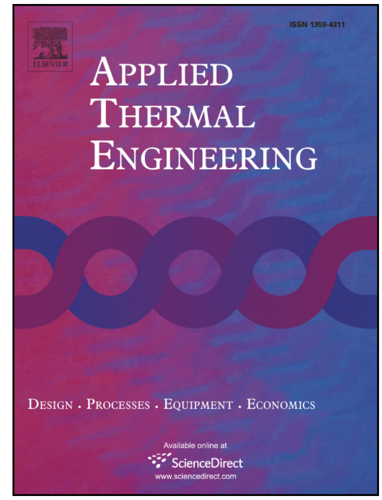
Accepted Manuscript



Experimental influence of an Internal Heat Exchanger (IHX) using R513A and R134a in a vapor compression system

Adrián Mota-Babiloni, Joaquín Navarro-Esbrí, Víctor Pascual-Miralles, Ángel Barragán-Cervera, Angelo Maiorino

PII: S1359-4311(18)32838-2
DOI: https://doi.org/10.1016/j.applthermaleng.2018.10.092
Reference: ATE 12836

To appear in: *Applied Thermal Engineering*

Received Date: 7 May 2018
Revised Date: 10 September 2018
Accepted Date: 21 October 2018

Experimental influence of an Internal Heat Exchanger (IHX) using R513A and R134a in a vapor compression system

Adrián Mota-Babiloni[1,a], Joaquín Navarro-Esbrí[a], Víctor Pascual-Miralles[a], Ángel Barragán-Cervera[a], Angelo Maiorino[b]

[a] ISTENER Research Group, Department of Mechanical Engineering and Construction, Universitat Jaume I, Campus de Riu Sec s/n, E-12071 Castellón de la Plana, Spain

[b] Department of Industrial Engineering, University of Salerno, Via Giovanni Paolo II 132, 84084, Fisciano, SA, Italy

Abstract

Coming refrigeration and air conditioning systems must include low GWP fluids and optimized components. An internal heat exchanger (IHX) is a common modification of the basic cycle to enhance its energy performance, and its benefits have been demonstrated with R134a and the recently developed hydrofluoro-olefin R1234yf. This paper assesses the experimental influence of a high effectiveness IHX using R134a, and the low GWP mixture R513A (a mixture of R134a and R1234yf) under different evaporating and condensing conditions (29 points tested in total). Discharge temperature has been increased up to 26 K for both fluids, and the greatest compression ratio is not feasible for R134a. The cooling capacity of the system results increased up to 5.6% for R513A whereas for R134a is around 3%. Furthermore, due to the minimum diminution of power consumption, COP also increases up to 8% for R513A and 4% for R134a. Because of the observed experimental results, high effectiveness IHX is recommended for R513A, especially for high compression ratio operations as long as the discharge temperature does not reach critical values. Finally, it has been found that Klein et al.'s and Hermes's correlations overestimate the COP benefit and the increase in power consumption should be considered.

Keywords: liquid-to-suction heat exchanger (LSHX); HFO/HFC mixture; low GWP alternative; refrigeration; energy performance; drop-in replacement.

Nomenclature

$c_{p,L}$	isobaric specific heat capacity [a], kJ kg^{-1} K^{-1}
C_r	specific heat ratio (-)
COP	coefficient of performance (-)
D	Klein et al.'s criterion coefficient (-)
h	specific enthalpy of refrigerant (kJ kg^{-1})
h_{lv}	refrigerant enthalpy of evaporation (kJ kg^{-1})
L	temperature lift (°C)
$\dot{m}_{ref}$	refrigerant mass flow rate (kg s^{-1})
N	compressor rotation speed (rpm)
P	pressure (MPa)
$\dot{P}$	Electric power consumption (kW)
PR	pressure ratio (-)
$\dot{Q}$	heat transfer (kW)
$\dot{q}_o$	evaporator refrigerating effect (kJ kg^{-1})
SCD	subcooling degree (K)
SHD	superheating degree (K)

[1] Corresponding author: Adrián Mota Babiloni, PhD
Tel: +34 964 729 151 Fax: +34 964728106
e-mail: mota@uji.es

T	temperature (°C)
v	specific volume (m^3 kg^{-1})
V_{comp}	compressor geometric volume (m^3)
x	refrigerant vapor quality (-)
Greek	
β	volume expansivity (K^{-1})
Δ	increment of associated parameter
ε	heat exchanger effectiveness (-)
η	compressor efficiency (-)
θ	refrigerant expansion factor (-)
ρ	density (kg m^{-3})
φ	available latent heat to produce additional refrigerating effect (-)
Subscripts	
ave	average
glo	global
in	inlet
iso	isentropic
k	condenser/condensing
L	liquid
o	evaporator/evaporating
out	outlet
sat	saturated
V	vapor
vol	volumetric
Abbreviatures	
EXV	electronic expansion valve
GWP	global warming potential
HFC	hydrofluorocarbon
HFO	hydrofluoro-olefin
IHX	internal heat exchanger
MAC	mobile air conditioning
OFF	IHX deactivated
ON	IHX fully activated

1. Introduction

Kigali Amendment to the Montreal Protocol is a transnational emission reduction initiative that projects to reduce the greenhouse gas emissions by 0.7 Gt_{CO2eq} from a baseline level of 1.3 Gt_{CO2eq} [1]. Hence, the most abundant hydrofluorocarbon (HFC) refrigerants, with Global Warming Potential (GWP) values of thousands, must be reduced to accomplish this target. In Europe, R134a is the most emitted fluorine-based fluid, being the average estimated emissions 20.1 (±6.3) Gg yr^{-1} over the period 2003–2014 [2]. Together with water chillers, food conservation, and household refrigerators, R134a is mainly used as a refrigerant (working fluid) in Mobile Air Conditioning (MAC) systems (48–59% in 2010). In China, the range for R134a emissions in 2010 is 10.5–22.7 Gg, and under a Business-as-usual (BAU) Scenario, R134a projected emissions would grow to 89.4 (57.9–123.9) Gg (about 75.3–161.1 Tg CO_{2-eq}) in 2030 [3]. Furthermore, in the USA, the estimated emissions for R134a (R^2>0.5) in 2012 were 69.6 (±18.4) Gg [4].

Unless feasible solutions are provided, it seems that the current situation about R134a emissions is not going to change. According to the United Nations Environment Programme's Report, the global R134a production was 273 kt in the year 2015 [5]. To avoid high costs associated with the Kigali Amendment, the replacement of R134a using low GWP refrigerants must also be made considering the refrigeration energy efficiency. The utilization of more energy efficient technologies with lower GWP refrigerants can reduce between 0.2% and 0.7% the expected global electricity consumption for the period 2018 to 2050 (net cost saving between 240 to 350 billion €) [6].

Lower replacement costs are associated with drop-in or retrofit alternatives. Very low GWP R1234yf and R1234ze(E) hydrofluoro-olefins (HFOs) have been proposed in different vapor compression applications to replace R134a with minor system modifications [7]. On the one hand, in drop-in tasks, R1234ze(E) presents around 30% lower capacity that can only be compensated by an enlargement of the volumetric cooling capacity [8]. On the other hand, R1234yf cooling capacity is closer to that of R134a (the cooling capacity drop is lesser than 10%), but the Coefficient of Performance (COP) is decremented performing this substitution in MACs [9], vending machines [10], and water cooled reciprocating chillers [11]; and therefore, it is not acceptable from an energetic point of view.

To overcome the energetic and flammability limitations, different mixtures of HFOs and R134a [7,12], which are compatible with most of the materials typically used with the HFCs, have been proposed [13]. Although low flammable mixtures with GWP limited to 150 have been tested [14], today, the most promising HFO/HFC mixtures alternatives to R134a are R450A (R134a/R1234ze(E) 42/58 in mass percentage) and R513A (R134a/R1234yf 44/56 in mass percentage). Both have been experimentally studied in a small capacity refrigeration system [15,16], in a commercial refrigeration system [17]; and then, R450A in a water-cooled refrigeration system [11] and R513A in water-cooled and air-cooled screw chillers [18]. From these experimental results, it is seen as this intermediate solution can provide comparable energy performance to R134a.

The internal heat exchanger (IHX), also known as the liquid-to-suction heat exchanger, could be used to increase the energy performance of the vapor compression systems. This additional component has been experimentally studied for R450A [19] with positive energetic results, closer to R1234ze(E) than R134a. However, no experimental results have been published yet about the benefit of IHXs in R513A vapor compression systems.

The study of the IHX on vapor compression systems using mixtures can be initially assessed by the analysis of the effect on its components, in this case R134a and R1234yf. McLinden et al. [20] predicted energy benefits using the IHX cycle with R1234yf because of its higher molar heat capacity. According to REFPROP v9.1 software calculations [21], the molar heat capacity of R1234yf at 0.1 MPa is 4% and 14% higher than R134a in liquid and vapor saturated state, respectively. Pottker and Hrnjak [22] studied the benefit of subcooling with and without IHX in an HVAC module and obtained maximum COP increases of 12% and 17% relative to the condition at zero condenser subcooling using R134a and R1234yf (indoor and outdoor temperatures of 35 °C). In a MAC, Cho et al. [23] improved the COP up to 4.6% by installing the IHX into the R1234yf system. In addition, the discharge temperatures resulted below those of R134a system without IHX. In the same application, varying the compressor speed from 1000 rpm to 2000 rpm and at the air stream temperatures of 27 and 35 °C, Direk et a. [24] increased the R1234yf COP between 6.4% and 9.9 % using an IHX. In a vapor compression system working with and without a 25% effectiveness IHX, Navarro-Esbrí [25] obtained a maximum COP increase of 10% for R1234yf, whereas for R134a it was approximately 6% (at the highest compression ratio tested). Sethi et al. [10] nearly obtained a match in performance between R134a and R1234yf with a 40% effectiveness IHX at the 30 °C ambient temperature in a vending machine (refrigeration plug-in) system. Finally, Aprea et al. [26] used a domestic refrigerator equipped with a capillary tube–suction line heat exchanger. After 24 h of operation, R1234yf can provide a 3% energy saving compared to R134a. This benefit is greater for a binary mixture of R134a and R1234yf (10/90% weight).

From previous studies, it can be extracted that IHX produces benefits for both R513A components, R134a and R1234yf. Therefore, the study the effects of this additional component using R513A, the recently developed low GWP mixture, becomes necessary. Given the lack of results of an IHX cycle using the lower GWP mixture R513A, this paper presents and discusses the effect of the inclusion of this heat exchanger in a fully monitored medium capacity refrigeration system at different evaporating and condensing conditions. To have a proper contrast of the cycle with and without the IHX, both have been designed to present a heat exchange effectiveness considered relatively high, around 80%. The effect of the IHX in the main operating and energetic parameters of the vapor compression cycle has been analyzed and results using both refrigerants have been compared.

2. Experimental setup

The vapor compression experimental setup is composed of the main refrigeration circuit, and two closed loop secondary circuits connected to the condenser and the evaporator. Figure 1 represents the schematic diagram of the experimental setup, containing the main components, the sensors, and the configuration.

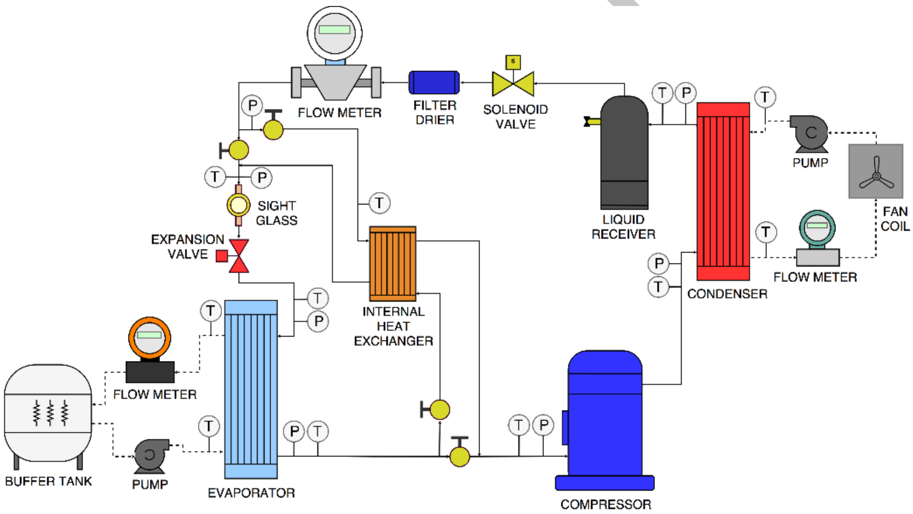
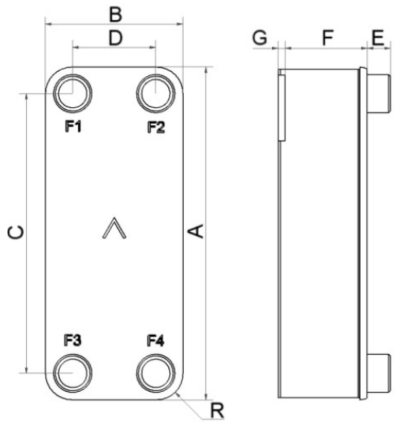


Figure 1. Experimental setup schematic diagram

The compressor is based on scroll technology with a suction volume of 114.5 cm^3 and uses 3.25 dm^3 of 32 cP viscosity POE oil as a lubricant. The three heat exchangers of the system are brazed plate types, having the condenser, evaporator, and IHX, a total of 40, 24 and 30 plates (with heat exchange areas of 2.39, 1.39, and 0.336 m^2), respectively. An electronic expansion valve (EXV) is used to cause the pressure drop between the condenser and the evaporator, and to control the evaporator superheating. The liquid recipient is placed immediately after the condenser and has a capacity of 7.1 dm^3. The IHX, which main characteristics are shown in Figure 2, can be activated using manual ball valves.



- **Dimensions (mm)**: A=193, B=76, C=154, D=40, F=71.2, G=6.2, R=18, and E=20.1
- **Channel volume**: 0.024 dm³
- **Number of plates**: 30
- **Max flow**: 4 m³ h^{-1}
- **Material**: 316 stainless steel plates, copper brazing

Figure 2. Internal heat exchanger main characteristics

The evaporator secondary circuit uses a commercial propylene glycol based secondary refrigerant with a freezing temperature of -37 °C as working fluid. It is heated by a set of resistances with a maximum power of 18 kW, regulated using a PID controller. The condenser secondary circuit uses water as secondary fluid and is cooled using a fan-coil controlled by a frequency inverter. The pumps installed in both secondary circuits have a frequency inverter to maintain the secondary fluid temperature difference between 4 K and 6 K.

The main sensors used to measure the operation of the refrigeration system are shown in Table 1 and their location is seen in Figure 1.

Table 1. Summary of sensors and their uncertainty associated

Measured parameter	**Sensor**	**Uncertainty**
Refrigerant and secondary fluids temperature	K-type thermocouple	± 0.3 K
Refrigerant pressure	Piezoelectric pressure transducer	± 0.15%, reading
Refrigerant mass flow rate	Coriolis mass flow meter	±0.1%, reading
Water volumetric flow rate	Electromagnetic flow meter	± 0.33%, reading
Commercial brine volumetric flow rate	Electromagnetic flow meter	± 0.114 $m^3 h^{-1}$
Power consumption	Digital wattmeter	±1.55%, reading
Compressor rotational speed	Frequency inverter	± 60 rpm

3. Operating conditions

The experimental tests performed to conclude about the feasibility of IHX with high effectiveness in an HFO/HFC mixture like R513A cover different evaporating and condensing conditions. Therefore, the targeted evaporating temperatures are -15 °C, -10 °C and -5 °C, and condensing temperatures of 32.5 °C and 40 °C. The combination of these conditions was performed with and without the IHX. Additional tests were performed at 40 °C and intermediate IHX effectiveness (regulating the by-pass through the opening of the valve located in the inlet IHX pipe).

The targeted evaporating superheating degree (SHD) was set at 11 K in the EXV, and the average values obtained for R134a and R513A were 10.6 K and 12.1 K, respectively. The condenser subcooling degree (SCD) was minimum due to the presence of the liquid receiver, and hence, the average value obtained was 2.4 K for R134a and 1.4 K for R513A.

Each steady-state test has been recorded for a minimum of 20 min and the period of 5 min with most stable parameters has been selected to calculate the average values of the operating condition. The sampling period was 2 seconds and therefore, the minimum number of measurements for each experimental test is 150. Thermodynamic states of refrigerants are calculated using REFPROP software [21].

4. Results and discussion

The experimental results for the main parameters measured and calculated are shown and discussed in this section. The influence of the condensing temperature and the IHX effectiveness are considered in different figures for each parameter. Equations can be consulted in a previous work [19] and Appendix.

4.1. Total superheating and subcooling degree

As it has been introduced in section 1, the activation of the IHX produces a heat exchange between the liquid and the suction lines that increases total SHD and SCD. This affects the compressor operation and the refrigerant vapor quality at the evaporator inlet. The experimental results for SHD and SCD are shown in Table 2 to complement the further analysis done in this section. Additionally, the resulting operating condensation and evaporation temperatures are also provided.

Table 2. IHX effectiveness and total superheating and subcooling degree measured.

Refrigerant	T_k (°C)	T_o (°C)	ε_{IHX}	$\dot{Q}_{IHX,ave}$ (kW)	SHD_{total} (K)	SCD_{total} (K)
R134a	32.4	-4.5	78.8 (±1.6)%	0.624	29.6	14.4
R134a	32.7	-9.8	80.2 (±1.2)%	0.615	34.7	17.9
R134a	32.6	-14.6	81.8 (±1.1)%	0.549	39.0	20.8
R134a	39.7	-4.7	79.6 (±1.2)%	0.800	35.7	18.1
R134a	40.1	-9.9	81.1 (±1.0)%	0.733	40.7	21.8
R134a	40.0	-15.0	[a]	[a]	[a]	[a]
R134a	39.8	-4.8	39.7 (±1.1)%	0.408	23.1	10.1
R134a	40.2	-9.8	39.6 (±1.0)%	0.362	25.4	11.5
R134a	40.1	-14.8	40.6 (±0.9)%	0.302	27.3	13.6
R513A	32.5	-5.1	76.7 (±1.5)%	0.779	31.2	13.1
R513A	32.6	-10.1	78.2 (±1.3)%	0.761	35.8	16.0
R513A	32.5	-15.1	79.7 (±1.1)%	0.684	40.0	19.1
R513A	40.1	-5.1	77.3 (±1.2)%	0.983	36.9	17.2
R513A	40.0	-10.0	78.8 (±1.0)%	0.901	41.4	19.9
R513A	39.8	-15.0	80.3 (±0.9)%	0.784	45.9	22.9
R513A	40.0	-4.9	41.0 (±1.1)%	0.543	24.9	9.3
R513A	39.9	-9.9	40.2 (±1)%	0.474	26.5	10.6
R513A	39.8	-14.9	39.2 (±0.9)%	0.400	28.3	11.5

[a] Test not performed due to discharge temperature limitations. According to compressor manufacturer, at T_k=40 °C and T_o=-15 °C, the total SHD must be limited to 30 °C.

The maximum IHX effectiveness is obtained when the valve in the IHX line is totally opened. Therefore, it can give an indication of the resulting IHX effectiveness in existing installations when R134a is replaced by R513A. Besides, the controlled intermedium IHX effectiveness helps to determine the benefits for the same targeted value of IHX effectiveness in new design installations. Slightly higher values for maximum IHX effectiveness are obtained for R134a and vary between 78.8% and 81.1% for R134a, whereas for R513A result between 76.7% and 80.3%. The maximum IHX effectiveness is increased at higher compression ratios (and hence, the difference is more significant between both operating temperatures). Furthermore, the intermediate IHX effectiveness has been controlled between 39.6% and 40.5% for R134a, and between 39.2% and 40.9% for R513A, and hence, the proposed standard operating IHX condition is comparable.

Total superheating and subcooling degrees of R513A are 1.2 K higher and 1.4 K lower than R134a. Thus, considering the average evaporator superheating and condenser subcooling degrees mentioned in Section 3, it can be concluded that the temperature variation of refrigerant vapor and liquid in the IHX are very similar between both fluids.

4.2. Discharge temperature

Apart from other effects, this parameter includes the heat exchanged in the IHX and results notably increased with its activation. Figure 3 shows the direct discharge temperature measurements in that line at different conditions.

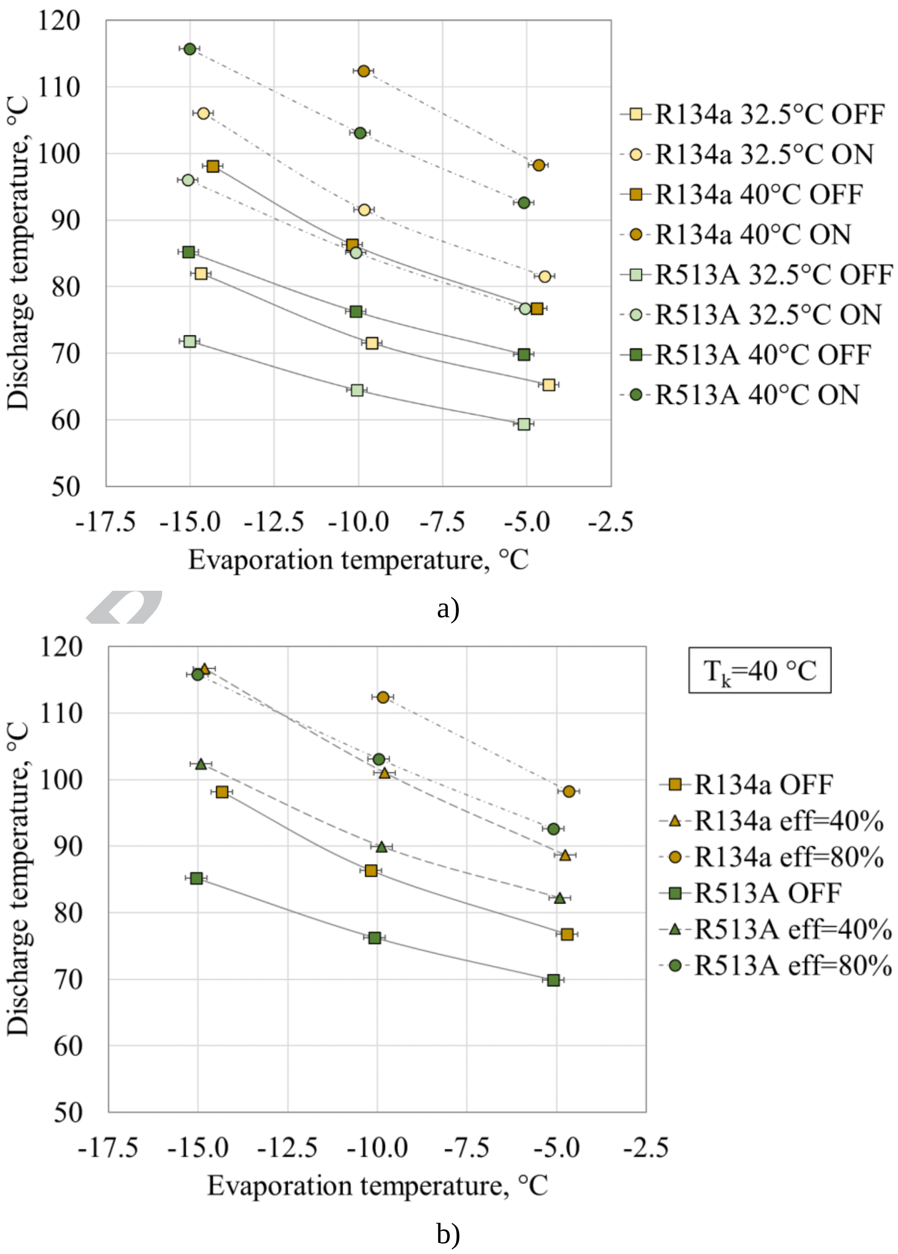


Figure 3. Discharge temperature versus evaporation temperature at different a) condensation temperatures, and b) IHX effectiveness

The discharge temperatures of R134a are higher than those of R513A in any case, between 5.9 K and 10.2 K without IHX and between 4.9 K and 10.1 K with 80% IHX effectiveness. Moreover, this reduction when using R513A allows the utilization of IHXs at relatively high compression ratios. Only at 40% IHX effectiveness, R134a discharge temperatures become similar to that of R513A with the IHX fully activated. Hence, a lower discharge temperature of R513A increments the operating envelope of the refrigeration system with and without IHX. Given the higher heat exchanged in the IHX for R513A (Table 2), the IHX has a slightly stronger influence on the discharge temperature for R513A, being the increase between 17.3 K and 26.8 K (T_o=-15 °C and T_k=42.5 °C not considered), and between 16.3 K and 26 K for R134a. Additionally, the trend in the discharge temperature results shows that if the T_o=-15 °C and T_k=42.5 °C condition was performed, a discharge temperature near 130 °C would be measured.

4.3. Compressor efficiencies

The additional superheating and pressure drop at the suction line influence the scroll-technology compressor efficiencies and hence affect the rest of the parameters here analyzed. Table 3 summarizes the compressor efficiency values (calculated using suction and discharge lines temperature) together with the pressure ratios measured. Volumetric efficiency is higher for R513A (as seen in rotary compressors [16]), but its increment is only visible at maximum IHX effectiveness. Additionally, the indirect calculation of compressor global efficiency indicates that this parameter is around 3% higher for R513A than R134a, being quite similar the compressor global efficiency values with and without IHX.

Table 3. Compressor efficiencies using line measurements

Refrigerant	**IHX**	**T_o (°C)**	**T_k (°C)**	**PR** (±0.01)	η_{vol}	η_{glo}
R134a	OFF	-4.4	32.7	3.25	0.884 (±0.027)	0.65
R134a	ON	-4.5	32.4	3.25	0.888 (±0.026)	0.66
R134a	OFF	-9.6	32.5	3.95	0.842 (±0.024)	0.59
R134a	ON	-9.8	32.7	4.01	0.853 (±0.025)	0.60
R134a	OFF	-14.7	32.5	4.87	0.770 (±0.022)	0.50
R134a	ON	-14.6	32.6	4.87	0.776 (±0.022)	0.51
R134a	OFF	-4.7	40.1	4.04	0.884 (±0.026)	0.63
R134a	ON	-4.7	39.7	4.00	0.823 (±0.024)	0.65
R134a	OFF	-10.2	40.1	4.99	0.824 (±0.024)	0.54
R134a	ON	-9.9	40.1	4.94	0.721 (±0.021)	0.56
R134a	OFF	-14.3	40.1	5.91	0.741 (±0.021)	0.45
R134a	ON	-15.0	40.0		[a]	[a]
R134a	ε_{IHX}=40%	-4.8	39.8	4.02	0.893 (±0.027)	0.63
R134a	ε_{IHX}=40%	-9.8	40.2	4.94	0.830 (±0.027)	0.55
R134a	ε_{IHX}=40%	-14.8	40.1	6.02	0.879 (±0.025)	0.44
R513A	OFF	-5.1	32.6	3.18	0.920 (±0.026)	0.67
R513A	ON	-5.1	32.5	3.18	0.922 (±0.026)	0.68
R513A	OFF	-10.1	32.6	3.81	0.887 (±0.026)	0.62
R513A	ON	-10.1	32.6	3.84	0.895 (±0.026)	0.64
R513A	OFF	-15.0	32.6	4.61	0.828 (±0.024)	0.55
R513A	ON	-15.1	32.5	4.61	0.839 (±0.024)	0.57
R513A	OFF	-5.1	40.1	3.88	0.909 (±0.026)	0.64
R513A	ON	-5.1	40.1	3.89	0.915 (±0.026)	0.66
R513A	OFF	-10.1	40.1	4.66	0.861 (±0.025)	0.58
R513A	ON	-10.0	40.0	4.62	0.871 (±0.025)	0.59
R513A	OFF	-15.1	39.9	5.61	0.802 (±0.023)	0.50
R513A	ON	-15.0	39.8	5.59	0.807 (±0.023)	0.51
R513A	ε_{IHX}=40%	-4.9	40.0	3.85	0.912 (±0.026)	0.66

R513A	ε_{IHX}=40%	-9.9	39.9	4.61	0.861 (±0.025)	0.58
R513A	ε_{IHX}=40%	-14.9	39.8	5.54	0.801 (±0.023)	0.51

[a] Test not performed due to discharge temperature limitations.

4.4. Mass flow rate

One of the direct consequences of the activation of the IHX is the decrease of the mass flow rate. Higher suction temperature and the additional pressure drop caused by this component decrease the suction density to a greater extent than the slight increase in volumetric efficiency. Hence, as the volumetric efficiency variation remains within ±1%, and the swept volume and compressor rotation speed is the same for all the tests, it can be said that in this case the mass flow rate variation directly reflects what happened for the suction density. This effect can be observed in Figure 4.

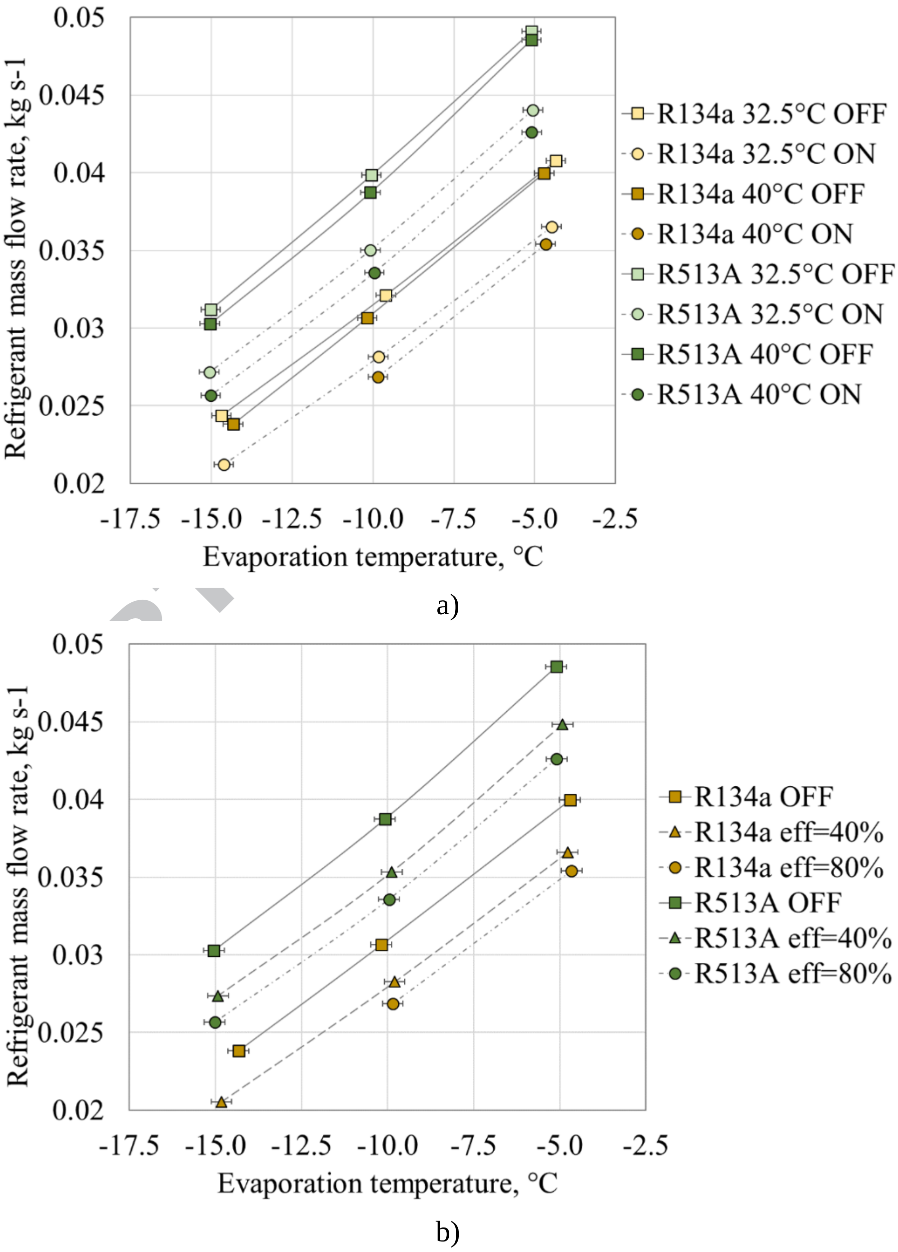
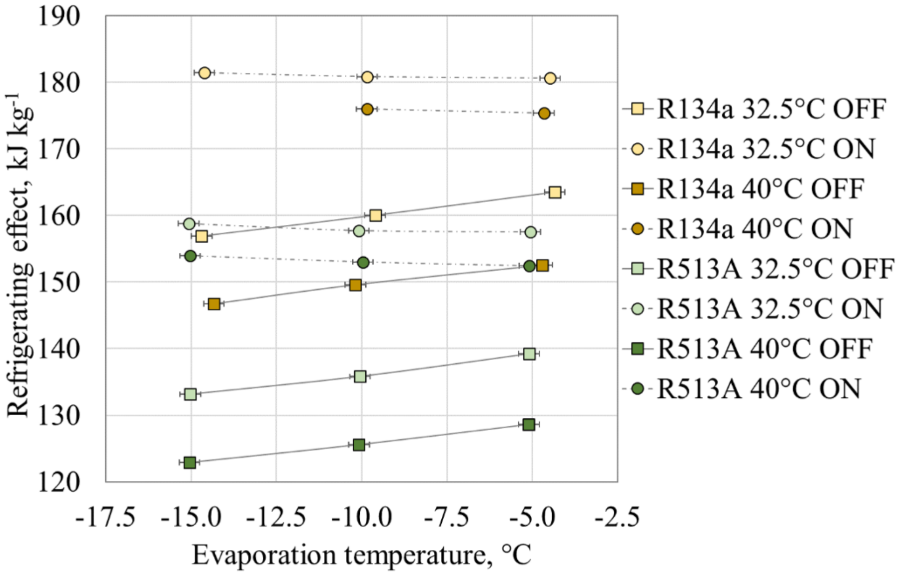


Figure 4. Refrigerant mass flow rate versus evaporation temperature at different a) condensation temperatures, and b) IHX effectiveness

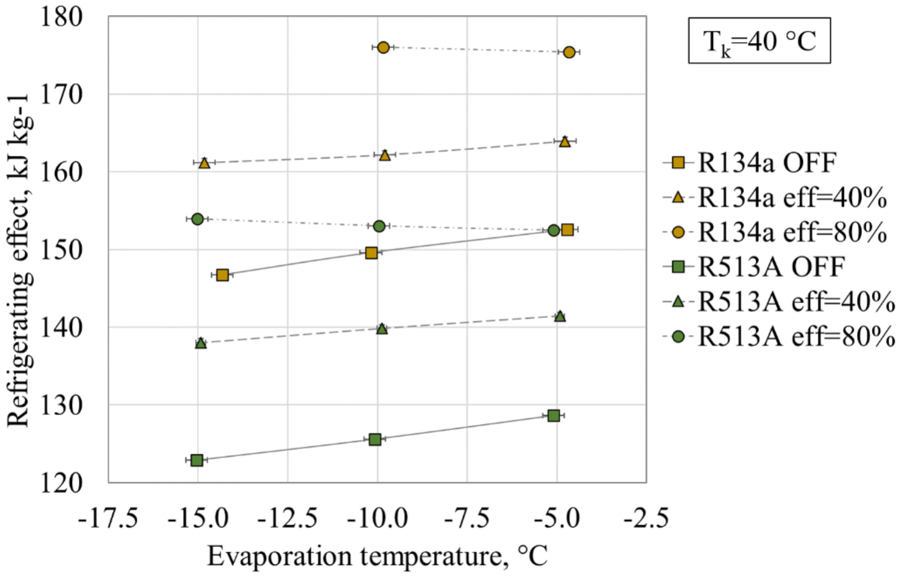
R513A mass flow rate is noticeably higher than that of R134a, between 20.4% and 28%. The decrease in mass flow rate caused by the IHX is quite similar between both fluids, and it varies between 10% and 15%. Under comparable conditions, the intermediate IHX position leads mass flow rate closer to the total IHX opening than the basic cycle configuration.

4.5. Cooling capacity

Before studying the cooling capacity results, the refrigerating effect (enthalpy difference at the evaporator) is shown in Figure 5. The refrigerating effect enhances the cooling capacity by the increase of total SCD, and hence, it reduces the enthalpy at the inlet of the evaporator. It should be highlighted that in this case, isenthalpic expansion is considered, and thus the enthalpy at the inlet of the expansion device is taken. The R513A refrigerating effect is between 13% and 16% lower than that of R134a, corresponding the lower reductions when the IHX is activated. Besides, it is also observed that when IHX is fully activated, this parameter is not affected by the evaporating temperature. The augment for R134a is between 10% and 18%, and that of R513A results between 13% and 22%. The more significant increase of refrigerating effect agrees with the research of Pottker and Hrnjak [27], which states that subcooling produces higher benefit in refrigerants with large liquid specific heat and smaller latent heat of vaporization. For the same operating temperatures, R513A has similar liquid specific heat but 11.3% smaller latent heat of vaporization.



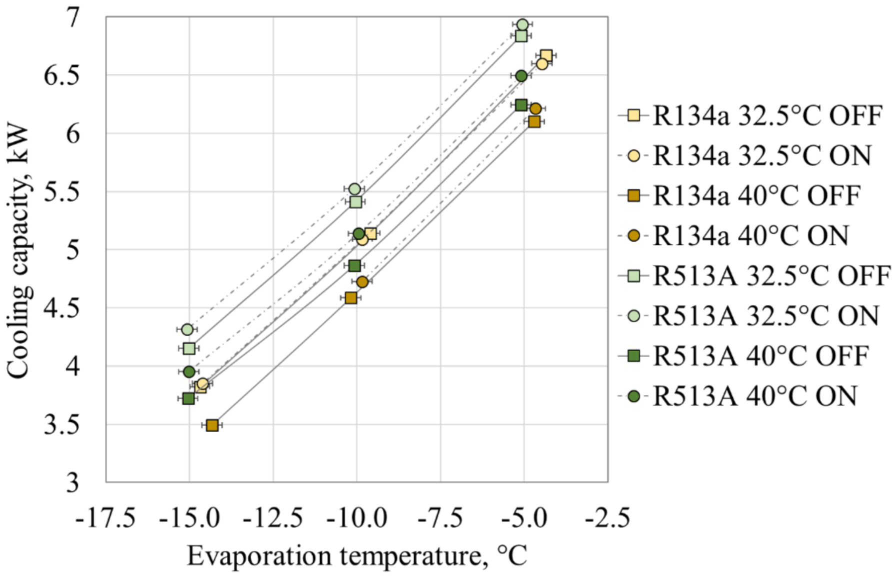
a)



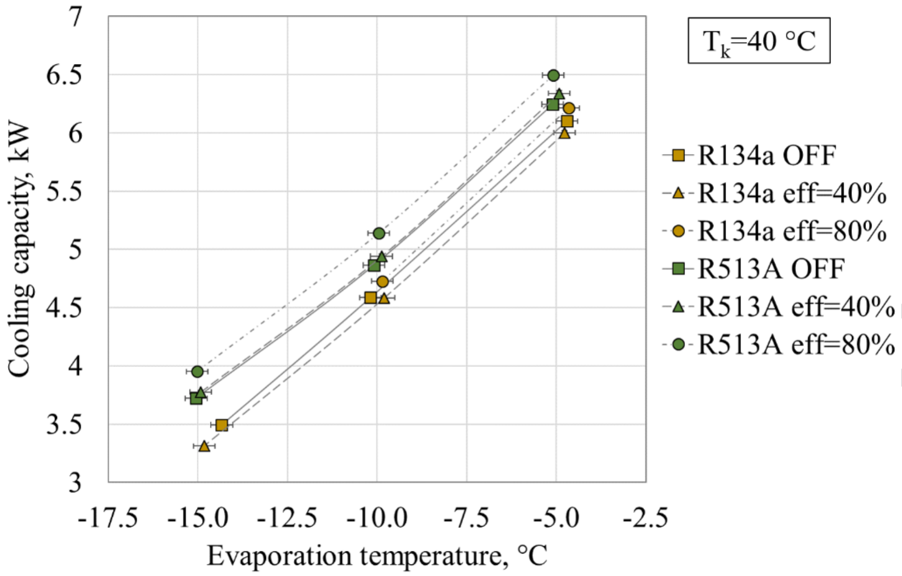
b)

Figure 5. Refrigerating effect versus evaporation temperature at different a) condensation temperatures, and b) IHX effectiveness

While the IHX decreases mass flow rate between 10% and 15%, the refrigerating effect is increased in the range of 13% and 22% for R513A. Therefore, as the cooling capacity is a product of both parameters, an improvement of the cooling capacity is expected in any case tested for the new alternative. Figure 6 represents experimental cooling capacity results.



a)



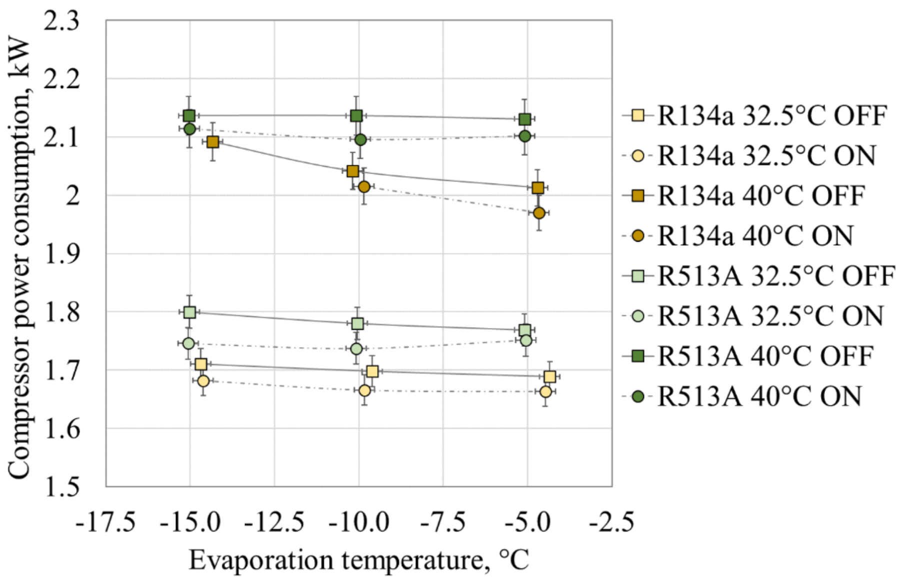
b)

Figure 6. Cooling capacity versus evaporation temperature at different a) condensing temperatures, and b) IHX effectiveness.

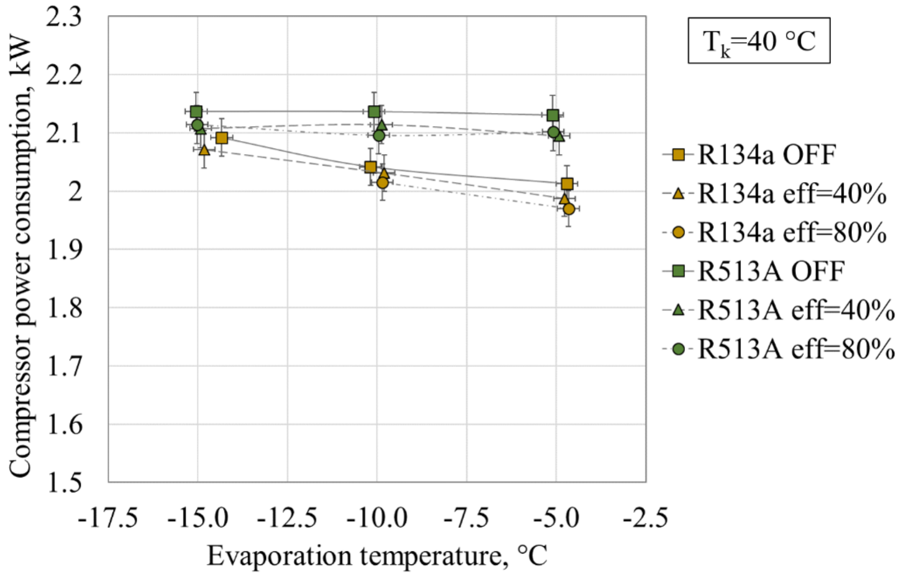
Firstly, Figure 6 reflects the cooling capacity enhancement when R513A is used instead of R134a because of the higher mass flow rate, especially at higher compression ratio conditions. While for R513A the IHX always increases the cooling capacity, between 1.5% and 5.6%, for R134a, it is only observed at higher compression ratio conditions (in this case, the increment is around 3%). Different IHX effectiveness does not vary the result of cooling capacity for R134a whereas R513A has a benefit from the maximum IHX effectiveness.

4.6. Compressor power consumption

To predict the IHX influence on compressor power consumption, two effects of opposite sign must be considered. Although the refrigerant mass flow rate is reduced, the specific work of compression results greater by the higher superheating degree at the compressor suction. Previous works [28] suggested that both effects are of a similar magnitude and hence, the variation in power consumption results imperceptible. Figure 7 contains the direct measurements of the compressor power consumption.



a)



b)

Figure 7. Compressor power consumption versus evaporation temperature at different a) condensation temperatures, and b) IHX effectiveness

Here, a minimum reduction of compressor power consumption is observed with any IHX effectiveness in all the conditions tested, since, as average, this parameter only diminishes 1.4%. In any event, the reduction is more noticeable for both refrigerants at higher compression ratios. Besides, the R513A average compressor power consumption is 4.7% higher than that of R134a. Compared to R134a, a 12% (on average) lower higher specific compression work of the new mixture is measured, caused by the R513A lower compression ratio and the higher isentropic slope. Hence, the increase in compressor power consumption is caused by the mass flow rate parameter. Additionally, as for R513A there is barely not influence of evaporation temperature, the R513A compressor power consumption difference with R134a increases at higher evaporation conditions. The minimum reduction of power consumption will favor the final COP increase using IHX.

4.7. Coefficient of Performance (COP)

In the studies included in the introduction (Section 1) of this paper, the energy performance of the R134a and R12334yf systems has been benefited by the IHX adoption. Figure 8 contains the COP experimental results obtained in this study.

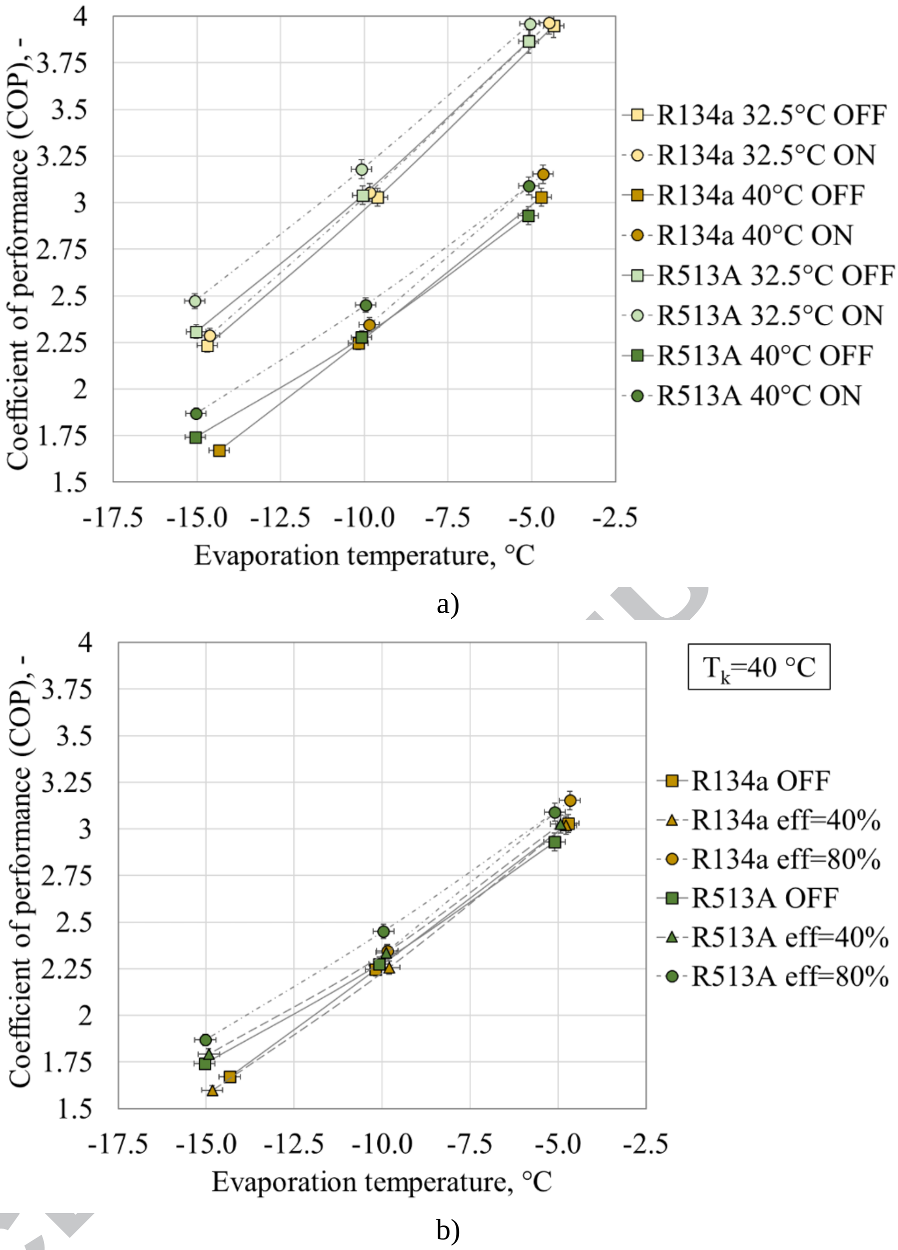


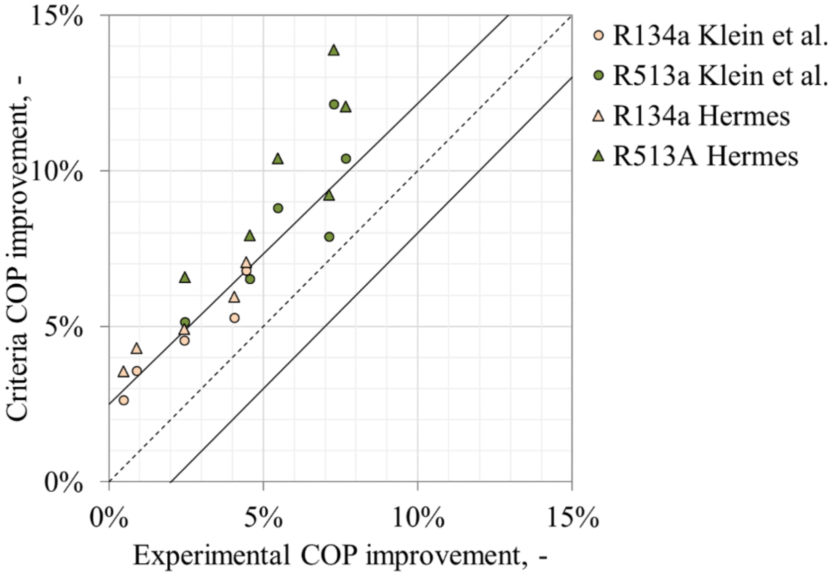
Figure 8. COP versus evaporation temperature at different a) condensation temperatures, and b) IHX effectiveness

IHX has an identifiable benefit on the energy performance of the system, quantified using the COP parameter. This increase is relevant at higher compression ratios (lower evaporating and higher condensation temperatures). Furthermore, the COP growth is higher using the lower GWP mixture R513A, between 2% and 8%, in contrast with R134a, which is only up to 4%. As a result, R513A presents better COP results at higher compression ratios and when the IHX is used.

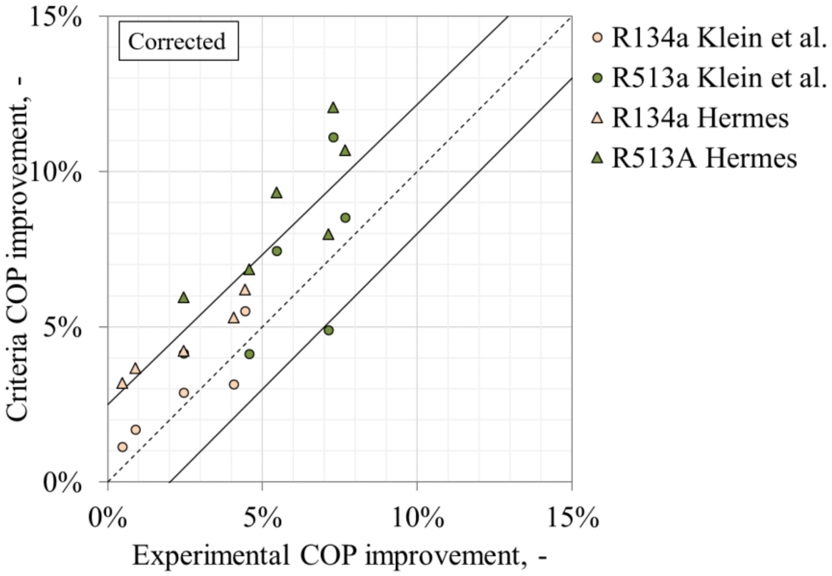
5. Criteria to predict COP improvement

Several authors have developed criteria to predict the COP improvement by the utilization of IHX. The Domanski et al.'s criterion [29] assumed isentropic compression, no-pressure drop at infinite heat exchangers and IHX. The criterion developed by Aprea et al. [30] supposed adiabatic devices, negligible pressure drops in the heat exchangers and the same compression isentropic efficiency in both configurations (with and without IHX). For the same purpose, Ziegler [31] used an adaptation of the dimensionless Stefan-number (considered absolute temperature instead of temperature difference), assuming no pressure drops and same pressure ratio. Despite those simplifications, these methods predict the positive sign of COP improvement in any condition, as seen in this experimental study. Additionally, Mastrullo et al. [32] suggested that higher the ratio of specific heat vapor molar at constant pressure and critical temperature is, higher the COP improvement. On average, this ratio is 4.4% higher for R513A than R134a.

On the other hand, the Klein et al.'s criterion [28] supposes that the IHX adoption has no influence on the compressor power consumption and provides a quantitative approximation to the COP improvement. Alternatively, Hermes [33] assumed constrained operating pressures, and isentropic compression, and obtained a similar expression to Domanski et al. [29]. Figure 9 shows the comparison between these methods and the experimental results presented.



a)



b)

Figure 9. Comparison of experimental COP and Klein et al.'s [28] and Hermes' [33] criteria a) original, and b) corrected.

In general, both criteria overpredicts the COP improvement. Firstly, Klein et al.'s criterion supposes that power consumption has no influence. In the experimental study, it has been seen that the power consumption is approximately decreased 2% and if this reduction is applied to the Klein's criterion, both values match within ±2.5%. Secondly, Hermes' criterion obtains closer values for R134a than R513A, and when the isentropic efficiency variation in experimental is applied to the Hermes' criterion results, the deviation is reduced as well.

6. Conclusions

R134a consumption is increasing and its emissions are very relevant compared to other common HFC gases. To provide a possible short-term replacement of R134a, R513A is proposed in refrigeration systems with and without IHX. This paper has presented and analyzed the first experimental values of the IHX influence on a vapor compression refrigeration system using R513A and R134a. Selected evaporation temperatures were -15 °C, -10 °C and -5 °C, and condensation temperatures, 32.5 °C and 40°C; and rated IHX effectiveness is 80%.

The total superheating degree was higher using R513A than R134a, whereas the contrary was observed for total subcooling degree values. Because of the total superheating degree variation measured, the R513A discharge temperature had higher increase, but these values were still is well below that of R134a. However, compressor efficiencies and refrigerant mass flow rate vary in a similar proportion for both refrigerants.

Due to the R513A smaller latent heat of vaporization, the refrigerant effect augmentation caused by the IHX is higher for this refrigerant, and thus produces a higher benefit for its cooling capacity. Power consumption is slightly reduced, and the resulting coefficient of performance is augmented when the IHX is used. COP gain is benefited from the maximum achievable IHX effectiveness, higher compression ratios, and R513A utilization. Additionally, results with intermediate effectiveness (40%) produced a lower benefit but allow to reach higher operating condensing conditions. Besides, for these experimental results, Klein et al.'s and Hermes' COP variation criteria overpredict the COP improvement because of the assumptions made.

Attending to the energy performance improvement achieved using R513A in a refrigeration system equipped with a high effectiveness IHX, and the reduction in its GWP value, it can be concluded that the combination of IHX configuration and this working fluid can lead to an environmental benefit if compared with a basic cycle configuration using the traditional R134a, especially at high pressure ratio.

Acknowledgements

Adrián Mota-Babiloni would like to acknowledge the funding received from the Spanish State Research Agency through the "Juan de la Cierva - Formación 2016" postdoctoral grant (FJCI-2016-28324) and from the Universitat Jaume I though its Research Stay Grants (E-2017-30).

References

[1] Roelfsema M, Harmsen M, Olivier JJG, Hof AF, Van Vuuren DP. Integrated assessment of international climate mitigation commitments outside the UNFCCC. Glob Environ Chang 2018;48:67–75. doi:10.1016/j.gloenvcha.2017.11.001.

[2] Graziosi F, Arduini J, Furlani F, Giostra U, Cristofanelli P, Fang X, et al. European emissions of the powerful greenhouse gases hydrofluorocarbons inferred from atmospheric measurements and their comparison with annual national reports to UNFCCC. Atmos Environ 2017;158:85–97. doi:10.1016/J.ATMOSENV.2017.03.029.

[3] Su S, Fang X, Li L, Wu J, Zhang J, Xu W, et al. HFC-134a emissions from mobile air conditioning in China from 1995 to 2030. Atmos Environ 2015;102:122–9. doi:10.1016/J.ATMOSENV.2014.11.057.

[4] Simmonds PG, Derwent RG, Manning AJ, McCulloch A, O'Doherty S. USA emissions

estimates of CH3CHF2, CH2FCF3, CH3CF3 and CH2F2 based on in situ observations at Mace Head. Atmos Environ 2015;104:27–38. doi:10.1016/J.ATMOSENV.2015.01.010.

[5] Maranion B, Pizano M, Woodcock A, Besri M, Carvalho S, Cathpole D, et al. 2016 TEAP Report. Volume I. Decision XXVII/4 Task Force Update Report Further Information on Alternatives to Ozone-Depleting Substances. Nairobi, Kenya: 2016.

[6] Höglund-Isaksson L, Purohit P, Amann M, Bertok I, Rafaj P, Schöpp W, et al. Cost estimates of the Kigali Amendment to phase-down hydrofluorocarbons. Environ Sci Policy 2017;75:138–47. doi:10.1016/J.ENVSCI.2017.05.006.

[7] Mota-Babiloni A, Makhnatch P, Khodabandeh R. Recent investigations in HFCs substitution with lower GWP synthetic alternatives: Focus on energetic performance and environmental impact. Int J Refrig 2017;82. doi:10.1016/j.ijrefrig.2017.06.026.

[8] Mota-Babiloni A, Navarro-Esbrí J, Mendoza-Miranda JM, Peris B. Experimental evaluation of system modifications to increase R1234ze(E) cooling capacity. Appl Therm Eng 2017;111. doi:10.1016/j.applthermaleng.2016.09.175.

[9] Zilio C, Brown JS, Schiochet G, Cavallini A. The refrigerant R1234yf in air conditioning systems. Energy 2011;36:6110–20. doi:10.1016/J.ENERGY.2011.08.002.

[10] Sethi A, Vera Becerra E, Yana Motta S. Low GWP R134a replacements for small refrigeration (plug-in) applications. Int J Refrig 2016;66:64–72. doi:10.1016/j.ijrefrig.2016.02.005.

[11] Mendoza-Miranda JM, Mota-Babiloni A, Ramírez-Minguela JJ, Muñoz-Carpio VD, Carrera-Rodríguez M, Navarro-Esbrí J, et al. Comparative evaluation of R1234yf, R1234ze(E) and R450A as alternatives to R134a in a variable speed reciprocating compressor. Energy 2016;114. doi:10.1016/j.energy.2016.08.050.

[12] Deveciо̆lu AG, Oruç V. Characteristics of Some New Generation Refrigerants with Low GWP. Energy Procedia, vol. 75, 2015, p. 1452–7. doi:10.1016/j.egypro.2015.07.258.

[13] Majurin J, Staats SJ, Sorenson E, Gilles W. Material compatibility of HVAC&R system materials with low global warming potential refrigerants. Sci Technol Built Environ 2015;21:491–501. doi:10.1080/23744731.2015.1009353.

[14] Meng Z, Zhang H, Lei M, Qin Y, Qiu J. Performance of low GWP R1234yf/R134a mixture as a replacement for R134a in automotive air conditioning systems. Int J Heat Mass Transf 2018;116:362–70. doi:10.1016/j.ijheatmasstransfer.2017.09.049.

[15] Makhnatch P, Mota-Babiloni A, Khodabandeh R. Experimental study of R450A drop-in performance in an R134a small capacity refrigeration unit. Int J Refrig 2017;84. doi:10.1016/j.ijrefrig.2017.08.010.

[16] Mota-Babiloni A, Makhnatch P, Khodabandeh R, Navarro-Esbrí J. Experimental assessment of R134a and its lower GWP alternative R513A. Int J Refrig 2017;74. doi:10.1016/j.ijrefrig.2016.11.021.

[17] Llopis R, Sánchez D, Cabello R, Catalán-Gil J, Nebot-Andrés L. Experimental analysis of R-450A and R-513A as replacements of R-134a and R-507A in a medium temperature commercial refrigeration system. Int J Refrig 2017;84:52–66. doi:10.1016/J.IJREFRIG.2017.08.022.

[18] Schultz K, Kujak S, Majurin J. Assessment of next generation refrigerant r513a to replace r134a for chiller products. Proc. 24th Int. Congr. Refrig., 2015. doi:10.18462/iir.icr.2015.0075.

[19] Mota-Babiloni A, Navarro-Esbrí J, Barragán-Cervera A, Molés F, Peris B. Drop-in analysis of an internal heat exchanger in a vapour compression system using R1234ze(E) and R450A as alternatives for R134a. Energy 2015;90. doi:10.1016/j.energy.2015.06.133.

[20] McLinden MO, Brown JS, Brignoli R, Kazakov AF, Domanski PA. Limited options for low-global-warming-potential refrigerants. Nat Commun 2017;8. doi:10.1038/ncomms14476.

[21] Lemmon EW, Huber ML, McLinden MO. NIST Standard Reference Database 23. Ref Fluid Thermodyn Transp Prop (REFPROP), Version 91 2013.

[22] Pottker G, Hrnjak P. Experimental investigation of the effect of condenser subcooling in R134a and R1234yf air-conditioning systems with and without internal heat exchanger. Int J Refrig 2015;50:104–13. doi:10.1016/j.ijrefrig.2014.10.023.

[23] Cho H, Lee H, Park C. Performance characteristics of an automobile air conditioning system with internal heat exchanger using refrigerant R1234yf. Appl Therm Eng 2013;61:563–9. doi:10.1016/j.applthermaleng.2013.08.030.

[24] Direk M, Kelesoglu A, Akin A. Drop-in performance analysis and effect of ihx for an automotive air conditioning system with R1234yf as a replacement of R134a. Stroj Vestnik/Journal Mech Eng 2017;63:314–9. doi:10.5545/sv-jme.2016.4247.

[25] Navarro-Esbrí J, Molés F, Barragán-Cervera Á. Experimental analysis of the internal heat exchanger influence on a vapour compression system performance working with R1234yf as a drop-in replacement for R134a. Appl Therm Eng 2013;59:153–61. doi:10.1016/j.applthermaleng.2013.05.028.

[26] Aprea C, Greco A, Maiorino A. An experimental investigation of the energetic performances of HFO1234yf and its binary mixtures with HFC134a in a household refrigerator. Int J Refrig 2017;76:109–17. doi:10.1016/j.ijrefrig.2017.02.005.

[27] Pottker G, Hrnjak P. Effect of the condenser subcooling on the performance of vapor compression systems. Int J Refrig 2015;50:156–64. doi:10.1016/j.ijrefrig.2014.11.003.

[28] Klein SA, Reindl DT, Brownell K. Refrigeration system performance using liquid-suction heat exchangers. Int J Refrig 2000;23:588–96. doi:10.1016/S0140-7007(00)00008-6.

[29] Domanski PA, Didion DA, Doyle JP. Evaluation of suction-line/liquid-line heat exchange in the refrigeration cycle. Int J Refrig 1994;17:487–93. doi:10.1016/0140-7007(94)90010-8.

[30] Aprea C, Ascani M, de Rossi F. A criterion for predicting the possible advantage of adopting a suction/liquid heat exchanger in refrigerating system. Appl Therm Eng 1999;19:329–36. doi:10.1016/S1359-4311(98)00070-2.

[31] Ziegler F. The multiple meanings of the Stefan-number (and relatives) in refrigeration. Int J Refrig 2010;33:1343–9. doi:10.1016/j.ijrefrig.2010.06.015.

[32] Mastrullo R, Mauro AW, Tino S, Vanoli GP. A chart for predicting the possible advantage of adopting a suction/liquid heat exchanger in refrigerating system. Appl Therm Eng 2007;27:2443–8. doi:10.1016/j.applthermaleng.2007.03.001.

[33] Hermes CJL. Alternative evaluation of liquid-to-suction heat exchange in the refrigeration cycle. Int J Refrig 2013;36:2119–27. doi:10.1016/J.IJREFRIG.2013.06.007.

Appendix

To complement the differences between refrigerants discussed along the paper, Table A.1. is also provided. It contains the main properties of R134a and R513A.

Table A.1. Main characteristics of R134a and R513A [21]

Property	**R134a**	**R513A**
Composition	pure R134a	R134a/1234yf 44/56 wt%
ANSI/ASHRAE Standard safety classification	A1	A1
AR5 GWP100-yr	1300	572
Relative molar mass, kg mol-1	102.03	108.43
Critical temperature, °C	101.06	96.50
Critical pressure, MPa	4.06	3.86
Boiling point at 0.1 MPa, °C	-26.36	-29.82
Glide at 0.1 MPa, °C	0.00	0.10
Liquid density [a], kg m^{-3}	1294.8	1221.9
Vapor density [a], kg m^{-3}	14.43	17.23
Liquid isobaric specific heat capacity [a], kJ kg^{-1} K^{-1}	1.34	1.31
Vapor isobaric specific heat capacity [a], kJ kg^{-1} K^{-1}	0.90	0.92
Liquid thermal conductivity [a], mW m^{-1} K^{-1}	92.01	79.21
Vapor thermal conductivity [a], mW m^{-1} K^{-1}	11.51	11.74
Liquid viscosity [a], μPa s	266.53	227.10
Vapor viscosity [a], μPa s	10.73	10.51

[a] At saturation, at 0 °C

Table A.2 contains Equations used in the calculation of parameters shown in this paper in order of appearance.

Table A.2. Equations used for indirect parameter calculations.

Parameter	**Equation**
Condensation temperature (°C)	$T_k = f\left(P = \frac{P_{k,in} + P_{k,out}}{2}, x = 0.5\right)$
Evaporation temperature (°C)	$T_o = f\left(P = \frac{P_{o,in} + P_{o,out}}{2}, x = 0.67\right)$
IHX effectiveness (-)	$\varepsilon_{IHX} = \frac{T_{V,out} - T_{V,in}}{T_{L,out} - T_{V,in}}$
Average IHX heat transferred (kW)	$\dot{Q}_{IHX,ave} = \dot{m}_{ref} \cdot average(\Delta h_V, \Delta h_L)_{IHX}$
Total superheating degree (K)	$SHD = T_{suc} - T_{o,sat\ vapor}$
Total subcooling degree (K)	$SCD = T_{xv,in} - T_{k,sat\ liquid}$
Pressure ratio (-)	$PR = {P_{disc}}/{P_{suc}}$
Volumetric efficiency (-)	$\eta_{vol} = \frac{\dot{m}_{ref}}{N/60 \cdot \rho_{suc} \cdot V_{comp}}$
Global efficiency (-)	$\eta_{glo} = \frac{\dot{m}_{ref} \cdot (h_{disc} - h_{suc})_{iso}}{\dot{P}_{comp}}$
Refrigerating effect (kJ kg^{-1})	$\dot{q}_o = h_{o,out} - h_{o,in}$
Cooling capacity (kW)	$\dot{Q}_0 = \dot{m}_{ref} \cdot \dot{q}_o$

Coefficient of performance, COP (-)	$COP = \dot{Q}_0 / \dot{P}_{comp}$
COP Klein et al.'s criteria [25,28]	$\left(\frac{COP_{IHX}}{COP_{without\ IHX}} - 1\right) = \varepsilon_{IHX}(-3.0468 + 19.3484D - 19.091D^2 + 1.2094L + 0.02101L^2 - 5.9980DL - 0.02797DL^2 + 5.52865D^2L)$ $D = \frac{h_{lv}}{c_{p,L}T_k},\ L = (T_k - T_o)$
COP Hermes' criteria [33]	$\frac{COP_{IHX}}{COP_{without\ IHX}} \approx \frac{1 + \varepsilon_{IHX}C_r\frac{\varphi}{1-\varphi}}{1 + \varepsilon_{IHX}\theta}$ $\varphi = c_{p,L}(T_k - T_o)/h_{lv},\ \theta = \beta(T_k - T_o),\ \beta = \left(\frac{1}{v}\right)\left(\frac{\partial v}{\partial T}\right)_p$

Figure A.1. shows the Ph diagram with the following R134a and R513A information:

- Liquid and vapor saturation lines.
- Basic cycle and basic cycle with IHX at evaporation temperature of -10 °C and condensation temperature of 40 °C, SHD and SCD of 11 and 2 °C, IHX effectiveness of 80%, isentropic efficiency of 0.8, no pressure loss in lines and components and no heat exchanged with the ambient.
- The isentropic lines using the specific entropy for the compressor suction in the case the IHX is activated.

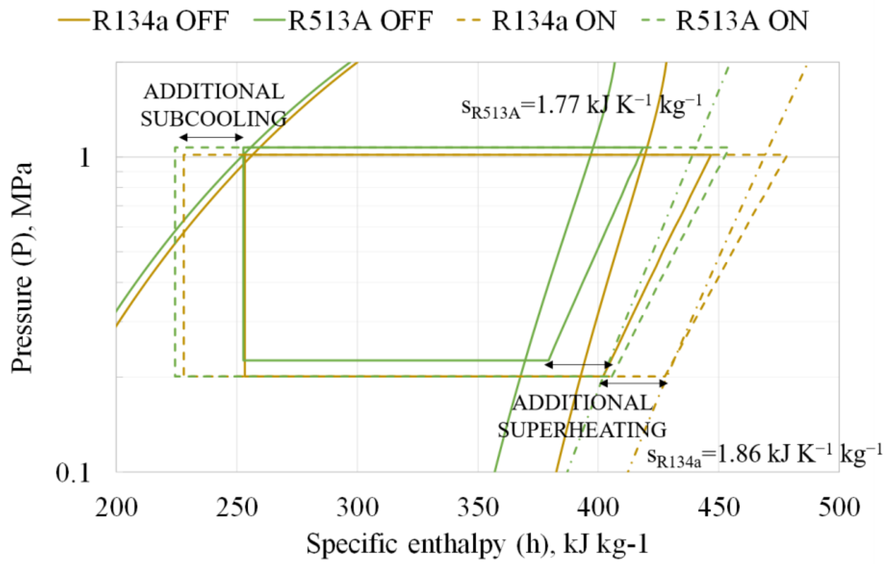


Figure A.1. Ph diagram of R513A and R134a, including cycles with and without IHX.

HIGHLIGHTS

- IHX influence is experimentally compared between R513A and R134a.
- The cooling capacity of the system results increased up to 5.6% for R513A.
- Average power consumption reduction is less than 2% for both fluids.
- COP increases up to 8% for R513A and 4% for R134a.
- High effectiveness IHX is recommended for R513A, especially at high compression ratios.