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Retrofit of lower GWP alternative R449A into an existing R404A indirect supermarket refrigeration system

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Highlights

- R449A is retrofitted into a R404A real supermarket refrigeration system.
- Cooling capacity is lower for R449A.
- COP of both refrigerants is comparable.
- R449A discharge temperature is higher but still admissible.
- TEWI analysis indicates that CO_2-eq. emissions of R449A are lower than R404A

Abstract

R404A is going to be phased out from most of the commercial refrigeration systems due to its high GWP value of 3943. R449A (GWP of 1282) has been proposed to replace R404A with only minor system modifications in supermarkets. This paper presents the measurements of a light retrofit replacement of R404A using R449A in a medium temperature indirect refrigeration system (secondary fluid temperature at the evaporator outlet between -9 and -4 °C). It has been demonstrated that with a slight expansion device adjustment and 4% increase of refrigerant charge, R449A can be used in this refrigeration system designed for R404A because of its suitable thermodynamic properties and acceptable maximum discharge temperature. At a secondary fluid temperature at condenser inlet of 30 °C, the COP of R449A nearly matches that of R404A (both were between 1.9 and 2.2), despite having approximately 13% lower cooling capacity. As a conclusion, attending to the GWP reduction and similar energy performance, it was demonstrated using the TEWI methodology that the use of the recently developed refrigerant R449A in these applications can reduce the total CO_2 equivalent emissions of an indirect supermarket refrigeration system designed for R404A refrigerant.

Keywords: supermarket refrigeration, R404A, R449A, HFC HFO mixture, low GWP, alternative refrigerant

Nomenclature

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c_P	Isobaric heat capacity (kJ kg^{-1} K^{-1})
c_V	Isochoric heat capacity (kJ kg^{-1} K^{-1})
$h_{evap,in}$	Refrigerant enthalpy at evaporator inlet (kJ kg^{-1})
$h_{evap,out}$	Refrigerant enthalpy at evaporator outlet (kJ kg^{-1})
k	Heat capacity ratio (-)
$\dot{m}_{ref}$	Refrigerant mass flow rate (kg s^{-1})
P	Pressure (Pa)
$\dot{P}_{comp}$	Power consumption (kW)
PR	Compression pressure ratio (-)
$\dot{Q}_{evap}$	Cooling capacity (kW)
SCD	Subcooling degree (K)
SHD	Superheating degree (K)
T	Temperature (°C)
$\dot{V}$	Compressor displacement (m^3 h^{-1})

Greek

η_{vol}	Compressor volumetric efficiency (-)
ρ_{suc}	Refrigerant density at the compressor suction line (kg m^{-3})

Subscripts

cond	condensing
disc	discharge
eq.	equivalent
sec.cond.in	secondary fluid at condenser inlet
sec.evap.out	secondary fluid at evaporator outlet
sat	saturated
suc	compression suction

Abbreviations

COP	Coefficient of performance
EXV	Electronic expansion valve
GHG	Greenhouse gas
GWP	Global warming potential
HFC	Hydrofluorocarbon
HFO	Hydrofluoroolefin
NBP	Normal boiling point
ODP	Ozone depletion potential
POE	Polyolester

TEWI	Total equivalent warming impact

1. Introduction

Hydrofluorocarbons (HFCs) were listed as greenhouse gases (GHGs) in the Kyoto Protocol (United Nations, 1997), and their use (together with other GHGs) must be strongly reduced to stop climate change. The European Union approved in 2014 the EU 517/2014 Regulation (F-gas) (European Parliament and the Council of the European Union, 2014) that established a calendar to phase out HFCs from different applications, depending on their global warming potential (GWP). The agreement on global HFC gases reduction has been reached recently as well (United Nations Environment Programme, 2016a).

Commercial refrigeration is the vapor compression system application with one of the higher HFC leakages (United Nations Environment Programme, 2016b) because of higher refrigerant amount of charge and probability of failure (Francis et al., 2017). Commercial refrigeration systems are used in supermarkets for food conservation and freezing under controlled conditions. As is the case for other vapor compression applications, the most commonly used refrigerants in developed countries are HFCs. The commercial refrigeration sector accounts for 40% of HFC consumption in the refrigeration/air-conditioning sector (in developing countries this is estimated to be 131 million metric tons of CO_2-eq.) and is known as one of the largest emitters of high GWP refrigerants in Europe (U.S. Environmental Protection Agency, 2010).

R404A and R507A (both present similar compositions and properties) are the refrigerants most commonly used in supermarket (commercial) refrigeration systems (Mota-Babiloni et al., 2015a). However, these fluids present very high GWP values, 3943 and 3985, respectively, and must be phased out from an extensive bank of existing refrigeration applications. There are different lower GWP options to replace R404A, especially low GWP carbon dioxide (R744) (Sawalha et al., 2017), ammonia and hydrocarbons (Antunes and Bandarra Filho, 2016); and HFC mixtures and HFC/HFO (Hydrofluoroolefin) mixtures (Mota-Babiloni et al., 2014). Among both synthetic alternatives, HFC/HFO mixtures can perform as a drop-in/light retrofit replacement with lower GWP values (Mota-Babiloni et al., 2015a).

HFC/HFO mixtures have been developed recently and some studies are currently being conducted to characterize their behavior in vapor compression systems compared to those of HFC refrigerants (Devecioğlu and Oruç, 2015), including R404A. Beshr et al. (2015) have compared the environmental impact of systems using Solstice N40 and L40, which provided relevant environmental benefits. Mota-Babiloni et al. (2014) theoretically studied six R404A alternatives (including four developmental HFC/HFO mixtures) and obtained higher efficiency for all simulated HFC replacements.

Among the HFC/HFO mixtures developed, R448A and R449A have been commercialized and are approved by the US Environmental Protection Agency as low GWP alternatives in commercial refrigeration applications since 2015 (US Environmental Protection Agency, 2015). Kedzierski and Kang (2016) have compared the heat transfer characteristics of both of these fluids with R404A in a 9.5 mm micro-fin tube. Even though both fluids present significantly lower heat transfer coefficients than R404A due to the refrigerant glide, the R449A coefficient is greater than the measured for R448A. Regarding the energy efficiency of both alternatives, Mota-Babiloni et al. (2015b) have presented positive experimental results for R448A for food conservation and freezing conditions that have been confirmed by Sethi et al. (2016) in a field trial evaluation. For R449A, Boscan and Sanchez (2015) have performed a compressor calorimeter test of this fluid and R404A but no field trial studies have been published.

While the new HFO/HFC mixture R449A has only one third of the R404A GWP value, there are still too few data available regarding the feasibility of using R449A in existing R404A refrigeration systems. Thus, the aim of this study has been to analyze the adaptation of this alternative refrigerant in an existing supermarket refrigeration system designed for R404A while performing only minimum modifications (drop-in replacement). The field measurements of the main operating parameters and the energy performance of both refrigerants are compared to qualitatively evaluate the proper adaptation of the new refrigerant in existing systems. This study has been conducted in an indirect medium evaporation temperature refrigeration system operating over a wide interval of time. In this work, a total equivalent warming impact (TEWI) analysis was included to assess the impact of R449A replacement on total CO_2 equivalent emissions.

2. R449A as R404A Alternative

An analysis of the main characteristics of R449A in comparison to R404A has been performed to assess the suitability of R449A to replace R404A in commercial refrigeration systems. Such a system has been modelled to predict the performance of a R449A system.

2.1. Main properties comparison

The new refrigerant R449A is a mixture of R32/R125/R134a/R1234yf at a mass percentage of 24.3/24.7/25.7/25.3, respectively, and developed to replace R404A in supermarket refrigeration systems. The main properties of both refrigerants are shown in Table 1.

Similar to R404A, R449A is a non-flammable and non-toxic refrigerant and is therefore classified as an A1 substance by the ANSI/ASHRAE Standard 34 (American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., 2013). Although the R449A hazardous thermal decomposition products may include carbon oxides, hydrogen fluoride, fluorocarbons and carbonyl fluoride (The Chemours Company, 2015a), these products are similar to those produced from R404A and other synthetic alternatives.

Moreover, the alternative fluid R449A is a non-ozone depleting refrigerant (chlorine free molecule) and its GWP is nearly three times lower than that of R404A, so the R449A negative climate change contribution under gas leakage (direct CO_2-eq. emissions) is less than that of R404A. Contrary to R404A, the GWP limitation for stationary refrigeration equipment established in the EU 517/2014 Regulation (European Parliament and the Council of the European Union, 2014) does not affect the use of R449A.

The normal boiling point (NBP) of R449A is close to that of R404A, so it accomplishes the freezing requirement (its operating evaporator pressure is above atmospheric pressure at low evaporating temperatures). The critical temperature and critical pressure values of this replacement are also higher, so the power required for compressing the vapor goes down. The R449A vapor density is lower and influences the mass flow rate, at a similar volumetric efficiency. Higher viscosity will imply higher energy consumption than ideally expected. Prior to the substitution, vapor line pipe sizing should be revised due to the great difference in density. The great difference in liquid thermal conductivity could affect heat exchangers design. Finally, the higher R449A latent heat of vaporization can compensate for the lower mass flow rate; therefore, the cooling capacity would be close to that of R404A.

In addition, at 0.1 MPa, the R449A temperature glide is 5.7 °C (4.2 °C at 2 MPa) whereas for R404A this value is 0.8 °C (0.3 °C at 2 MPa). This phenomenon implies that R449A cannot be simplified as a near-azeotropic mixture (as usually occurs for R404A). For this reason the redesign of R404A systems (or the consideration of other technologies) to adapt non-azeotropic mixtures such as R449A can lead to an energy efficiency improvement. Furthermore, R449A can present problems in vapor compression system components where liquid can collect.

2.2. Previous analysis

A theoretical vapor compression cycle analysis has been done using the recent refrigerant property data. Both refrigerants are compared at the conditions of the middle evaporating temperature of -13 °C (calculated as the sum of 1/3 of the dew point temperature and 2/3 of the bubble temperature at the evaporating pressure) and the middle condensing temperature of 40 °C (vapor quality 0.5). Other assumptions are 2 K subcooling, 7 K superheating and 65% isentropic efficiency of compression. The modelled results are shown in Table 2.

Under the modelled conditions it is expected that the R449A system will perform with a 6.1% higher coefficient of performance (COP) than the R404A system. The reduction of vapor density at the compressor inlet is compensated for by the higher cooling effect of R449A, so that the volumetric cooling capacity is only slightly reduced. This indicates that the system retrofitted with R449A will deliver a slightly lower cooling capacity. The discharge temperature is expected to be 15 K higher with the R449A refrigerant.

According to the R449A manufacturer, R449A can provide lower energy consumption and similar cooling capacity (The Chemours Company, 2015b). The compressor manufacturer Embraco® published that, although the discharge temperature of R449A was around 6 K higher than R404A, it was lower than those of other R404A alternatives such as R448A, R407A and R407F (Marek and Sedliak, 2015). Kedzierski and Kang (2016) proved that the R449A heat transfer coefficient is 31 to 48% below that of R404A in micro-fin tubes (qualities tested between 10 and 70%).

3. Supermarket Refrigeration Systems

The field trial comparison between refrigerants was done using data measured from a real supermarket medium temperature refrigeration system that provides cooling to the display cases at temperatures between -9 and -5 °C through a secondary fluid (an indirect refrigeration system). Indirect refrigeration systems are used when refrigerant charge amount reduction is a priority (for example to lower refrigerant cost and leakage). A secondary fluid (in this case a mixture of propylene glycol and water) is used to cool the products located in the cabinets of supermarkets or retail stores (Figure 1). The studied supermarket refrigeration system is located in Södertälje (south-central part of Sweden, latitude of 59.2° N), and represents a typical supermarket installation in Sweden.

The analyzed refrigeration system is part of three medium temperature vapor compression systems that are combined in parallel to cover a 264 kW rated medium temperature cooling load. The heat from the connected cabinets is transferred to the systems through a common secondary fluid circuit, and rejected to the ambient through another secondary fluid loop.

The main components of each vapor compression circuit (Figure 2) are the semihermetic reciprocating compressor (8 cylinders) designed for R404A, an electronic expansion valve (EXV), and three plate heat exchangers: condenser, evaporator (both volume per channel of 0.201 liters) and subcooler (0.095 liters). Both the condenser and evaporator exchange heat with a water/ propylene glycol secondary fluid (62/38 per volume).

4. Methodology

4.1. Measurements and data acquisition

The analyzed supermarket system has been monitored over a period of 172 days. During this time, R404A normal operation was established and then R449A was retrofitted into the system. The retrofit procedure followed the refrigerant manufacture recommendation and included a polyolester (POE) oil lubricant substitution, the EXV controller settings adjustment to obtain the correct superheating (approximately 7 K), and the refrigerant charge amount adjustment (12.0 kg for R404A and 12.5 kg for R449A, an increase of 4%).

As shown in Figure 2, the temperatures at the inlet and outlet of the compressor and the heat exchangers were measured using thermocouples PT1000 class A, with an accuracy of ±0.15 K at 0 °C. In order to record the low and high pressures of the vapor compression system, the suction and discharge lines pressures (0-10 and 0-35 bar (g), respectively) were measured using a pressure transducer with an accuracy of ±1%. Temperature and pressure measurements are both used to obtain main refrigerant properties (enthalpy, density, etc.) at different points of interest. Besides, to obtain the compressor electricity consumption an EP Pro Class B power meter (with an accuracy of ±1%) was used.

The measured data have been recorded at 30 seconds intervals and then the main operating parameters have been calculated over 31 days of baseline R404A operation, followed by 46 days of R449A system operation.

4.2. Equations

The COP is expressed as the ratio of the cooling capacity and the measured compressor power consumption, Equation 1.

$$COP = \dot{Q}_{evap}/\dot{P}_{comp} \tag{1}$$

The cooling capacity of the refrigeration system is the product of the evaporator enthalpy difference and the refrigerant mass flow rate, Equation 2.

$$\dot{Q}_{evap} = \dot{m}_{ref}\left(h_{evap,out} - h_{evap,in}\right) \tag{2}$$

Using the volumetric efficiency of the compressor, the suction point density and the compressor geometrical information (displacement of 185 m^3 h^{-1} at nominal frequency, 50Hz), the refrigerant mass flow rate can be calculated, Equation 3.

$$\dot{m}_{ref} = \eta_{vol}\rho_{suc}\dot{V} \tag{3}$$

In a real supermarket refrigeration system some energy parameters are usually calculated using regressions obtained from the compressor manufacturer’s data (Bagarella et al., 2014). Thus, in the case of R404A, the compressor volumetric efficiency is obtained using a third degree polynomial that depends on the evaporating and condensing temperatures for a given superheating degree.

Since the volumetric efficiency of the used compressor is not given for R449A, this value is obtained using Equation 4 which takes into account the difference in thermodynamic properties between both refrigerants. For further analysis, the uncertainty related to the volumetric efficiency is considered 1%.

$$\eta_{vol,R449A} = 1 - \frac{\left(PR_{R449A}{}^{\frac{1}{k_{R449A}-1}}\right)}{\left(PR_{R404A}{}^{\frac{1}{k_{R404A}-1}}\right)}\left(1 - \eta_{vol,R449A}\right) \tag{4}$$

Pressure ratio is obtained dividing the discharge pressure by the suction pressure (Equation 5) and the heat capacity ratio dividing the heat capacity at constant pressure by the heat capacity at constant volume (Equation 6).

$$PR = P_{disc}/P_{suc} \tag{5}$$
$$k = c_P/c_V \tag{6}$$

5. Results and Discussion

In this section, the operating parameters during the stable operation of a real supermarket refrigeration system using R404A and its lower GWP alternative, R449A, are provided. The parameters presented are middle evaporation and condensing temperatures, secondary fluid temperatures at the evaporator outlet and condenser inlet, and superheating and subcooling degrees. Next, the energy parameters of the refrigeration system using both fluids, COP and cooling capacity, are discussed and compared.

Only parameters measured at steady state conditions have been represented (10712 data points for R404A and 6307 data points for R449A). The system was considered stable if under a 10 minutes duration of time the compressor discharge temperature variation was not greater than ±0.1 K.

5.1. System operating conditions

The middle evaporating and condensing (averages of the inlet and outlet of the evaporator and condenser, respectively) temperatures are presented in Figure 3. The lower resulting operating temperatures for R449A are due to its higher temperature glide.

It can be seen that both systems maintained comparable secondary fluid temperatures at the evaporator outlet (Figure 4.a) whereas the secondary fluid temperature range at the condenser inlet covered a wide range of operation conditions (Figure 4.b), with values during R404A operation slightly lower than those during the R449A operation. This was because the R449A system was evaluated during a period of colder ambient temperatures, compared to R404A system.

In addition, the different measured operating temperatures resulted in higher observed compression pressure ratio values for R449A (4.85 on average at stable conditions) compared to that of R404A (4.40 on average at stable conditions). The R449A measured pressure ratio values were higher because of the lower evaporating temperature, glide effects and the influence of suction and discharge pressure drops (R449A is more viscous than R404A). These factors could imply lower compression efficiency, as well as higher compressor isentropic work and discharge temperatures than expected.

The measured discharge temperatures were greater for R449A than for R404A because of the higher compression ratio and specific compression work. At a middle condensing temperature of 35 °C, the discharge temperature of R449A was approximately 15 K higher than that measured using the baseline R404A. In addition, the maximum compressor discharge temperature reached using R449A was approximately 91.3 °C.

Thus, the observed values were within the safe limit defined by compressor manufacturers. It should be noted that the higher temperatures for the R449A system imply greater heat losses from the compressor and consequently a reduction in its overall energy efficiency. Although the limiting value of 120 °C was not reached, at a higher condensing or lower evaporating temperatures than presented, or if a higher total superheating degree occurred, the stability of the lubricants and compressor components could be affected.

As shown in Table 3, during the reference periods the system operated with comparable subcooling (the difference between the bubble point and condenser outlet temperatures) and superheating temperatures (the difference between the dew point and the evaporator outlet temperatures) because of the electronic device and refrigerant charge amount adjustment.

Although both subcooling and superheating degrees were slightly lower for the alternative refrigerant (approximately 1 K lower on average during stable operation), both values were enough to maintain correct operating conditions. The lower subcooling degree suggests that the alternative would have benefited from a slightly higher refrigerant charge amount than that which was established and tested according to the manufacturer's retrofit guideline. The reliable R449A superheating value is due to the presence of the electronic expansion valve in the system, which contributes to a more efficient operation than the refrigeration system with thermostatic expansion valve.

Given the effects of subcooling and superheating degree on refrigeration system performance, the energetic comparisons in this study were performed at 3±2 K subcooling degree and 7±2 K superheating degree values that represented the majority of the measured values.

5.2. Energetic Results

In this subsection the energetic results (COP and cooling capacity) of both refrigerants are discussed at a 30 °C secondary fluid temperature at the condenser inlet over a representative range of secondary fluid temperatures at the evaporator outlet and controlled superheating and subcooling degrees.
The aim of the comparison at the same secondary fluid temperature at the condenser inlet for both fluids is to take into account the effects of temperature glide and heat transfer properties of the refrigerants when analyzing the performance of the refrigeration system. For instance, at the secondary fluid temperature at evaporator outlet of -7 °C, the observed values of middle evaporating temperature for R404A and R449A refrigerants have been established at -12.8 °C and -16.2 °C, respectively. Similarly, at the secondary fluid temperature at condenser inlet of 30 °C, the observed values of middle condensing temperature for R404A and R449A refrigerants have been established at 39.2 °C and 38.8 °C, respectively.

The COP represents the energy performance of a vapor compression system. In refrigeration systems, it is calculated as the ratio of the cooling capacity and the electric power consumption of the compressor, Equation 1. The COP uncertainty values for R404A and R449A were 3.0 and 2.5%, respectively. Figure 5 shows that COP values of both refrigerants are between 1.9 and 2.2 with COP of R449A being on average 3.6% lower at the presented conditions.

Despite the fact that R449A shows a lower cooling capacity (shown below), it consumes less electricity for the compressor so its COP closely matches that of R404A. From another point of view, it can also be said that although R449A works at a higher compression ratio, its higher refrigerating effect nearly balances the greater specific compressor work. Thus, accounting for the uncertainty of COP calculation, it can be concluded that the utilization of R449A in a R404A refrigeration system results in comparable energy performance.

To strengthen the conclusion about COP, the cooling capacity values of both working fluids are shown in Figure 6. The cooling capacity is the product of the evaporator refrigerating effect and the refrigerant mass flow rate, Equation 2. The process of expansion has been considered isenthalpic so the enthalpy at the evaporator inlet has been calculated using the high pressure value and the temperature before the EXV. Moreover, it is considered that all the heat exchanged in the evaporator is transferred to the secondary fluid.

The cooling capacity uncertainty values of R404A and R449A were 4.0% and 3.5%, respectively. The R449A cooling capacity values (between 65.3 and 90.3 kW) were on average 12.8% below those observed using R404A (between 77.7 and 99.0 kW). This reduction was due to the lower R449A mass flow rate (approx. 32%) which was partly compensated for by the higher R449A refrigerating effect (29%).

The observed COP and cooling capacity reduction is greater than that is theoretically predicted when comparing R404A and R449A at same middle evaporating and middle condensing temperatures (as presented in Table 2). This is due to the adverse effect of temperature glide on temperature profile in heat exchangers that influences suction density, compression pressure ratio, refrigerating effect and specific compression work.

Given that no system modifications have been performed in this study, the observed refrigeration system performance can be further improved through engineering optimization. A properly sized compressor, optimized heat exchangers and other improvements would result in better performance of the system.

5.3. $CO_{2\text{-}eq}$ emissions comparison

In this subsection a total equivalent warming impact (TEWI) analysis is presented to assess the CO_2-eq. emissions saved by replacing R404A with R449A. The TEWI analysis gives a good understanding of the phenomenon because it takes into account both direct (due to leakages) and indirect (electricity consumption) emissions. As demonstrated by Makhnatch and Khodabandeh (2014), the TEWI analysis is precise enough to compare the CO_2-eq. contribution of refrigeration systems that use different refrigerants and the contribution of additionally accounted emissions of further analysis as life cycle climate performance is negligible.

The TEWI analysis has been performed for both refrigerants according to the methodology listed in EN 378-1:2008 standard (European Committee for Standardization/Technical Committee, 2008). The main assumptions are used in accordance with the values suggested by the LCCP guideline (International Institute of Refrigeration, 2016) and the annual energy consumption of both supermarkets obtained from experimental observations. Two annual leakage rate scenarios are analyzed: 12%, according to LCCP guideline, and 0.4%, which is the lowest leakage rate reported for the small supermarket refrigeration system (Infante Ferreira et al., 2013).

The TEWI analysis results are presented in the Table 4 and these results prove that the use of the R449A refrigerant in this refrigeration system brings a relevant environmental benefit in both scenarios as it significantly reduces the total CO_2-eq. emissions of the system. While at 12% annual leakage rate the greatest reduction is due to the impact of leakage losses, the greatest reduction at 0.4% annual leakage rate is due to the recovery losses.

5.4. Comparison with other similar works

This subsection compares the results obtained in this paper with those published before with experimental test rigs or real systems. One experimental work found in the open literature was authored by Boscan and Sanchez (2015). As the Boscan and Sanchez comparisons have been presented for the same saturated vapor temperatures, these results cannot be directly compared to those presented in this study due to the effect of glide. However, the main findings of both works are summarized in Table 5.

6. Conclusions

Due to climate change concerns, refrigerants with higher GWP values will be replaced by more environmentally friendly alternatives. The refrigerant R449A has been proposed as an alternative to R404A in supermarket refrigeration systems. The aim of this study has been to provide field test results derived from the utilization of this new fluid in an indirect medium temperature refrigeration system designed for R404A and compare its operation with both refrigerants. The primary conclusions of the paper are presented below.

R449A can replace R404A because of the comparable thermodynamic properties of both refrigerants. The main differences regarding refrigerants properties are observed in the glide (+4.9 K for R449A at 0.1 MPa), vapor density and liquid thermal conductivity (-26% and +26%, respectively).

The operation of both refrigerants in the analyzed middle temperature indirect refrigeration system did not present significant differences. The replacement of R404A in the indirect refrigeration system with R449A required only the EXV controller and refrigerant charge adjustments. The subcooling and superheating degrees of R449A were slightly lower, as also occurred for the mean heat exchangers operating temperatures. The discharge temperature of R449A was greater than that of R404A and reached values of up to 91 °C. Thus, when retrofitting R404A systems attention should be given to the higher compressor discharge temperatures (e.g., in low temperature systems or systems where high superheating is maintained).

Establishing a secondary fluid temperature at the condenser inlet of 30 °C for the energy comparison, R449A benefited from a higher refrigerating effect such that the R449A COP value was similar to that of R404A, despite having a lower cooling capacity. Thus, even though R449A use in a R404A system does not result in greater energy performance, there is an environmental benefit if R449A is retrofitted that comes from the reduction of the CO_2-eq. emissions due to refrigerant leakage and recovery losses.

Therefore, regarding the safety, similarity in operation and comparable energy performance observed through this real case study, it can be concluded that the utilization of R449A can provide environmental and energy benefits. Therefore, R449A could be considered to be a lower GWP alternative to R404A in indirect refrigeration systems where a decrease in cooling capacity is acceptable.

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Figure 1. Simplified schematic of analyzed indirect refrigeration system.

Figure 2. Schematic of the vapor compression system.

Figure 3. Observed middle a) evaporating and b) condensing temperatures.

Figure 4. Secondary fluid temperature a) at the evaporator outlet and b) at the condenser inlet.

Figure 5. COP at secondary fluid temperature at the condenser inlet of 30 °C.

Figure 6. Cooling capacity at secondary fluid temperature at the condenser inlet of 30 °C.

Table 1. Main characteristics of R404A and its alternative R449A (Lemmon et al., 2013). .

	R404A	R449A	Rel. dev.
Molecular mass (g mol^{-1})	97.6	87.2	
ASHRAE safety classification	A1	A1	
ODP	0	0	
AR5 $GWP_{100\text{-year}}$	3943	1282	-67%
Critical Temperature (°C)	345.2	357.0	3%
Critical Pressure (kPa)	3729	4662	25%
NBP (°C)	226.7	227.2	0%
Liquid density [a] (kg m^{-3})	1151	1198	4%
Vapor density [a] (kg m^{-3})	30.32	22.43	-26%
Liquid c_P [a] (kJ kg^{-1} K^{-1})	1.388	1.417	2%
Vapor c_P [a] (kJ kg^{-1} K^{-1})	1.000	0.976	-2%
Liquid therm. cond. [a] (mW m^{-1} K^{-1})	73.15	91.83	26%
Vapor therm. cond. [a] (mW m^{-1} K^{-1})	12.82	12.03	-6%
Liquid viscosity [a] (μPa s)	179.7	190.5	6%
Vapor viscosity [a] (μPa s)	11.00	11.29	3%
Latent heat of vaporization [a] (kJ kg^{-1})	166.0	198.8	20%

[a] at 0 °C conditions

Table 2. Theoretical comparison of R449A and R404A.

	R404A	R449A	Deviation
Refrigerating effect (kJ kg^{-1})	109.1	145.5	33.4%
Vapor density at the compressor inlet (kg m^{-3})	19.90	14.74	-26.0%
Volumetric cooling capacity (kJ m^{-3})	2171	2144	-1.3%
Discharge temperature (°C)	66.8	81.8	15.0 °C
COP (-)	2.22	2.35	6.1%

Table 3. Observed superheating and subcooling values.

	Share of observed stable data points						
Subcooling degree, K	0	1	2	3	4	5	6
R404A	0%	0%	12%	77%	9%	1%	1%
R449A	4%	38%	38%	16%	3%	1%	0%
Superheating degree, K	≤4	5	6	7	8	9	≥10
R404A	6%	3%	8%	25%	30%	16%	12%
R449A	1%	13%	46%	27%	8%	3%	3%

Table 4. Total equivalent warming impact.

	R404A		R449A	
Annual leakage rate, %	12	0.4	12	0.4
kg CO_2-eq. impact due to:				
- recovery losses	4731		1602	
- energy consumption	37783		35701	
- leakage losses	85164	2839	28838	961
TEWI, kg CO_2-eq.	127679	45353	66142	38265

Table 5. Boscan and Sanchez (2015) and the present paper experimental results.

Boscan and Sanchez (2015)		**The present study**	
Comparison at same saturated vapor temperature		Comparison at same secondary fluid at the condenser inlet temperature	
Conditions	Variation of R449A compared to R404A	Conditions	Average variation of R449A compared to R404A
$T_{\mathrm{sat.suc}}$=[-45,0] °C $T_{\mathrm{sat.cond}}$=[20, 60] °C $T_{\mathrm{return\ gas}}$=20 °C SCD=0 K	$\dot{Q}_{evap}$=[-19,5]% $\dot{P}_{comp}$=[-16,10]% COP=[-4,17]% T_{disc}=[9,17]%	$T_{\mathrm{sec.cond.in}}$=30 °C $T_{\mathrm{sec.evap.out}}$=[-8.3, -5.7] °C SCD=3±2 K SHD=7±2 K	$\dot{Q}_{evap}$= -12.8% $\dot{P}_{comp}$= -10.6% COP= -3.6% T_{disc}=+15.2 K

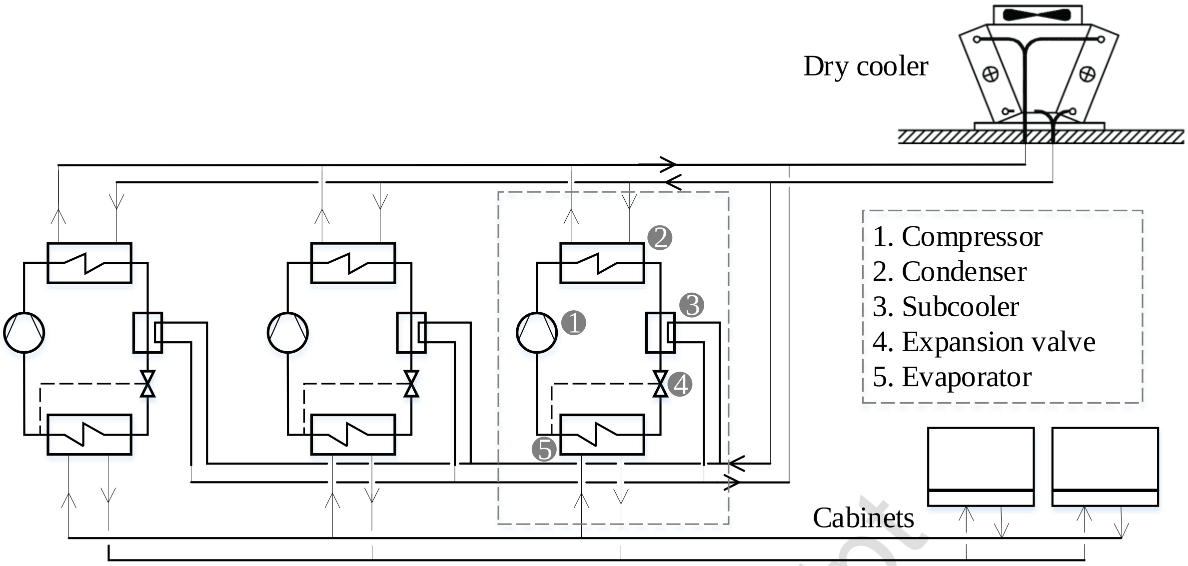

Dry cooler
1. Compressor
2. Condenser
3. Subcooler
4. Expansion valve
5. Evaporator
Cabinets

Figure 1

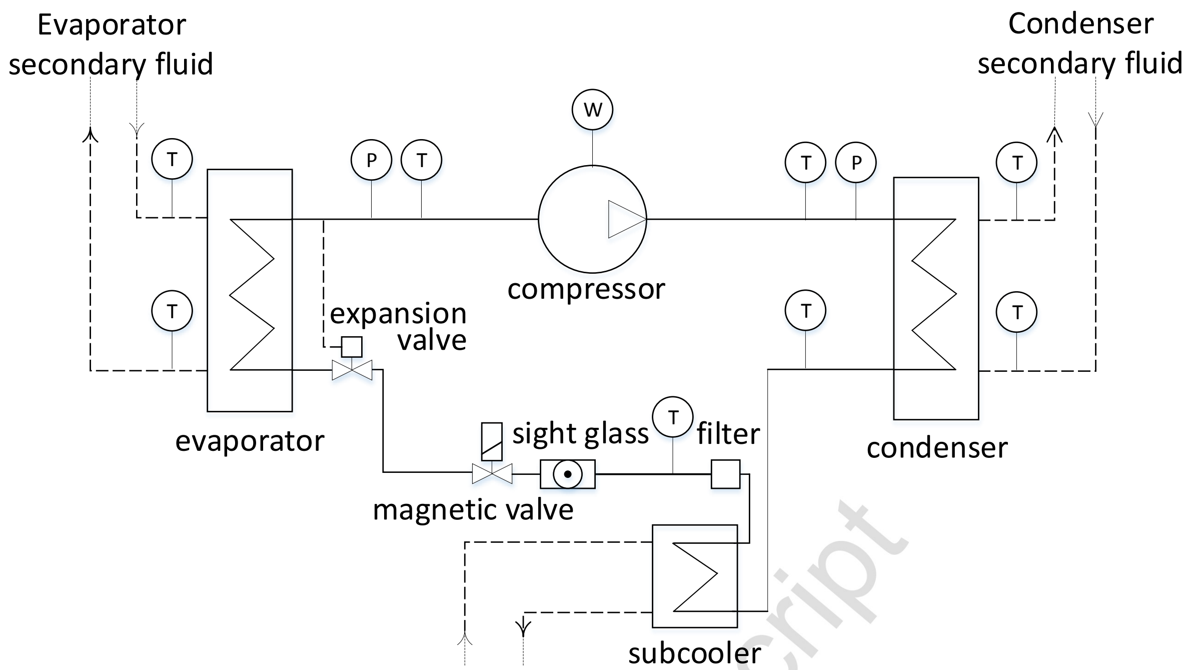
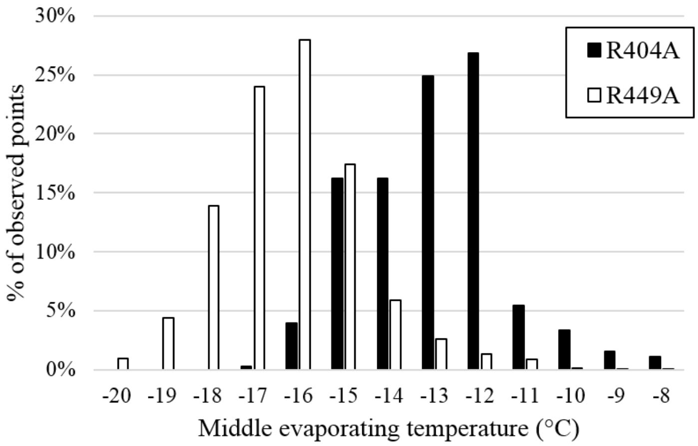
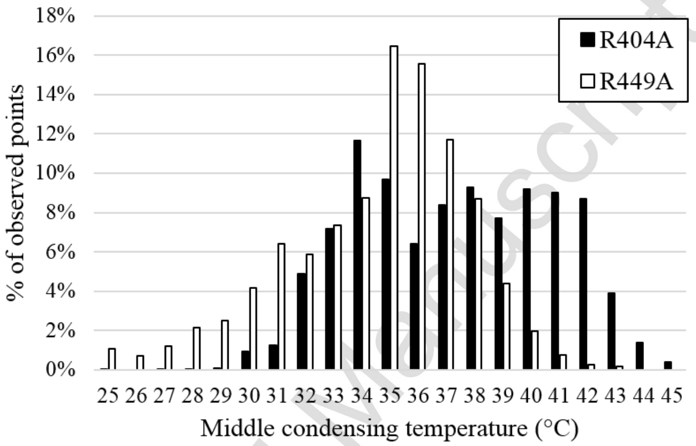

Evaporator
secondary fluid
Condenser
secondary fluid
W
T
P
T
T
P
T
compressor
T
T
T
expansion
valve
evaporator
sight glass
T
filter
condenser
magnetic valve
subcooler

Figure 2

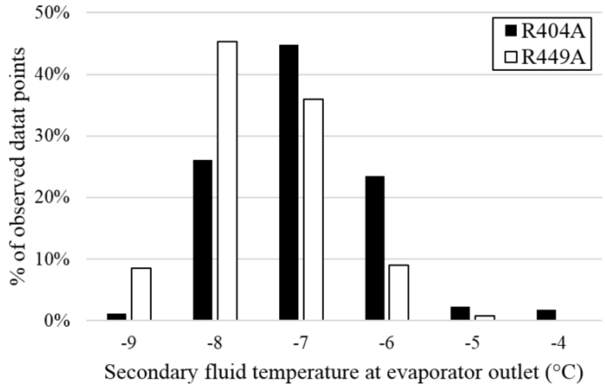

30%
25%
20%
15%
10%
5%
0%
% of observed points
R404A
R449A
-20 -19 -18 -17 -16 -15 -14 -13 -12 -11 -10 -9 -8
Middle evaporating temperature (°C)

a)

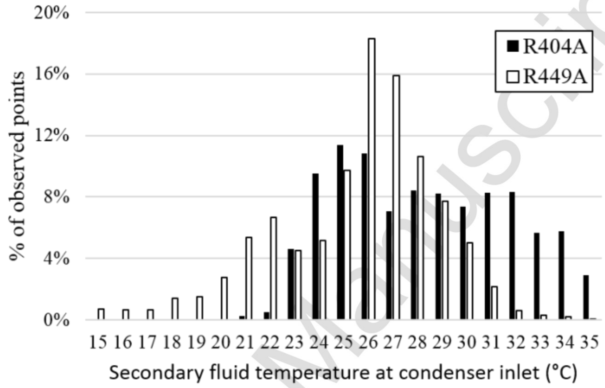

18%
16%
14%
12%
10%
8%
6%
4%
2%
0%
% of observed points
R404A
R449A
25 26 27 28 29 30 31 32 33 34 35 36 37 38 39 40 41 42 43 44 45
Middle condensing temperature (°C)

b)

Figure 3

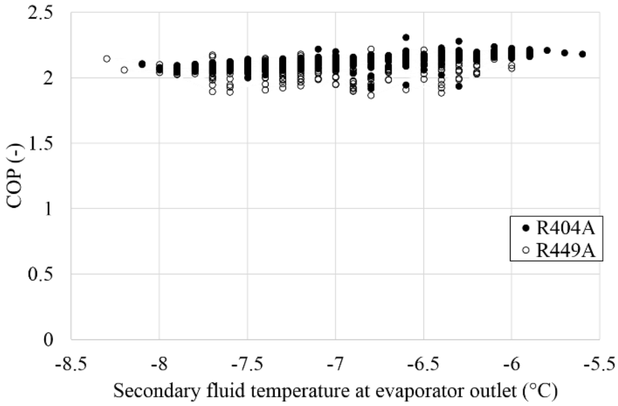

50%
40%
30%
20%
10%
0%
% of observed datat points
R404A
R449A
-9
-8
-7
-6
-5
-4
Secondary fluid temperature at evaporator outlet (°C)

a)


20%
16%
12%
8%
4%
0%
% of observed points
R404A
R449A
15 16 17 18 19 20 21 22 23 24 25 26 27 28 29 30 31 32 33 34 35
Secondary fluid temperature at condenser inlet (°C)

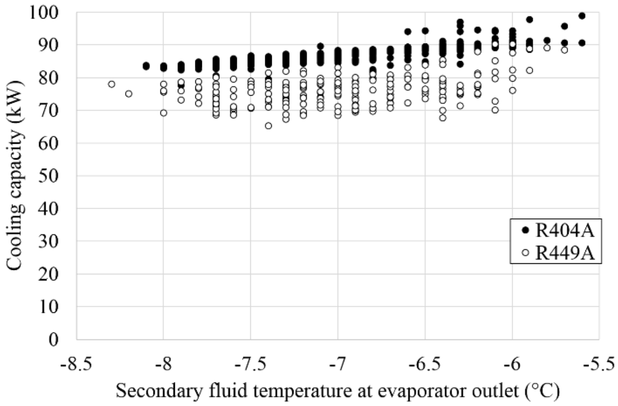
b)

Figure 4



2.5
2
1.5
1
0.5
0
COP (-)
-8.5
-8
-7.5
-7
-6.5
-6
-5.5
Secondary fluid temperature at evaporator outlet (°C)
R404A
R449A

Figure 5



100
90
80
70
60
50
40
30
20
10
0
Cooling capacity (kW)
-8.5
-8
-7.5
-7
-6.5
-6
-5.5
Secondary fluid temperature at evaporator outlet (°C)
R404A
R449A

Figure 6