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Research Article

Thermodynamic and Environmental Analysis of Hydrocarbon Refrigerants as Alternatives to R134a in Domestic Refrigerator

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Abstract

The process of phasing out of medium and high global warming potential refrigerants is accelerating in all areas of refrigeration, particularly since the European F-Gas Regulation No. 517/2014 and the ensuing Kigali amendment went into effect. Hydrocarbon refrigerants are being considered as suitable alternatives due to their low global warming potential and excellent thermal properties, but due to their flammability, safety precautions must be followed. This theoretical study contributes to the evaluation of the thermal and environmental impact of hydrocarbon refrigerants as drop-in alternatives to R134a in domestic refrigerator. In order to conduct an analysis of energy, exergy, and environmental factors, R134a and all hydrocarbons refrigerants proposed by ASHRAE—R290, R600, R600a, R601, R601a, and R1270—were examined as operating fluids used in a domestic refrigerator with a cooling capacity of 157 W and constant condenser temperature of 40°C and variable evaporator temperature every 5°C between -5 and -30°C. The results revealed that all the alternative refrigerants except R601 and R601a have higher thermal and environmental performance than R134a and can be used after refrigerator compressor replacement.

Keywords: *Domestic refrigerator; R134a; R290; R600; R600a; R601; R601a; R1270; TEW1.*

1. Introduction

Due to its low cost, excellent thermodynamic and thermophysical properties, and lack of ozone depletion potential (ODP), R134a is a type of HydroFluoroCarbons (HFCs) that is still used as a refrigerant in refrigerators, particularly in developing countries. On the other hand, it has a high Global Warming Potential (GWP) [1]. Therefore, HFCs are being phased out according to the Kyoto Protocol [2]. HydroCarbons (HCs) are being considered as potential alternatives due to their low GWP and also have good thermophysical properties [3]. In reality, HCs were utilized as refrigerants in refrigerating units in the beginning of the 20th century. Nevertheless, nonflammable ChloroFluoroCarbons CFCs took their place due to the technical and safety issues with the usage of HCs at the given moment [4]. Their flammable properties are the fundamental disadvantage of HCs, which restrict their usage as refrigerants. Experts advise using a small amount of refrigerant in a refrigerating unit to help reduce this issue. Before the installation of large-volume refrigeration equipment, there are a number of safety rules and measures that should be followed. Some of the primary precautionary guidelines that have to be followed while utilizing HCs include containing them in a sealed system, decreasing the charge of HCs for particular uses, reducing the level of concentration of HCs in the ambient air (less than the flammability limit), utilizing a suitable ventilating source, and removing any potential source of ignition [5, 6].

In order to assess the possibility of using HC refrigerants as the best performing alternatives to HFC refrigerants, many researchers have conducted thermodynamic, environmental, or both investigations of these refrigerants as they are used in refrigeration systems [7-11]. In an identical refrigeration facility with a hermetic compressor, under the same operating conditions, Sánchez et al. [12] carried out experiment research to examine the performance of low-GWP refrigerants as alternatives to R134. This included two evaporating temperature levels (0 and -10ºC) at three condensing temperatures (25, 35, and 45ºC) for every one of them. Among these refrigerants, R290 and R600 were tested as HCs. The results showed that the R290 has the best results in terms of cooling capacity and coefficient of performance, but it is not preferred to be utilized as a direct drop-in substitute due to the difference of a displacement to decrease the similar cooling capacity, this applies to R600. Shaik and Srinivas [13] conducted a thermodynamic analysis on a domestic refrigerator that used R134a as a refrigerant, and compared it with alternative environmentally friendly refrigerants, including R290 and R600a without any modification to the refrigerator. Their results showed that these refrigerants had a performance slightly lower than R134a at various evaporator and condenser temperatures. Hastak and Kshirsagar [14] presented an experimental study to evaluate the thermal performance of R600a and R436a (R290:R600a, 54:46 by weight) in a household refrigerator using R134a refrigerant, an HC compressor in place of an HFC compressor and optimal capillary. The results revealed that alternative refrigerants have lower power consumption and higher COP, and that the mixture refrigerant is a better alternative, especially in the long term. Siddegowda et al. [15] predicted some thermodynamic properties for a group of hydrocarbon refrigerants as alternatives to replace R134a

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using SRK EOS software. They performed a simulation of an 89 W domestic refrigerator using a ten state point vapour compression cycle with condenser temperature at 55°C and evaporator temperatures ranging from -5 to -30°C. Their findings indicate that R290 and R1270 are suitable and suggested as R134a replacements with smaller compressors. de Paula et al. [16] published a mathematical modeling of a vapor compression refrigerating unit with a minimal capacity for cooling that uses many different refrigerants, including R290 and R600a as substitutes for R134a. They completed the environmental analysis using the entire equivalent warming impact, and the thermo-economic evaluation using the performance coefficient, the exergy efficacy, and the overall plant cost ratio. According to their examination of the thermodynamic, economic, and environmental factors, the system including R290 performs better than the other systems under the examined thermodynamic circumstances in terms of energy, efficiency, the environment, and the economy. Using three different vapor compression system configurations—a single-stage cycle, a cycle having an inner heat exchanger, and a two-stage cycle having vapor injection—Ghanbarpour et al. [17] presented an empirical investigation for evaluating the energy and exergy performances as well as the environmental impact. All operating with HCs R290, R600a, and R1270 as alternatives to R134a. According to their findings, alternative HC refrigerants might perform as well as R134a in every configuration while reducing carbon emissions by 50% when used. Sharma and Dwivedi [18] compared the performance of the R134a and R600a experimentally in a similar vapor compression refrigeration system. In order to compare the refrigerants, the performance coefficient, refrigeration effect, and Carnot coefficient of performance were computed. The exergy destruction and second law efficiency for the main components were then estimated. They discovered that R600a is a better refrigerant.

Following this in-depth review of the literature, it becomes clear that HC refrigerants are excellent alternatives to R134a in vapour compression refrigeration systems, especially domestic refrigerators, but the focus is usually on R290 and R600a, with few studies that take into account both thermodynamic (energy and exergy) and environmental analysis. This study presents a theoretical evaluation of the performance of all hydrocarbon refrigerants recommended by ASHRAE: R290, R600, R600a, R601, R601a, and R1270 as drop-in alternatives for R134a in a domestic refrigerator with cooling capacity of 157 W. Evaluation is done by performing an energy analysis based on compression ratio, compressor work, volumetric refrigeration capacity, and coefficient of performance, followed by an exergy analysis based on total exergy destruction rate, exergy efficiency, and sustainability index, and finally an environmental analysis based on total equivalent warming impact.

2. Comprehensive Properties of Hydrocarbon Refrigerants

When choosing a refrigerant, take into account its thermophysical characteristics as well as environmental, economic, and technological considerations [19]. ODP and GWP are important environmental features that help decrease environmental effect, but they must also have strong thermophysical and economic properties in order to be used. Table 1 shows the definitions of HCs and R134a used in this study as well as their thermophysical, safety and environmental properties. As a first impression after taking a look at the table, all refrigerants have optimal thermophysical properties for vapor compression refrigeration systems except R601 and R601a because of their high natural boiling temperature. Refrigerants are classified in terms of safety, toxicity and flammability [20]. The refrigerants' flammability is indicated by numbers, as follows: (1) is non-flammability, (2) is medium flammability, and (3) is higher flammability, while letters are an indication of toxicity, (A) is lower toxicity, and (B) is higher toxicity. Therefore, all HC refrigerants have low toxicity, but the problem of flammability remains, which must be taken into account when using these refrigerants according to the instructions, some of which were mentioned in the introduction section. Finally, in view of the environmental properties, HC refrigerants are very suitable because they do not have ODP and have a low GWP (less than 150 [2]).

3. Configuration and Assumptions

The performance is analyzed on the basis of data, most of which were taken from a 200-liter domestic refrigerator with a working fluid R134a, as shown in Table 2. The refrigerator operates on the actual vapor-compression refrigeration cycle and with some assumptions that will be mentioned later. Can Figure 1shows the path of the cycle on P-h diagram compared to the ideal cycle.

To make the analysis simpler, some of the presumptions from related earlier studies have been used as follows [26, 27]:

(1) Each component is in a steady state and is flowing steadily.

(2) The isentropic efficiency of a compressor is a function of the compression ratio.

(3) The throttling processes in the expansion device are isenthalpic.

Table 1. Thermophysical, safety and environmental properties of the studied refrigerants. [20-23]

Property	R134a	R290	R600	R600a	R601	R601a	R1270
Chemical Name	1,1,1,2-Tetrafluo-roethane	Propane	Butane	Isobutane	Pentane	Isopentane	Propene (propylene)
Chemical Formula	CF_3CH_2F	$CH_3CH_2CH_3$	$CH_3CH_2CH_2CH_3$	$CH(CH_3)_2CH_3$	$CH_3(CH_2)_3CH_3$	$(CH_3)_2CHCH_2CH_3$	$CH_3CH{=}CH_2$
Molecular Mass (gmol^{-1})	102.03	44.096	58.122	58.122	72.15	72.15	42.08
Critical Temperature (°C)	101.06	96.74	151.98	134.66	196.5	187.2	91.061
Critical Pressure (kPa)	4059.3	4251.2	3796.0	3629.0	3367.0	3378	4554.8
Normal Boiling Point (°C)	–26.074	–42.11	–0.49	–11.75	36.1	27.8	–47.62
Safety Group	A1	A3	A3	A3	A3	A3	A3
ODP	none	none	none	none	none	none	none
GWP_{100} (kg CO_2-eq/kg refrigerant)	1300	5	4	20	11	20	1.8
Atmospheric life time (year)	13.4	0.034	-	0.016	-	0.009	0.001

(4) No pressure drop and heat losses in the cycle are considered.
(5) The volumetric efficiency of the compressor is 1.
(6) The refrigerator's freezer is the subject of analyses.

Table 2. Data adopted in thermodynamic and environmental analysis.

Parameter	Unit	Data
Cooling capacity (Q_e)	W	157
Condenser temperature (T_c)	°C	40
Evaporator temperature (T_e)	°C	-30 to -5
Subcooling temperature	°C	5
Superheat temperature	°C	5
Cooled air temperature in the freezer cabinet (T_{ca})	K	8+T_e
Ambient temperature (T_a)	K	303
Mechanical efficiency of the compressor (η_{mech})	-	0.81
Electrical efficiency of the compressor (η_{elec})	-	0.93
Refrigerant weight of R134a (N)	kg	0.135
Leakage rate (L)	kg/year	10% for R134a [24]
Refrigerator lifetime (n)	year	15
Recovery factor (α)	-	0.7 [17]
Carbon emission factor (β)	kg CO_2-eq. / kWh	0.5 Middle East [25]

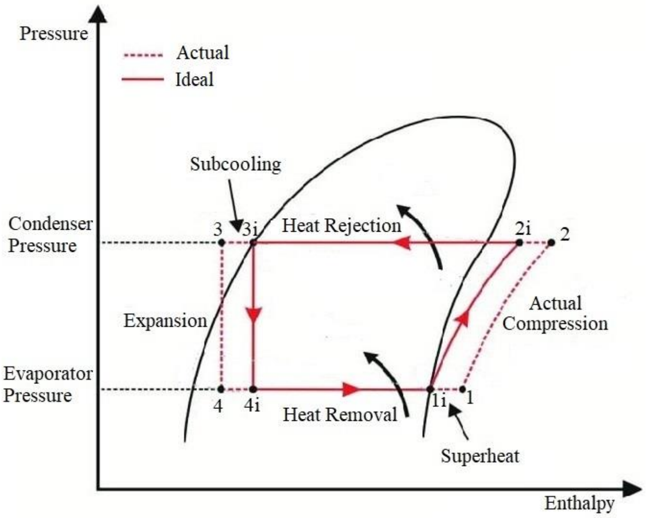

Figure 1. The proposed cycle path on P-h diagram.

4. Mathematical Analysis

4.1 Energy Analysis

Performance is evaluated in terms of energy aspects based on the first law of thermodynamics. According to the assumptions that were mentioned in the previous section, the energy expenditure is represented in the compressor work, which can be expressed by Eq. (1) [28]:

$$W_{com} = \frac{m(h_2 - h_1)}{\eta_{mech}\,\eta_{elec}} = \frac{m(h_{2s} - h_1)}{\eta_{mech}\,\eta_{elec}\,\eta_{is}} \quad (1)$$

The refrigerant mass flow rate is determined depending on the value of cooling capacity and is given by Eq. (2):

$$m = \frac{Q_e}{h_1 - h_4} \quad (2)$$

Eq. (3) is used to express the isentropic efficiency, which depends on the compression ratio [29]:

$$\eta_{is} = 0.874 - 0.0135 P_r \quad (3)$$

The compression ratio is an important parameter for determining the compressor design and size. In this cycle, it represents the ratio of high pressure (condenser pressure) to the ratio of low pressure (evaporator pressure) as expressed in Eq. (4):

$$P_r = \frac{P_c}{P_e} \quad (4)$$

The volumetric refrigeration capacity is another parameter to evaluate performance, especially concerning the compressor design and size, it can be expressed in Eq. (5) [27]:

$$VRC = \frac{Q_e}{m\,v_1} \quad (5)$$

In general, the vapor-compression refrigeration cycle performance is expressed by the coefficient of performance, which represents as the ratio of the heat removal by the evaporator (cooling capacity) to work rate done in the compressor as expressed in Eq. (6) [30]:

$$COP = \frac{Q_e}{W_{com}} \quad (6)$$

4.2 Exergy Analysis

It is possible to assess the effectiveness of the vapor-compression refrigeration cycle using the energy analysis approach. It cannot, however, assess the cycle's severity of irreversibility. In order to further assess the irreversibility of the cycle while replacing the refrigerant, the exergy analysis approach is used.

Typically, Eq. (7) is used to represent the exergy balance for the steady process control volume [28]:

$$EX_d = \sum EX_{in} - \sum EX_{out} + \sum Q\left[1 - \frac{T_a}{T_s}\right]_{in} - \sum Q\left[1 - \frac{T_a}{T_s}\right]_{out} + \sum W_{in} - \sum W_{out} \quad (7)$$

Eq. (8) gives the exergy rate of every state point within the cycle, assuming that kinetic as well as potential energy variations were ignored [31], [32]:

$$EX = m[(h - h_o) - T_a(s - s_o)] \quad (8)$$

Where, h_o and s_o are the enthalpy and entropy at ambient temperature.

According to Eq. (7) and Eq. (8), the exergy destruction rate of the cycle components are measured as follows:

In the compressor

$$EX_{d,com} = EX_1 - EX_2 + \sum W_{in} = m[(h_1 - h_o) - T_a(s_1 - s_o)] - m[(h_2 - h_o) - T_a(s_2 - s_o)] + W_{com} = m[(h_1 - h_2) - T_a(s_1 - s_2)] + W_{com} \quad (9)$$

In the condenser

$$EX_{d,c} = EX_2 - EX_3 - \sum Q\left[1 - \frac{T_a}{T_s}\right]_{out} = m[(h_2 - h_o) - T_a(s_2 - s_o)] - m[(h_3 - h_o) - T_a(s_3 - s_o)] - Q_c\left[1 - \frac{T_a}{T_a}\right] = m[(h_2 - h_3) - T_a(s_2 - s_3)] \quad (10)$$

In the expansion device

$$EX_{d,exp} = EX_3 - EX_4 = m[(h_3 - h_o) - T_a(s_3 - s_o)] - m[(h_4 - h_o) - T_a(s_4 - s_o)] = m\,T_a(s_4 - s_3) \tag{11}$$

In the evaporator

$$EX_{d,e} = EX_4 - EX_1 + \sum Q\left[1 - \frac{T_a}{T_s}\right]_{in} = m[(h_4 - h_o) - T_a(s_4 - s_o)] - m[(h_1 - h_o) - T_a(s_1 - s_o)] + Q_e\left[1 - \frac{T_a}{T_{ca}}\right] = m[(h_4 - h_1) - T_a(s_4 - s_1)] + Q_e\left[1 - \frac{T_a}{T_{ca}}\right] \tag{12}$$

The total exergy destruction rate can be expressed in Eq. (13):

$$EX_{d,tot} = EX_{d,com} + EX_{d,c} + EX_{d,exp} + EX_{d,e} \tag{13}$$

The total exergy efficiency of the cycle can be expressed in Eq. (14) [33]:

$$\eta_{ex,tot} = 1 - \frac{EX_{d,tot}}{W_{com}} \tag{14}$$

The effective use of resources is necessary for sustainability evaluation, which is carried out using the sustainability index approach, which is connected to energy efficiency. Exergy evaluation can be crucial for increasing productivity because it enables any user to fully utilize the advantages of their resources while reducing drawbacks like environmental harm [34]. In order to acquire the sustainability evaluation represented in Eq. (15), the sustainability index approach based on energy efficiency is a valuable tool. [35]:

$$SI = \frac{1}{1 - \eta_{ex}} \tag{15}$$

4.3 Environmental Analysis

There are several metrics for evaluating the environmental impact left by the systems, one of which and the most used is the total equivalent warming impact metric. Such environmental metric, which has a mathematical expression in Eq. (16), includes both direct as well as indirect greenhouse gas releases from a refrigerating system [36]:

$$TEWI = \left[GWP\,L\,n + GWP\,N(1 - \alpha) + E_y\,\beta\,n\right]10^{-3} \tag{16}$$

5. Results and Discussion

In this section, the results are presented and discussed in three stages: energy analysis results, exergy analysis results, and environmental analysis results. The results were built considering that the condenser temperature is constant at 40°C, While the evaporator temperature ranges between -30 and -5°C, performance parameters are calculated every 5°C. The thermal properties of the cycle (P,v,h,s) are determined using Coolselector®2 software, and the equations are simulated using MATLAB software.

Figure 2 shows the compression ratio of R134a and alternative refrigerants over a range of evaporator temperatures. For all refrigerants, the compression ratio increases with the increase in the temperature difference between the condenser and the evaporator, and this increase requires a larger compressor work and thus higher costs. For R134a, the minimum compression ratio is 4.1742 and the maximum compression ratio is 12.0379 at evaporator temperatures of -5 and -30°C, respectively. According to such results, the results of the alternative refrigerants can be divided into three levels: excellent, represented by R290 and R1270, acceptable, represented by R600 and R600a, and bad, represented by R601 and R601a. For R290 and R1270, the minimum and maximum compression ratios are 3.3719 and 3.2855, and of 8.1585 and 7.7883 at evaporator temperatures of -5 and -30°C, respectively. That is, the minimum and maximum compression ratios are reduced by 19% and 21%, and by 32% and 35% when using R290 and R1270, respectively. For R600 and R600a, the minimum and maximum compression ratios are 4.4477 and 4.0550, and of 13.4220 and 11.3991 at evaporator temperatures of -5 and -30°C, respectively. That is, the minimum and maximum compression ratios are reduced by 3% and 5%, respectively when using R600a, they increase by 6% and 10%, respectively when using R600, this is an acceptable result. For R601 and R601a, the minimum and maximum compression ratios are 5.9639 and 5.4693, and of 22.6863 and 19.4231 at evaporator temperatures of -5 and -30°C, respectively. That is, the minimum and maximum compression ratios are increased by 30% and 24%, and by 47% and 38% when using R601 and R601a, respectively. This means that it is difficult to use R601 and R601a in instead of R134a without mixing, because it requires a compressor with a very high displacement to secure the required cooling capacity.

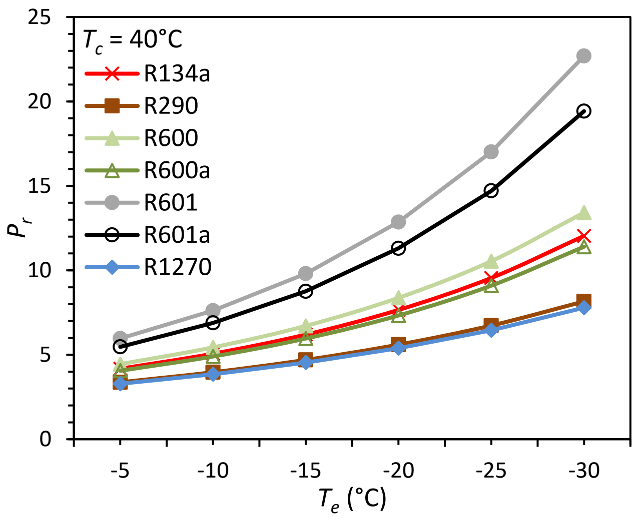

Figure 2. Variation of compression ratio with evaporator temperature for several refrigerants.

Figure 3 shows the volumetric refrigeration capacity of R134a and alternative refrigerants at different evaporator temperatures. This parameter gives a visualization of the compressor displacement and required refrigerant charge to secure the cooling capacity. The volumetric refrigeration capacity decreases with the increase in the temperature difference between the condenser and the evaporator for all refrigerants. For R134a, the maximum volumetric refrigeration capacity is 1779.80 kJ/m3 and the minimum volumetric refrigeration capacity of 583.22 kJ/m3 at evaporator temperatures of -5 and -30°C, respectively. According to this result, the results of the alternative refrigerants can be divided into three levels: excellent, represented by R290 and R1270, acceptable, represented by R600 and R600a, and bad, represented by R601 and R601a. For R290 and R1270, the maximum and minimum volumetric refrigeration capacities are 2475.20 and 3007.00 kJ/m3, and are 963.36 and 1209.10 kJ/m3 at evaporator temperatures of -5 and -30°C, respectively. That is, the maximum and minimum volumetric refrigeration capacities are increased by 28% and 41%, and by 39% and 52% when using R290 and R1270, respectively. This means that it is possible to use R290 and R1270 instead of R134a, but with use a compressor with a lower displacement (smaller in size) and change in the charge amount to secure the same performance. For R600 and R600a, the maximum and minimum volumetric refrigeration capacities are 681.17 and 957.10 kJ/m3, and of 214.93 and 320.08 kJ/m3 at evaporator temperatures of -5 and -30°C, respectively. That is, the maximum and minimum volumetric refrigeration capacities are decreased by 61% and 46%, and by 63% and 45% when using R600 and R600a, respectively. This means that when using R600 and R600a instead of R134a, a higher displacement compressor and a different charge amount should be used to guarantee the same performance. For R601 and R601a, the maximum and minimum volumetric refrigeration capacities are190.08 and 255.59 kJ/m3, and are 48.06 and 68.61 kJ/m3 at evaporator temperatures of -5 and -30°C, respectively. That is, the maximum and minimum volumetric refrigeration capacities are decreased by 89% and 86%, and by 92% and 88% when using R600 and R600a, respectively. This means that it is difficult to use R601 and R601a instead of R134.

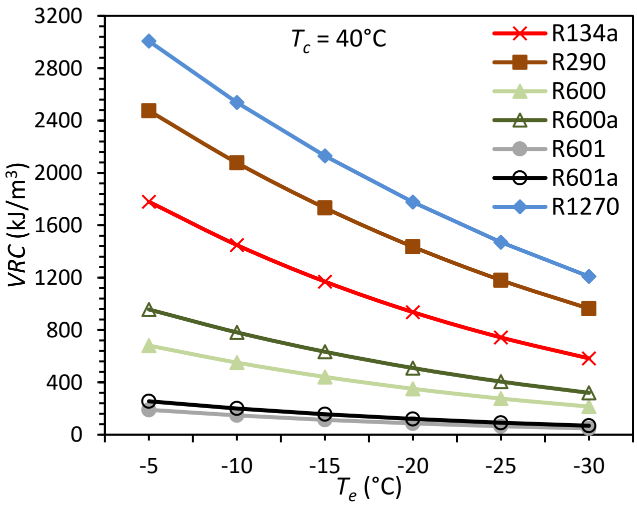

Figure 3. Variation of volumetric refrigeration capacity with evaporator temperature for several refrigerants.

Figures 2 and 3 provide more design indicators than thermal ones. To clearly evaluate the thermal performance when using alternative refrigerants, compressor work and coefficient of performance results are investigated. Figure 4 shows compressor work behavior at different evaporator temperatures using R134a and alternative refrigerants. It is clear that the compressor work increases with the increase in the temperature difference between the condenser and the evaporator for all refrigerants, in other words, power consumption increases as the evaporator temperature decreases. All refrigerants show convergent values at an evaporator temperature from -5 to -20°C, with preference given to the R600 and R600a, it recorded the lowest values for the compressor work. Differences begin to appear more after that, especially at -30°C, the compressor work reaches to the maximum, and there are clear differences between work values of the refrigerants. At the evaporator temperature of 30°C, the compressor work arrived to 115.5 W using R134a, while arrived to 108.7, 112.9, 111.4, 135.4, 127.2, and 107.8 W using R290, R600, R600a, R601, R601a, and R1270, respectively. This means that the compressor work decreased by 5.88%, 2.25%, 3.55%, and 6.66% using R290, R600, R600a, and R1270, respectively, while increased by 14.69% and 9.19% using R601 and R601a, respectively.

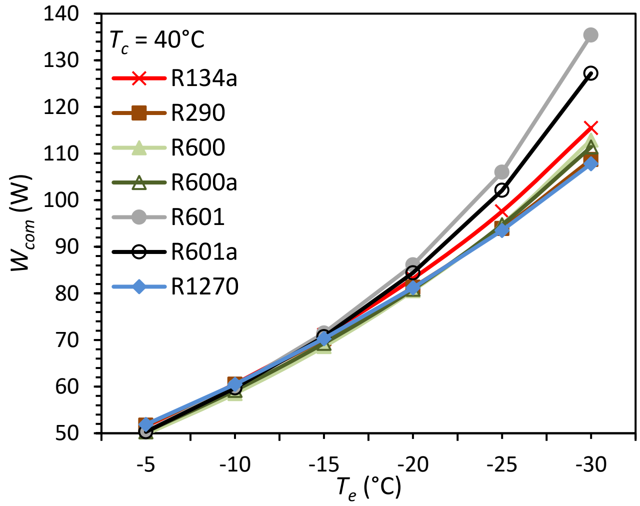

Figure 4. Variation of compressor work with evaporator temperature for several refrigerants.

Figure 5 shows coefficient of performance with different evaporator temperatures using R134a and alternative refrigerants. It is clear that the coefficient of performance decreases with the increase in the temperature difference between the condenser and the evaporator for all refrigerants, this is due to the increased work required from the compressor. All refrigerants show convergent values at an evaporator temperature from -5 to -20°C, with preference given to the R600 and R600a, it recorded the highest values for the coefficient of performance. Differences begin to appear more after that, especially at -30°C where the coefficient of performance is 1.3591 using R134a, while it is 1.4448, 1.3912, 1.4097, 1.1593, 1.2344, and 1.4565 using R290, R600, R600a, R601, R601a, and R1270, respectively. This means that the coefficient of the performance increased by 5.93%, 2.30%, 3.59%, and 6.68% using R290, R600, R600a, and R1270, respectively, while decreased by 14.70% and 9.17% using R601 and R601a, respectively.

Exergy analysis was performed based on three parameters: total exergy destruction rate, exergy efficiency, and sustainability index. Figure 6 shows the total exergy destruction rate as a function of different evaporator temperatures using R134a and HCs. It is clear that the exergy destruction rate increases with the increase in the temperature difference between the condenser and the evaporator, in other words, thermal losses increase due to irreversibility as the evaporator temperature decreases. All refrigerants have convergent values of the total exergy destruction rate at evaporator temperature from -5 to -20°C, with preference given to the R600 and R600a, it recorded the lowest values. Differences begin to appear more after that, especially at -30°C, the destruction rates reach their highest levels due to the increased intensity of irreversibility. At an evaporator temperature of -30 °C, it is observed that the total exergy destruction rate using R134a is 83 W, while it is 76.1, 80.3, 78.8, 106.6, 94.7, and 75.3 W when using R290, R600, R600a, R601, R601a, and R1270. This means that the total exergy destruction rate was decreased by 8.31%, 3.25%, 5.33%, and 10.22% using R290, R600, R600a, and R1270 respectively, while increased by 22.13% and 12.35% using R601 and R601a, respectively.

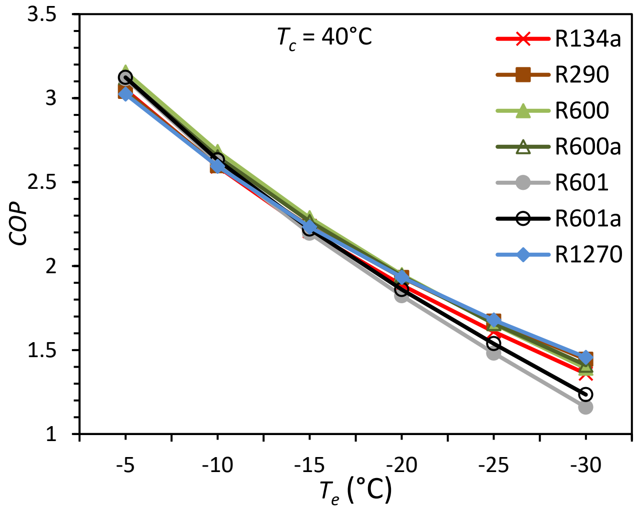

Figure 5. Variation of coefficient of performance with evaporator temperature for several refrigerants.

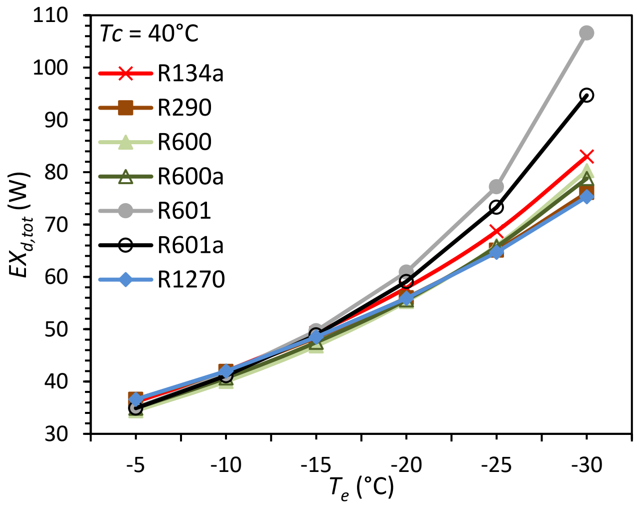

Figure 6. Variation of total exergy destruction rate with evaporator temperature for several refrigerants.

Figure 7 shows the change in efficiency with evaporator temperatures when R134 and alternative HC refrigerants are used. Contrary to all the parameters that were presented, the exergy efficiency did not show direct or inverse behavior, it increased with the evaporator temperature to a certain extent and then decreased, as this behavior applies to all refrigerants. This behavior was reported in [9], and it is explained by the fact that, according to Eq. (14), the exergy efficiency depends on both the compressor work and the total exergy destruction rate. When increasing, it means that the increase in the compressor work is greater than the increase in the total exergy destruction rate, and vice versa when decreasing. At evaporator temperatures of -15°C, R134a's maximum exergy efficiency was recorded to be 0.3071, while that of R290, R600, and R600a was 0.3109, 0.3183, and 0.3152. At evaporator temperatures of -20°C, R1270's maximum exergy efficiency was recorded to be 0.3112, while that of R601 and R601a was 0.3091 and 0.3107, respectively. At 30°C, the exergy efficiency of alternative refrigerants is higher than the exergy efficiency of R134a, except for R601 and R601a.

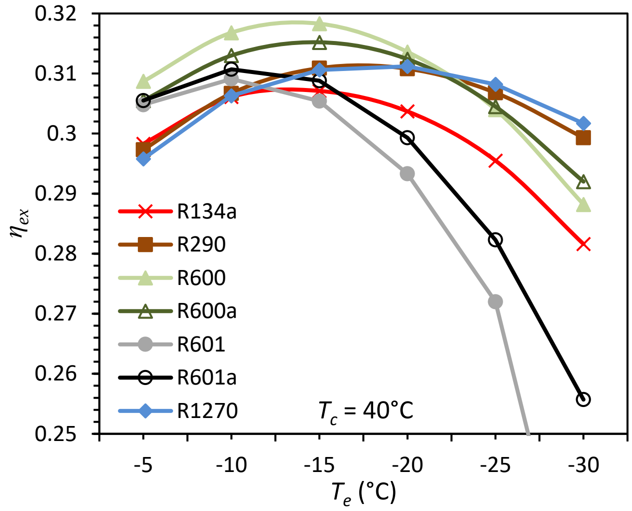

Figure 7. Variation of exergy efficiency with evaporator temperature for several refrigerants.

Depending on the exergy efficiency values, the sustainability index is determined, which showed different values between a same refrigerant and between refrigerants according to change evaporator temperatures, as shown in Figure 8.

The best sustainability index was observed for R134a to be 1.4431, while that of R290, R600, and R600a was 1.4513, 1.4669, and 1.4602 respectively at evaporator temperatures -15°C, while was observed for R1270 to be 1.4517 at evaporator temperatures -20°C, and that of R601 and R601a was 1.4473 and 1.4508 respectively at evaporator temperatures -10°C.

Total equivalent warming impact is an effective parameter in evaluating the environmental impact of refrigeration systems because it diagnoses carbon emissions from the system itself as well as from the energy source supplied to the system. Therefore, reducing power consumption when replacing the refrigerant has environmental benefits in addition to economic benefits. According to Eq. (16), the first and second terms represent direct emissions for which the refrigerator is responsible, and the third term represents indirect emissions. All refrigerants are considered to have the same weight of charge (135 g). As for the leakage rate of HC refrigerants, it is determined according to the leakage rate of R134a listed in Table 2, where the leakage rate of HC refrigerant represents the ratio of its high pressure to high pressure of R134a multiplied by the leakage rate of R134a. While the annual energy consumption is determined depending on the compressor work, considering that the refrigerator operates 24 hours a day.

Figure 9 shows the change in total equivalent warming impact of the used refrigerants with the evaporator temperature. It is clear that the total equivalent warming impact increases with decreasing the evaporator temperature due to indirect emissions that increase as a result of power consumption and irreversibility losses. Except for R601, which records higher values at -25 and -30°C evaporator temperatures, and R601a, which records higher value at -30°C, all HC refrigerants have a lower total equivalent warming impact than R134. Overall, In the evaporator

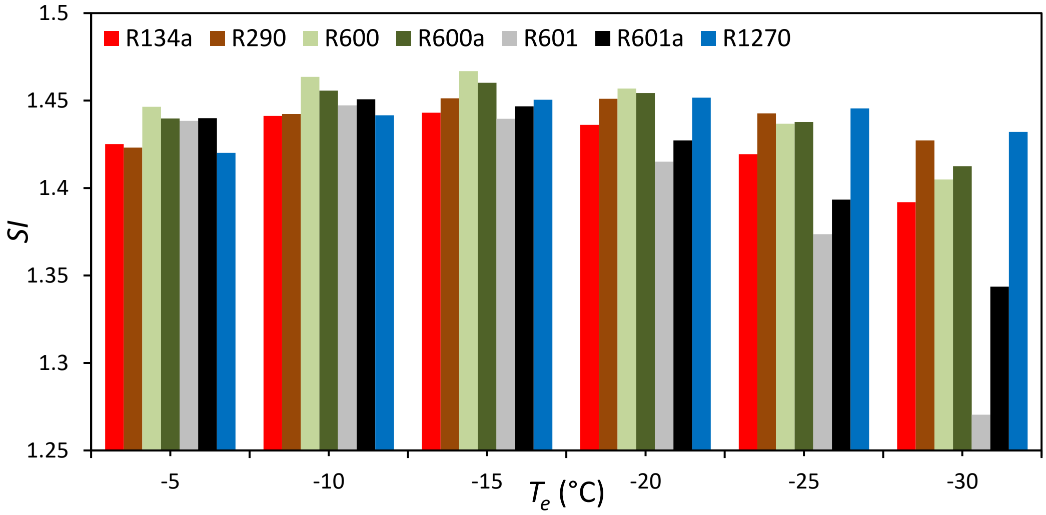

Figure 8. Variation of sustainability index with evaporator temperature for several refrigerants.

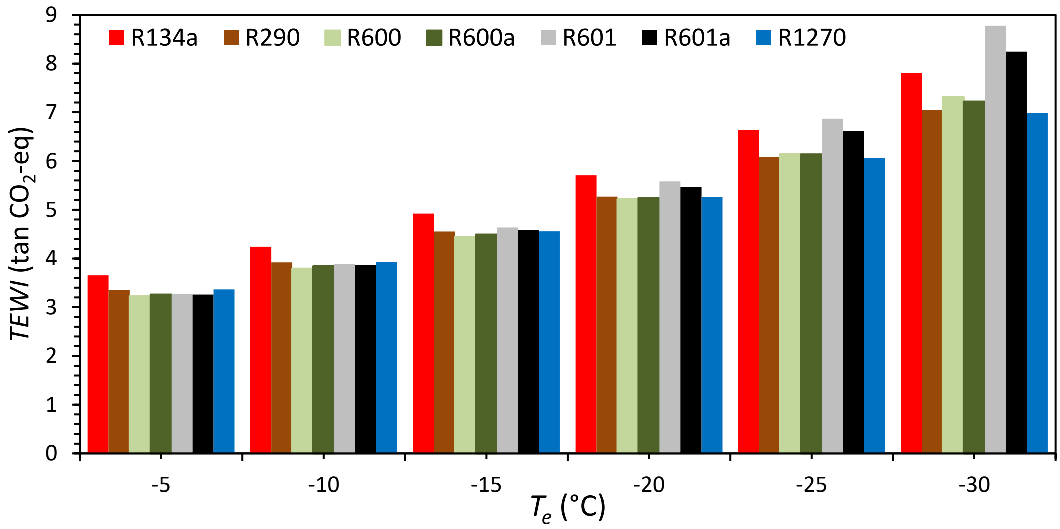

Figure 9. Variation of Total equivalent warming impact with evaporator temperature for several refrigerants.

temperature range of -5 to -20°C, R600 was shown to have the lowest total equivalent warming impact, followed by R600a, and at -25 and -30°C, R1270 was shown to have the lowest total equivalent warming impact, followed by R290.

6. Conclusions and Recommendations

This study contributes to the thermal and environmental evaluation of HC refrigerants: R290, R600, R600a, R601, R601a, and R1270 as working fluids in a domestic refrigerator instead of high-GWP R134a. A triple analysis of energy, exergy, and environmental was performed with constant operating conditions for ambient and condenser temperatures of 30 and 40°C, respectively, and variable operating conditions for evaporator temperatures from -5 to -30°C. The results can be summarized according to the calculated parameters:

(1) There are prominent differences in the compression ratios, and the refrigerants can be arranged from lowest to highest compression ratio under the same conditions as follows: R1270, R290, R600a, R134a, R600, R601a, and R601.

(2) There are prominent differences in the volumetric refrigeration capacities, and the refrigerants can be arranged from the highest to lowest volumetric refrigeration capacity under the same conditions as follows: R1270, R290, R134a, R600a, R600, R601a, and R601.

(3) In the evaporator temperature range of -5 to -20°C, R600 has the lowest compressor work (power consumption) and total exergy destruction rate, followed by R600a. However, at -25 and -30°C, R1270 has the lowest compressor work and total exergy destruction rate followed by R290.

(4) In the evaporator temperature range of -5 to -20°C, R600 has the highest coefficient of performance followed by R600a. However, at -25 and -30°C, R1270 has the highest coefficient of performance, followed by R290.

(5) In the evaporator temperature range of -5 to -20°C, R600 and R600a had the best energy efficiency and sustainability index, while R1270 and R290 had the best results at -25 and -30°C.

(6) With the exception of R601, which recorded higher values at -25 and -30°C evaporator temperatures and also R601a, which recorded a higher value at -30°C, all HC refrigerants have a lower total equivalent warming impact than R134 under the same conditions. R1270 has the lowest total equivalent warming impact, followed by R290 at -25 and -30°C, and R600 also has the lowest total equivalent warming impact, followed by R600a in the -5 to -20°C evaporator temperature range.

According to the results of this study, it is recommended to use R600 or R600a instead of R134a but with a higher displacement compressor and a change in amount of the charge. Also, it is recommended to use R1270 or R290 instead of R134a, but with a lower displacement compressor and a change in amount of the charge. With safety instructions adherence when using HC refrigerants. It is not recommended to use R601 and R601a refrigerants in domestic refrigerators.

Nomenclature

COP	Coefficient of performance
EX	Exergy rate (W)
E_y	Annual energy consumption (kWh/year)
h	Specific enthalpy (kJ/kg)
L	Leakage rate (kg/year)
m	Refrigerant mass flow rate (kg/s)
N	Refrigerant weight (kg)
n	Refrigerator lifetime (year)
p	Pressure (bar)
P_r	Pressure ratio
Q	Heat rate (W)
Q_e	Cooling capacity (W)
SI	Sustainability index
s	Specific entropy (kJ/kg.K)
T	Temperature (°C)
$TEWI$	Total equivalent warming impact (tan CO_2-eq)
VRC	Volumetric refrigeration capacity (kJ/m^3)
v	Specific volume (m^3/kg)
W	Work rate (W)

Greeks Symbol

α	Recovery factor at the end of life
β	Carbon emission factor (kg CO_2-eq./kWh)
η	Efficiency

Subscripts

o	Reference state
1-4	State points of refrigerant for actual cycle
$2s$	State point at constant entropy
1i-4i	State points of refrigerant for ideal cycle
a	Ambient
c	Condenser
ca	Cooled air in the freezer cabinet
com	Compressor
d	destruction
e	Evaporator
$elec$	Electrical
ex	Exergy
exp	Expansion device
in	Inlet
is	Isentropic
$mech$	Mechanical
out	Outlet
s	Space
tot	Total

Abbreviations

ASHRAE	American Society of Heating, Refrigerating and Air-Conditioning Engineers
CFCs	ChloroFluoroCarbons
GWP	Global Warming Potential
HCs	HydroCarbons
HFCs	HydroFluoroCarbons
ODP	Ozone Depletion Potential

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