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Research Article

Thermal Performance Analysis of Low-GWP Refrigerants in Automotive Air-Conditioning System

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The energy and exergy of low-global warming potential (GWP) refrigerants were investigated experimentally and theoretically. Refrigerants with a modest GWP_{100} of ≤ 150 can be sufficient for bringing down emissions which were concerned for the automotive air-conditioning system. Three types of low-GWP refrigerants, R152a, R1234yf, and R1234ze(E), were examined with particular reference to the current high-GWP of R134a. The effect of different evaporating and condensing temperatures in addition to compressor speed was considered. The purpose was to bring a clear view of the performance characteristics of possible environment friendly alternatives of R134a. The analysis was carried out with compressor power, cooling capacity, coefficient of performance, exergy destruction, and exergy efficiency. It was noted that the total exergy destruction of R1234yf was reduced by 15% compared to that of R134a. The refrigerant R1234ze(E) has the highest energetic and exergetic performance compared with the other investigated refrigerants.

1. Introduction

The greenhouse gases of the refrigerants became the most issue in scientific research due to their impact on the environment. The industrial gases which are released in the environment are comparable with CO_2 gas, which is identified by a high global warming potential (GWP_{100}) equal to one. The gas with a higher value of global warming potential warms the earth more than the CO_2 gas. The current third generation of hydrofluorocarbon refrigerants in the automotive industry is characterized by a zero ozone depletion potential and a high global warming potential when released to the atmosphere. So, there was a great need to find out an alternative to the current R134a ($GWP_{100} = 1430$) under the Kyoto protocol and the Montreal protocol [1]. Under the agreement of the Montreal Protocol, the phasedown of R134a was agreed by a reduction of the production rate as 7% in 2016, 37% in 2018, 55% in 2021, 69% in 2024, and 76% in 2027, with the phaseout virtually complete in 2030 [2]. From 2017, according to a directive of the European Union, automotive manufacturers manufactures cars with environment-friendly alternative refrigerants with $GWP_{100} \leq 150$. In the United States, by 2020 newly manufactured cars will be equipped with air-conditioning system containing environment friendly refrigerants with $GWP_{100} \leq 150$ [2]. Globally, there are more demands than ever to reduce greenhouse gases to protect the environment and to find out a new generation of environmentally friendly refrigerants of low GWP, which is investigated here. So, the energy and exergy analysis of the automotive air-conditioning cycle which is based on the first law of thermodynamics to analyze the use of energy and on the dynamic analysis which is based on the second law of thermodynamics provides an alternative means of assessing energy efficiency for each part of the refrigeration cycle and compares operations rationally [3].

Many researchers in the past decade have concerned about the energy and exergy analysis of the refrigerants with a low global warming potential ($GWP_{100} < 150$) according to Europe's recommendation. Cho and Park [4] studied experimentally the energy and exergy analysis of the refrigerant R1234yf and compared with R134a used in automotive air conditioning. The experiments were performed with a variable speed compressor, and the refrigeration cycles contained a heat exchanger. Both the cooling capacity and the coefficient of performance of R1234yf were reduced by 4–7% and 3.6–4.5%, respectively, compared with the R134a system. Jemaa et al. [5] performed a theoretical study using the Engineering Equation Solver (EES) software to analyze the chiller refrigeration cycle using R1234ze(E) as an alternative to R134a. Both the energy and exergy were analyzed at different evaporating and ambient temperatures. Zhang et al. [6] studied the energy and exergy of a zeotropic mixture of R32 and R236fa refrigerants used in a 4 kW chiller. The experiments with different concentration ratios of R32 to R236fa were carried. The exergy loss for each component of the chiller refrigeration cycle was discussed.

A review of alternatives to the R134a refrigerant was carried out by Verma et al. [7]. From the environmental point of view, the refrigerant with a low total equivalent warming impact factor (TEWI) of the investigated alternative refrigerants was the most suitable one in a straight drop-in substitute for R134a. Garcia et al. [8] performed a comparison study of the transient response for the R1234yf refrigeration cycle as a replacement for R134a. It was concluded that there was a similar dynamic behavior between the refrigerants R134a and R1234yf.

The performance of the automotive air-conditioning system for the three types of refrigerants, R134a, R290, and R1234yf, at different operating conditions was studied by Navarro et al. [9]. It was concluded that a significant improvement in a compressor and volumetric efficiencies was obtained with R290 compared with R134a. Navarro-Esbrí et al. [10] investigated experimentally the R1234yf refrigerant in a refrigeration system as a replacement for R134a at different evaporating temperatures, condensing temperatures, and compressor speeds with and without a heat exchanger. There was a reduction in cooling capacity by 9% when compared with R134a at the same operating conditions. Gomaa [11] carried out a comparative study between R152a, R1234yf, and R1234ze(E) with a baseline of R134a in an automotive air-conditioning system. It was noted that the performance of R1234yf was very close to that of R134a when compared with the performance of R152a and R1234ze(E), respectively. The performance of the R1234yf refrigeration system in automotive air conditioning was studied by Lee and Jung [12]. They noted that there was a reduction in COP of the refrigeration system with R1234yf by 4.0% lower than that of R134a.

Belman-Flores et al. [13] performed an energetic and exergetic study on a domestic refrigerator with R1234yf as a replacement for R134a. They developed a thermodynamic computational model which enables to calculate the refrigeration cycle parameters at different operation conditions involving the exergy destruction ratio and exergy efficiency. The exergy destruction was mainly concentrated in the compressor, especially for the refrigerants R1234yf and R134a. Joybari et al. [14] conducted an exergetic study for a domestic refrigerator with R134a. It was found that the highest exergy destruction takes place in the compressor followed by the condenser, capillary tube, and evaporator.

Sánchez et al. [15] presented an experimental study on the energy performance evaluation of four low-GWP refrigerants compared with a high GWP of R134a as baseline. The four refrigerants R1234yf, R1234ze(E), R600a, R290, and R152a were tested at different condensing and evaporating temperatures. Shaik and Ashok Babu [16] performed a theoretical study using the MATLAP code on four low-GWP alternatives of R22 in residential air conditioners. The thermodynamic performance of the investigated refrigerants was compared at different evaporating temperatures. Li et al. [17] performed a comparative study on energy efficiency of R717, R600a, and R1234yf as low-GWP refrigerants compared with R134a in domestic refrigerators. They concluded that R1234yf has similar performance to R134a which can be considered as a drop-in alternative.

Vali et al. [18] performed an analytical study on the performance parameters of a refrigeration system with R22, R32, R134a, R152a, R290, and R1270. From environmental point of view, the R1270 was a more suitable refrigerant to replace R22.

In the present study, the experimental and theoretical investigation was carried out for three different types of refrigerants, which are considered as one of the most environmentally friendly refrigerants in automotive air conditioning applications. Due to the multiple operating conditions during the year of the automotive air conditioning, the study was extended to cover a wide range of compressor speed, refrigerant flow rate, and evaporating and condensing temperatures. The energetic and exergetic performance with a low-GWP_{100} equal to 150 or less was the main point of interest. The low-GWP refrigerants of hydrofluorocarbon (R152a) and the two of a very low-GWP of the fourth generation refrigerants which are hydrofluoroolefins of R1234ze(E) and R1234yf were investigated in this research, with particular references to the present HFC-R134a (R134a). A weak double bond in hydrofluoroolefin refrigerants allows for short atmospheric life [19] while maintaining stability in the system, as illustrated in Figure 1. The atmospheric lifetime of the refrigerants is useful to measure the time it takes to leave the atmosphere as greenhouse gases. Table 1 describes the thermodynamic and environmental properties of the investigated refrigerants [19, 20].

2. Experimental Setup and Procedure

The experimental test rig comprises of a closed-loop circuit of the R134a refrigeration system, a open-loop circuit of the ducted air-cooled condenser, and a open-loop circuit of ducted air passing through a fin and tube evaporator as shown in Figure 2. The closed-loop refrigeration cycle of R134a consists of a variable speed semihermetic compressor, air-cooled condenser, expansion valve, liquid receiver, filter drier, flowmeter, and evaporator. To control and vary the compressor speed, a frequency inverter was connected to the

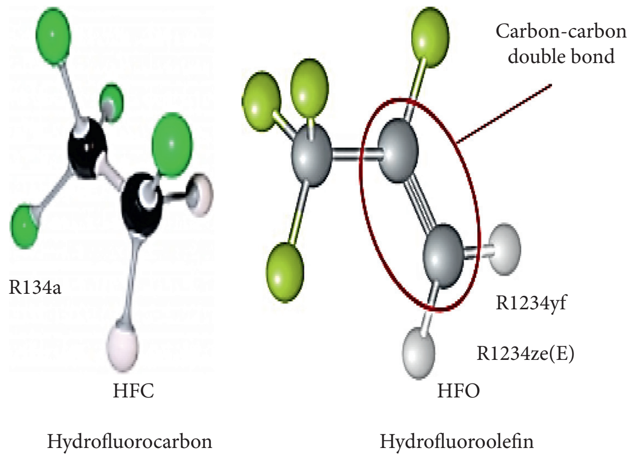

Figure 1: Atomic bond of the HFC and HFO refrigerants [19].

Table 1: Thermodynamic and environmental properties of the investigated refrigerants [19, 20].

Item	R152a	R1234yf	R1234ze(E)	R134a
Molecular weight (kg/kmol)	66	114	114	102
ASHRAE safety classification	A2	A2L	A2L	A1
Critical temperature (°C)	113	95	109	101
Boiling point (°C)	−24.0	−29	−19	−26
Critical pressure (kPa)	4580	3382	3636	4059
ODP	0	0	0	0
100-year GWP (GWP_{100})	140	4	6	1430
Atmospheric lifetime (years)	0.6	0.03	0.05	14

electric drive of the compressor. The power utilization of the compressor was measured with a wattmeter having an accuracy of ±1%. The second circuit was the ducted air-cooled condenser, which was equipped with measuring instruments to allow measurement of air temperature and velocity on the condenser airside. The duct was incorporated with a variable speed axial fan. The inlet temperature condition of the condenser varied according to the heat supplied from the electric heater which was inserted before the condenser coil.

The third open-loop circuit was the duct-containing evaporator. The cooled air supplied from the evaporator coil passed through the duct in which the air duct was equipped with a variable speed axial fan and an electric heater to control the evaporator load. Two voltage regulators were used to adjust the airspeed and air temperature to a required value through the evaporator by controlling the voltage across the DC motor of the fan and the heater, respectively. Twenty thermocouples (type-k) of accuracy ±0.5°C were inserted upstream and downstream of the evaporator and condenser, respectively, in accordance with ASHRAE recommendation. A data acquisition system connected to the thermocouples was used to measure the temperature. The air velocity was measured in both evaporator and condenser ducts by a hotwire anemometer with an accuracy of ±0.1%. The refrigerant flowmeter with an accuracy of ±1% was connected through the refrigeration cycle to measure the refrigerant flow rate at different compressor speeds. The refrigerant pressure before and after the compressor was recorded with a high- and low-pressure gauge with an accuracy of ±1%.

3. Uncertainty Analysis

The implication of the experimental error specifies the error of the measuring and calculated quantities. For the different parameters, the uncertainty analysis was established according to Holman [21]. For independent variables (X_1, $X_2, X_3, \ldots, X_n$), given $Y_1, Y_2, Y_3, \ldots, Y_n$ uncertainties and W_R was the uncertainty in the result , which can be calculated as

$$Y_R = \left[\left(\frac{\partial R}{\partial X_1} Y_1\right)^2 + \left(\frac{\partial R}{\partial X_2} Y_2\right)^2 + \left(\frac{\partial R}{\partial X_3} Y_3\right)^2 + \cdots + \left(\frac{\partial R}{\partial X_n} Y_n\right)^2\right]^{1/2}. \tag{1}$$

The uncertainty values of measuring and calculating parameters are given in Table 2.

4. Energy Analysis

The energy balance was a statement of the energy conservation law (first law of thermodynamics) while the exergy balance was a statement of the energy degradation (second law of thermodynamics) [22]. The representative of thermodynamic refrigeration cycle is illustrated in Figure 3 in which the cooling capacity (Q_{evap}) was given as

$$Q_{\text{evap}} = \dot{m}_{\text{ref}} \left(h_1 - h_4\right), \tag{2}$$

in which the refrigerant mass flow rate depends on the volumetric efficiency, stock volume, and specific volume of the refrigerant at the suction point as

$$\dot{m}_{\text{ref}} = \frac{60\, \eta_v V_{\text{th}}}{v_1\, \text{RPM}}. \tag{3}$$

The volumetric cooling capacity (VCC) is defined as the cooling capacity per unit refrigerant volume at the exit of the evaporator, and it can be calculated as follows:

$$\text{VCC} = \rho_1 \left(h_1 - h_4\right). \tag{4}$$

The compressor power was given by

$$W_{\text{comp}} = \dot{m}_{\text{ref}} \left(h_2 - h_1\right). \tag{5}$$

At the compressor outlet, the actual specific enthalpy of the superheated vapor refrigerant (h_2) is calculated as

$$h_2 = h_1 + \frac{\left(h_{2,\text{is}} - h_1\right)}{\eta_{\text{is,comp}}}. \tag{6}$$

The isentropic efficiency of the compressor ($\eta_{\text{is,comp}}$) was taken as 0.65 [23].

The coefficient of performance was defined as

$$\text{COP} = \frac{Q_{\text{evap}}}{W_{\text{comp}}}. \tag{7}$$

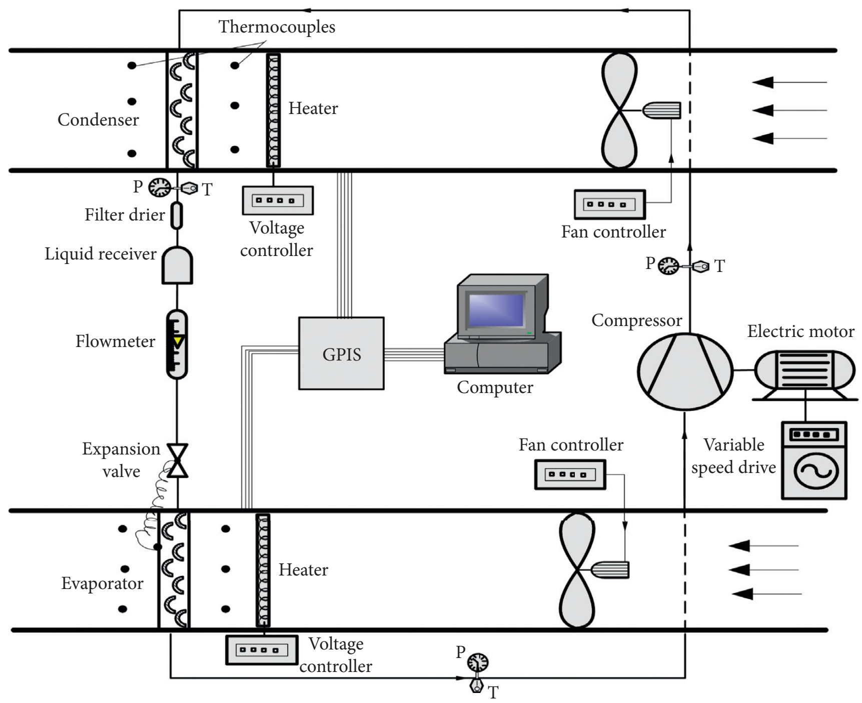

FIGURE 2: Schematic diagram of the experimental test rig.

TABLE 2: Uncertainties of the measuring instruments and calculated parameters.

Item	Uncertainty (%)
Thermocouples (type-k)	±2.1
Hotwire anemometer	±3.3
Refrigerant flowmeter	±2.9
Refrigerant pressure gauge	±3
Refrigeration capacity	±6.3
Compressor power	±7.2
COP	±5.6
Heat rejection	±7.3
Total energy destruction	±7.9

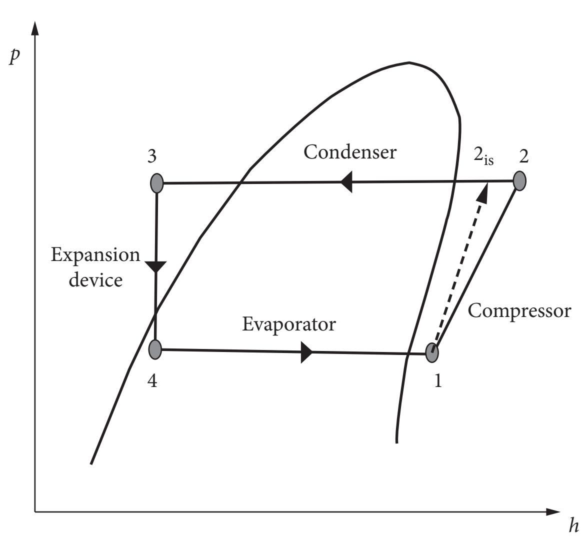

FIGURE 3: Pressure-enthalpy diagram of the air-conditioning refrigeration cycle.

5. Exergy Analysis

Exergy analysis is a method for determining the availability of the energy that can be used in a certain system in which the deviation of the refrigeration cycle state from a given situation to the reference situation. Exergy analysis is an effective tool to find out where and how much of the input energy of a system was lost. The exergy balance was a statement of energy degradation (second law of thermodynamics). The following assumptions were considered in the system exergy analysis:

(i) The steady-state conditions were satisfied for all system components.
(ii) The pressure losses in the pipelines were neglected.
(iii) The kinetic energy, potential energy, and exergy losses were not considered.
(iv) The heat gain and heat loss from the system were ignored.

A graphical presentation of exergy balance through the refrigeration cycle of the automotive air-conditioning system is illustrated in Figure 4. The mathematical representative of the exergy (second law analysis) can be expressed as

$$\dot{E}x_{dest} = \sum \dot{E}x_{in} - \sum \dot{E}x_{out} + \sum\left[\dot{Q}\left(1 - \frac{T_o}{T}\right)\right]_{in} - \sum\left[\dot{Q}\left(1 - \frac{T_o}{T}\right)\right]_{out} + \sum \dot{W}_{in} - \sum \dot{W}_{out}. \quad (8)$$

The exergy efficiency η_{Ex} is a very useful measure of the extent to which the cycle approaches an ideal behavior. The exergy efficiency is the ratio of the actual COP to the maximum possible COP at the same operating condition [24]:

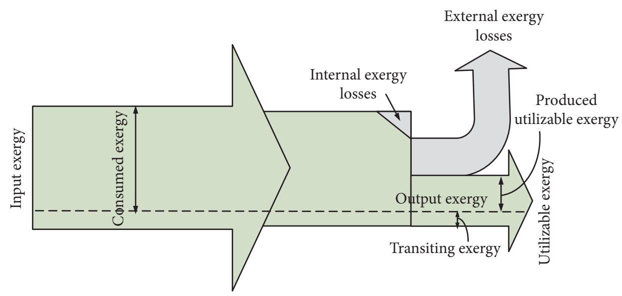

FIGURE 4: Graphical presentation of exergy balance [24].

$$\eta_{Ex} = \frac{\dot{E}x_{out}}{\dot{E}x_{in}} = 1 - \frac{\dot{E}x_{dest}}{\dot{E}x_{in}}. \tag{9}$$

When the thermodynamic process is reversible, the exergy efficiency $\eta_{Ex} = 1$ and the exergy efficiency < 1 in other cases. Thermodynamically, the exergy in any state was given by

$$\dot{E}x = (h - h_o) - T_o(s - s_o). \tag{10}$$

The above equations were applied to each thermodynamic process of the refrigeration cycle in which the exergy destruction and efficiency can be expressed as follows.

Compressor:

$$\begin{aligned} \dot{E}x_{dest\,Comp} &= \dot{W}_{comp} + \dot{E}x_1 - \dot{E}x_2, \\ \eta_{Ex\,Comp} &= 1 - \left(\frac{\dot{E}x_{dest\,Comp}}{\dot{W}_{in}}\right). \end{aligned} \tag{11}$$

Condenser:

$$\begin{aligned} \dot{E}x_{dest\,Cond} &= \dot{E}x_2 - \dot{E}x_3, \\ \eta_{Ex\,Cond} &= 1 - \left(\frac{\dot{E}x_{dest\,Cond}}{\dot{E}x_2 - \dot{E}x_3}\right). \end{aligned} \tag{12}$$

Expansion valve:

$$\begin{aligned} \dot{E}x_{dest\,Exp} &= \dot{E}x_3 - \dot{E}x_4, \\ \eta_{Ex\,Exp} &= 1 - \left(\frac{\dot{E}x_3}{\dot{E}x_4}\right). \end{aligned} \tag{13}$$

Evaporator:

$$\begin{aligned} \dot{E}x_{dest\,Evap} &= \left(\dot{E}x_4 - \dot{E}x_1\right) + \left[\dot{m}(h_1 - h_4)\left(1 - \frac{T_o}{T_1}\right)\right], \\ \eta_{Ex\,Evap} &= \frac{\dot{E}x_{dest\,Evap}}{\dot{E}x_1 - \dot{E}x_4}. \end{aligned} \tag{14}$$

The total energy destruction for all refrigeration cycle components can be expressed as

$$\dot{E}x_{dest,total} = \dot{E}x_{dest.Comp} + \dot{E}x_{dest.Cond} + \dot{E}x_{dest.Exp} + \dot{E}x_{dest.Evap}. \tag{15}$$

The Engineering Equation Solver [25] software was used with the previous equations to develop a solution model of each investigated refrigerant with different input parameters at all state points of temperatures and refrigerant mass flow rate corresponding to and similar to that of experiments.

6. Results and Discussion

The results of the low-GWP refrigerants in the automotive air-conditioning system with R134a as baseline were presented as an energetic performance involving cooling capacity, compressor power, and coefficient of performance for different compressor speeds and evaporating and condensing temperatures. The exergetic performance is presented in the form of exergy destruction of each cycle component, total exergy destruction, and exergy efficiency at different cases of refrigerant flow rate and evaporating and condensing temperatures.

6.1. The Effect of Varying Compressor RPM. The air conditioning in the automotive application and in most cases the compressor is semihermetic, which is usually connected to the engine crankshaft by a belt via two pulleys in which the compressor RPM varies according to crankshaft RPM, which in turn affects the refrigerant flow rate. The effect of varying compressor rotation of R134a on the compressor power at various condensing temperatures is shown in Figure 5. The compressor power consumption decreases with a lower value of the compressor speed and condensing temperature. The ambient air temperature affects the compressor power directly. As the condensing temperature increased by 5°C, the compressor power increased by 13% in which there was an increase in compressor power by 17% when the RPM of the compressor speed accelerated by 15%. Figure 6 illustrates experimentally both the cooling capacity and COP of R134a with the compressor RPM at various condensing temperatures. The cooling capacity increases with the increase of compressor RPM due to the increase in refrigerant mass flow rate. An increase in the condensing temperature by 5°C led to a reduction in the cooling capacity and the COP by 9% and 27%, respectively. The decrease in compressor RPM led to increase in COP of the refrigeration cycle, which can be revealed to the decrease in compressor RPM which yielded less friction between moving parts and consequently a higher isentropic efficiency of the compression process was obtained.

The investigation of a wide range of operating conditions on the automotive air-conditioning system with different refrigerant types of $GWP_{100} < 150$ was studied using Engineering Equation Solver software (EES, 2017). A validation between the experimental and theoretical (EES) results was performed at the same operating condition which was in fair agreement. Therefore, the characteristics of the refrigerants

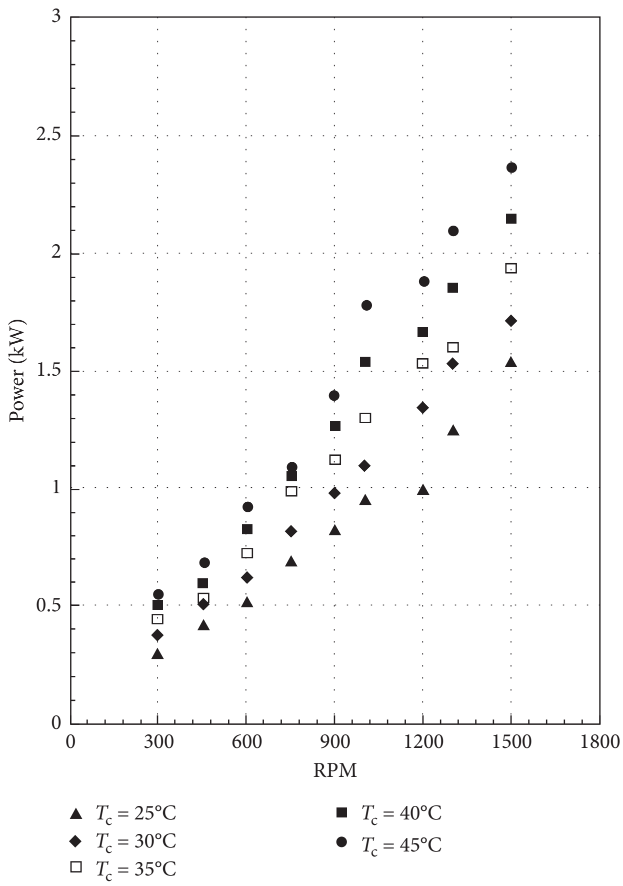

Figure 5: Compressor power versus RPM of R134a (experimental results).

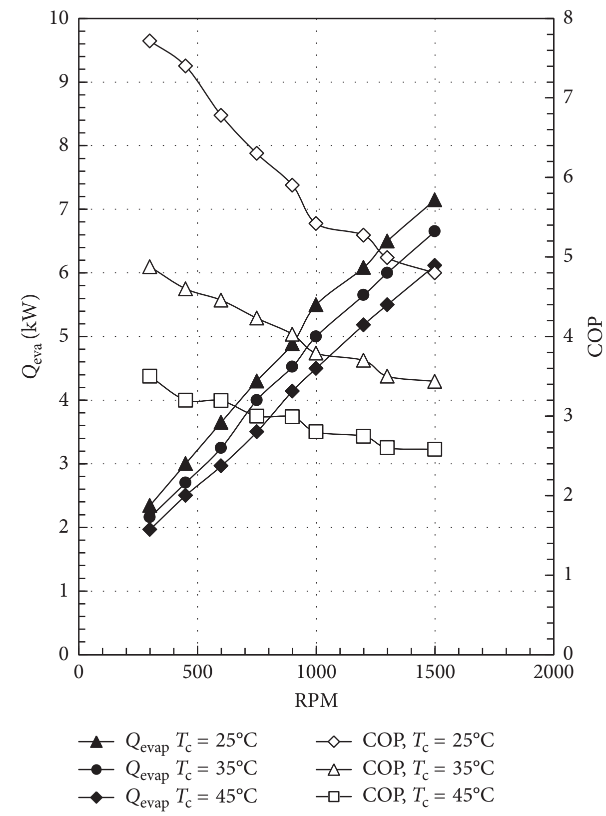

Figure 6: Cooling capacity and COP versus RPM of R134a (experimental results).

R152a, R1234yf, and R1234ze(E) were investigated in comparison with a current high-GWP refrigerant of R134a.

6.2. Performance Criteria of Investigated Refrigerants. The performance criteria of the low GWP refrigerants of R152a, R1234yf, and R1234ze(E) compared with R134a were specified to the effect of evaporating temperature, condensing temperature, and refrigerant flow rate on the energetic and exegetic parameters of the automotive air conditioning system. In particular, the most possible drop-in replacement refrigerant to R134a in automotive air conditioning application was R152a (GWP_{100} less 10 times), R1234yf (GWP_{100} less 358 times), and R1234ze(E) (GWP_{100} less 238 times).

6.2.1. The Effect of Condensing Temperature. In particular, the variation of the condensing temperature through the day, month, and season affects the system performance; therefore, a wide range of condensing temperature $20^\circ\mathrm{C} \leq T_c \leq 45^\circ\mathrm{C}$ was considered. Figure 7 illustrates the cooling capacity of different refrigerant types at $T_e = 10^\circ\mathrm{C}$ and $\dot{V} = 0.0031\ \mathrm{m^3/s}$. At the same operating condition, it was noted that a reduction in cooling capacity was recorded for R152a, R1234yf, and R1234ze(E) compared with R134a by 3.6%, 3.8%, and 19%, respectively. The compressor power was impacted by changing of condensing temperature for different refrigerant types, which is illustrated in Figure 8. At the same operating condition, the compressor power consumption of R134a was higher than that of R152a, R1234yf, and R1234ze(E) by 8.5%, 1.6%, and 28%, respectively. It has been shown that the refrigerant R134a has higher values of both refrigeration capacity and compressor power than the proposed refrigerants at the same operating condition. In this case, the coefficient of performance (COP) is the most important factor to determine the performance characteristics of the proposed refrigerants, which is illustrated in Figure 9. The COP of R134a was lower than that of R1234ze(E) and R152a by 10.8% and 5.6%, respectively. It was confirmed that the COP of the refrigerant R1234yf wasvery close to the performance of R134a in which the COP of R134a was higher by 2%.

The exergy destruction of the system compressor with different condensing temperatures is shown in Figure 10. The highest exergy destruction through the compressor was obtained for R1234yf followed by R134a while R152a has the lowest values of exergy destruction. Compared with the base refrigerant of R134a, the compressor exergy destruction of the refrigerant R1234ze(E) was higher by 6% while a reduction in the compressor exergy destruction was obtained by 14% and 29% for the refrigerants R1234yf and R152a, respectively, compared with R134a. The exergy destruction of the evaporator, expansion valve, and condenser was also calculated to form the total exergy destruction. The variation of the total exergy destruction of all refrigeration cycle components with the condensing temperature is illustrated

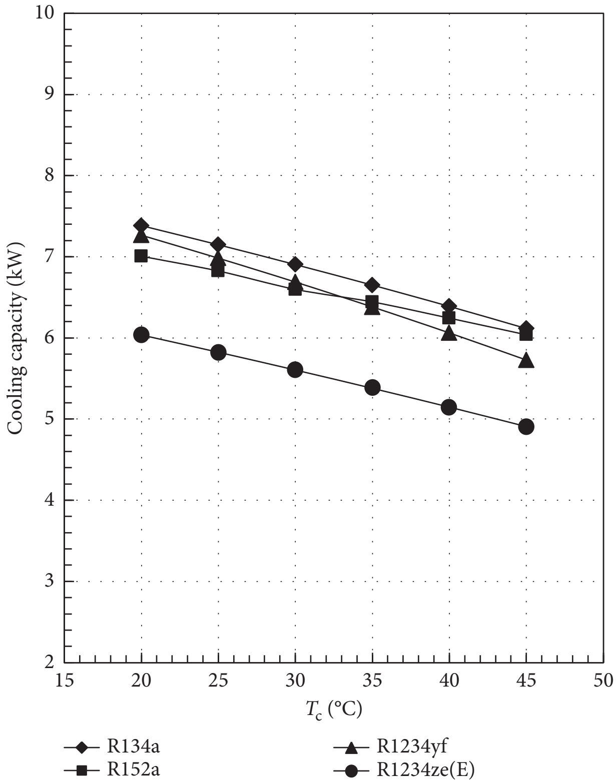

Figure 7: Effect of condensing temperature on cooling capacity for investigated refrigerants.

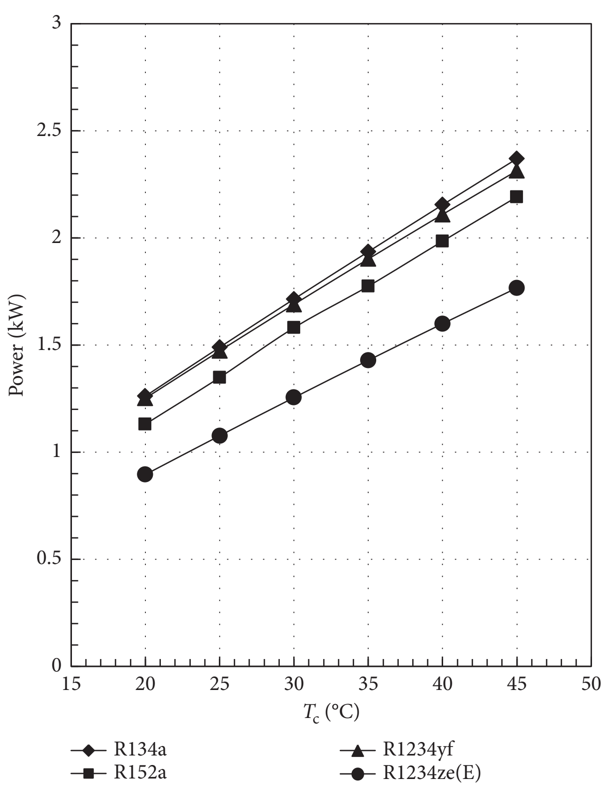

Figure 8: Effect of condensing temperature on compressor power for investigated refrigerants.

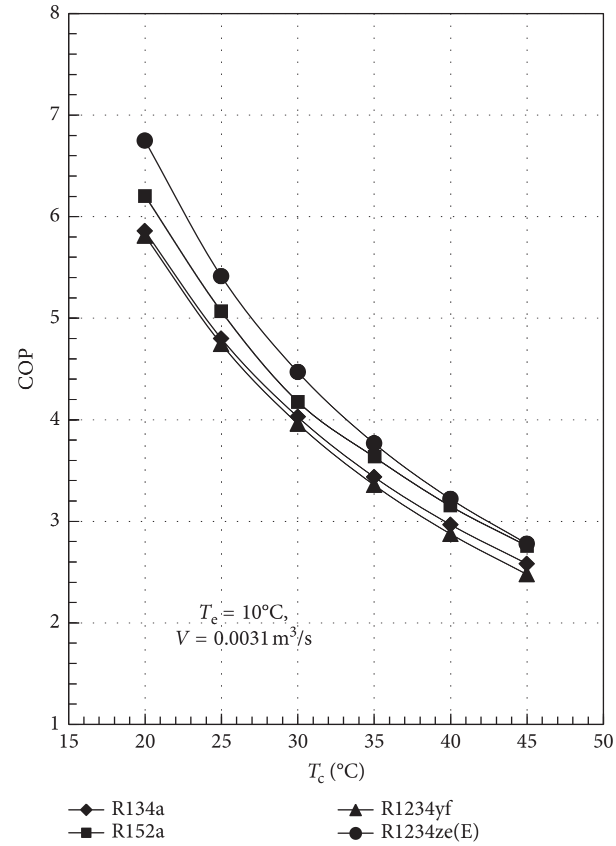

Figure 9: Effect of condensing temperature on COP for investigated refrigerants.

in Figure 11. The total exergy destruction of R1234yf and R1234ze(E) was quite similar in which the total exergy destruction of both refrigerants were higher than that of R134a by 12% while the total exergy destruction of R152a was lower than that of R134a by 26%.

The effect of varying condensing temperature on the energy and exergy parameters is summarized as a sample of results in Table 3.

6.2.2. The Effect of Evaporating Temperature. The evaporating temperature was different from one application to another, and it should be noted that the evaporating temperature affects the system performance positively. A wide range of evaporating temperatures which were applied in many air conditioning applications (–15°C to 15°C) was tested for different refrigerant types of R134a, R152a, R124yf, and R1234ze(E). The compressor power consumption was influenced by varying evaporating temperature for all investigated refrigerants, as illustrated in Figure 12. Although the enthalpy difference $(h_2 - h_1)$ in Equation (4) decreases as the evaporating temperature increases, the values of the compressor power increase as the evaporating temperature increases. This can be explained on the basis of equations (2) and (4), where the enthalpy difference $(h_2 - h_1)$ decreases with the increase in the evaporating temperature, while the volumetric efficiency increases and the specific volume decreases in accordance with

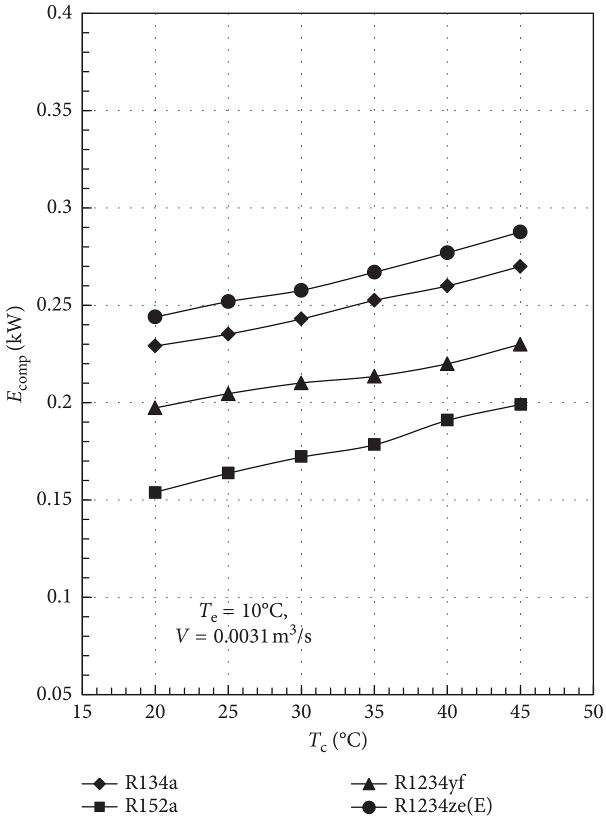

Figure 10: Exergy destruction in compressor versus condensing temperature.

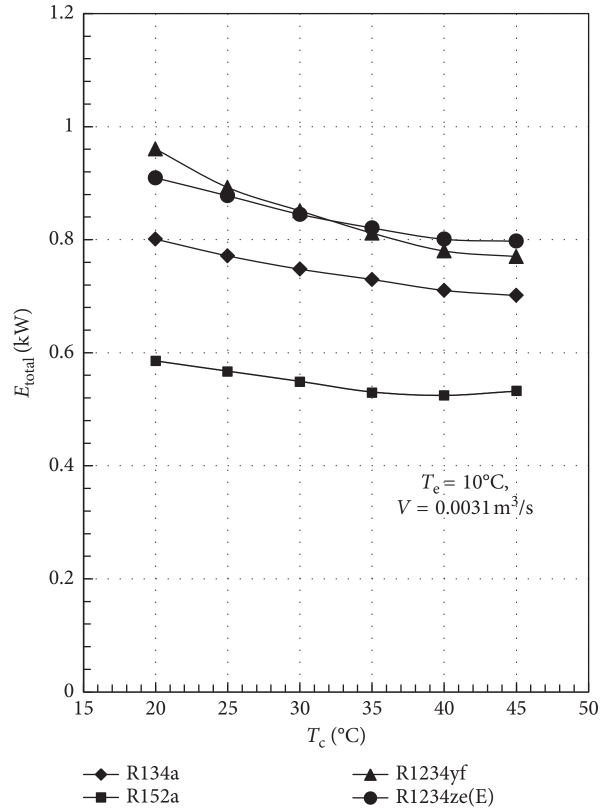

Figure 11: Total exergy destruction versus condensing temperature.

Equation (2). This led to an increase in mass flow rate being greater than the decrease in the enthalpy difference ($h_2 - h_1$), and consequently the compressor power consumption increases with evaporating temperature. This trend curve confirmed with Li et al. [17] and Llopis et al. [26].

The compressor power of R134a and R1234yf was quite similar. A reduction in a compressor power for R152a and R1234ze(E) by 8% and 26%, respectively, was occurred when compared with R134a.

There are two important parameters that can characterize the most appropriate alternative refrigerants to R134a: volumetric cooling capacity and discharge temperature. Figure 13 illustrates the volumetric refrigeration capacity (VCC) and the discharge temperature versus the evaporating temperature for all investigated refrigerants. The volumetric refrigeration capacity expresses the cooling capacity per unit volume at the exit of the evaporator. It indicates the volume of refrigerants handled by the compressor. It was noted that R134a has the highest volumetric cooling capacity followed by R152a and R1234yf while the R1234ze(E) has lowest values of volumetric cooling capacity. This mean that the refrigerants R152a and R1234yf can be replaced by R134a on the same compressor size in which the refrigerant R1234ze(E) needs to resize the compressor for a given duty.

It is necessary to study the discharge temperature for the low-GWP refrigerant compared with R134a in order to clarify the steadiness and lifetime of the compressor. Referring to Figure 13 which illustrates the discharge temperature versus the evaporating temperature for all investigated refrigerants, it was noted that the refrigerant R152a has the highest discharge temperature, and it is impediment in replacement between R152a and R134a, while the discharge temperature of R1234ze(E) and R1234yf is lower than that of R134a which is considered an advantage in replacement between R1234ze(E), R1234yf, and R134a.

The changing of the evaporating temperature affects the COP of the system positively, which is illustrated in Figure 14. As the evaporating temperature increased, the COP of the refrigeration system increased, which can be attributed to the increase in cooling capacity and was more rapid than the increase in compressor power. The refrigerant R1234ze(E) has the highest values of the COP among all the investigated refrigerants. It was confirmed that the thermal performance of the R1234yf refrigerant (GWP_{100} = 358 times less than that of R134a) was the closest to the thermal performance of the R134a system and therefore a more environmentally sustainable refrigerant for automotive air conditioning.

The exergy destruction for the system compressor with different evaporating temperatures is illustrated in Figure 15. The highest exergy destruction through the compressor was obtained for R1234ze(E) followed by R134a, while R152a has the lowest values of exergy destruction. Compared with the base refrigerant of R134a, the compressor exergy destruction of the refrigerant R1234ze(E) was higher by 6.5% while the compressor exergy destruction of R1234yf and R152a was reduced by 11% and 40%, respectively. The total exergy

Table 3: Results of the investigated refrigerants at different condensing temperatures.

Item	Tc, °C	R152a	R1234yf	R1234ze(E)	R134a
Cooling capacity	30	6.60	6.69	5.61	6.90
	35	6.44	6.38	5.38	6.65
	40	6.24	6.06	5.15	6.39
	45	6.04	5.73	4.91	6.12
Compressor power	30	1.58	1.69	1.25	1.71
	35	1.774	1.90	1.43	1.94
	40	1.984	2.11	1.60	2.15
	45	2.191	2.31	1.77	2.37
COP	30	4.18	3.96	4.47	4.03
	35	3.63	3.36	3.77	3.44
	40	3.15	2.88	3.22	2.97
	45	2.76	2.48	2.78	2.58
Total exergy destruction, E_{total}	30	0.549	0.851	0.845	0.748
	35	0.530	0.812	0.821	0.730
	40	0.524	0.780	0.801	0.710
	45	0.532	0.770	0.797	0.701
Total exergy efficiency, η_{total}	30	0.322	0.28	0.357	0.34
	35	0.299	0.26	0.336	0.32
	40	0.27	0.25	0.31	0.3
	45	0.255	0.24	0.29	0.28

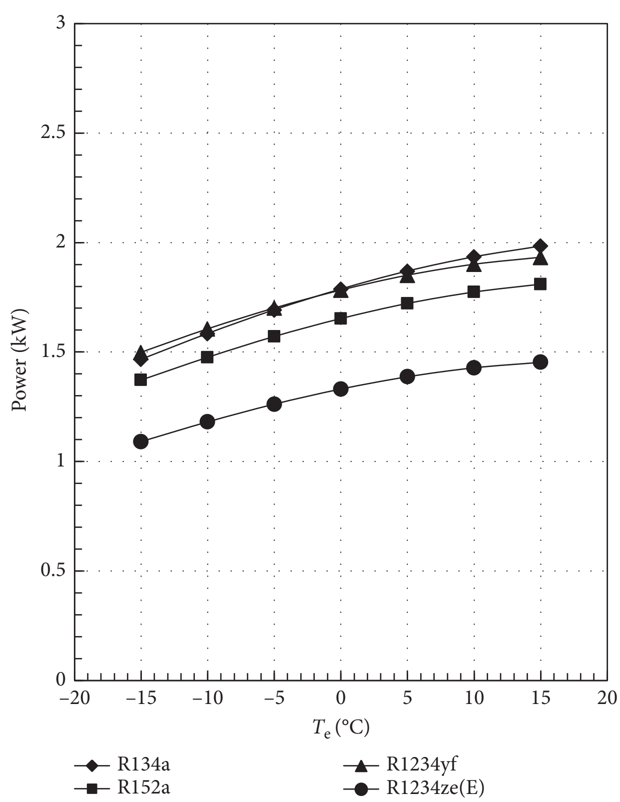

Figure 12: Compressor power versus evaporating temperature.

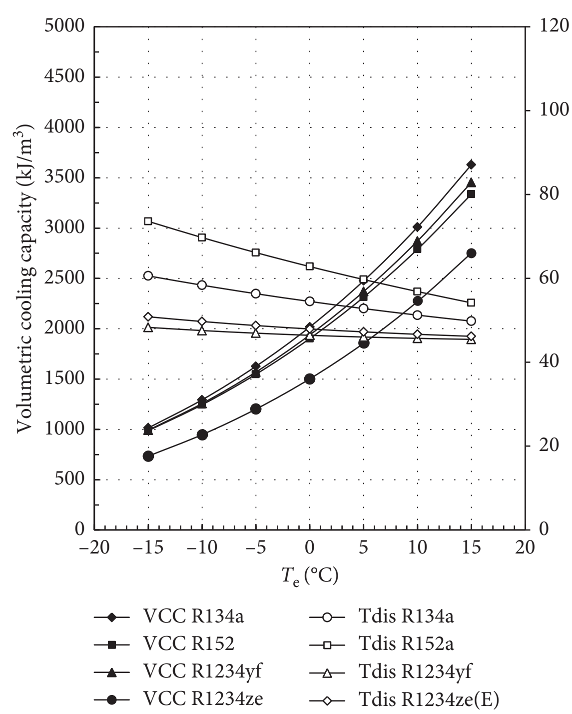

Figure 13: Effect of evaporating temperature on volumetric cooling capacity and discharge temperature.

destruction of the all-cycle components for different types of refrigerants and the evaporating temperature is illustrated in Figure 16. The total exergy destruction of R1234ze(E) was higher than that of R134a by 5.4% while the total exergy destruction of R1234yf and R152a was lower than that of R134a by 15% and 45%, respectively.

6.2.3. The Effect of Refrigerant Flow Rate. The refrigerant volume flow rate variation which was produced as a result of varying compressor speed which in turn was originated from automotive crankshaft speed affects the performance of the automotive air conditioning. The effect of varying volume flow rate on the cooling capacity and the coefficient of performance is illustrated in Figures 17 and 18 for a typical condition of $T_e = 10$°C and $T_c = 35$°C. The refrigerant mass flow rate varied as the RPM of the crankshaft was changed, which depend on

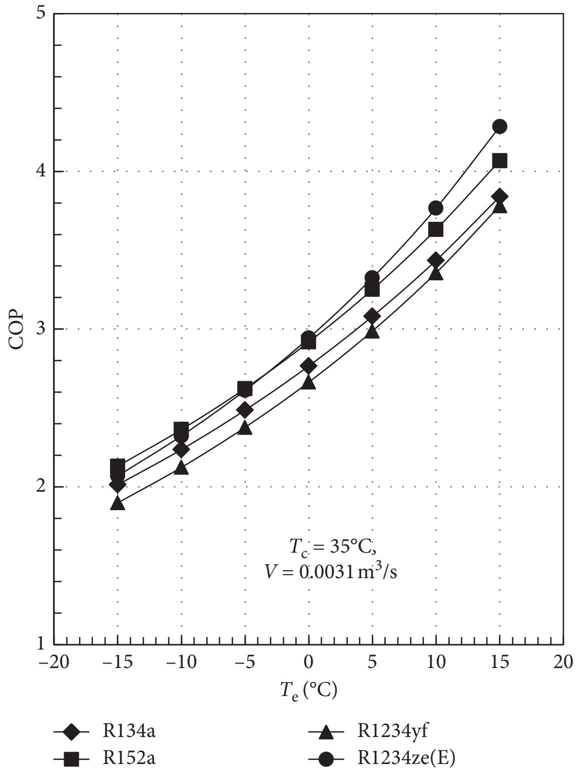

Figure 14: COP versus evaporating temperature for $T_c = 35^{\circ}$C.

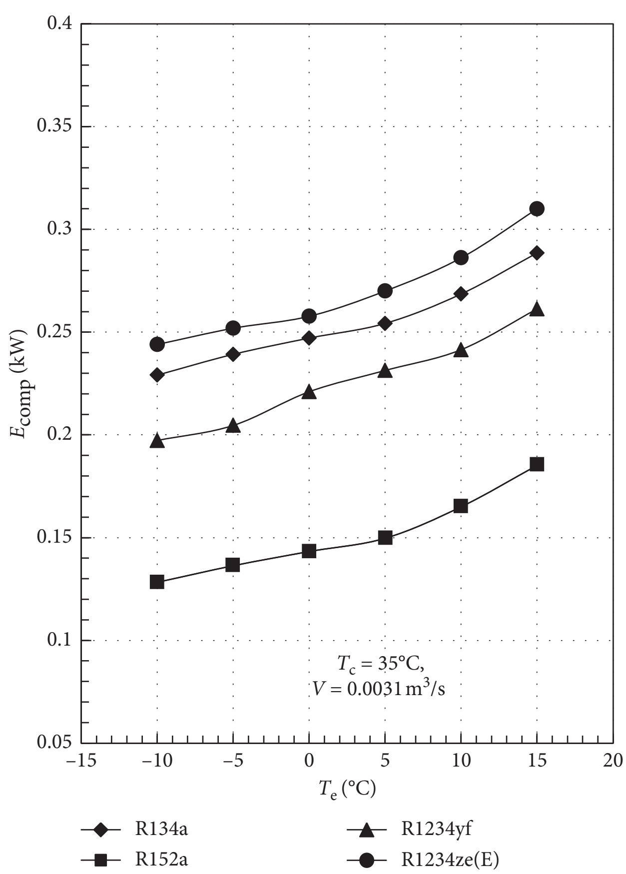

Figure 15: Exergy destruction in compressor versus evaporating temperature.

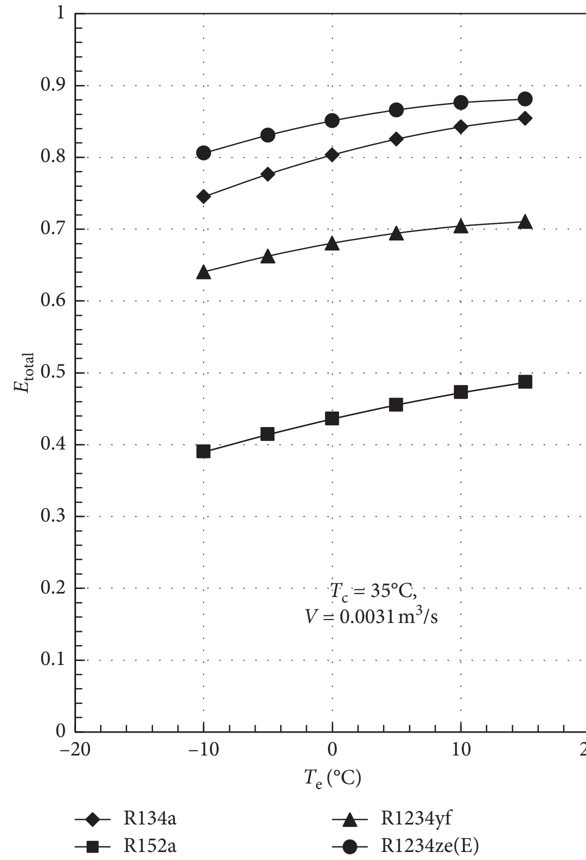

Figure 16: Total exergy destruction versus evaporating temperature.

the rate of fuel consumption of the automotive engine. In practice, the refrigerant mass flow rate ($\dot{m} = \rho\, \dot{v}$) which is the refrigerant density multiplied by volume flow rate affects the cooling capacity and compressor power according to equations (1) and (2) in which the density varies according to variation in evaporating temperature. Therefore, the influence of the refrigerant flow rate was evident to the coefficient of performance. As the refrigerant flow rate increases, the COP decreases which can be explained by the increase in compressor power with a flow rate greater than the increase in cooling capacity. It is noted that the refrigerant R1234ze(E) has the highest COP between investigated refrigerants.

The total exergy destruction of the compressor, condenser, expansion valve, and evaporator for the different refrigerant types is illustrated in Figure 19. The highest exergy destruction of the compressor was obtained for R1234ze(E), and the highest exergy destruction of the condenser was obtained for R1234fy, while the lowest value of exergy destruction for the compressor and condenser was obtained for R152a. The highest values of the exergy destruction for the evaporator and expansion valve were obtained for R1234ze(E) and R1234fy, respectively.

Exergy efficiency can give more logical ways to improve the energy performance of automotive air conditioning. For that reason, the exergy analysis which was performed for each cycle component should be considered integrated [6]. Figures 20 and 21 show the total exergy efficiency with both

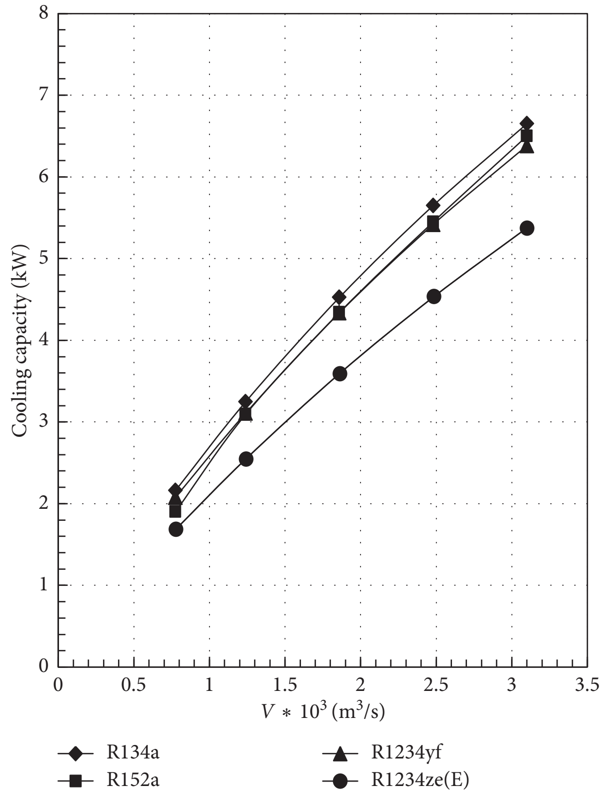

Figure 17: Cooling capacity versus refrigerant flow rate at $T_c = 35°C$.

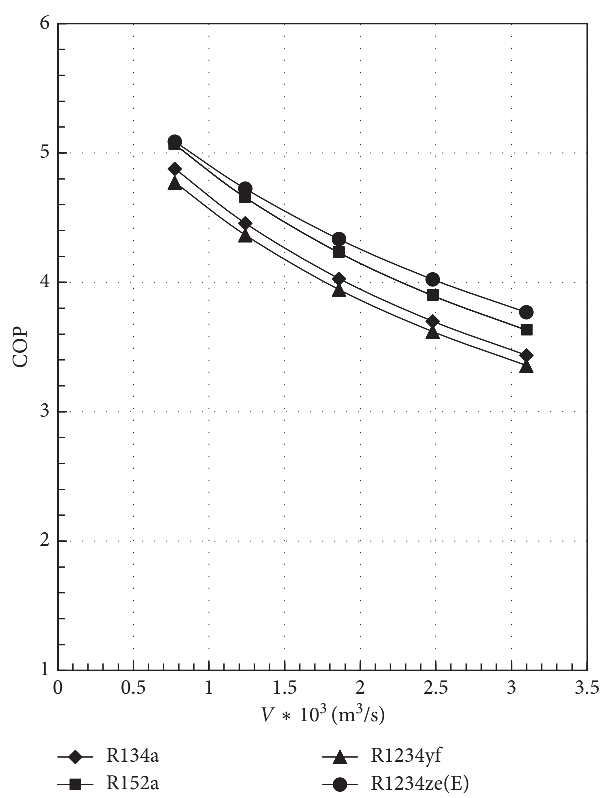

Figure 18: COP versus refrigerant volume flow rate.

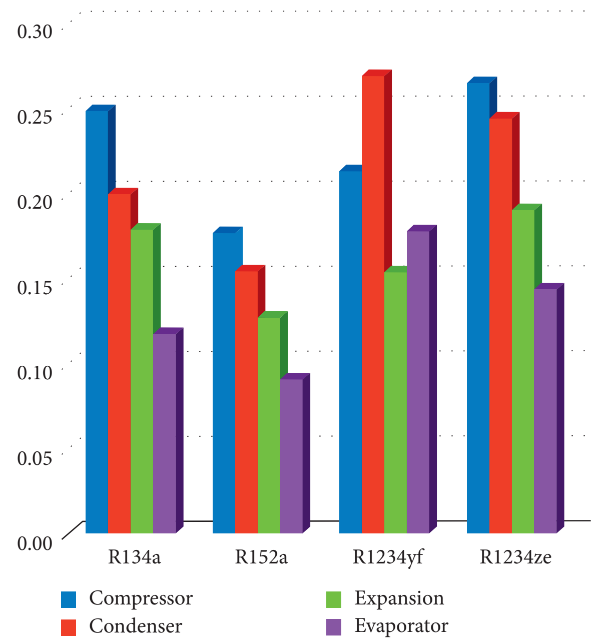

FIGURE 19: Total exergy destruction of each component for different refrigerant types.

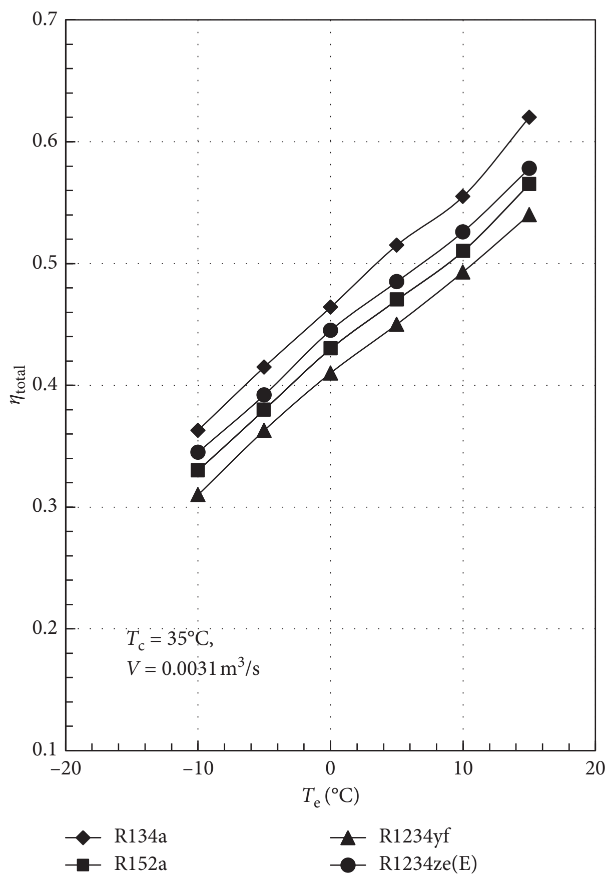

FIGURE 20: Total exergy efficiency versus condensing temperature.

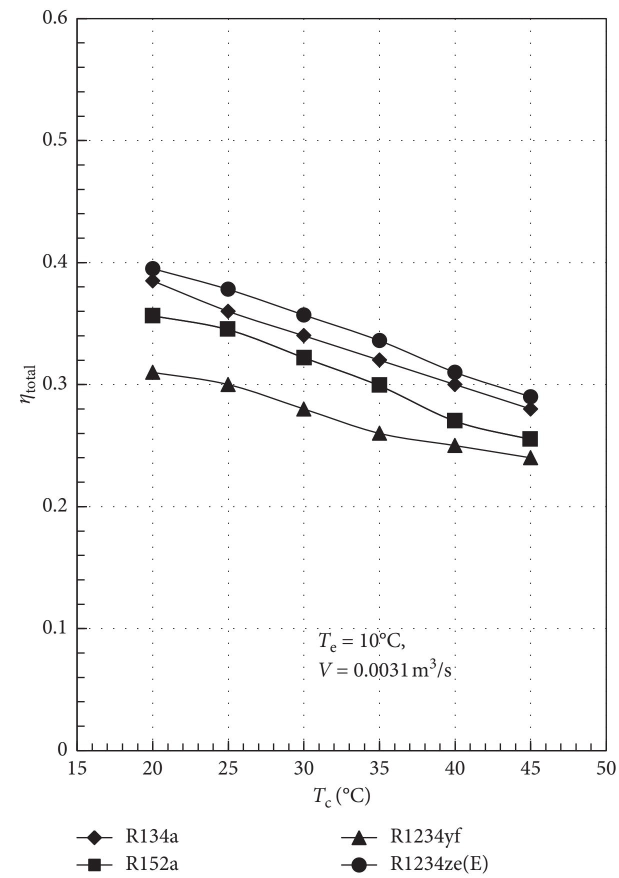

FIGURE 21: Total exergy efficiency versus evaporating temperature.

condensing and evaporating temperatures. The total exergy efficiency was decreased with the condensing temperature while it was increased with the evaporating temperature. For both condensing and evaporating temperatures, the refrigerant R1234yf has the lowest exergy efficiency followed by R152a. The highest exergy efficiency was obtained with R1234ze(E) at different condensing temperatures while at different evaporating temperatures, the highest exergy efficiency was obtained for R134a. From the environmental, thermal, and exergy point of view, the refrigerant R1234yf has the best performance among all refrigerants that have been investigated to replace R134a in an automotive air-conditioning system.

7. Conclusion

Energy and exergy analysis was presented for many environmentally friendly refrigerants as a drop-in replacement of current high-GWP of R134a in the automotive air-conditioning system. Three alternative refrigerants which is distinguished by zero ODP and $GWP_{100} < 150$ were investigated, with particular reference to the current R134a refrigerant ($GWP_{100} = 1430$). The exergy destruction of each component and the exergy efficiency at different condensing temperatures, evaporating temperatures, and refrigerant flow rates were presented, and the main conclusions are summarized as follows:

(i) For all values of condensing and evaporating temperatures, a higher system COP was obtained at a lower compressor speed which was produced from slow crankshaft RPM.

(ii) A reduction in the cooling capacity by 9% and in COP by 27% was confirmed when a condensing temperature increased by 5°C.

(iii) Based on the test results, the refrigerant R1234ze(E) had the highest coefficient of performance among all investigated refrigerants.

(iv) The refrigerant R1234yf was considered the closest in thermal performance to the refrigerant R134a.

(v) The total exergy destruction of R1234ze(E) was higher than that of R134a by 5.4% while the total exergy destruction of R1234yf and R152a was lower than that of R134a by 15% and 45%, respectively.

(vi) The refrigerant R1234ze(E) was the most environmentally acceptable and had the best energetic and exergetic performance among all the tested refrigerants.

(vii) The highest exergy efficiency was obtained for R1234ze(E) at different condensing temperatures, while at different evaporating temperatures, the highest exergy efficiency was obtained for R134a.

Nomenclature

COP: coefficient of performance
CP: specific heat, kJ/kg·K
Ex: exergy, kW
h: specific enthalpy, kJ/kg
$\dot{m}_{ref}$: refrigerant flow rate, kg/s
p: refrigerant pressure, kPa
Q: rate of heat transfer, kW
RPM: revolution per minute, min^{-1}
s: entropy, kJ/kg·K
T: temperature, K
$\dot{V}$: refrigerant flow rate, m^3/s
VCC: volumetric cooling capacity, kJ/m^3
W: compressor power, kW
GWP_{100}: global warming potential for 100 years
P: density, kg/m^3
c: cooling
comp: compressor
cond: condenser
exp: expansion
evap: evaporator
dest: destruction
dis: discharge
in: inlet
is: isentropic
mech: mechanical
o: dead state
out: outlet.

Data Availability

The data that support the findings of this study are available upon request from the corresponding author.

Conflicts of Interest

The authors declare that they have no conflicts of interest.

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