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

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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 greater 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 of 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

COP	coefficient of performance (-)
T	temperature (°C)
P	pressure (MPa)
Q	heat transfer (kW)
SCD	subcooling degree (K)
SHD	superheating degree (K)
Greek	
ε, eff	heat exchanger effectiveness (-)
η	compressor efficiency (-)
Subscripts	
ave	average
iso	isentropic
glo	global
k	condenser/condensing
o	evaporator/evaporating
vol	volumetric
Abbreviatures	

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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 $CO_{2\text{-eq}}$) in 2030 [3]. Furthermore, in the USA, the estimated emissions for R134a (R^2>0.5) in 2012 was 69.6 (±18.4) Gg [4].

Unless feasible solutions are provided, it seems that the current situation about R134a emission 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 to the Kigali Amendment, the replacement of R134a using low GWP refrigerants must be made also 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 to drop-in or retrofit alternatives. Very low GWP R1234yf and R1234ze(E) hydrofluoro-olefins (HFO) has been proposed in different vapor compression applications to replace R134a with minor system modifications [7]. On the one hand, in direct replacement, R1234ze(E) present 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 R134a (the cooling capacity drop is lesser than 10%), but the Coefficient of Performance (COP) of the system is also decremented using the HFO in MACs [9], vending machines [10], and water cooled reciprocating chillers [11]; and therefore is not acceptable from an energetic point of view.

To overcome the energetic and flammability limitations, mixtures of HFOs and R134a has been proposed [7,12] that are also compatible with most of the materials typically used with the HFCs [13]. Even though low flammable mixtures with GWP limited to 150 has been tested [14,15], today, the most promising HFO/HFC mixtures alternatives to R134a are R450A (R134a/R1234ze 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 [16,17], in a commercial refrigeration system [18]; and then, R450A in a water-cooled refrigeration system [11] and R513A in water-cooled and air-cooled screw chillers [19]. From these experimental results, it is seen as this intermediate solution can provide comparable energy performance to R134a in the different systems studied.

A common possibility to increase the energy performance of the vapor compression systems is the adoption of an internal heat exchanger (IHX), also known as liquid-to-suction heat exchanger.

This additional component has been studied for R450A [20] with positive energetic results, closer to its forming component R1234ze than R134a. However, no experimental results have been published yet about the benefit of IHXs in R513A vapor compression systems.

An adequate way to predict the feasibility of an IHX to increase the energy performance is to study the effect on its forming components. McLinden et al. [21] predicted energy benefits using IHX cycle with R1234yf because of its higher molar heat capacity. According to REFPROP v9.1 software calculations the molar heat capacity of R1234yf at 0.1 MPa is 4 and 14% higher than R134a in liquid and vapor saturated state, respectively [22]. Pottker and Hrnjak [23] 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. [24] improved the COP of a MAC by up to 4.6% by installing the IHX into the R1234yf refrigeration system keeping the discharge temperatures below the R134a system without IHX. In the same application and varying the compressor speed from 1000 to 2000 rpm and at the air stream temperatures of 27 and 35 °C, Direk et a. [25] increased the R1234yf COP between 6.4% and 9.9 % using IHX. In a vapor compression system working with and without a 25% effectiveness IHX, Navarro-Esbrí [26] obtained a maximum 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 [27] used a domestic refrigerator equipped with a capillary tube–suction line heat exchanger. After 24 h of working, R1234yf can obtain a 3% energy saving compared to R134a.

From previous studies, it can be extracted that IHX produces benefits for both R513A components, R134a and R1234yf and thus there is a need of knowledge about the effects of this additional components in the recently developed low GWP mixture. Given the lack of results of 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, it has been designed to present a heat exchange effectiveness quite high, around 0.8. The effect of the IHX in the main operating and energetic parameters of the vapor compression cycle has been analyzed.

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.

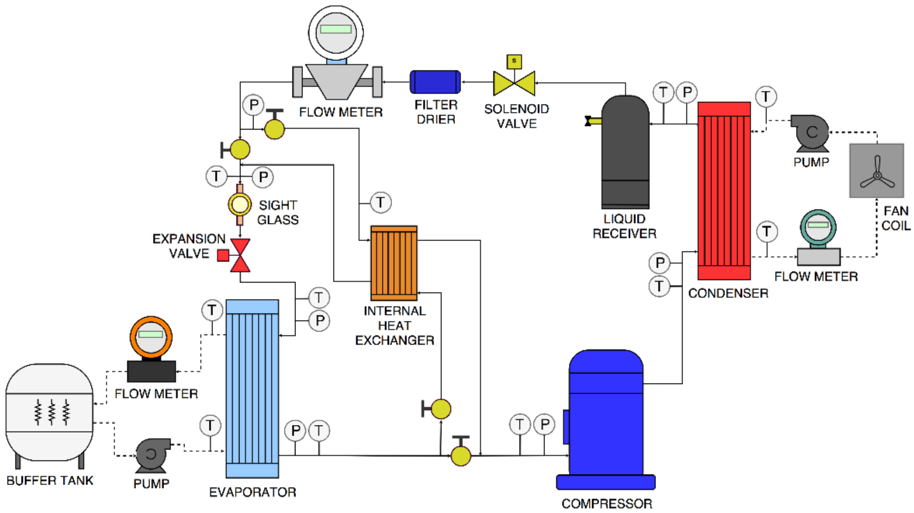
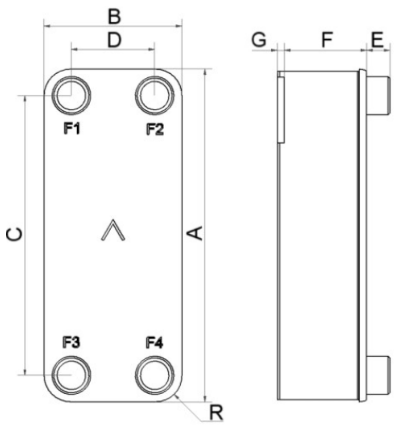


Figure 1. Experimental setup schematic diagram

The compressor is 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 dm^3
- **Number of plates**: 30
- **Max flow**: 4 m^3 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 by means of a PID controller. The condenser secondary circuit uses water as secondary fluid and is cooled using a fan-coil controlled by a frequency inverter. Pumps present in both secondary circuits have a frequency inverter to maintain the secondary fluid temperature difference between 4 and 6 K.

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

Table 1. Summary of sensors and their uncertainty associated

Measured parameter	**Sensor**	**Uncertainty**

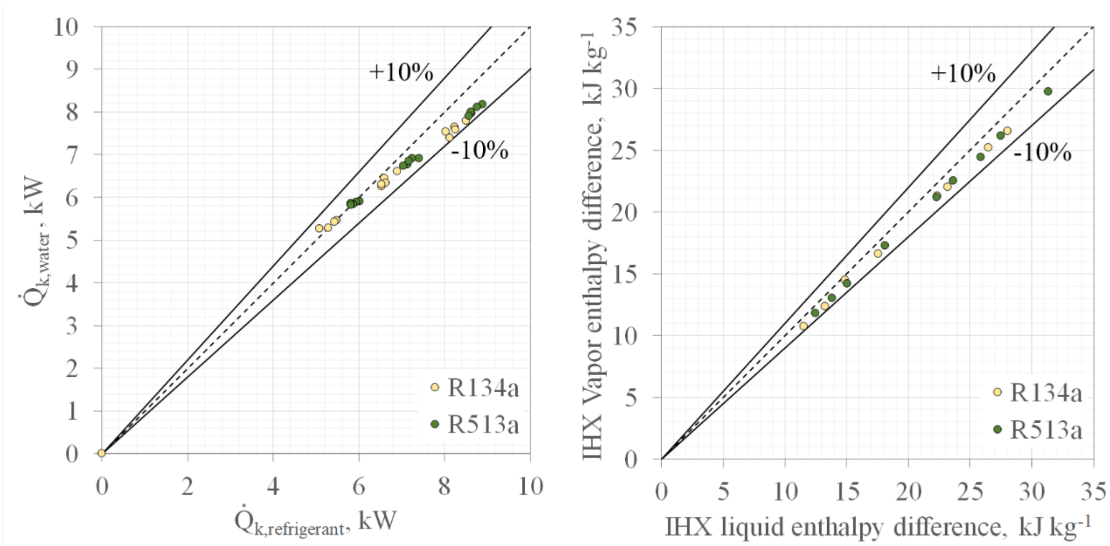
Refrigerant and secondary fluids temperature	K-type thermocouple	± 0.3 K
Refrigerant pressure	Piezoelectric pressure transducer	± 0.15%, reading
Refrigerant mass flow rate	Coriolis mass flow meter	±0.1%, reading
Water volumetric flow rate	Electromagnetic flow meter	± 0.33%, reading
Commercial brine volumetric 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 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, -10 and -5 °C, and condensing temperatures of 32.5 and 40 °C. The combination of this conditions was performed with and without the IHX. Additional tests were performed at 40 °C and intermediate IHX effectiveness (regulating the by-pass throug 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 was 10.6 K and for R513A, 12.1 K. 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 secs and therefore the minimum number of measurements for each experimental test is 150. Thermodynamic states of refrigerant are calculated using REFPROP software [22]. The accuracy of the measurements can be observed in Figure 3, that shows the heat balance in the IHX between the liquid and the vapor. The average heat balance standard deviation in the condenser is 4.2% for both fluids, and in the IHX is 5.5% for R134a and 4.1% for R513A.



a) b)

Figure 3. Heat balance in the a) condenser, and b) IHX

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 is considered in different figures for each parameter. Equations can be consulted in a previous work [20].

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 results in an increase in total superheating and subcooling that affects the compressor operation and the refrigerant quality at the evaporator inlet. The total SHD and SCD achieved by means of the full and partial activation of the IHX are shown in Table 2 since it is going to complement the further analysis in this section. Additionally, the resulting operating condensation and evaporation temperatures are given.

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	32.40	-4.48	78.8 (±1.6)%	0.624	29.55	14.43
R134a	32.71	-9.84	80.2 (±1.2)%	0.615	34.67	17.86
R134a	32.64	-14.61	81.8 (±1.1)%	0.549	39.03	20.83
R134a	39.67	-4.66	79.6 (±1.2)%	0.800	35.70	18.10
R134a	40.10	-9.85	81.1 (±1.0)%	0.733	40.71	21.75
R134a	40.00	-15.00	[a]	[a]	[a]	[a]
R134a	39.76	-4.77	39.7 (±1.1)%	0.408	23.06	10.10
R134a	40.23	-9.80	39.6 (±1.0)%	0.362	25.36	11.48
R134a	40.14	-14.83	40.6 (±0.9)%	0.302	27.30	13.57
R513A	32.47	-5.05	76.7 (±1.5)%	0.779	31.15	13.11
R513A	32.64	-10.08	78.2 (±1.3)%	0.761	35.76	15.96
R513A	32.52	-15.07	79.7 (±1.1)%	0.684	40.02	19.05
R513A	40.08	-5.09	77.3 (±1.2)%	0.983	36.88	17.22
R513A	39.95	-9.96	78.8 (±1.0)%	0.901	41.37	19.87
R513A	39.82	-15.03	80.3 (±0.9)%	0.784	45.91	22.88
R513A	39.97	-4.92	41.0 (±1.1)%	0.543	24.88	9.33
R513A	39.93	-9.88	40.2 (±1)%	0.474	26.48	10.59
R513A	39.77	-14.93	39.2 (±0.9)%	0.400	28.30	11.50

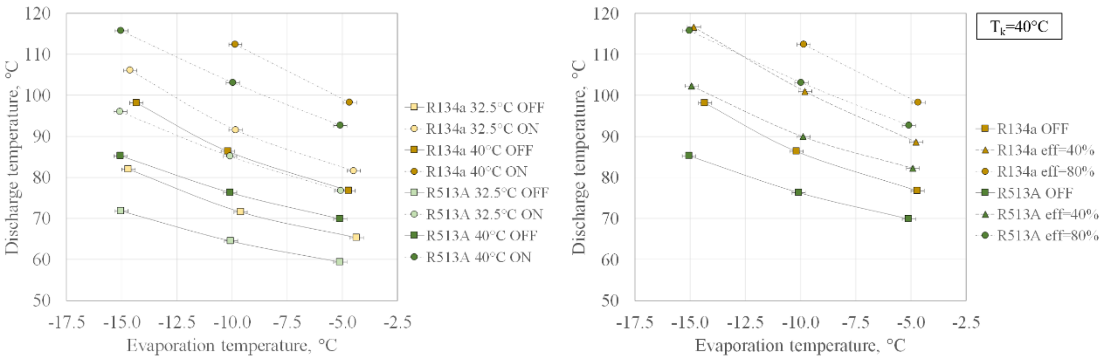
[a] Test not performed due to discharge temperature limitations. According to compressor manufacturer, at T_k=40 °C and T_o=-15 °C, the total SHD must be limited to 30 °C.

Maximum IHX effectiveness can give an idea of the substitution of R134a using R513A in an existing installation. And the controlled intermedium IHX effectiveness can give an idea of the benefits for the same value targeted. Slightly higher values for maximum IHX effectiveness are obtained for R134a and varies between 78.8 and 81.1% for R134a, whereas for R513A results between 76.7 and 80.3%. The maximum IHX effectiveness is increased for higher the compression ratios (and hence, greater difference between the two operating temperatures). Furthermore, the intermedium 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 intermediate operating IHX condition is comparable.

Total superheating and subcooling degrees of R513A is 1.2 higher and 1.4 K lower than R134a. So, 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

The compressor discharge temperature reflects the heat absorbed in the evaporator (latent and sensible), in the suction line superheat, and during the compression process. Therefore, this parameter includes the heat exchanged in the IHX and it is increased. Figure 4 shows the direct discharge temperature measurements in the discharge line at different conditions (discharge temperature values are approximately 40 K higher).



a) b)

Figure 4. Discharge temperature versus evaporation temperature at different a) condensation temperatures, and b) IHX effectiveness

The discharge temperatures of R134a are greater than those of R513A in any case, between 5.9 and 10.2 K without IHX and between 4.9 and 10.1 K with 80% IHX effectiveness. and allow the utilization of this additional component at high compression ratios. Only at 40% IHX effectiveness, R134a discharge temperatures become similar to that of R513A with the IHX fully activated. Hence, the lower discharge temperature of R513A increments the operating envelope of the refrigeration system with and without IHX. Given the greater heat exchanged in the IHX for R513A (Table 2), the IHX has a stronger influence on discharge temperature for R513A, being the increase between 17.3 and 26.8 K for R513A for this refrigerant (T_o=-15 °C and T_k=42.5 °C not considered), and between 16.3 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 were performed, a discharge temperature near 130 °C would be measured.

4.3. Compressor efficiencies

The additional superheating influences the scroll-technology compressor efficiencies and hence affects the rest of the parameters here analyzed. Table 3 summarizes the calculated average test values. Volumetric efficiency is always higher for R513A, and the increment of volumetric efficiency is only visible when IHX is fully activated. Isentropic efficiency of both refrigerants is comparable in any case. Additionally, as usually seen in IHX experimental studies, this component noticeably enhances the isentropic efficiency, remaining in an intermediate value when IHX is partially activated. Finally, measured global efficiency is around 3% higher for R513A than R134a in any case and the values with and without IHX are quite similar, and it can be concluded that the isentropic efficiency increase compensate the electromechanical efficiency reduction.

Table 3. Scroll compressor efficiencies using line measurements

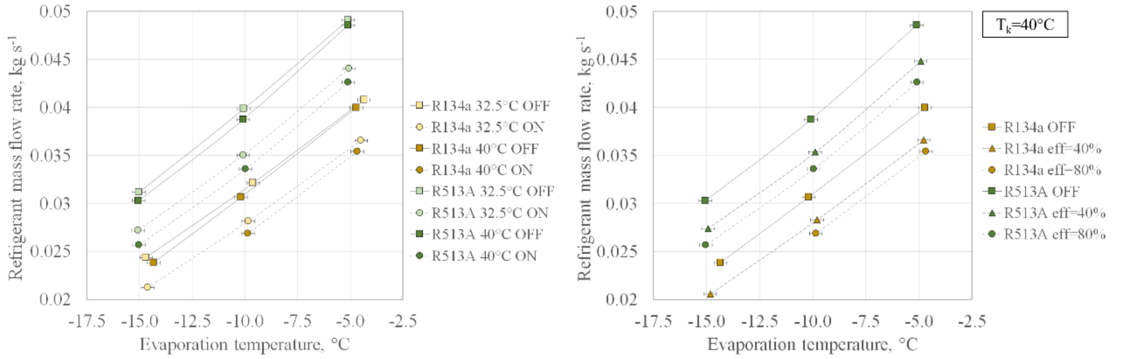
Refrigerant	**IHX**	**T_o (°C)**	**T_k (°C)**	η_{vol}	η_{iso}	η_{glo}
R134a	OFF	-4.35	32.66	0.88	0.60	0.65
R134a	ON	-4.48	32.40	0.89	0.67	0.66

R134a	OFF	-9.61	32.50	0.84	0.56	0.59
R134a	ON	-9.84	32.71	0.85	0.66	0.60
R134a	OFF	-14.69	32.50	0.77	0.51	0.50
R134a	ON	-14.61	32.64	0.78	0.59	0.51
R134a	OFF	-4.71	40.05	0.88	0.59	0.63
R134a	ON	-4.66	39.67	0.89	0.66	0.65
R134a	OFF	-10.18	40.09	0.82	0.54	0.54
R134a	ON	-9.85	40.10	0.83	0.61	0.56
R134a	OFF	-14.33	40.11	0.74	0.47	0.45
R134a	ON	-15.00	40.00	[a]	[a]	[a]
R134a	ε_{IHX}=40%	-4.77	39.76	0.88	0.62	0.63
R134a	ε_{IHX}=40%	-9.80	40.23	0.82	0.57	0.55
R134a	ε_{IHX}=40%	-14.83	40.14	0.72	0.50	0.44
R513A	OFF	-5.10	32.62	0.92	0.61	0.67
R513A	ON	-5.05	32.47	0.92	0.67	0.68
R513A	OFF	-10.05	32.58	0.89	0.57	0.62
R513A	ON	-10.08	32.64	0.89	0.66	0.64
R513A	OFF	-15.02	32.58	0.83	0.52	0.55
R513A	ON	-15.07	32.53	0.84	0.61	0.57
R513A	OFF	-5.10	40.13	0.91	0.59	0.64
R513A	ON	-5.09	40.08	0.92	0.65	0.66
R513A	OFF	-10.08	40.13	0.86	0.54	0.58
R513A	ON	-9.96	39.95	0.87	0.61	0.59
R513A	OFF	-15.05	39.94	0.80	0.49	0.50
R513A	ON	-15.03	39.82	0.81	0.56	0.51
R513A	ε_{IHX}=40%	-4.92	39.97	0.91	0.62	0.66
R513A	ε_{IHX}=40%	-9.88	39.93	0.86	0.58	0.58
R513A	ε_{IHX}=40%	-14.93	39.77	0.80	0.52	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 that happened for the suction density. This effect can be observed in Figure 5.



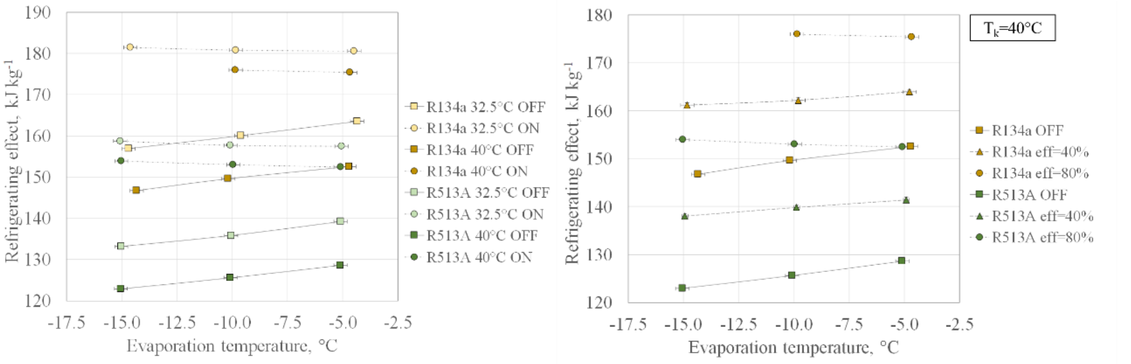
a) b)

Figure 5. 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 total IHX opening than basic cycle configuration.

4.5. Cooling capacity

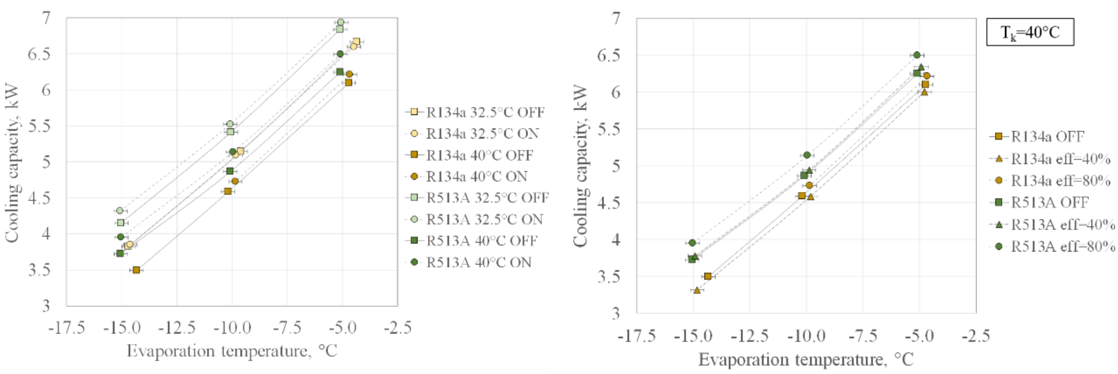
Before studying the cooling capacity results, the refrigerating effect (enthalpy difference at the evaporator) is shown in Figure 6. The refrigerating effect enhance the refrigerating effect by the increase of total SCD and hence the reduction of the enthalpy at the inlet of the evaporator (in this case isenthalpic expansion is considered and thus the enthalpy at the inlet of the EXV is taken). 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 larger increase of refrigerating effect agrees with the research of Pottker and Hrnjak [28], that 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 6. 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 7 represents experimental cooling capacity results.



a) b)

Figure 7. Cooling capacity versus evaporation temperature at different a) condensing temperatures, and b) IHX effectiveness.

Firstly, Figure 7 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 increase the cooling capacity, between 1.5 and 5.6%, for R134a this increase is only observed at higher compression ratio conditions, and in this case, it is around 3%. Different IHX effectiveness do not vary the result of cooling capacity for R134a whereas R513A have a benefit from the maximum IHX effectiveness achievable.

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 suction of more superheated vapor. Previous works [29] suggested that both effects are of a similar magnitude and hence the variation in power consumption results imperceptible. Figure 8 contains the direct measurements of the compressor power consumption.

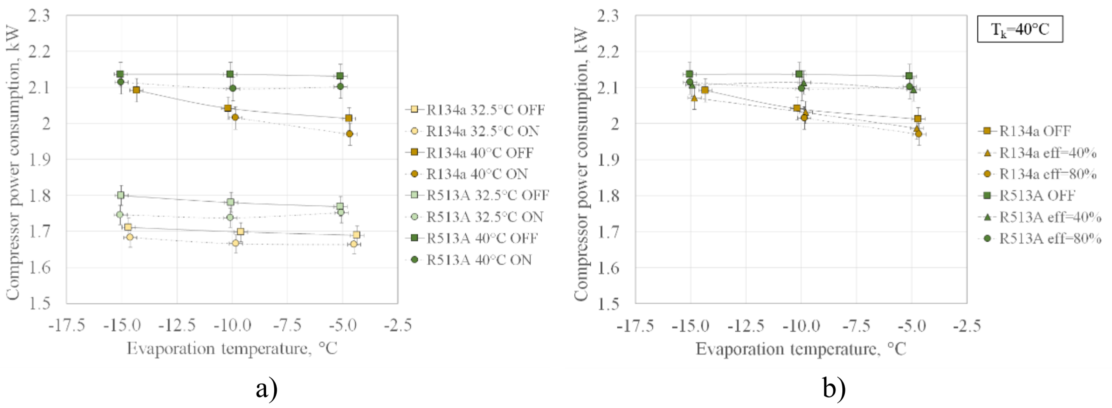
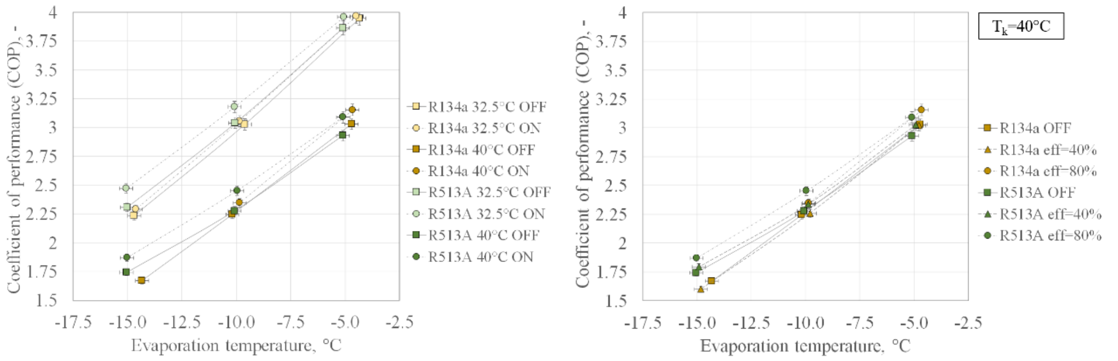


Figure 8. Compressor power consumption versus evaporation temperature at different a) condensation temperatures, and b) IHX effectiveness

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 and is caused by the greater specific compression work of the new mixture (in Ph diagram of both fluids, it can be seen that the R513A isentropic slope in vapor phase is lower). Additionally, as for R513A there is barely not influence of evaporation temperature, the R513A compressor power consumption difference with R134a increase 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, it has always been seen a benefit for the energy performance of the system because of the IHX adoption. Figure 9 contains the COP experimental results obtained in this study.



a) b)

Figure 9. 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 greater for higher compression ratios (lower evaporating temperatures and higher condensation temperatures). Furthermore, the COP growth is higher for the lower GWP mixture R513A, between 2 and 8%, in contrast with R134a, that results between 0 and 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 criteria [30] assumed isentropic compression, no-pressure drop infinite heat exchangers and no-pressure drop IHX. That developed by Aprea et al. [31] 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 [32] used an adaptation of the dimensionless Stefan-number (used absolute temperature instead of temperature difference), assuming no pressure drops and same pressure ratio. Despite the simplifications, these methods predict the positive sign of COP improvement in any condition, as seen in this experimental study. Additionally, Mastrullo et al. [33] suggested that higher the ratio of specific heat vapor molar at constant pressure and critical temperature is, higher the COP improvement. As average, this ratio is 4.4% higher for R513A than R134a.

On the other hand, the Klein et al.'s criteria [29] supposes that the IHX adoption has no influence on the compressor power consumption and provides a quantitative approximation to the COP improvement. Alternatively, Hermes [34] assumed constrained operating pressures, and isentropic compression, and obtained a similar expression to Domanski et al. [30]. Figure 10 shows the comparison between this method and the experimental results presented.

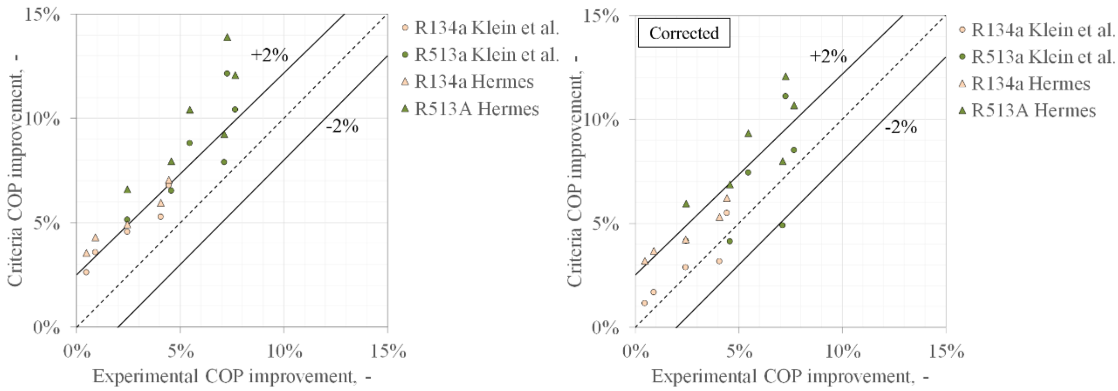


Figure 10. Comparison of experimental COP and Klein et al.'s and Hermes' criteria [29] a) original, and b) corrected.

In general, both criteria overpredicts the COP improvement. Firstly, Klein et al.'s criteria 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 criteria, both values matches within ±2.5%. Secondly, Hermes' criteria obtains closer values for R134a than R513A, and when the isentropic efficiency variation in experimental is applied to the Hermes' criteria results, the deviation is reduced.

6. Conclusions

R134a consumption is increasing and its emissions are one of the most relevant among all HFC gases. To provide a feasible short-term replacement of it, R513A is proposed in systems with and without IHX. This paper has presented the first experimental values of the influence of the IHX in an experimental system using R513A and R134a. Selected evaporation temperatures was -15, -10 and -5 °C, and condensation temperatures, 32.5 and 40°C; and rated IHX effectiveness is 80%.

The heat transfer in the IHX 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, discharge temperature produces higher increase for R513A discharge temperature but 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 increases caused by the IHX is higher on this refrigerant and thus produces higher benefit for cooling capacity. Power consumption is slightly reduced and resulting coefficient of performance is augmented when the IHX is used. COP gain is benefited from maximum achievable IHX effectiveness, higher compression ratios, and R513A utilization. Additionally, results with intermediate effectiveness (80%) produced lower benefit but allows 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.

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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] Devecioǐlu 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] 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.

[16] 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.

[17] 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.

[18] 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.

[19] 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.

[20] 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.

[21] 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.

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

[23] 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.

[24] 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.

[25] 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.

[26] 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.

[27] Aprea C, Greco A, Maiorino A. An experimental investigation on the substitution of HFC134a with HFO1234YF in a domestic refrigerator. Appl Therm Eng 2016;106:959–67. doi:10.1016/j.applthermaleng.2016.06.098.

[28] 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.

[29] 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.

[30] 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.

[31] 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.

[32] 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.

[33] 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.

[34] 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.