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Assessment of using different ozone-friendly R22 alternative refrigerants in residential air conditioners in a high-ambient temperature country

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Abstract

The performance of a vapor compression refrigeration system (VCRS)-based residential air conditioner operating in a high-ambient temperature (HAT) country was investigated using six zero-ODP (ozone depletion potential) refrigerants as replacements to R22. The non-flammable alternative refrigerants considered in the present research were R134a, R404A, R407C, R410A, R448A, and R507A. Using the basic conservation laws, the VCRS was modeled during steady-state operation and solved using engineering equation solver (EES) software. Coefficient of performance (COP), pressures and temperatures at compressor suction and discharge, Global Warming Potential (GWP), critical pressure and temperature, compressor pressure ratio, volumetric cooling capacity (VCC) specific cooling capacity (SCC), and refrigeration effect were utilized as assessment criteria for the alternative refrigerants considered. From these refrigerants, the highest values of suction pressure, discharge temperature, and condenser pressure were attained by R410A. In addition, the discharge temperatures for all refrigerants, except R134a, were all higher than their corresponding critical values, causing a quicker drop in the VCRS's performance. As an alternative refrigerant, R407C showed the highest SCC of 141.0 kJ/kg followed directly by 139.2 and 138.0 kJ/kg for R410A and R448A, respectively. A reverse trend was found for VCC with respective values of 4722 and 3775 kJ/m^3 for R410A and R448A. Lower volume flow rates and smaller-sized compressors are expected for higher VCC refrigerants. The same trend was found for the compressor's specific work input and condenser's specific heat transfer with values of (51.14, 46.82, and 45.38 kJ/kg) and (190.3, 187.8, and 183.4 kJ/kg) for R410A, R407C, and R448A, respectively. For applications in HAT countries, larger condenser's specific heat transfer makes the refrigerant more applicable. Conversely, with respect to COP, refrigerant R134a with a value of 3.075 was the superior alternative followed by R448A and R407C with respective COPs of 3.042 and 3.011. Based on the overall assessment in terms of environmental obligation, COP, compressor input power, refrigerant flow rate required, and all the evaluations made in this research, refrigerant R448A was recommended as the most appropriate substitute to R22 which can effectively be used in residential air conditioners in a HAT country.

Keywords: R22 alternative refrigerants, Ozone-friendly, Air conditioner, GWP, High-ambient temperature, R448A

Introduction

As the world's temperature continues to rise, particularly in hot regions, the need for air conditioning has become increasingly important [1–3]. With temperatures reaching uncomfortable and even dangerous levels, utilization of air conditioning to provide relief from the heat is not just a luxury but a necessity for many individuals and communities. Furthermore, air conditioning can also be useful in enhancing indoor air quality and promoting better health. But with the rapidly increased demand for air conditioning, the harmful environmental impact should be carefully considered, and efforts should be focused on more sustainable approaches [4–6]. Through the gradual reduction in the production and utilization of substances that dangerously deplete the stratospheric ozone layer, known as ozone-depleting substances, an international agreement referred as the Montreal Protocol was established in 1987 [7–9]. In vapor compression refrigeration system (VCRS)-based air conditioning devices, refrigerant R22, with an ozone depletion potential (ODP) of 0.055, was mostly used. Accordingly, the gradual switching of refrigerant R22 to more environment-friendly refrigerants, i.e., zero-ODP, has commenced in many countries. Unfortunately, this transition produced a new environmental crisis since some alternative refrigerants have high Global Warming Potential (GWP). In VCRS-based air-conditioners, one of the popular zero-ODP alternative refrigerants to R22 is R410A. However, due to its relatively high GWP of 2088, it has a higher harmful impact on global warming than R22 with a GWP of 1810. In accordance with Kyoto Protocol, the emission of gases responsible for global warming, i.e., with relatively high-GWP, should be reduced. Due to their good performance in VCRS-based air conditioners, these high-GWP alternative refrigerants are still widely used, but their impact on the environment cannot be ignored. As the world continues to focus on reducing greenhouse gas emissions and slowing global warming, there is a growing motivation to adopt alternative refrigerants that have both zero ODP and low GWP [10–12]. In any VCRS, the selection of a refrigerant to be used as a working fluid is based on some properties which are desirable, such as relatively high values of critical temperature, thermal conductivity, and latent heat of vaporization/condensation; ozone-friendly with comparatively low GWP; toxic-free; and non-flammable. In fact, none of the available refrigerants can meet all these desirable properties [13]. Several researchers reviewed and investigated various flammable and non-flammable alternative refrigerants [14–16]. However, the focus of the present research will be limited only to non-flammable alternatives to refrigerant R22. Shaik and Babu [17] theoretically investigated the performance of a VCRS-based window-type air conditioner at 7.2 °C evaporating temperature and 54.4 °C condensing temperature. Investigation was performed using pure and blends of Hydrocarbon and Hydrofluorocarbon refrigerants, i.e., R290, R32, R1270, R134a, R125, and R152a, as possible replacements for R22. Refrigerants investigated were environment-friendly and have GWP values in the range of 0.0244–1.685 times that of R22. In comparison to R22 and other refrigerants investigated, refrigerant mixture R290/R1270/R134a with a mass% of 45/5/50 showed lower discharge temperature, 2.1% higher coefficient of performance (COP), 2.01% lower power consumption, and higher overall thermal performance. Kasera and Bhaduri [18] carefully reviewed the effectiveness of R407C, a zero-ODP refrigerant, as a substitute for R22. In terms of COP, energy, and exergy analysis, it was observed that R22 was slightly better than R407C. Using blends of R134a, R125, R32, R290, and R152a as alternative refrigerants to R22, Talanki and Shaik [19] numerically investigated the thermal performance of an actual VCRS at evaporating and condensing temperatures of 7.2 and 54.4 °C, respectively. A MATLAB code was developed, and results demonstrated that COP of the refrigerant mixture R152a/R290 with a mass% of 5/95 was 0.2%-higher COP than R22 with slightly lower GWP and compressor's discharge temperature of 10 and 6.6 °C, respectively. Using engineering equation solver (EES) software, Altinkaynak et al. [20] theoretically studied the energy and exergy efficiency of several alternative refrigerants to R22, i.e., R448A, R410A, and R407C, in a VCRS. For the various evaporator temperatures and refrigerants considered, R22 showed the highest values for COP of 2.81 and exergy efficiency of 52.2% followed by R448A with 2.63 and 48.8%, respectively. Among other substitutes, R448A was selected as the most suitable refrigerant instead of R22. At an evaporating and condensing temperature ranges of -15–15 °C and 35–60 °C, respectively, Shaik et al. [10] numerically evaluated the environmental effect and thermal performance of 27 refrigerant and refrigerant mixtures as substitutes to R22 in a VCRS-based air conditioner. The calculated values of energy efficiency ratio (EER) showed that R22 was superior to the best refrigerant investigated, i.e., R1270 which is a highly flammable low-GWP refrigerant, with about 0.95%. At indoor and outdoor temperature ranges of 17–27 °C and 30–55 °C, respectively, Tarish et al. [21] performed energy and exergy analyses for a VCRS split unit using EES software and R161, which is a flammable refrigerant [22], as an alternative refrigerant to commonly used refrigerants R22 and R134a. Results revealed enhancement of 7.3% in COP and 8.6% in exergy efficiency for R161 when compared with those for R22 and R134a under similar working conditions. Using the experimental data from a VCRS-based water-chiller working R-134A, R404A, R22, and R407C at various ambient and water temperatures, an exergy analysis was performed by Al-Nadawi [23]. The total irreversibility increased with the increase in water flow rate and ambient temperature. Refrigerants R407C and R134a were found as a good replacement for R22. Alawadhi and Phelan [24] reviewed VCRS-base residential air conditioners working in high-ambient temperatures, i.e., higher than 40 °C, and using low-GWP alternatives refrigerants. In comparison with R22 and in terms of COP and cooling capacity, all non-flammable (A1) refrigerants showed lower performance, low-flammable (A2 and A2L) refrigerants attained similar performance, while higher-flammable (A3) refrigerants showed higher performance. An experimental evaluation of two R22 alternative refrigerants, i.e., R458A and R453A, in a VCRS-based split air-conditioner in high-ambient temperatures (between 35 and 52 °C) was presented by Al-Ragom et al. [4]. Both alternative refrigerants showed lower cooling capacity and COP than R22 with somewhat better values for R458A than R453A.

The above published literature investigated, numerically and experimentally, the performance of various VCRSs using R22 and different ozone-friendly alternative refrigerants, both pure and mixtures. These refrigerants can be categorized as non-, low-, or high-flammable refrigerants. However, in high ambient temperature (HAT) countries where high cooling capacities are required with the resulting relatively large refrigerant charges, the utilization of high-flammable refrigerants is not recommended [10, 25]. Moreover, in utilizing VCRSs in HAT countries, where comparatively elevated condensing temperatures are expected, the compressor discharge temperature should always be examined whether it is higher or lower than the refrigerant critical temperature. This may be attributed to the fact that in any VCRS, when refrigerant discharge temperature exceeds its critical temperature, faster drop in the COP of the cycle is expected [11, 13]. Accordingly, the aim of the present research is to investigate, numerically, the thermal performance of a VCRS-based residential air conditioner working in a HAT country using R22 and selected non-flammable, non-toxic, and zero-ODP alternative refrigerants, i.e., R134a, R404A, R407C, R410A, R448A, and R507A. An EES software code was developed which is used for the thermodynamic analysis of the VCRS and the calculation of the various parameters required.

Methods

This section presents a description of the methodology utilized in the present research. This contains detailed information about the refrigerants used, the mathematical model, and the developed EES software code.

Characteristics of the selected R22 alternative refrigerants

In the present research, six alternative refrigerants to R22 were considered, i.e., R134a, R404A, R407C, R410A, R448A, and R507A. Their various properties, such as critical temperature (T_{crit}), critical pressure (P_{crit}), latent heat of vaporization/condensation (h_{fg}), molecular weight, boiling point (BP), category of refrigerant (pure, azeotropic, near azeotropic, and zeotropic), temperature-glide (T_{glide}), safety group, and values for ODP and GWP are given in Table 1. Based on their temperature-glide, refrigerant mixtures are normally classified as: azeotropic refrigerants with zero-temperature glide, near-azeotropic refrigerants with less than 1 °C temperature-glide, and zeotropic (non-azeotropic) refrigerants with temperature-glide higher than 1 °C [26]. It is clear from Table 1 that R410A is the superior refrigerant in the value of latent heat followed closely and respectively by refrigerants R407C and R448A. Consequently, when compared with R22 at the same refrigerant flow rate, a relatively higher cooling capacity can be produced. Conversely, refrigerants R134a, R404A, and R507A will have a lower cooling effect due to their lower latent heat.

Operating conditions

In the present research, Iraq was selected as an example of a high-ambient temperature country. Choosing Baghdad as one of the Iraqi cities, the design indoor and outdoor air temperatures were set according to the Iraqi Cooling Code [28, 29] to be 24 °C and 48 °C, respectively. Knowing that the recommended value of the condensing temperature over the ambient (CTOA) is in the range of 8 to 16 °C [30], an average value of 12 °C was selected resulting in a saturation temperature of 60 °C in the condenser coil. On the other hand, a saturation temperature of 5 °C was considered in the evaporator coil. Furthermore, it was assumed that the compressor isentropic efficiency was 80%, refrigerant pressure losses were negligible, and the process in the expansion device was isenthalpic. In addition, the degree

Table 1 Properties of the various refrigerants investigated in the present research [26, 27]

Property	Refrigerant						
	R22	**R134a**	**R404A**	**R407C**	**R410A**	**R448A**	**R507A**
Composition (Mass %)	R22 (100)	R134a (100)	R125 + R143a + R134a (44 + 52 + 4)	R32 + R125 + R134a (23 + 25 + 52)	R32 + R125 (50 + 50)	R32 + R125 + R134a + R1234ze + R1234yf (26 + 26 + 21 + 7 + 20)	R125 + R143a (50 + 50)
Category of refrigerant	Pure	Pure	Near Azeotropic	Zeotropic	Near Azeotropic	Zeotropic	Azeotropic
Critical Temp. (°C)	96.13	101	72.12	86.2	71.34	82.68	70.62
Critical Pressure (kPa)	4989	4059	3735	4632	4901	4595	3705
Latent Heat of Vaporization / Condensation (kg/kg)[a]	233.75	216.97	200.94	249.08	272.97	241.5	196.94
Molecular Weight (g/mol)	86.5	102.0	97.6	86.2	72.6	86.28	98.9
Normal Boiling Point (°C)[a]	−40.8	−26.1	−46.2	−43.6	−51.4	-46.12	−46.7
Normal Bubble Point (°C)[a]	–	–	−46.6	−43.8	−51.6	-46.39	–
Normal Dew Point (°C)[a]	–	–	−45.8	−36.7	−51.5	-40.21	–
T_{glide} (Nominal) (°C)[b]	0.00	0.00	0.75	7.00	0.08	6.17	0.00
Safety Group	A1	A1	A1	A1	A1	A1	A1
ODP	0.055	0	0	0	0	0	0
GWP_{100} (AR5)[c]	1810	1430	3940	1620	1920	1273	3990

[a] At a pressure of 1 atm

[b] T_{glide} is calculated as the average value of temperature-glide at various pressures

[c] According to the Fifth Assessment Report (AR5) of the Intergovernmental Panel on Climate Change (IPCC)

of superheat at the evaporator outlet was set at 8 °C while the degree of subcooling at the condenser outlet was set at 6 °C.

Thermodynamic analysis of the VCRS-based residential air conditioner

The VCRS considered in the present research consists of the following main components: a hermetic compressor, an air-cooled condenser, an expansion device, and an air-cooled evaporator, as presented schematically in Fig. 1. In addition, the schematic representation of the VCRS on the pressure-enthalpy (P-h) diagram is displayed in Fig. 2. From which and using EES software, it can be found that the refrigerant temperature and specific enthalpy at the evaporator outlet (or compressor inlet) can be expressed as:

$$T_{evap\,out} = T_{comp\,in} = T_{sat\,evap} + \Delta T_{SH} \Rightarrow T_1 = T_7 + \Delta T_{SH} \tag{1}$$

and,

$$h_{evap\,out} = h_{comp\,in} = h_1 = h\,@\left(P_{evap}\right)\&\left(T_{evap\,out}\right) = h\,@\left(P_1\right)\&\left(T_1\right) \tag{2}$$

The specific enthalpy (h_{2s}) at the compressor outlet considering an isentropic compression process can be found as:

$$h_{comp\,out\,isen} = h_{2s} = h\,@\left(P_{cond}\right)\&\left(s_{evap\,out}\right) = h\,@\left(P_2\right)\&\left(s_1\right) \tag{3}$$

Using the definition of compressor isentropic efficiency ($\eta_{comp\ isen}$), the actual values of the specific enthalpy at the outlet of the compressor (h_2) and the compressor work in (kJ/kg) can be evaluated as:

$$\eta_{comp\,isen} = \frac{W_{comp\,isen}}{W_{comp}} \tag{4}$$

$$W_{comp\,isen} = h_{comp\,out\,isen} - h_{comp\,in} = h_{2s} - h_1 \tag{5}$$

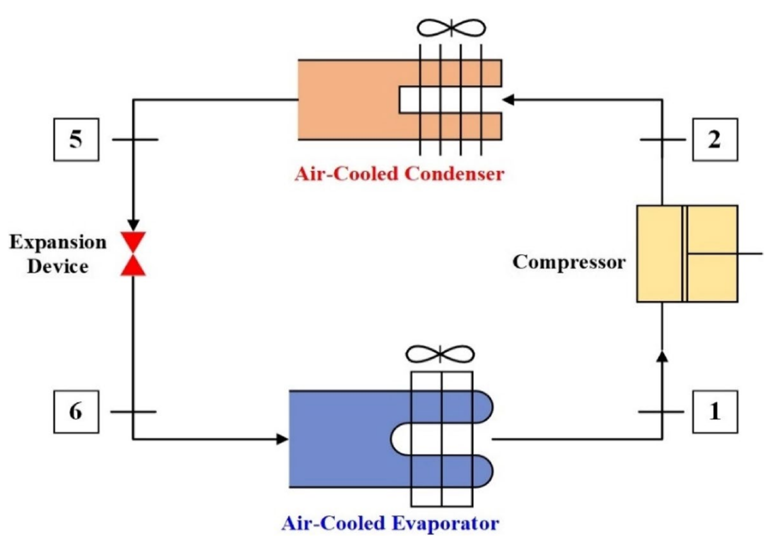

Fig. 1 Schematic diagram of the VCRS considered in the current work

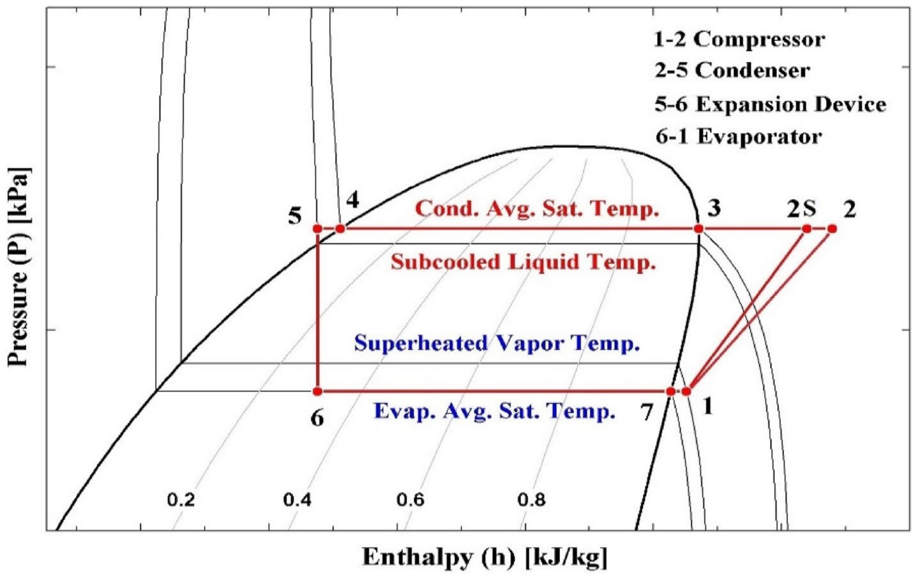

Fig. 2 Pressure-enthalpy (P-h) schematic diagram of the VCRS considered in the present analysis

$$W_{comp} = h_{comp\,out} - h_{comp\,in} = h_2 - h_1 \tag{6}$$

$$\Rightarrow h_{comp\,out} = h_{comp\,in} + \frac{W_{comp\,isen}}{\eta_{comp\,isen}} \Rightarrow h_2 = h_1 + \frac{h_{2s} - h_1}{\eta_{comp\,isen}} \tag{7}$$

$$T_2 = T_{comp\,out} = T_{cond\,in} = T\,@\,(P_{cond})\,\&\,(h_{comp\,out}) = T\,@\,(P_2)\,\&\,(h_2) \tag{8}$$

The refrigerant temperature and specific enthalpy at the condenser outlet (or expansion device inlet) can be expressed as:

$$T_{cond\,out} = T_{ED\,in} = T_{sat\,cond} - \Delta T_{SC} \Rightarrow T_5 = T_4 - \Delta T_{SC} \tag{9}$$

and,

$$h_5 = h_{cond\,out} = h_{ED\,in} = h\,@\,(P_{cond})\,\&\,(T_{cond\,out}) = h\,@\,(P_5)\,\&\,(T_5) \tag{10}$$

Through the expansion device, the process is considered isenthalpic as follows:

$$h_{evap\,in} = h_{ED\,out} = h_{ED\,in} \Rightarrow h_6 = h_5 \tag{11}$$

The equation for calculating the specific cooling capacity (SCC) in the evaporator (q_{evap}) in (kJ/kg) can be determined as follows:

$$q_{evap} = h_{evap\,out} - h_{evap\,in} = h_1 - h_6 \tag{12}$$

For any cooling capacity (Q_{evap}) in ton of refrigeration, the refrigerant mass flow rate ($\dot{m}_{ref}$) in (kg/s) in the VCRS can be evaluated using the following equation:

$$\dot{m}_{ref} = \frac{Q_{evap} \times 3.517}{q_{evap}} = \frac{Q_{evap} \times 3.517}{h_1 - h_6} \tag{13}$$

Using the value of refrigerant vapor density at the compressor inlet ($\rho_{comp\ in}$), the volumetric cooling capacity (VCC) in the evaporator ($q_{evap\ vol}$) was computed as:

$$q_{evap\,vol} = \rho_{comp\,in}\, q_{evap} = \rho_1 (h_1 - h_6) \tag{14}$$

The COP for the VCRS is expressed as:

$$COP = \frac{q_{evap}}{W_{comp}} = \frac{h_1 - h_6}{h_2 - h_1} \tag{15}$$

The above model is applicable for pure refrigerants and azeotropic refrigerant mixtures. For non-azeotropic mixtures, some modifications should be considered. Assuming the refrigerant pressure is constant in the evaporator and condenser and using the EES software with an iterative procedure, the average value of the bubble and dew point temperatures under the evaporating and condensing pressures is used to calculate the saturation temperature in the evaporator and condenser, respectively, as follows:

$$T_{evap\,bub} = T\ @\ (P_{evap}\ and\ X = 0) \tag{16}$$

$$T_{evap\,dew} = T\ @\ (P_{evap}\ and\ X = 1) \tag{17}$$

where X is the vapor quality of the refrigerant.

$$T_{evap} = \frac{T_{evap\,bub} + T_{evap\,dew}}{2} \tag{18}$$

Similarly,

$$T_{cond\,bub} = T\ @\ (P_{cond}\ and\ X = 0) \tag{19}$$

$$T_{cond\,dew} = T\ @\ (P_{cond}\ and\ X = 1) \tag{20}$$

and,

$$T_{cond} = \frac{T_{cond\,bub} + T_{cond\,dew}}{2} \tag{21}$$

The temperature-glide in the condenser and evaporator was calculated as:

$$T_{evap\,glide} = T_{evap\,dew} - T_{evap\,bub} \tag{22}$$

$$T_{cond\,glide} = T_{cond\,dew} - T_{cond\,bub} \tag{23}$$

The temperatures at the outlet of the evaporator and condenser are as follows:

$$T_{evap\,out} = T_1 = T_{evap\,dew} + \Delta T_{SH} \tag{24}$$

$$T_{cond\,out} = T_5 = T_{cond\,bub} - \Delta T_{SC} \tag{25}$$

Results and discussion

Using the above-mentioned operating conditions, the mathematical model was solved using EES software for the various refrigerants considered in the present research and the data collected is presented in this section. With the aim of comparing the refrigerants studied, several evaluation indicators were used including GWP; evaporator pressure; discharge pressure and temperature; critical temperature and pressure; compressor pressure ratio; refrigeration effect; COP; and volumetric cooling capacity (VCC). Figure 3 depicts the values of GWP listed in Table 1. From which, it is obvious that from the various R22 alternative refrigerants with zero-ODP considered in this research, refrigerant R448A showed the lowest GWP value of 1273 followed in an ascending order by R134a, R407C, and R410A with GWPs of 1430, 1620, and 1920, respectively. The values of suction and discharge pressure for the different refrigerants in addition to the refrigerant temperature at compressor exit and PR calculated using EES software are presented in Fig. 4. The trend of variation of both suction (Fig. 4a) and discharge pressure (Fig. 4b) versus refrigerant type is similar for all refrigerants. However, monitoring the pressure in the discharge side is more important than that in the suction since it is associated with building air-conditioners with thicker copper-tubes to withstand higher condensing pressures. Accordingly, from Fig. 4b, relatively higher condensing pressures is observed for R410A followed by R507A, R404A, R448A, R407C, R22, and R134a with values of 3839, 2948, 2878, 2829, 2646, 2428, and 1683 kPa, respectively. These values can be expressed with reference to R22 discharge pressure as ($P_{cond}/P_{cond\,R22}$) to give 1.581, 1.214, 1.185, 1.165, 1.090, and 0.693 for R410A, R507A, R404A, R448A, R407C, and R134a, respectively. The assessment process of R22 alternative refrigerants needs to cover the effect of each individual parameter first and then combining the overall effects of all parameters to achieve the overall evaluation. Therefore, the compressor pressure ratio (PR), which directly affects the magnitude of power input to the compressor, is displayed in Fig. 4c. From which, refrigerants with relatively high discharge pressure, i.e., R410A, R507A, R404A and R448A, show comparatively low values of PR, which indicates minimum increase in compressor power when

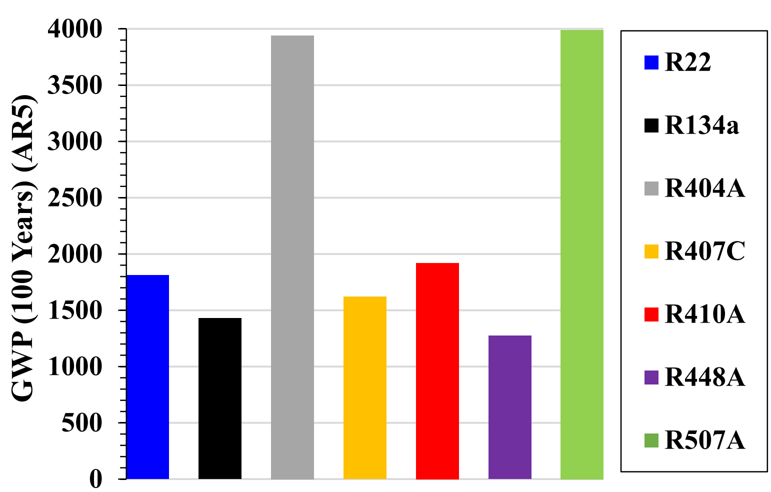

Fig. 3 The values of GWP according to the Fifth Assessment Report (AR5) [27] for the various refrigerants considered in the present research

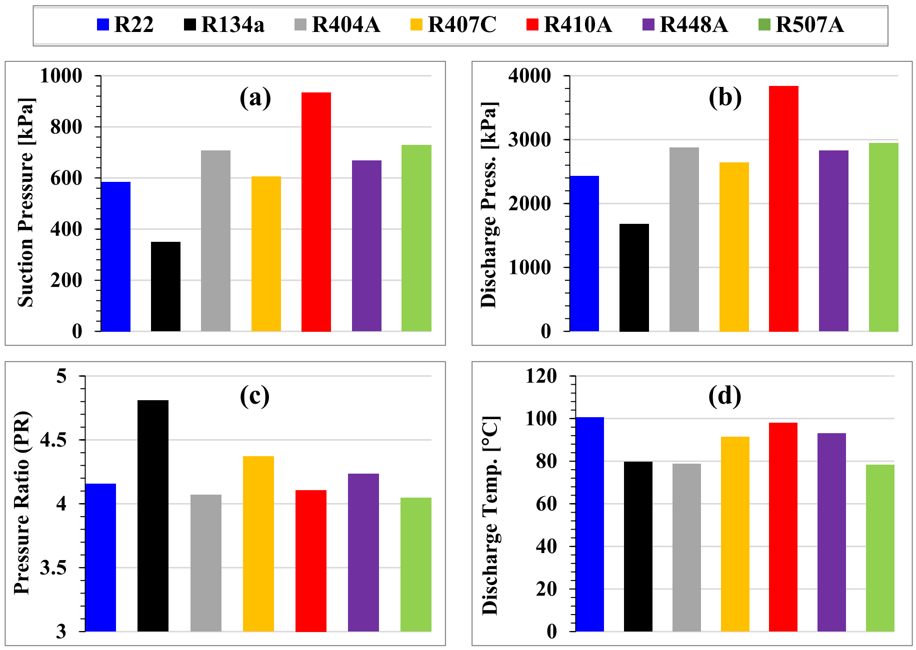

Fig. 4 The calculated values of **a** suction pressure, **b** discharge pressure, **c** PR, and **d** discharge temperature for the different refrigerants investigated in this research

compared with R22. For all the alternative refrigerants investigated, the temperature at the compressor exit is lower than that for R22, as shown in Fig. 4d. However, the discharge temperature alone will not provide the suitable criterion for comparison if not accompanied with its corresponding critical value. Thus, the value of critical temperature and pressure are displayed in Fig. 5 for the various refrigerants studied along with their values at compressor exit. From which, it is obvious that the critical temperature for all alternative refrigerants except R134a are lower than that for R22. In addition, since a HAT country is selected for the utilization of the residential air conditioner in the present research, the discharge temperatures for all refrigerants except R134a are

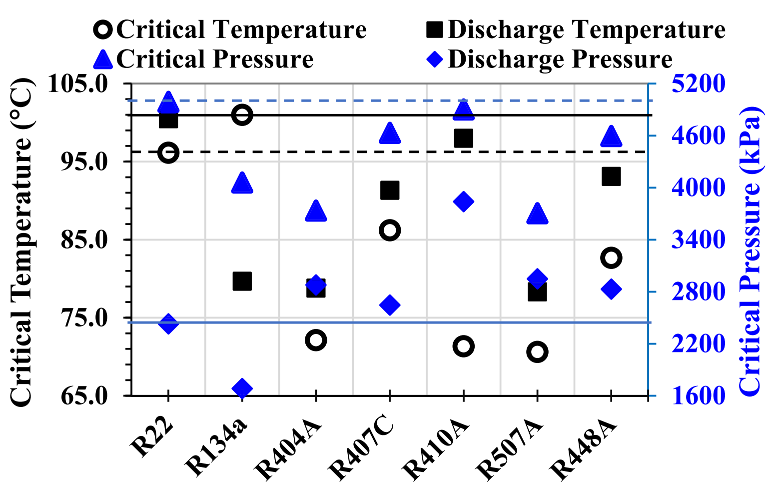

Fig. 5 The calculated values of discharge temperature and pressure versus their corresponding critical values for the various refrigerants considered in the present research

found to be higher than their critical values. Accordingly, a faster drop in the performance of the air conditioner is expected, and this drop is in a relationship with the extent level of discharge temperature over its corresponding critical value [13, 31]. This point will be further discussed in another part of this section. Moreover, Fig. 5 shows that the critical pressure for all alternative refrigerants is lower than that for R22. However, the values of discharge pressure are lower than their critical values for all refrigerants considered in this study.

For the different refrigerants investigated in this research, the calculated values of evaporator specific cooling capacity (SCC) and VCC are shown in Fig. 6. With reference to R22 with a SCC of 144.0 kJ/kg, all selected alternative refrigerants have lower respective values of 141.0, 139.2, 138, 131.0, 91.96, and 88.01 kJ/kg for R407C, R410A, R448A, R134a, R404A, and R507A (Fig. 6a). But these number doesn't mean that the refrigerant with a higher SCC has a better evaporator cooling capacity and superior performance because the cooling capacity is highly dependent on the refrigerant mass flow rate, which is in direct proportion with the refrigerant density at compressor inlet and volume flow rate of the compressor. Hence, to consider the effect of refrigerant density at compressor inlet, the logical comparison of refrigerants should be based on the VCC shown in Fig. 6b. This figure clearly shows a different trend for the VCC than that for the SCC. From which, R410A is the superior refrigerant with a VCC of 4722 kJ/m^3 followed by 3775, 3469, 3422, 3150, 3138, and 2156 kJ/m^3 for R448A, R407C, R22, R507A, R404A, and R134a, respectively. Refrigerants with higher VCCs will require compressors with smaller sizes, i.e., lower volume flow rate and power consumption, to produce the same cooling capacity of other alternative refrigerants or can produce higher cooling capacities utilizing the same-sized compressors of other alternative refrigerants. However, examining the cooling capacity should always be associated with checking the required power input to the compressor to give the overall evaluation of the cycle using COP. In Fig. 7a, the estimated values of the compressor's specific work input for the various refrigerants investigated in the present research are presented. From which, the respective values with a descending order of 51.14, 46.88, 46.82, 45.38, 42.61, 35.62, and 34.72 kJ/kg for R410A, R22, R407C, R448A, R134a, R404A, and R507A were found. The refrigerant with the comparatively highest specific work input was R410A, which may be attributed to the fact

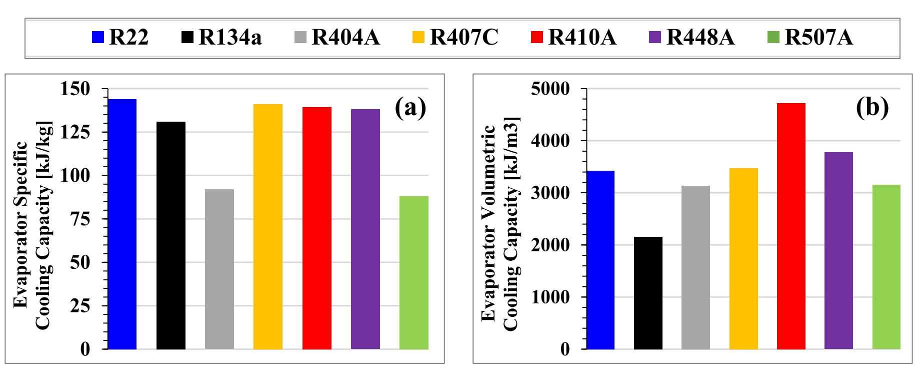

Fig. 6 The calculated values of **a** specific and **b** volumetric cooling capacity in the evaporator for the different refrigerants investigated in this research

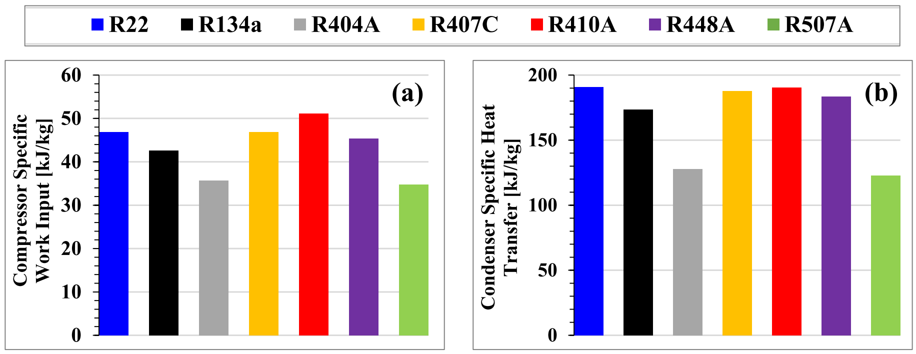

Fig. 7 The calculated values of the **a** specific work input to the compressor and **b** specific heat transfer in the condenser for the various refrigerants considered in this research

that R410A was the refrigerant with the relatively highest condenser pressure and discharge temperature over the critical temperature. For the different refrigerants considered in this research, the predicted values of the specific heat transfer in the condenser are depicted in Fig. 7b. From which, the values of specific heat transfer are 190.9, 190.3, 187.8, 183.4, 173.6, 127.6, and 122.7 kJ/kg for R22, R410A, R407C, R448A, R134a, R404A, and R507A, respectively. The first three alternative refrigerants, i.e., R410A, R407C, and R448A, reveal very close values of condenser's specific heat transfer which make them suitable for applications with VCRSs in HAT countries. Furthermore, the contrasting effects of all the aforementioned factors, either negative or positive, on the thermal performance of the VCRS are grouped and presented in Fig. 8 in the form of the COP of the cycle. In a descending order, refrigerants R134a, R22, R448A, R407C, R410A, R404A, and R507A have calculated COP values of 3.075, 3.072, 3.042, 3.011, 2.721, 2.581, and 2.535. Here we find that a refrigerant with relatively high values of SCC and VCC, such as R410A, shows somewhat low value of VCRS's COP due to its high specific work input. While R448A with its

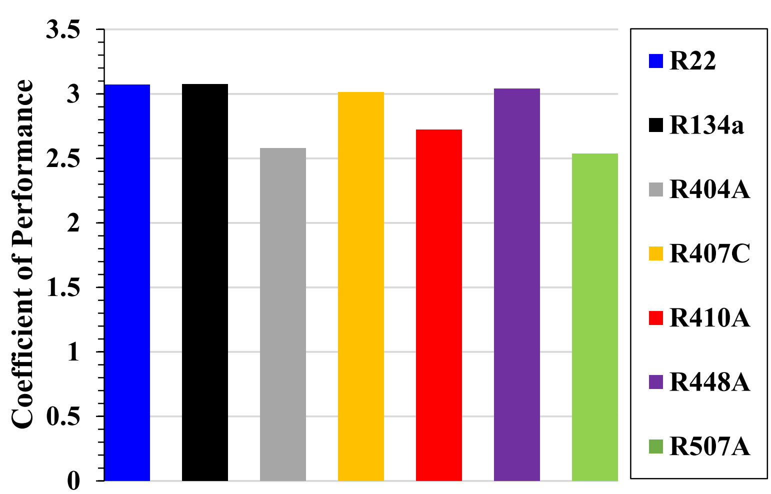

Fig. 8 The calculated values of COP for the different refrigerants investigated in this research

slightly lower values of SCC and VCC showed higher COP owing to its lower specific work input.

With reference to the values for R22, the calculated values of evaporator's VCC, compressor volumetric work input, COP, and compressor pressure ratio for the various refrigerants are presented versus each other in different forms in Fig. 9. For selecting the most appropriate refrigerant, a fast inspection of the three figures included will reveal that refrigerants R410A, R448A, and R407C have a VCC ratio higher than 1, i.e., 1.380, 1.103, and 1.014, respectively, which is a positive point. However, in comparison with R22, these three refrigerants have lower COP ratio, as a negative point, with ascending order values of 0.886, 0.980, and 0.990 for R410A, R407C, and R448A, respectively. At this point, it can be realized that R448A is the superior refrigerant of the aforesaid three refrigerants with respect to COP ratio in addition to its lower ratio for compressor volumetric work input of 0.968, which is lower than those for R407C and R410A with 1.034 and 1.557, respectively. Furthermore, the COP value for R134a is nearly equals that for R22, but with very low ratio for VCC of 0.630 and higher ratio value for compressor pressure ratio of 1.157, as negative points, with a lower ratio for compressor volumetric work input of 0.629, as positive point. In comparison with R22 and with respect to the compressor pressure ratio, refrigerants R410A, R448A, and R407C prevail values for pressure ratios of 0.988, 1.019, and 1.052, respectively. In terms of compressor pressure and COP ratios (Fig. 9c), the priority goes for R448A when compared with other alternative refrigerants considered. Furthermore, for further comparisons between R22 and its selected non-flammable alternative refrigerants in the present research, the refrigerant flow rate required to develop a cooling capacity in the evaporator of 2-Ton was calculated and the values are presented in Fig. 10a. Lower flow rate of refrigerant means the requirement for smaller-sized compressor to

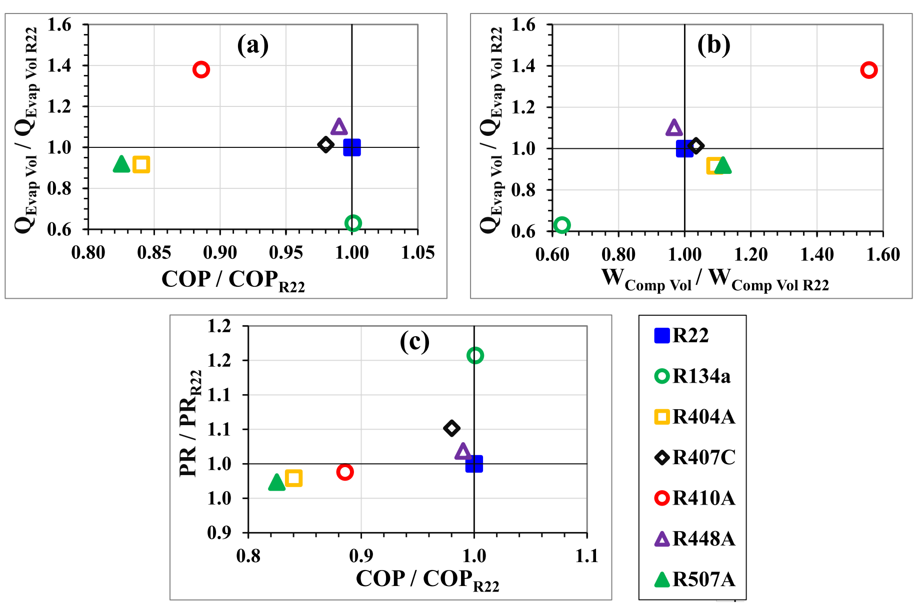

Fig. 9 The calculated values of the evaporator volumetric cooling capacity versus **a** COP and **b** compressor volumetric work input in addition to **c** the compressor pressure ratio versus COP, all referenced to R22 values, for the different refrigerants investigated in this research

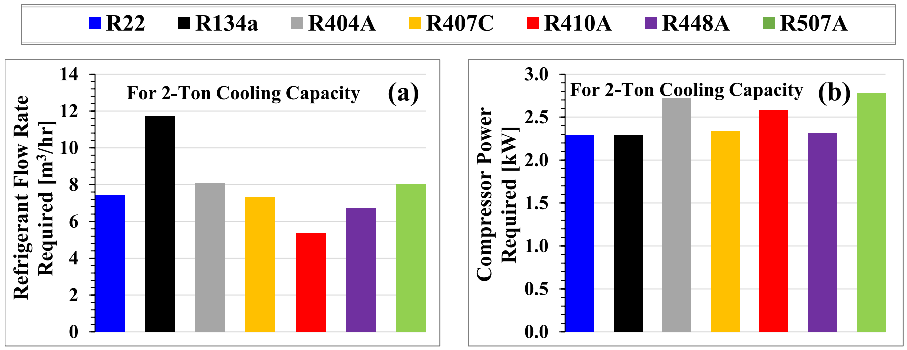

Fig. 10 The calculated values of the required **a** refrigerant flow rate and **b** compressor power for a 2-Ton evaporator cooling capacity for the various refrigerants considered in the present research

reach the expected cooling capacity. From the lower to the higher flow rate, refrigerants can be arranged as R410A, R448A, R407C, R22, R507A, R404A, and R134a with values of 5.363, 6.708, 7.300, 7.400, 8.039, 8.070, and 11.745 m^3/hr. But the compressor power required, which is an essential factor in calculating COP of any VCRS, depends not only on the refrigerant flow rate but also on the compressor's specific work input and refrigerant's density at the inlet to the compressor. Accordingly, the required compressor power to operate a 2-Ton air-conditioner in the selected operating conditions of HAT country was computed and presented in Fig. 10b. From which, the values in an ascending order are 2.287, 2.290, 2.312, 2.336, 2.584, 2.726, and 2.776 kW for R134a, R22, R448A, R407C, R410A, R404A, and R507A, respectively. Except R134a, all the other alternative refrigerants to R22 showed slightly higher power consumption required to operate the compressor. In order to make an overall evaluation for the superior alternative refrigerant to R22 from the ones considered in the present research, all the abovementioned factors should be considered. In terms of thermal performance, all the abovementioned factors and discussions clearly showed that, from the six R22 alternative refrigerants accessed in the present research, i.e., R134a, R407C, R410A, R404A, R448A, and R507A, refrigerants R448A, R134a, and R407C showed higher values for COP ratio. However, these three refrigerants when compared with respect to compressor size, compressor power required, and refrigerant flow required, refrigerants R407C and R448A require minimal values with the positive consequences on air conditioner size and initial cost. Furthermore, it was previously shown that R448A has lower-GWP than R407C, which is an essential point in selecting any substitute refrigerant. Accordingly, for residential air conditioners in HAT countries, it can be ultimately concluded that R448A is the most appropriate substitute refrigerant to R22 that can effectively be employed from the six alternative refrigerants investigated in the present research.

Conclusions

In the present research, six ozone-friendly refrigerants, i.e., R134a, R404A, R407C, R410A, R448A, and R507A, were assessed as non-flammable alternatives to R22 in a residential air conditioner utilized in a HAT country. The evaluation was performed in terms of COP, suction and discharge pressures and temperatures, GWP, critical temperature and pressure, pressure ratio, SCC, VCC, and refrigeration effect. A mathematical model for the VCRS was developed and solved using EES software. From the data collected, presented, and discussed above, the following conclusions can be summarized:

- Refrigerant R410A showed the highest values of suction pressure, discharge temperature, and condenser pressure among the refrigerants investigated.
- For all alternative refrigerants considered excluding R134a, the discharge temperatures were all higher than their corresponding critical values resulting in a faster drop in the performance of the air conditioner.
- R22 showed the highest SCC followed directly by R407C, R410A, and R448A with respective values of 144.0, 141.0, 139.2, and 138.0 kJ/kg. But for a comparison based on VCC, a different trend was found with a maximum value of 4722 kJ/m^3 for R410A followed by R448A with 3775 kJ/m^3. Higher VCCs means the necessity for smaller volume flow rates and lesser-sized compressors.
- Within the alternative refrigerants, R410A, R407C, and R448A showed relatively higher values of compressor's specific work input (51.14, 46.82, and 45.38 kJ/kg) and condenser's specific heat transfer (190.3, 187.8, and 183.4 kJ/kg), respectively. Greater specific heat transfer in the condenser makes the refrigerant more appropriate for applications in HAT countries. However, R407C and R410A showed lower COP values than R448A with respective ascending values of 2.721, 3.011, and 3.042. Refrigerant R410A demonstrated a somewhat lower value of COP owing to its slightly higher values of SCC and VCC with the resulting higher specific work input.
- From the alternative refrigerants with the highest COP values, i.e., R134a, R448A, and R407C, and in terms of energy efficiency, refrigerant flow rate required, compressor size, environmental obligation, and all the comparisons made, it can be assessed that refrigerant R448A is the most promising replacement for R22 which can be successfully used in residential air conditioners in HAT countries.

Nomenclature

h Specific enthalpy (kJ/kg)

$\dot{m}_{ref}$ Refrigerant flow rate (kg/s)

P Pressure (kPa)

Q_{evap} Cooling capacity (Ton of Refrigeration)

q Specific heat transfer (kJ/kg)

s Specific entropy (J/(kg/K))

T Temperature (°C)

W Compressor work (kJ/kg)

X Vapor quality (-)

Greek symbols

Δ Difference

η Efficiency

ρ Density (kg/m^3)

Subscripts

bub Bubble point
comp Compressor
dew Dew point
ED Expansion Device
evap Evaporator
fg Latent heat
glide Temperature-glide
in Inlet
isen Isentropic
out Outlet
sat Saturation
SC Subcooling
SH Superheating
1–7 & 2s Specified in Fig. 2

Abbreviations

COP Coefficient of performance
CTOA Condensing temperature over the ambient
EES Engineering Equation Solver
GWP Global Warming Potential
HAT High ambient temperature
ODP Ozone depletion potential
SCC Specific cooling capacity
VCC Volumetric cooling capacity
VCRS Vapor compression refrigeration system

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Authors' contributions
The author confirms sole responsibility for the following: study conception and design, data collection, analysis and interpretation of results, and manuscript preparation.

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Availability of data and materials
The datasets used and/or analyzed during the current study are available from the corresponding author on reasonable request.

Declarations

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

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