Experimental drop-in replacement of R404A for warm countries using the low GWP mixtures R454C and R455A

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HIGHLIGHTS

- R454C and R455A have similar properties to R404A and low global warming potential.
- The experimental operation and performance of these mixtures and R404A are compared.
- The average alternatives cooling capacity is comparable to that of R404A.
- The R454C and R455A energy performance is higher than that of R404A.
- An internal heat exchanger does not provide significant benefits to R454C and R455A.

Experimental drop-in replacement of R404A for warm countries using the low GWP mixtures R454C and R455A

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Abstract

This article presents an analysis of the feasibility of R454C and R455A, two new low global warming potential (GWP of 148) refrigerants, in vapor compression refrigeration systems as alternatives to R404A for warm countries. R454C and R455A can be the most viable low GWP options to perform a direct replacement of R404A because of the similar characteristics. They only present meaningful differences in flammability, critical temperature, temperature glide and vapor density. The analysis is based on an experimental comparison of R404A with R454C and R455A, using a fully instrumented experimental setup equipped with Internal Heat Exchanger (IHX) at condensation temperatures that represent operating conditions of warm countries. The experimental results show that cooling capacity of the replacements is slightly lower than R404A, being the Coefficient of Performance (COP) of the new mixtures 10-15% higher than that of R404A, especially at higher condensation temperatures. The results also show that the adoption of an IHX is not recommended with the alternatives due to the discharge temperature increase and the low energy performance benefit.

Keywords

greenhouse effect; refrigeration; vapor compression system; HFO/HFC mixtures; energetic performance; internal heat exchanger

Nomenclature

COP	coefficient of performance, -
c_p	specific isobaric heat capacity, kJ kg^{-1} K^{-1}
DP	differential pressure, kPa
$\dot{E}$	power consumption in the motor-compressor set, kW
$\dot{m}_{ref}$	refrigerant mass flow rate, kg s^{-1}

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PR	compressor pressure ratio, -
Q	heat transfer, kW
SCD	subcooling degree, K
SHD	superheating degree, K
T	temperature, °C
ε_{IHX}	heat exchanger effectiveness, %
η	compressor efficiency, -
Subscripts	
Glo	global
in	inlet of the component
k	condensation or condenser
L	liquid phase
o	evaporation or evaporator
out	outlet of the component
V	vapor phase
vol	volumetric
Abbreviatures	
IHX	internal heat exchanger
GHG	greenhouse gas
GWP	global warming potential, measured in kg of CO_2-eq.
HFC	hydrofluorocarbon
HFO	hydrofluoroolefin
IHX	internal heat exchanger
ODP	ozone depletion potential
ON	internal heat exchanger activated
PID	proportional-integrative-derivative
sat liq	saturated liquid
sat vap	saturated vapor
TXV	thermostatic expansion valve
VFC	variable frequency controller

1. Introduction

The average Earth temperature will be increased by 5 °C by 2100 if the emission of Greenhouse Gasses (GHGs) follows such market predictions. This phenomenon would produce irreversible and negative effects on global climate, as multiple natural disasters [1]. Given the contribution of vapor compression refrigeration systems to the global warming [2], the European Union has approved the EU Regulation No 517/2014 and established controls and regulations for HydroFluoroCarbons (HFCs) management [3].

Commercial refrigeration [4] is one of the sectors pointed out by this regulation because of the high accidental leakage ratio, the high amount of refrigerant required and the high global warming potential (GWP) of the refrigerant most commonly used, R404A [5,6]. R404A has good thermophysical and transport properties to operate at low and medium evaporating temperatures, but its GWP is 3922 (3922 times the GWP of CO_2 based on a 100-year time horizon) [8]. Due to the recently imposed restrictions, the most relevant chemical companies are developing alternatives to R404A [8]. These alternatives should be chosen considering different factors such as low environmental degradation, safety, and adaptation to required operating temperatures [9].

Furthermore, Kigali Amendment to the Montreal Protocol established that HFC consumption should be drastically reduced during the coming decades to avoid 0.5 °C increase in the global mean temperature by 2100. The parties also recognized that warmest countries require special measures because the higher ambient temperatures increase the electricity consumption of refrigeration systems [10,11]. Beshr et al. [12] demonstrated that the required reduction of GHG emissions in refrigeration could only be reached by using very low GWP fluids.

The low GWP alternatives to HFCs in vapor compression systems can be categorized into natural and synthetic refrigerants. Natural refrigerants are the hydrocarbons (i.e. R290, R600a or mixtures) [13], ammonia (R717), CO_2 (R744) [14] and water. These refrigerants require the highest initial investment cost and presents higher flammability, toxicity or pressure. On the other hand, synthetic refrigerants include pure fluids, but also mixtures with lower GWP than commonly used HFCs [15]. Three subgroups can be differenced in lower GWP synthetic alternatives to R404A: HFC mixtures, A1 HFO/HFC mixtures, and A2L HFO/HFC mixtures.

HFC mixtures are mainly composed by R407 series refrigerants and R410A. Bortolini et al. [16] recommended R410A for mid-temperature systems because of the higher Coefficient Of Performance (COP). For low-temperature systems, R407F resulted in comparable COP to R404A and increased the cooling capacity. For a set point temperature of 0 °C and a leakage rate of 10%, Cascini et al. [17] also found that R410A results in the best environmental performance but above this leakage rate, and depending on the set point temperature, R407F is the most appropriate option. In current marine refrigeration systems for passenger ships, Pigani et al. [18] concluded that R407F substitution shows an excellent compromise of efficiency, volumetric cooling capacity, GHG emissions, and safety. Llopis et al. [19] found that the use of R407H in R404A low-temperature systems gives an increase in evaporator performance, reduction of the compressor and overall system power consumption and acceptable discharge temperature.

The HFC/HFO mixtures include HFCs seen in other mixtures, like R134a or R32, and the hydrofluoroolefins (HFOs) R1234yf and R1234ze. The final flammability classification and GWP value depend on the percentage of higher GWP A1 refrigerants and lower GWP A2L refrigerants. Wang and Amrane [20] have identified several alternatives with lower GWP that can be a good alternative for drop-in and light retrofit replacement of HFCs. Mota-Babiloni et al. [21] proposed six HFO and HFC mixtures that presented greater performance than R404A. A1 HFC/HFO mixtures are those considered “no flame propagation” and “lower toxicity” refrigerants [22]. Three A1 HFC/HFO mixtures have been studied for stationary refrigeration systems: R448A and R449A, and R452A. Mota-Babiloni et al. [23], used R448A in a medium capacity refrigeration test bench and presented lower cooling capacity than R404A but higher COP. In measurements from an indirect supermarket refrigeration system [24], R449A also showed lower cooling capacity and comparable COP to R404A. However, Sethi et al. [25] showed that in a walk-in freezer/cooler, R448A matches the capacity with 4 to 8% higher COP compared to R404A. Vaitkus and Dagilis [26] obtained poor results for R448A compared to R507A (refrigerant with thermodynamic behavior comparable to R404A) in a eutectic refrigerating system. R452A, in transport refrigeration, achieved an emission reduction of 11 and 12% compared to R404A for medium and low evaporation temperatures, respectively [27]. Kedzierski and Kang [28], in a micro-fin tube, obtained an 8% larger convective boiling heat transfer coefficient for R449A than R448A (but between 26 and 48% lesser than R404A). Righetti et al. [29] reviewed the state of the art of the two-phase heat transfer (boiling and condensation). Despite the promising energetic results, A1 HFC/HFO mixtures are only acceptable at the medium-term because the GWP value (above 1250) is not low enough to stop climate change.

A2L HFC/HFO mixtures are those considered “lower flammability refrigerants with a maximum burning velocity of ≤ 10 cm s^{-1}” and “lower toxicity” refrigerants [22] that results in GWP values below 150. They could be used in hermetically sealed equipment, walk-in coolers/freezers, secondary systems or condensing units, among others. Recently, the new mixtures R454C, R455A, R457A, and R459B have been identified as low GWP candidates to replace R404A [30]. Unlike A1 HFC and HFC/HFO mixture alternatives to R404A, A2L HFC/HFO mixtures have not been yet widely studied, and only a few reports are available with results published in limited conditions, neither of them in scientific papers. The Air-Conditioning, Heating, and Refrigeration Institute published dew point-based results of compressor calorimeter tests of R454C [31] and R455A [32] in R404A reciprocating compressors. Sethi et al. [25] used R455A in a R404A selfcontained freezer, and the 24-h energy consumption was 6% lower with the alternative. Minor et al. [33] reported similar energy consumption between R404A and R454C in a reach-in freezer.

Recently developed low GWP synthetic mixtures alternatives to R404A could offer low electricity consumption for warm climates, but the knowledge about their behavior is still limited [34]. Therefore, this paper presents an experimental analysis of the operation of R454C and R455A, two recently developed GWP<150 mixtures to replace R404A, in commercial refrigeration systems. Due to the special necessities of warm climates, the operating conditions have been selected according to their ambient conditions for food freezing and conservation. The analysis is based on different aspects and using experimental results to provide reliable data about the operation of these low GWP alternatives. The influence of the Internal Heat Exchanger (IHX) on the system is also studied.

2. Characteristics of the proposed A2L HFC/HFO mixtures

The low GWP refrigerants considered in this paper to replace R404A are the mixtures R454C and R455A. Table 1 shows the main information of these fluids that it is used to compare some of their characteristics used to predict the possible behavior of them in refrigeration systems. The thermodynamic and transport properties have been calculated using REFPROP software [35].

Table 1. Properties of A2L HFO/HFC mixtures and R404A

Refrigerant	**R404A**	**R454C**	**R455A**
Composition, mass percentage	R125: 44% R143a: 52% R134a: 4%	R32: 21.5% R1234yf: 78.5%	R744: 3.0% R32: 21.5% R1234yf: 75.5%
Developer	Various	Chemours	Honeywell
Safety classification [22]	A1	A2L	A2L
Critical temperature, °C	72.04	88.47	87.53
Critical pressure, kPa	3728.8	4553.4	4821.8
Normal boiling point, °C	-46.22	-45.56	-52.02
Temperature glide [a], K	0.75	7.80	12.85
Molecular weight, g mol^{-1}	97.6	87.5	90.8
Latent heat of vaporization [a], kJ kg^{-1}	201.1	227.5	239.6
Liquid density [b], kg m^{-3}	1150.0	1136.3	1128.8
Vapor density [b], kg m^{-3}	30.32	20.43	20.98
Liquid c_p [b], kJ kg^{-1} K^{-1}	1.388	1.410	1.433
Vapor c_p [b], kJ kg^{-1} K^{-1}	1.001	0.975	0.975
Liquid thermal conductivity [b], mW m^{-1} K^{-1}	73.11	86.16	87.99
Vapor thermal conductivity [b], mW m^{-1} K^{-1}	12.86	11.90	12.05
Liquid viscosity [b], µPa s	179.3	174.4	170.6
Vapor viscosity [b], µPa s	11.01	10.92	11.07
ODP	0	0	0
AR4 GWP	3922	148	148

[a] at 100 kPa

[b] at 0°C

2.1. Composition

Both R404A replacements, R454C and R455A, show very similar composition, being the base of the composition the HFO R1234yf (A2L and GWP of 0.4) in a proportion of 75.5 and 78.5%, used to maintain the GWP below the threshold of 150. Then, the new mixtures use 21.5% of the HFC R32 (A2L and GWP of 675). In the case of R455A, the mixture is completed using a small percentage of the natural refrigerant R744 (A1 and GWP of 1).

2.2. Security

ANSI/ASHRAE 34 [23] classifies R454C and R455A in the 2L flammability group since the main components of these mixtures, R1234yf and R32, are also included in this group. The small percentage of R744 in R455A, an A1 refrigerant, does not affect its final security classification. Therefore, these fluids exhibit flame propagation when tested at 60 °C and 101.3 kPa, have a maximum burning velocity below 10 cm s^{-1} when tested at 23 °C and 101.3 kPa, the lower flammability limit is greater than 0.10 kg m^{-3} (3.5% in air by volume), and the heat of combustion of less than 19 kJ kg^{-1}. Additional information about the component flammability can be found in [36]. In the event of leakage, the flammability of the component R32 has been considered very low under the general operating conditions of wall-mounted air conditioners [37]. Additionally, R455A possesses a flammability range of 1.1% at 23 °C, that can be considered narrow compared to pure HFOs or other HFO/HFC mixtures [38]. In the same way, all the replacement refrigerants are included in the toxicity group A "lower toxicity". Therefore, their toxicity has not been identified at concentrations less than or equal to 400 ppm by volume.

2.3. Operation

The critical temperature of R404A is approximately 16 °C lower than the alternatives. Thus, a better performance of the system with the R404A alternatives at warm ambient temperatures is expected [39]. Besides, attending to the pressure-temperature relationship (Figure 1), it is seen that the operating pressures of the alternatives are bellow R404A, so the same security devices and adjustments can also be used for these new refrigerants when operating at similar temperature ranges.

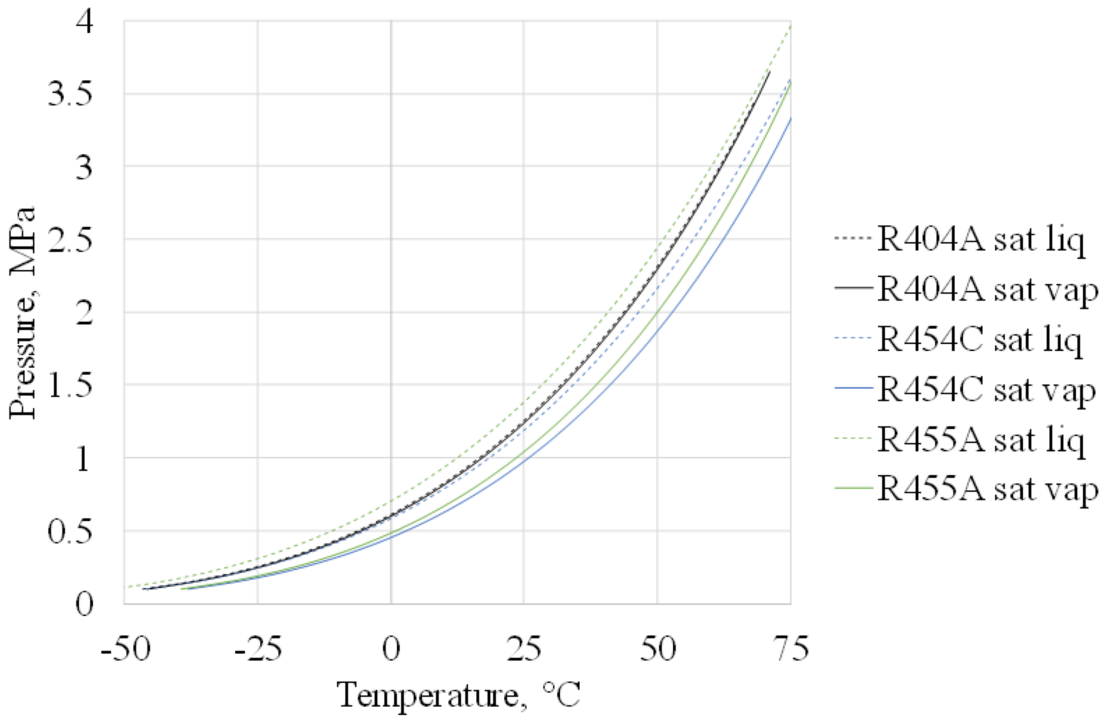

Figure 1. Pressure-temperature diagram for R404A and the two low GWP alternatives

The normal boiling point of the alternatives is below the lower temperature expected in low temperature refrigeration applications (food freezing), -52 °C for R455A, and -45.6 °C for R454C. Therefore, the proposed refrigerants can be used at similar R404A evaporation temperatures for commercial refrigeration systems.

It was common to give priority to azeotropic or near-azeotropic refrigerants as R404A (R507A) in commercial refrigeration systems. However, in the fourth generation of refrigerants, this requisite is not going to be very strict, and the non-azeotropic behavior is tolerable (7.8 and 12.9 K of temperature glide for R454C and R455A, respectively), trying to have a benefit from this behavior in heat exchangers. The pinch point analysis in the adaptation of alternative refrigerants to the existing systems is going to become very relevant. Besides, the glide can introduce a mass transfer resistance that reduces the heat transfer coefficients in the evaporator and the condenser [27, 40, 41]. It produces additional irreversibility due to interactions between the components that reduce the thermodynamic effectiveness [42].

POE type lubricant oils, plastics and elastomers compatibility, water solubility and dielectric properties of R454C are comparable to R404A [34]. Karnaz [43] found that R455A is fully miscible with POE32 lubricant from 0 to 40% lubricant concentration at temperature measurements from 65 to -60 °C.

2.4. Thermodynamic and transport properties

The liquid density is slightly lower, and therefore, the amount of refrigerant charge is going to be like that of R404A, and it is recommended to perform the initial charge at 85% of the previous baseline value in case of retrofit or drop-in substitution. The greatest difference is present in vapor density that reduces the mass flow rate and depending on the refrigerating effect, could also produce lower cooling capacity and higher time-on of the refrigeration system. The alternatives vapor density reduction is around 32% compared to that of R404A.

Besides, the specific heat capacity in the liquid phase is slightly higher, and that property in the vapor phase is slightly lower, so a great influence on heat transfer performance and heat exchanger design is not expected based on the analysis of these properties. The effects observed in the liquid phase and the vapor phase regarding the thermal conductivity property are opposite. While that of liquid is between 18 and 20% around higher, that of vapor phase is approximately 7% lower. Considering that liquid fluid properties dominate in condensation heat transfer; liquid viscosity can compensate the negative effect of glide on alternative refrigerants thermal performance.

The difference observed in viscosity between R404A, and the alternative refrigerants cannot be considered great enough to influence the heat transfer and pressure losses. The vapor viscosity of the alternatives is comparable to that of R404A, and the reduction in liquid viscosity is only 3 and 5% for R454C and R455A, respectively.

2.5. Environmental effects

All the refrigerants accomplish with the requisite of not being harmful to the ozone layer since they do not have chlorine atoms in the molecule. Additionally, the very low GWP values of the alternative refrigerants can reduce the direct contribution substantially to the greenhouse effect in the case of accidental leakage. The GWP value of the alternative refrigerants is 148 (approximately 4% of that shown by R404A).

Considering only the GWP value, the environmental benefit could be very clear. However, the greenhouse gas liberation of fossil fuels burning for electricity generation must be accounted to calculate the total contribution to the greenhouse effect [44]. In the following, the methodology followed to obtain the experimental performance of the system using R404A and its alternatives is described. The energetic performance of the refrigeration system will help to conclude about the feasibility of the substitution of R404A using low GWP alternative refrigerants.

3. Experimental setup

Figure 2 shows the schematic representation of the experimental setup used to study the behavior of the refrigerants in vapor compression systems. This scheme contains all the main components, sensors and instrumentation; and the most relevant secondary components, necessary for a correct operation of the setup.

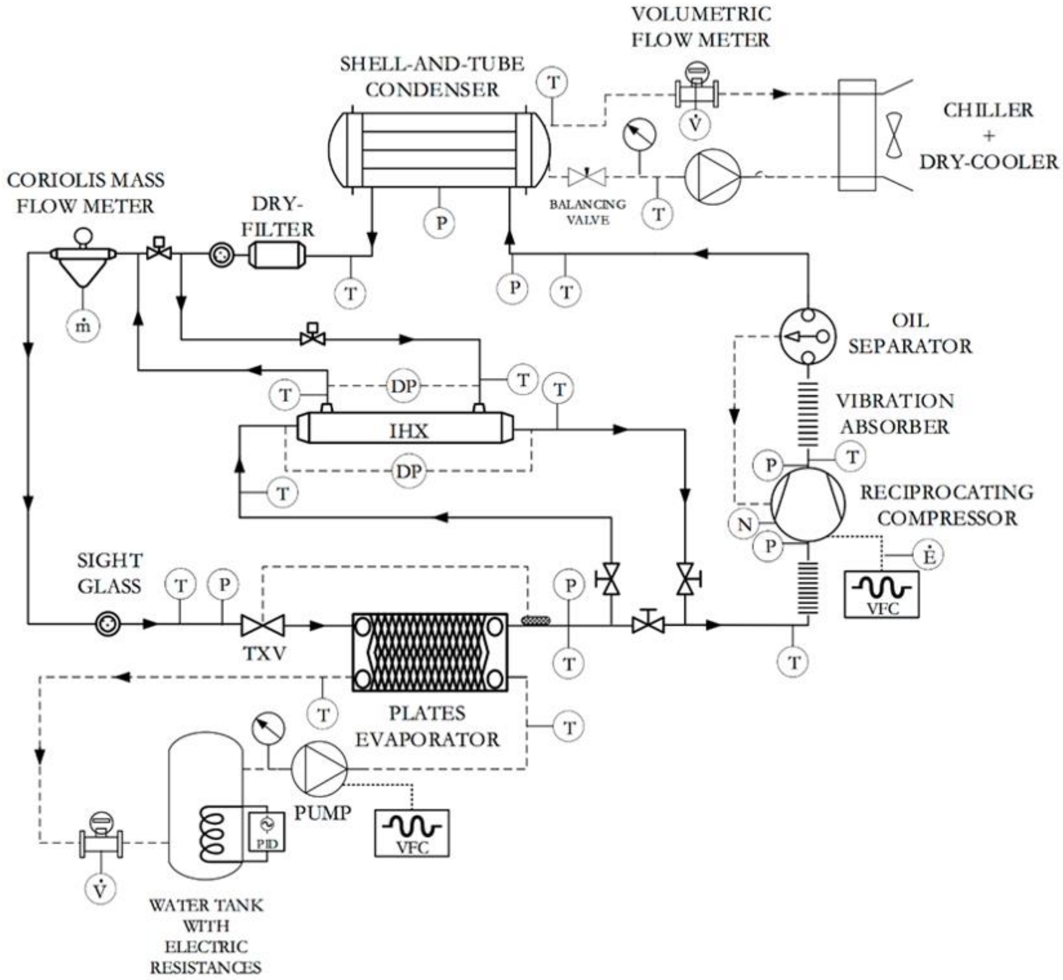

Figure 2. Schematic representation of the experimental setup.

The vapor compression circuit consists of a Bitzer V open type reciprocating compressor (2 cylinders, 681 cm^3 and 573±2 rpm at 50Hz), which is driven by a 7.5 kW variable frequency drive; a SWEP B25Tx20 plate evaporator (20 plates and heat transfer area of 1.13 m^2); a COFRICA RS25 shell and tube condenser (two passes and external heat transfer area of 2.87 m^2), and a Danfoss TS2 Thermostatic Expansion Valve (TXV). Additionally, a tube-in-tube countercurrent IHX can be activated through a set of electronic and manual valves. The main components of the circuit and the refrigerant lines are insulated using flexible elastomeric foam.

Besides, two secondary circuits enable the simulation of the heat source (heat removal circuit) and the heat sink (heat load circuit):

- The heat load circuit uses a commercial propyleneglycol/water brine that is heated through a set of 16 kW nominal power resistances controlled by a PID module. These resistances are in 100 litres insulated tank. The flow of the secondary refrigerant can be controlled using a variable frequency pump. The tubes of this circuit are insulated.
- The heat removal circuit uses water as the secondary fluid that it is cooled using a combination of 7.9 kW chiller and variable frequency dry cooler. The flow rate can be reduced using a manual balancing valve.

K type thermocouples and pressure transducers are located before and after each main component of the vapor compression and secondary circuits. Both parameters are used to calculate the thermodynamic states of the refrigerant using the latest available version of the software REFPROP [34]. The measuring instrumentation also includes a Coriolis mass flow meter for the refrigerant, two electromagnetic volumetric flow meters for the secondary circuits, a differential pressure transducer at both sides of the IHX, and the motor-compressor power consumption measurement from the frequency inverter. Table 2 summarizes the characteristics of the measurements.

Table 2. Measured parameters and details of the sensors

Measured parameters	Sensor	Maximum reading uncertainty	Range
Temperatures	K-type thermocouples	±0.3 K	[-40,100] °C
Pressures	Piezoelectric pressure transducers	±7 kPa	[0,400] kPa
Mass flow rate	Coriolis mass flow meter	±0.22%	[0,0.756] kg s^{-1}
Power input to the motor	Frequency inverter output	±0.45 kW	[0,7.5] kW
Compressor rotation speed	Capacitive sensor	±1%	[0-700] rpm
Pressure drops in the IHX	Differential pressure transducers	±0.01 kPa	Liquid [0,60] kPa Vapor [0,100] kPa

4. Methodology and experimental tests

The methodology proposed is based on the analysis of cooling capacity, the COP and discharge temperature obtained from the experimental tests carried out in the vapor compression test bench presented above where the evaporation and condensation temperatures are varied in the range of conditions for frozen and chilled food observable in warm countries. This analysis has also been extended to the convenience of the adoption of an IHX, evaluating the same energetic parameters from experimental tests.

The middle condensation temperatures (T_k), are 32.0, 39.5 and 47.0 °C, and the targeted middle evaporation temperatures (T_o), are –30.0, –21.5 and –13.0 °C. Both operating temperatures were calculated considering the temperature glide in heat exchangers. Therefore, the average vapor quality is used: (qualities of 0.5 and 0.67 for the condenser and evaporator, respectively), and the average of the inlet and outlet pressures. The highest evaporation and lowest condensation temperature condition has not been tested due to secondary circuits limitation.

The condensation temperature is fixed, and the deviation was always below the thermocouple uncertainty. The evaporator superheating degree was adjusted using the TXV screw to be approximated to 7.5 K, and the condenser subcooling degree was set at intermediate conditions at 2.5 K optimizing the refrigerant charge. Table 3 summarizes the resulting values for the operating conditions. Extended information about the operating temperatures calculation is available in [22].

Table 3. Operating conditions summary

T_k, °C	47			39.5			32		
Fluid	T_o, °C	SHD, K	SCD, K	T_o, °C	SHD, K	SCD, K	T_o, °C	SHD, K	SCD, K
R404A	-12.9	7.6	2.1	-13.0	7.6	2.0			
	-22.5	7.3	2.5	-21.6	7.1	2.2	-21.7	7.7	2.3
	-30.4	7.2	2.7	-30.3	7.9	2.4	-30.3	7.4	2.6
R454C	-12.8	7.4	1.8	-13.1	7.6	1.0			
	-21.7	7.4	2.2	-21.8	7.8	2.4	-21.5	7.3	1.6
	-29.8	7.5	3.1	-29.7	7.8	2.5	-29.7	7.3	1.9
R455A	-12.8	7.4	1.7	-14.8	7.3	2.1			
	-21.8	7.3	2.4	-21.8	7.7	2.5	-25.3	7.6	2.1
	-30.4	7.7	2.4	-30.4	7.6	3.1	-29.9	7.3	2.5

The tests are carried out in steady-state conditions, recording them for 20 minutes and the period between measurements is 0.5 seconds. Then, the most stable 5 minutes period of each test is selected to extract its average values. Each condition is repeated three times, and the standard deviation of condensation temperature was always below 0.51% and that of the evaporation temperature, below 1.04%.

In the evaporator and condenser, the flow rate measurement deviation was within ±5% of the specified one. Then, the maximum variation of the secondary fluid temperature at the outlet of both heat exchangers was less than 0.3 °C. Besides, the voltage measurements of the electrical machines have a tolerance of no more than ± 10%, and the frequency measurements must have a tolerance of greater than ±1% [45]. Furthermore, both the evaporation and condensation pressures are within a range of ±2.5 kPa and the refrigerant mass flow rate within ±0.0005 kg s^{-1}.

The performance of the vapor compression system using R404A and the two alternatives, R454C and R455A, is analyzed in detail through different energetic and operating parameters. Then, the influence of the IHX on the different refrigerants is also studied.

5. Experimental results and discussion

5.1. Cooling capacity

The experimental cooling capacity is calculated using Equation (1) (the maximum uncertainty for the averaged data is ±0.70%, the method for determining this uncertainty propagation is described by Taylor and Kuyatt [46]). The refrigerant mass flow rate is directly measured; the evaporator outlet enthalpy uses the pressure and temperature outlet evaporator measurements, but that at the inlet is based on the measurements at the TXV inlet, assuming isenthalpic expansion.

$$\dot{Q}_o = \dot{m}_{ref}(h_{out} - h_{in})_o \quad (1)$$

Figure 3 presents the experimental cooling capacity of R404A, R454C, and R455A at the different tested evaporation and condensation conditions. The cooling capacity of the refrigerants is greater at higher evaporation temperatures and lower condensation temperatures because the increase of the mass flow rate and the refrigerating effect (this term is calculated as the enthalpy difference at the evaporator), considering comparable superheating and subcooling degree conditions. The cooling capacity values of the alternative refrigerants are close to those of R404A and, following the trend seen in mass flow rate and refrigerating effect, they are benefited from higher condensation temperatures. However, at the lower and intermediate condensation temperatures, the alternatives show slightly lower cooling capacity results.

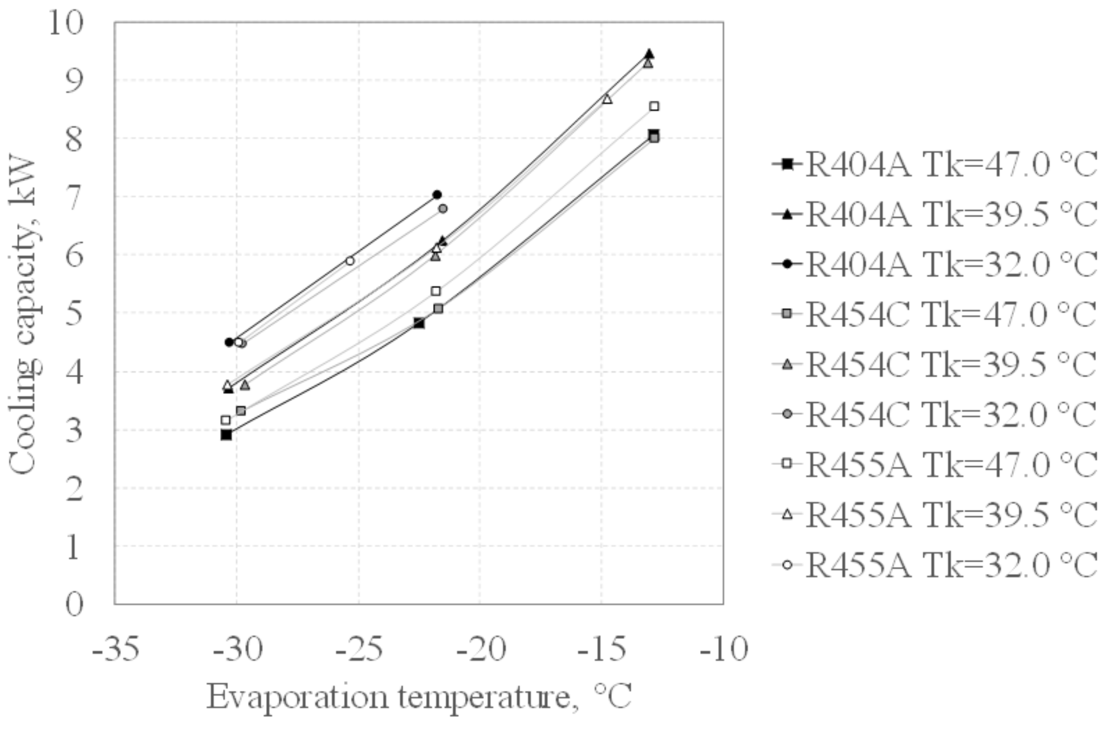

Figure 3. Experimental results for the cooling capacity

The mass flow rate (Figure 4) increase is highly dependent on the suction density and hence, depends on the evaporation temperature and superheating degree (approximately constant). This parameter grows at lower compression ratios (higher evaporation and lower condensation temperatures) because of the volumetric efficiency increase (Table 4). The relative average mass flow reduction compared to R404A is 21% and 23% considering R454C and R455A. The global efficiency values presented in Table 4 are calculated considering the product of the measured mass flow rate and the isentropic compression work divided by the measured electric power consumption of the compressor.

Table 4. Experimental pressure ratios and compressor volumetric and global efficiencies

Tk, °C	47			39.5			32		
Fluid	PR, -	η_{vol}, -	η_{Glo}, -	PR, -	η_{vol}, -	η_{Glo}, -	PR, -	η_{vol}, -	η_{Glo}, -
R404A	5.46	0.72	0.62	4.61	0.76	0.61			
	7.78	0.64	0.63	6.29	0.71	0.64	5.27	0.73	0.62
	10.64	0.53	0.64	8.89	0.62	0.64	7.38	0.68	0.65
R454C	5.98	0.77	0.65	5.04	0.83	0.67			
	8.35	0.71	0.67	7.03	0.77	0.71	5.75	0.80	0.67
	11.65	0.67	0.74	9.73	0.69	0.70	8.06	0.77	0.72
R455A	5.96	0.74	0.64	5.36	0.75	0.63			
	8.39	0.68	0.65	7.02	0.71	0.66	6.68	0.74	0.65
	12.01	0.59	0.68	9.98	0.64	0.67	8.10	0.70	0.69

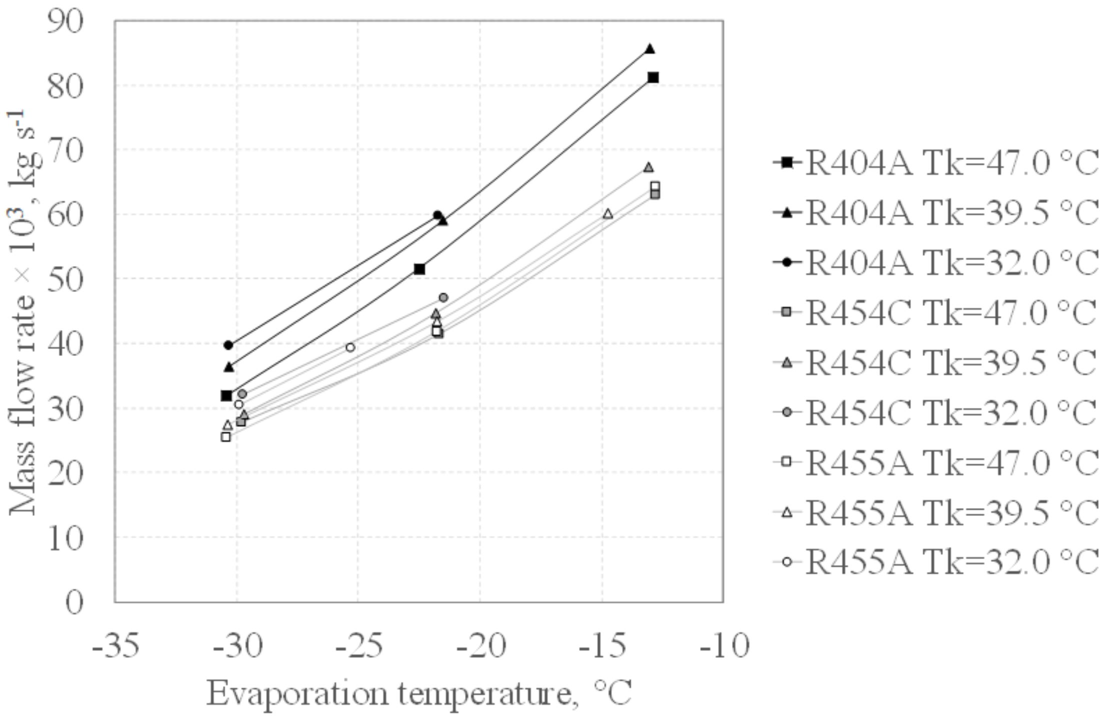

Figure 4. Experimental results for the refrigerant mass flow rate

The variation of condensation temperature, considering similar subcooling degree, affects the vapor quality at the evaporator inlet. In the same way, the variation of evaporation temperature, considering similar superheating degree, also affects the vapor quality at the evaporator inlet. Therefore, the slope of the saturated liquid and vapor lines justifies the greater influence of condensation temperature on the refrigerating effect than the mass flow rate, as reported in

Figure 5. Then, the specific heat of vaporization determines the difference between refrigerants. In this case, the average vapor quality of the alternative refrigerants is greater by 0.9 and 0.11 (R454C and R455A, respectively) and the relative average refrigerating effect increase compared to R404A is 27% and 34% considering R454C and R455A.

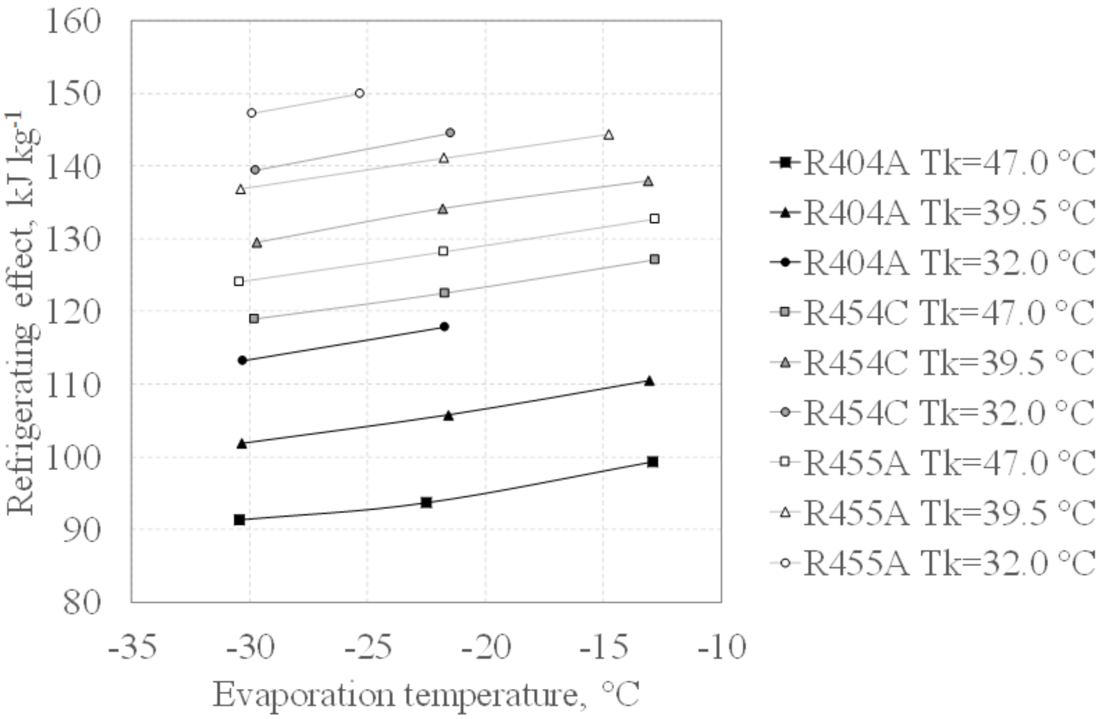

Figure 5. Experimental results for the refrigerating effect

Considering the comparable cooling capacity results of R404A and the low GWP alternatives R454C and R404A, there is no need of compressor substitution to retrofit R404A using the new mixtures. In existing installations, the cooling requirements and hence, the compressor time-on parameters are not going to be affected by the replacement.

5.2. Coefficient of performance

The COP is the ratio between the heat removed from the ambient in the evaporator and the electrical power supplied to the motor-compressor (Equation 2) (the maximum uncertainty for the averaged data is ±0.8% [46]). Therefore, in addition to the previous cooling capacity analysis, this section also discusses section the power consumption results.

$$COP = \dot{Q}_o/\dot{E} \qquad (2)$$

Figure 6 shows the power consumption results directly measured from the frequency inverter connected to the motor. The power consumption augments with the increase of evaporation and condensation temperatures. The power consumption of the alternative refrigerants, R454C and R455A, is 13% and 8% lower than that measured using R404A. The difference between R404A and the alternatives is evident at higher condensation and evaporation temperatures because of the increase of the specific compression work parameter.

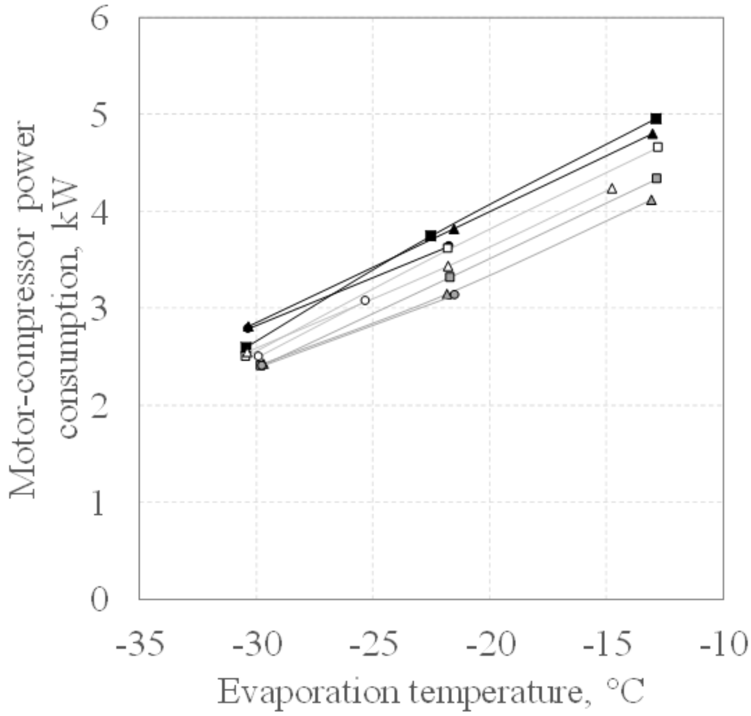

Figure 6. Experimental results for the power consumption

The specific work in the compressor (compressor enthalpy difference) is presented in Figure 7 to complete the power consumption analysis. The higher compression ratio of the alternatives justifies the higher values for this parameter. Although the values of R454C and R455A are 15 and 21% higher than that of R404A, the compressor power consumption does not reflect, in fact, this effect because of the lower mass flow rate values, presented in section 5.1., and the global compressor efficiency (Table 4).

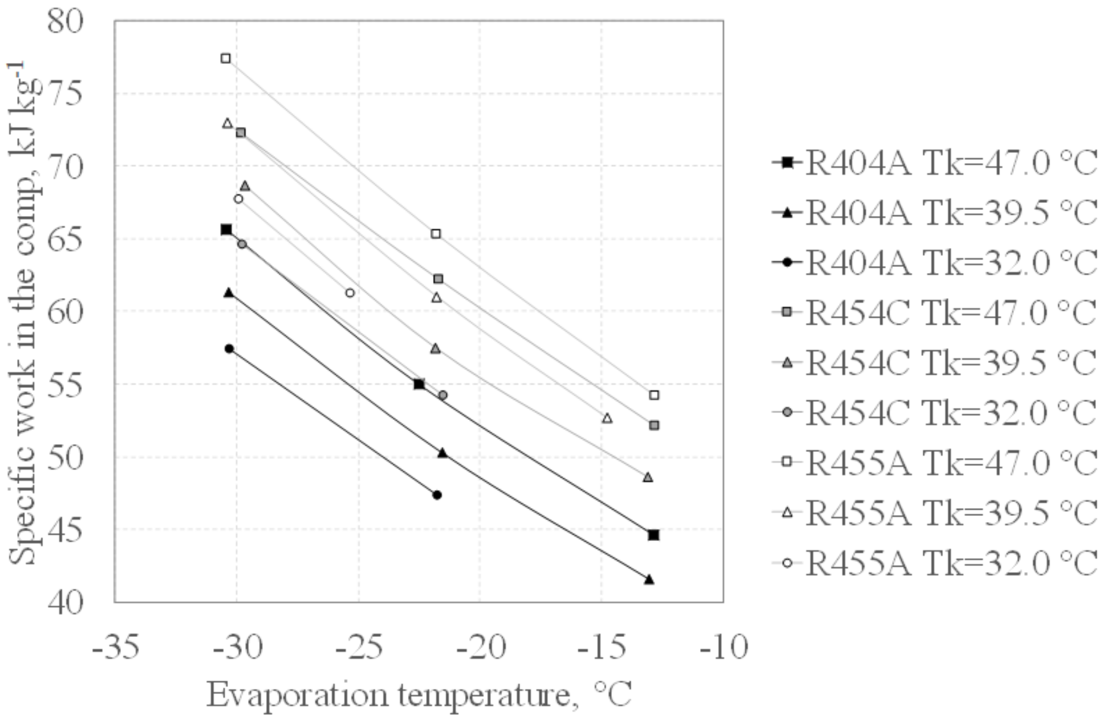

Figure 7. Experimental results for the compressor specific work

Finally, Figure 8 represents the experimental COP values. The COP follows the same trend as the cooling capacity parameter, so COP values are greater at lower condensation temperatures. Better performance values at higher ambient conditions agree with that observed by the two drop-in tests (at 21.1 and 32.2 °C ambient temperature) performed by Minor et al. [34] using R454C in a commercial freezer.

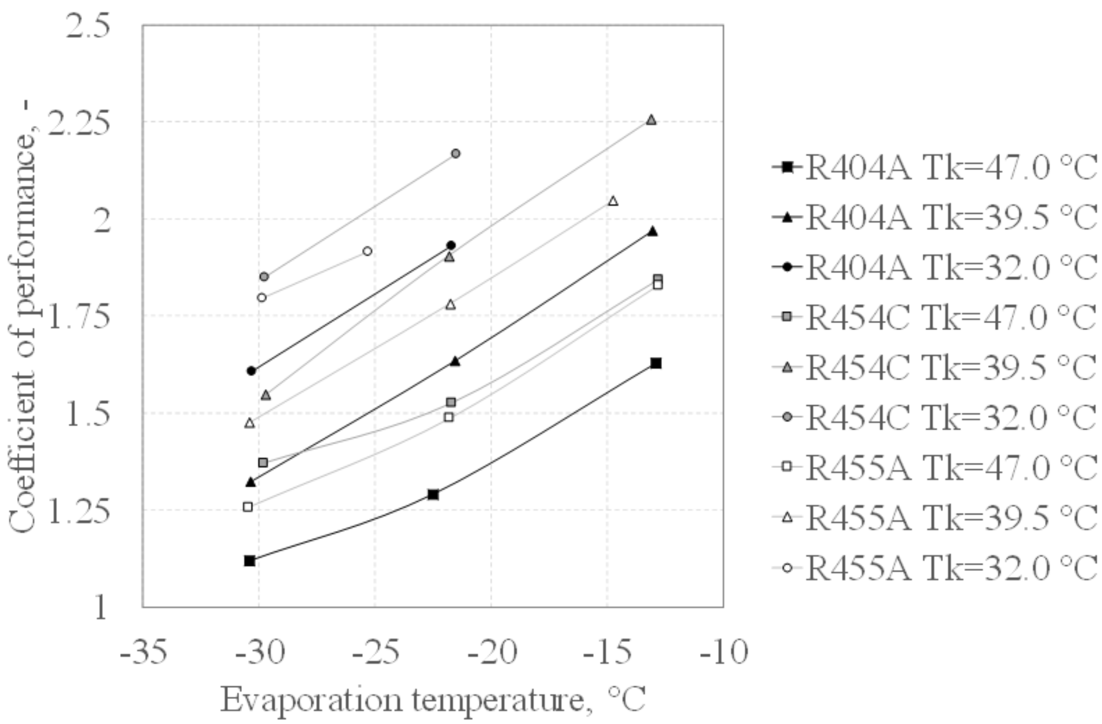

Figure 8. Experimental results for the COP

For all the conditions tested, the energy performance of the alternatives is greater than that of R404A, with an increase about 13% in COP values for higher condensation temperatures. R454C show the best results for all the operating conditions tested. Therefore, the higher COP values of the alternatives can result in energetic, environmental and economic benefits in the case of drop-in replacement.

5.3. Discharge temperature

The discharge temperature must be considered in refrigeration vapor compression systems since a high value can degrade the oil lubricant used in the compressor and this component can fail. Figure 9 shows the compressor discharge temperature directly measured in the discharge line.

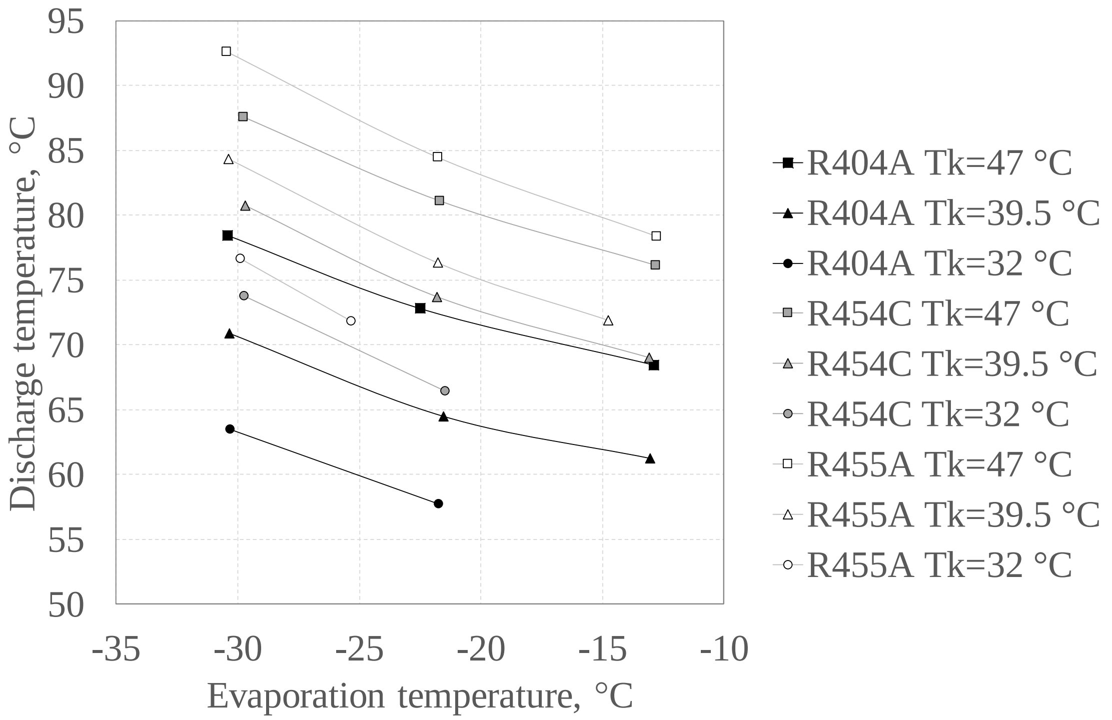

Figure 9. Experimental results for the discharge temperature

For all the refrigerants, the greatest discharge temperatures observed at higher compression ratios. The difference between the alternative refrigerants of R404A is also augmented with the increase of this parameter. R455A shows the highest discharge temperatures of all refrigerants tested because of the higher compression ratio, being 8.9 K the average difference with R404A. Besides, the average difference between R454C and R455A is 3.2 K. For none of the conditions analyzed the discharge temperatures are extreme, and therefore the alternatives can be used in R404A without lubricant oil degradation risk. However, additional cooling for the more extreme operation conditions would be recommended to ensure and enhance the compressor lifetime.

6. Internal heat exchanger

The addition of an internal heat exchanger (liquid-to-suction heat exchanger) can enhance the cooling capacity and COP of a refrigeration system [47]. The final effect (positive or negative) and the benefit vary on the type of refrigerant and the operating conditions. As the size and design of the commercial fridges and freezers ease the installation of an IHX with low investment cost, this section includes a study of the appropriateness of its utilization on the vapor compression system using R404A and its alternatives. The results of the basic cycle with and without the IHX are analyzed and compared to conclude about the suitability of this additional component.

The operating conditions proposed and, if possible, tested, are the intermediate and lower condensation temperatures and the higher and lower evaporation temperatures shown before. Table 5 shows the resulting effectiveness (ε_{IHX}) of the heat exchanger (calculated using Equation 3) and the measured pressure drops. Equation 3 is obtained considering that the minimum heat capacity corresponds to that of the vapor side.

$$\varepsilon_{IHX} = \frac{T_{V,out} - T_{V,in}}{T_{L,out} - T_{V,in}} \tag{3}$$

Table 5. Heat exchange efficiency and liquid and vapor pressure drops in the IHX

T_k, °C	39.5				32			
Fluid	T_o, °C	ε_{IHX}	DP_L, kPa	DP_V, kPa	T_o, °C	ε_{IHX}	DP_L, kPa	DP_V, kPa
R404A	-12.8	20.2%	3.78	7.04	-18.5	21.8%	3.02	7.41
	-29.9	26.0%	2.23	6.57	-30.4	25.9%	2.42	8.86
R454C	-13.2	23.2%	4.41	16.42	-26.7	27.3%	2.69	6.53
	-29.9	28.4%	2.93	8.43	-29.8	28.0%	2.95	7.29
R455A	-14.8	22.8%	3.02	8.69	-26.5	27.2%	2.45	6.24
	-30.3	28.5%	2.49	6.80	-29.7	29.0%	2.82	7.60

The ε_{IHX} depends on the operating conditions and the thermophysical properties of the fluid, considering the same IHX. The higher ε_{IHX} is obtained at the greatest compressor pressure ratio condition. Among the tested refrigerants, R404A presents the lowest ε_{IHX}, and the difference with its alternatives is within 2.1% and 5.5%. The ε_{IHX} of both alternatives is comparable, and the difference between them is equal or below 1%. The additional pressure drops could be detrimental to the energy performance of the system.

Besides, as Figure 10 presents, the discharge temperature is significantly augmented, between 7.1 and 11.8 K for R404A, 7.0 and 10.3 K for R454C and 7.0 and 9.9 K for R455A. Greater differences between discharge temperature with and without the IHX and higher discharge temperatures with IHX are observed at greater pressure ratios. The absolute temperatures reached are close to the maximum recommended limit, and hence the usage of IHX at higher compression ratios must be specifically studied for each case.

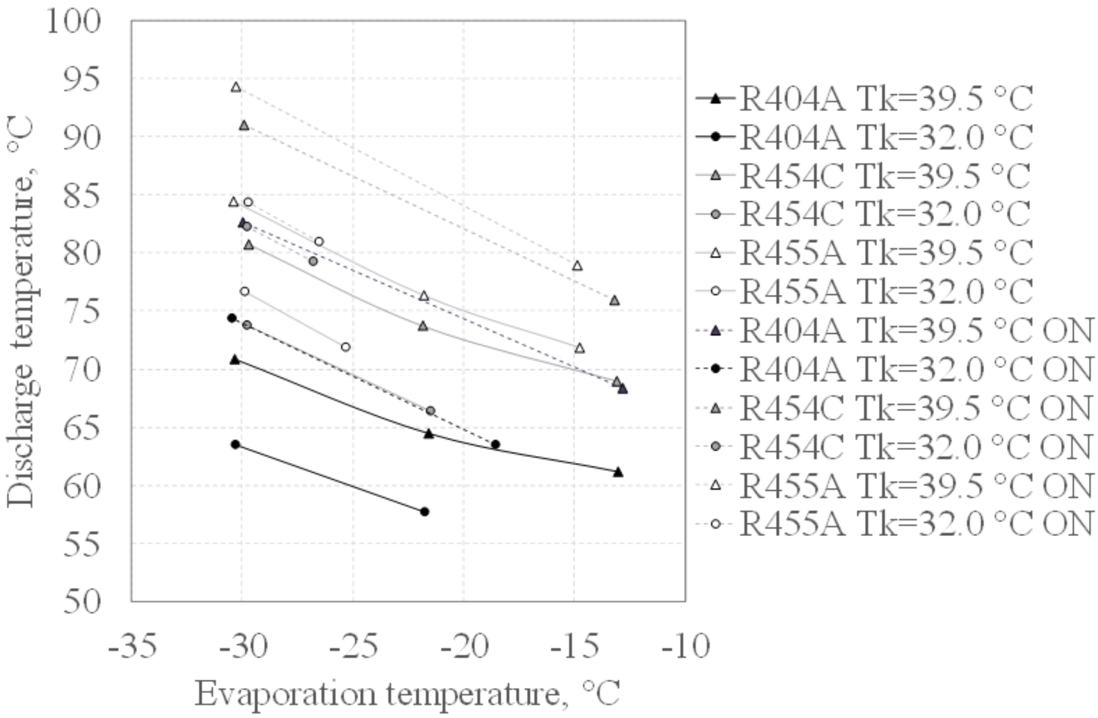

Figure 10. Experimental results for the discharge temperature without and with the IHX

Then, the influence of the IHX on the energetic performance of the vapor compression system is assessed again through the cooling capacity (Figure 11), and COP (Figure 12) experimental analysis. The cooling capacity grows because of the additional subcooling that enhances the refrigerating effect, despite the lighter mass flow rate reduction, and the major increase is observed using R404A. The cooling capacity increase of the alternatives is between 1 and 5%. In the same way, the influence on COP is again higher for R404A than for the alternatives, being the COP increase up to 4% for R454C and R455A. Therefore, the performance of the R454C and R455A refrigeration systems is not benefited from the inclusion of the IHX. Besides, the evolution of COP demonstrates that the power consumption using the IHX is almost the same than without this component.

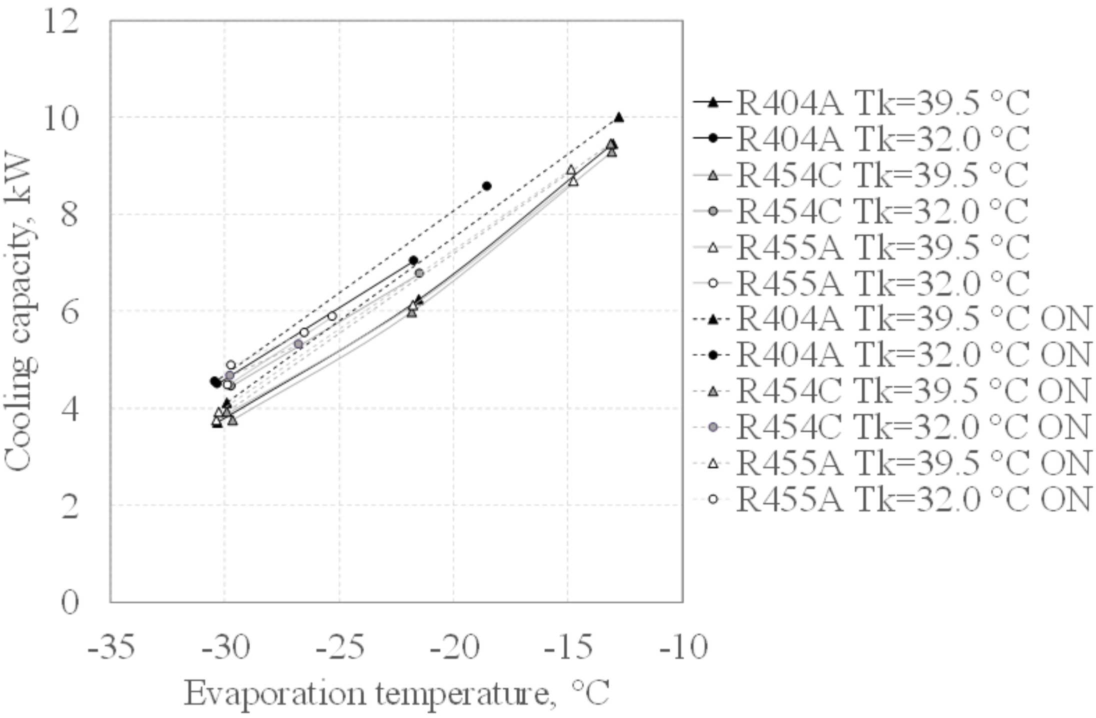

Figure 11. Experimental results for the cooling capacity without and with the IHX

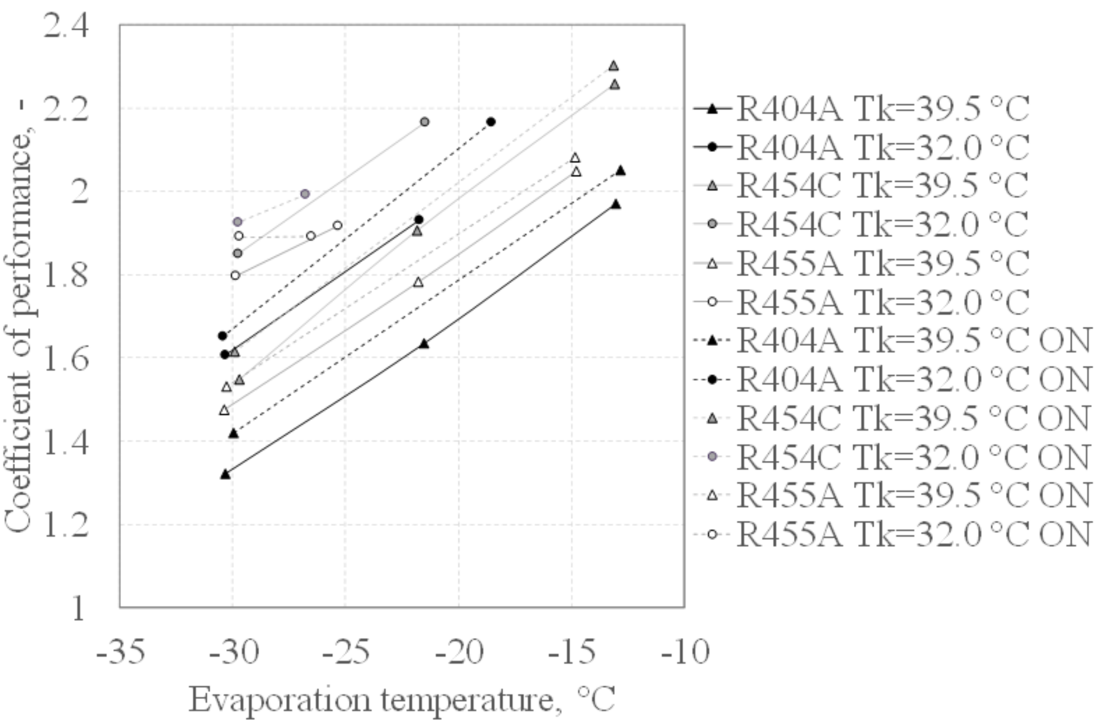

Figure 12. Experimental results for the COP without and with the IHX

Given that for the alternatives R454C and R455A, the greatest increase of COP and cooling capacity is equal or below 4% and involves a strong discharge temperature augmentation, the utilization of IHX is not recommended in these systems.

7. Conclusions

Due to the limitations imposed in the HFC phase-down calendar, this paper proposes and analyzes the drop-in replacement of R404A using the new mixtures R454C and R455A in vapor compression refrigeration systems for warm countries. Considering the lack of research for these refrigerants and their low GWP (below 150), this paper includes a study of the properties and a discussion of the experimental results using basic cycle and IHX configurations.

R454C and R455A present comparable thermodynamic and transport properties to R404A, except for the vapor density and the latent heat of vaporization. Furthermore, both alternatives are lower flammability refrigerants and show remarkable temperature glide (around 7 and 13K, respectively). However, their critical temperatures are higher than that of R404A and make both alternatives promising in high ambient conditions.

The experimental analysis has been carried out using a vapor compression refrigeration test bench at condensing temperatures of 32.0, 39.5 and 47.0 °C, and evaporation temperatures of –30.0, – 21.5 and –13.0 °C. The conclusions reached from the experimental results favor the utilization of R454C and R455A. In comparison with R404A, the average cooling capacity of the alternatives is within the uncertainty of the parameter and the COP is 15% and 10% higher for R454C and R455A, respectively. Although the performance results using IHX are positive with R404A, the maximum COP increase is only 4% for the new mixtures, and their discharge temperatures are near the operating limit.

The similar thermodynamic properties between R404A and the two alternative refrigerants, R454C and R455A, the considerable reduction of GWP, and better energetic performance results, makes these new mixtures appropriate alternatives to R404A in vapor compression refrigeration systems, especially for higher condensation conditions and without IHX.

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