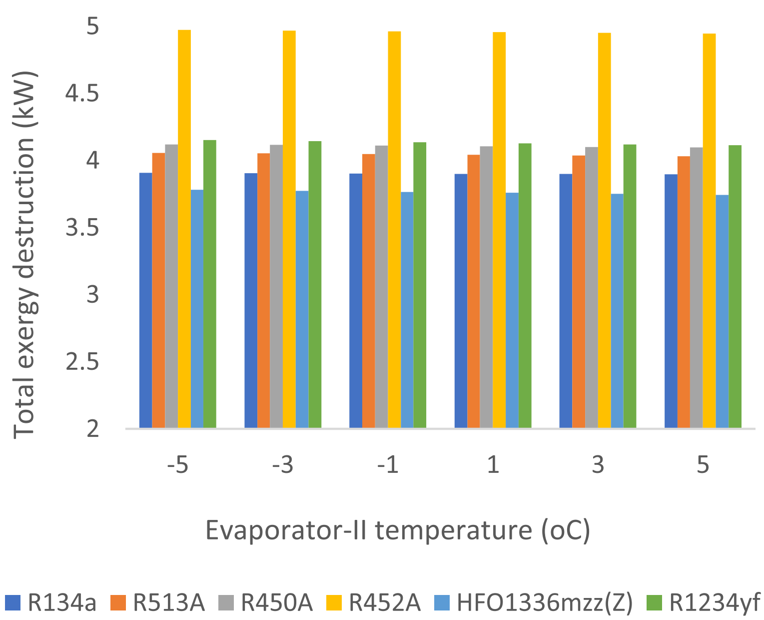

Fig. 10 Variation of total exergy destruction as a function of evaporator-II temperature

evaporator-II temperature. An increase in evaporator temperature not only decreases the mass flow rate for constant refrigerant capacity but also reduces the temperature difference between the evaporator and indoor air. This reduction causes a decrease in exergy destruction. Further, total exergy destruction increases with an increase in condenser temperature as the higher the temperature difference between the ambient and condenser temperature the higher are the exergy losses. The maximum exergy destruction is obtained for R452A followed by R450A, R513A, R1234yf, R134a, and HFO1336mzz(Z) throughout the temperature variation. The total exergy destruction of R513A, R450A, R1234yf, and R452A is obtained for evaporator-I are 3.84–3.85%, 5.45–5.11%, 6.27–6.29%, and 27.25–25.46%, respectively, higher than R134a; for evaporator-II are 3.84–3.49%, 5.45–5.13%,

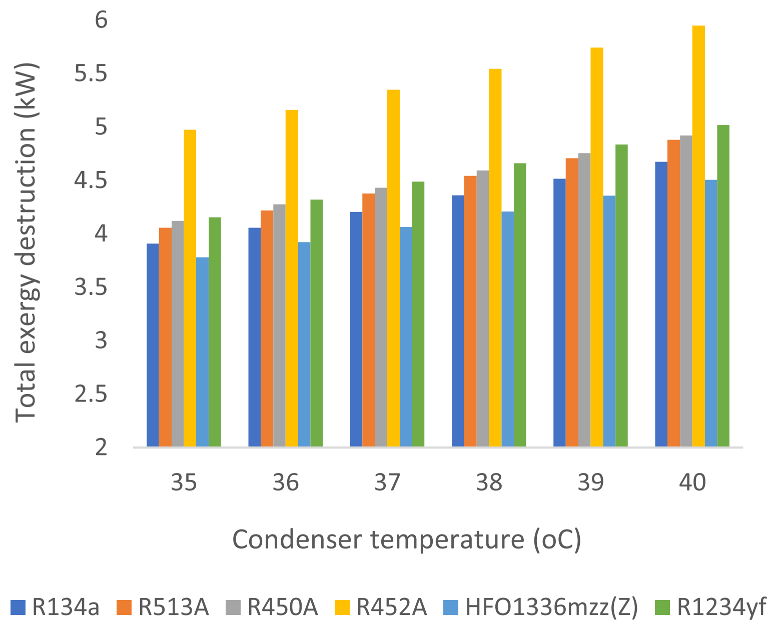

Fig. 11 Variation of total exergy destruction as a function of condenser temperature

6.27–5.57%, and 27.25–26.94%, respectively, higher than R134a and for condenser are 3.84–4.45%, 5.45–5.29%, 6.27–7.38%, and 27.25–27.31%, respectively, higher than R134a. But total exergy destruction of HFO1336mzz(Z) is 3.25–3.17%, 3.25–3.93%, and 3.25–3.57% lower than R134a at evaporator-I, evaporator-II, and condenser temperature, respectively. Therefore, the total exergy destruction of R1234yf and R450A are comparable to each other for evaporator-I and evaporator-II. As per the total exergy destruction analysis, the alternatives to R134a are HFO1336mzz(Z), R513A, R1234yf, and R450A.

Figures 12, 13, and 14 demonstrate the exergy efficiency of the system as the function of evaporator-I, evaporator-II, and condenser temperature. The exergy efficiency decreases with a decrease in evaporator-I temperature owing to an increase in the total exergy destruction. However, there is no remarkable influence of evaporator-II temperature on exergy efficiency of the

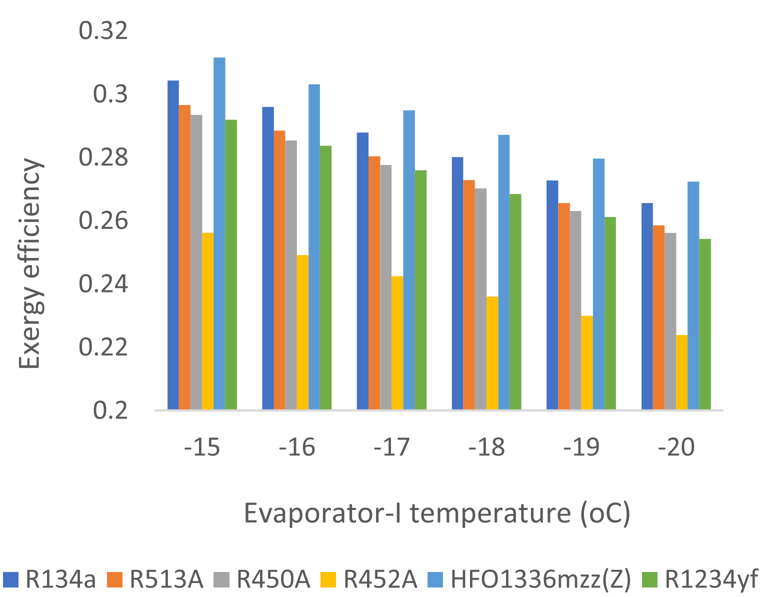

Fig. 12 Variation of exergy efficiency as a function of evaporator-I temperature

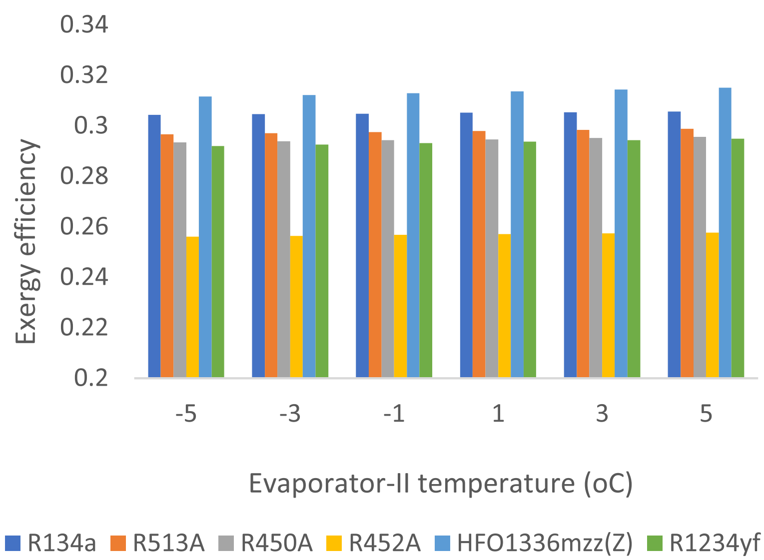

Fig. 13 Variation of exergy efficiency as a function of evaporator-II temperature

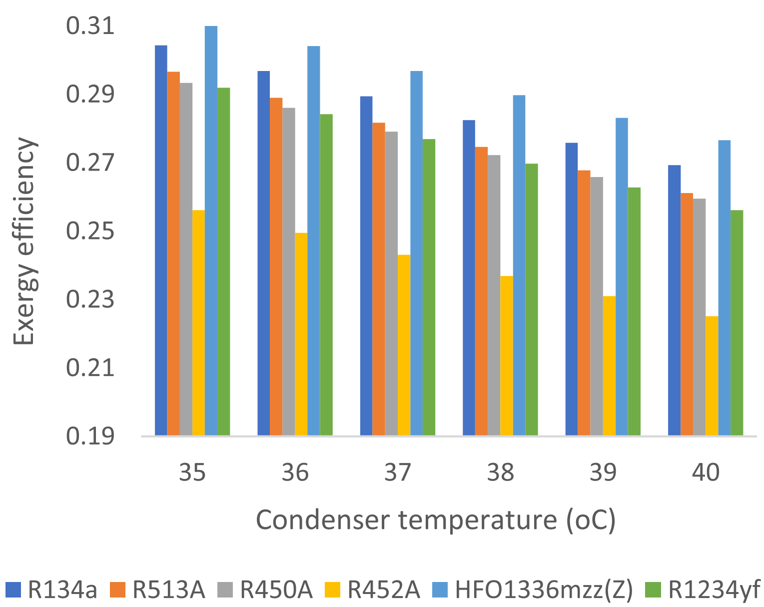

Fig. 14 Variation of exergy efficiency as a function of condenser temperature

system. This can be attributed to the fact that exergy efficiency depends on two factors—total exergy destruction and compressor power, and both of these factors vary negligibly with evaporator-II temperature. In addition, exergy efficiency decreases with an increase in condenser temperature as the difference between evaporator and condenser temperature increases, leading to a decrease in exergy efficiency. The increase in exergy efficiency can be attributed to the higher overall COP of the system. The higher exergy efficiency is obtained with HFO1336mzz(Z) followed by R134a, R513A, R450A, R1234yf, and R452A throughout temperature variation. Among these refrigerants, the exergy efficiency performance of R513A, R450A, and R1234yf is comparable to each other. The relative exergy efficiency of R513A, R450A, R1234yf, and R452A are found to be 2.53–2.67%, 3.58–3.54%, 4.07–4.25%, and 15.84–15.70%, respectively, lower than R134a for evaporator-I; 2.53–2.23%, 3.58–3.27%, 4.07–3.50%, and 15.84–15.67%, respectively, lesser than R134a for evaporator-II and 2.53–3.01%, 3.58–3.64%, 4.07–4.90%, and 15.84–16.41%, respectively, lower than R134a for condenser. However, the exergy efficiency of HFO1336mzz(Z) for evaporator-I, evaporator-II, and condenser temperature is higher than R134a by 2.40–2.52%, 2.40–3.08%, and 2.40–2.71%, respectively.

4.2.1 Component-wise exergy destruction for two evaporator vapor compression system

An analysis to determine the energy loss in each component of the two evaporator VCRS is carried out. The percentage contribution of each system component to the total exergy destruction is plotted in Fig. 15. The compressor is the highest contributor, accounting for more than half of the total exergy destruction (56.00 to 61.84%), whereas the expansion valve is the lowest (0.42 to 0.54%). The exergy destruction in the compressor is highest for HFO1336mzz(Z) and lowest for R452A, respectively. The exergy destruction in the expansion valve is the least for R452A. The absolute value of exergy destruction for the compressor varies from 2.338 to 2.785 kW. The second highest contributor is the flash chamber (FC), accounting for 9.38 to 11.84% of total exergy destruction for refrigerant R513A, R450A, R1234yf, and R452A while the highest value was obtained for R452A. But for R134a and HFO1336mzz(Z), the second highest is obtained for PRV. The maximum total exergy destruction is obtained with R452A and minimum for HFO1336mzz(Z) as shown in Table 6. Therefore, it can be concluded that the compressor, flash chamber, and PRV play a key role in determining the performance of the refrigerants and the whole system.

5 Conclusion

This study accounts for the emerging need for refrigeration in automobiles and theoretically evaluates the performance of a two-evaporator system to fulfill the need for both air conditioning and refrigeration. The comparative analysis of environmentally

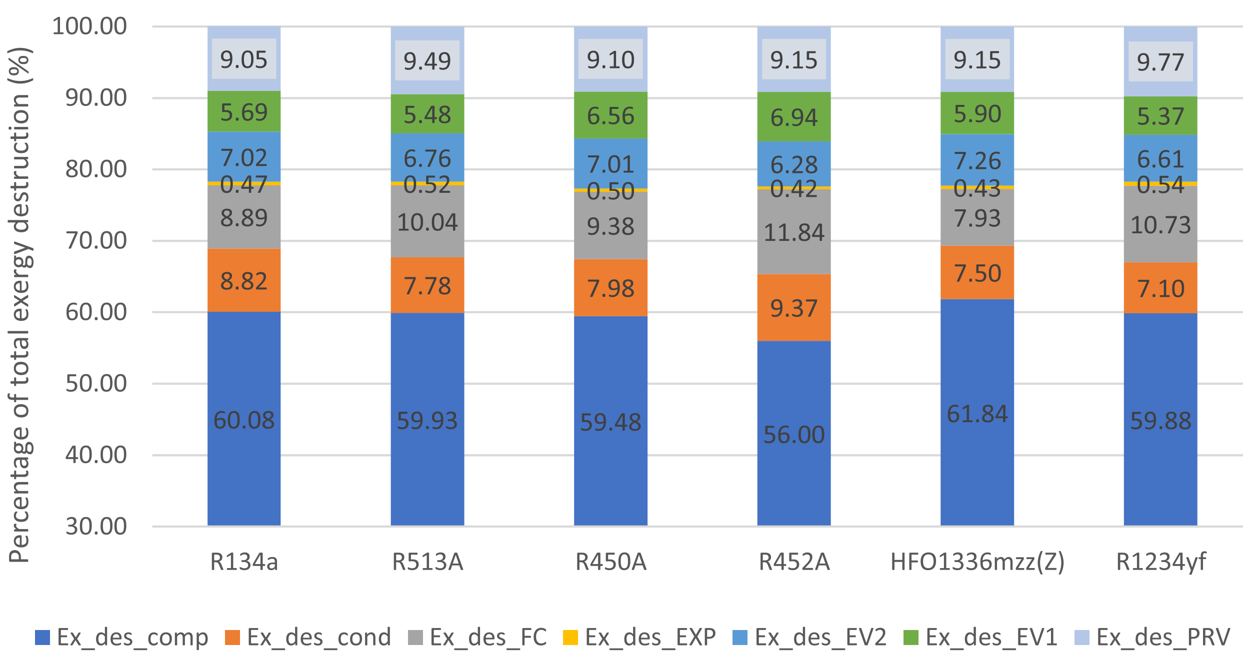

Fig. 15 Percentage contribution of total exergy destruction in each component ($T_{ev\text{-}I} = -15°C$, $T_{ev\text{-}II} = -5°C$ and $T_{cond} = 35°C$)

Table 6 The results of exergy destruction of each component ($T_{ev\text{-}I} = -15°C$, $T_{ev\text{-}II} = -5°C$, $T_{cond} = 35°C$) with the help of EES

Refrigerants	Comp.	Cond.	EV-II	EV-I	FC	EXP	PRV	Total
R134a	2.348	0.3445	0.2744	0.2224	0.3474	0.01839	0.3537	3.908
R513A	2.432	0.3159	0.2742	0.2225	0.4075	0.02118	0.3851	4.058
R450A	2.451	0.3287	0.2887	0.2703	0.3866	0.02074	0.3751	4.121
R452A	2.785	0.4659	0.3121	0.3449	0.589	0.02089	0.455	4.973
HFO1336mzz(Z)	2.338	0.2837	0.2746	0.2231	0.2998	0.01619	0.3459	3.781
R1234yf	2.487	0.2949	0.2746	0.223	0.4456	0.02224	0.4059	4.153

friendly refrigerants such as R513A, R450A, R1234yf, and HFO1336mzz(Z) is also performed against R134a and R452A. The effect of operating parameters such as evaporator-I, evaporator-II, and condenser temperatures on performance variables including power consumption, overall COP, total exergy destruction, and exergy efficiency is investigated. Finally, exergy destruction in each component of the system is determined. The following conclusions are drawn from this study:

- The COP and exergy efficiency of the system varies from 2.46 to 1.99 and 31.16 to 25.43% respectively for refrigerants HFO1336mzz(Z), R134a, R1234yf, R450A, and R513A throughout temperature variation with the lowest obtained at −20°C evaporator-II temperature.
- Work consumption and total exergy destruction of the system varies from 5.49 to 6.75 kW and 3.78 to 5.04 kW respectively for HFO1336mzz(Z), R134a, R513A, R450A, and R1234yf throughout temperature variation with the highest obtained at −20°C evaporator-II temperature.
- HFO1336mzz(Z) shows the best performance among all selected refrigerants. By using it, there is an increase in COP and exergy efficiency of the system by 2.34–2.62% and 2.52–2.95%, while there is a reduction in power consumption and exergy destruction by 2.53–2.26% and 3.25–3.17% respectively compared to R134a.
- R452A exhibits the worst thermodynamic performance at all evaporator-I, evaporator-II, and condenser temperatures.
- The thermodynamic performance of R1234yf and R450A are comparable to each other and these refrigerants are better alternatives to R134a because of their very low GWP.
- R134a shows the lowest total exergy destruction among the selected refrigerants after HFO1336mzz(Z).
- R134a shows the highest COP and exergy efficiency after HFO1336mzz(Z), while the exergy efficiency and COP of R450A and R513A are comparable to each other.
- The compressor and expansion valve contribute the highest and lowest, respectively, to the total exergy destruction.

In conclusion, a two-evaporator vapor compression refrigeration system is a better alternative to two separate systems in automobiles to achieve both refrigeration and air-conditioning. The merit of the cycle is that it runs with single compressor to give two different evaporation temperature - for air conditioning at evaporator-II and refrigeration at evaporator-I. This design is more space-efficient considering the space constraints in automobiles, more economical due to the high initial cost of using two systems, and more energy-efficient and hence, more environmentally friendly than the two individual systems.

HFO1336mzz(Z), R1234yf, R450A and R513A exhibit the optimum energy and exergy performance and are more environmentally friendly refrigerants. Therefore, as per this study, they can be the better alternatives to R134a in two evaporator vapor compression systems.

6 Nomenclature

Symbols

$\dot{Q}$Heat transfer rate [kW]

$\dot{W}$Work rate [kW]

$\dot{m}$Mass flow rate [kg s^{-1}]

sSpecific entropy [kJ kg^{-1} K^{-1}]

hSpecific enthalpy [kJ kg^{-1}]

TTemperature [°C]

$\dot{E}_x$Exergy destruction rate [kW]

ηEfficiency

PPressure [kPa]

Subscripts

Ddestruction

ininlet

outoutlet

actactual

maxmaximum

mechmechanical

eleelectrical

exexergy

isisentropic

compcompressor

condcondenser

evevaporator

expexpansion valve

Abbreviations

COP	Coefficient of performance
VCRS	Vapor compressor refrigeration system
EER	Energy efficiency ratio
AAC	Automobile air conditioning
HFO	Hydrofluoro-olefin
HFC	Hydrofluorocarbon
NBP	Normal boiling point
ODP	Ozone depletion potential
GWP	Global warming potential

Authors' contributions

The authors read and approved the final manuscript.

Funding

There is no funding support for this research.

Availability of data and materials

All the figures and data are included in the manuscript.

Declarations

Competing interests
The authors declare that they have no competing interests.

Received: 25 April 2022 Accepted: 13 December 2022
Published online: 11 January 2023

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