Experimental comparison between R152a and R134a working in a refrigeration facility equipped with a hermetic compressor

R. Cabello [a], D. Sánchez [a*], R. Llopis [a], I. Arauzo[a], E. Torrella [b]

[a]*Jaume I University, Dep. of Mechanical Engineering and Construction, Campus de Riu Sec s/n E-12071, Castellón, Spain*
[b]*Polytechnic University of Valencia, Dep. of Applied Thermodynamics, Camino de Vera, 14 E-46022 Valencia, Spain*

*Corresponding author: sanchezd@uji.es Tel.: +34 964728142; Fax.: +34 964728106

HIGHLIGHTS

- An experimental comparison between R134a and R152a is presented and commented.
- The same refrigeration plant with a hermetic compressor was used with both refrigerants.
- A reduction of about 10% in cooling capacity was registered by using R152a.
- A reduction up to 16% in electrical power consumption was registered by using R152a.
- An improvement up to 13% in COP was obtained using R152a.
- Aside safety issues R152a was used successfully as a "drop-in" refrigerant for R134a.

ABSTRACT

The EU Regulation 517/2014 has recently been approved in a further attempt to curb the effects of Global Warming. As a consequence, the refrigeration sector is moving towards refrigerants with a low Global Warming Potencial (GWP_{100}) in accordance with the limit fixed by these regulations (150). In this regard, the old refrigerant R152a attracts renewed interest due to its low GWP (138) and its similarity to R134a.

The present work shows the results of using R152a in a vapour compression plant equipped with a hermetic compressor and an IHX designed for R134a. The refrigerant was replaced by a conventional "drop-in" process in order to carry out an energy comparison. The results have revealed an improvement in the COP with R152a up to 13% despite a reduction in the cooling capacity of about 10%. During the test campaign, R134a hermetic compressors have been shown to be capable of operating with R152a.

KEYWORDS
R152a, R134a, F-gas, hermetic compressor, Internal Heat Exchanger, GWP

Nomenclature

e	relative error
COP	coefficient of performance
GWP	global warming potential (at 100 years)
h	enthalpy ($kJ \cdot kg^{-1}$)
HFC	hydrofluorocarbons
HOC	heat of combustion ($MJ \cdot kg^{-1}$)
IHX	internal heat exchanger
$\dot{m}$	mass flow rate ($kg \cdot s^{-1}$)
MW	molecular weight ($kg \cdot kmol^{-1}$)
N	compressor rotation speed (rpm)
NBP	normal boiling point (ºC)
ODP	ozone depletion potential
P	pressure (MPa)
P_C	compressor power consumption (W)
POE	polyolester oil
$\dot{Q}$	heat transfer rate (W)
qv	volumetric specific capacity ($kJ \cdot m^{-3}$)
RCL	refrigerant concentration limits ($gr \cdot m^{-3}$)
SC	subcooling (K)
SH	superheating (K)
T	temperature (ºC)
TEV	thermostatic expansion valve
v	specific volume ($m^3 \cdot kg^{-1}$)
V_G	geometrical volume (m^3)
$\dot{V}_G$	geometrical flow rate ($m^3 \cdot s^{-1}$)
w	specific compression work ($kJ \cdot kg^{-1}$)

Greek Symbols

Δ	increment
ε	thermal effectiveness
η_V	compression volumetric efficiency
η_G	compression global efficiency
λ	latent heat of phase change ($kJ \cdot kg^{-1}$)

Subscripts

C/comp	compressor
crit	critical point
dis	discharge

ev	evaporator
env	environment
hp	high pressure side
i	inlet / inner
ihx	internal heat exchanger
k	condenser
liq	liquid
lp	low pressure side
o	outlet / outer
ref	refrigerant
s	isentropic
sat	saturation
suc	suction
suf	compressor surface
v	vapour

1. Introduction

Fluorinated gases commonly used as refrigerants present high Global Warming Potential (GWP) values that make a strong contribution to global warming in case of leakage. For example, HFC-134a presents a GWP_{100} of 1300; HFC-404A a GWP_{100} of 3943; HFC-507A a GWP_{100} of 3985; HFC-410A a GWP_{100} of 1924 and HFC-407C a GWP_{100} of 1704, according to the latest report from IPCC AR5 (2013). The direct consequence of this problem is that the refrigeration industry is moving towards low GWP refrigerants, which may be either natural or artificial.

In Europe, this environmental concern has been translated into new regulations. On the one hand, there are those that limit the use of refrigerants with high GWP_{100} in a short time threshold, such as Directive 2006/40/EC on mobile air conditioning and Regulation 517/2014 on fluorinated greenhouse gases. On the other hand, some regulations impose environmental taxes on HFCs increasing its price (as in the case of Spain, Norway and Denmark), or promoting fiscal incentives that could encourage the adoption of natural refrigerants (this is the case of Germany and Austria) (Maratou *et al.*, 2013). Against this background, a new fourth generation of refrigerants is considered by Calm (2012). Their most remarkable new features are environmentally friendly (zero ODP and low/very low GWP), non-toxic and non-flammable properties.

In this situation, a small number of current HFC refrigerants could be used without contradicting the future new regulations or without being penalized with upcoming taxes. These regulations are driving the refrigeration industry to explore new environmentally friendly fluids, although in some cases they offer lower levels of security than existing refrigerants. Furthermore, these new refrigerants should not cause more indirect emissions of CO_2 related to electricity consumption than direct equivalent emissions of CO_2 due to leakages of refrigerant. Consequently, a deep analysis must be performed of its working conditions before making a decision and changing the refrigerant in a refrigeration facility.

Among those fluids with a GWP lower than the value set by the regulations (150), HFC-152a can be considered a good candidate, since its GWP_{100} is 138 (IPCC, 2013). This fluid has been used for a long time as an aerosol spray propellant and foam-blowing agent, as well as a component in some refrigerant blends (R401A, R415A, R430A, R500, etc.). However, its level of flammability designated by ASHRAE as A2 (ASHRAE, 2013) could be the reason why it has not been used as a pure refrigerant until now.

The automotive industry was the first to consider R152a as an alternative to R134a. The theoretical work developed by Ghodbane (1999) determined a coefficient of performance that was 10% higher for R152a than the value for R134a. The work of Kim *et al* (2008) based on experimental data, revealed an improvement in COP and cooling capacity of more than 20% using R152a instead of R134a at the same refrigeration test bench equipped with a swash-plate open-type compressor. Bryson *et al* (2011) stated that R152a presents improvements of up to 2% in cooling capacity and up to 9% in COP with respect to R134a, using an open-type compressor and adjusting the compressor rotation speed and expansion device. These last three references report an increase in discharge temperature as well as a reduction in the mass charge using R152a.

Apart from mobile air conditioning, little experimental work has been carried out in other fields. Bolaji (2010) presented an experimental study where R134a was directly replaced by R152a in a domestic refrigerator with a hermetic compressor.

In order to provide more experimental information on R152a, this work presents and analyses experimental data results obtained through a fully monitored vapour compression plant under a wide range of operating conditions. The plant was first charged and tested with R134a and then with R152a without any changes in lubricant or regulation. Apart from the safety issue, the analysis of the data evidences the better energy performance of the facility working with R152a, with an increment in COP of up to 13%.

2. Comparison of R134a and R152a fluid properties

Like other artificial substances, HFC refrigerants are obtained from natural but chemically modified substances. R134a and R152a are produced from ethane but each of them has a different composition as can be seen in Table 1. The R134a molecule contains four fluorine atoms and two hydrogen atoms, while R152a has only two fluorine atoms and four hydrogen atoms. The greater the number of carbon-fluorine chemical bonds, the higher the GWP level is. In the same way, the greater the number of hydrogen atoms is, the more flammable it is. This is the reason why R152a is more environmentally friendly than R134a but less safe, as can be noticed in its heat of combustion (HOC) compared to R134a. However, it is important to note that HOC level of R152a is notably lower than that of hydrocarbons such as propane (50.4 MJ/kg) or isobutene (49.4 MJ/kg), which have an ASHRAE security level of A3 (ASHRAE, 2013).

In Table 1 it can be observed that the Normal Boiling Point (NBP) of R152a is slightly higher than that of R134a, and the Molecular Weight (MW) in R152a is about 35.3% lower than that of R134a. The first implies a higher critical temperature, lower pressures for equal phase change temperatures and a higher compression ratio than R134a, whilst the second means that R152a presents a higher latent heat of vaporization or condensation (λ), in application of the Trouton rule (Wisniak, 2001). Combining its higher latent heat with its higher specific volume means the volumetric cooling capacity for R152a could be expected to be similar to that of R134a.

The main differences mentioned above are depicted in Figure 1 for two ideal vapour compression cycles working with R134a and R152a under the same operating conditions: isentropic compression, T_{ev}: -10ºC, T_k: 40ºC, no superheating at the evaporation outlet and no subcooling at the condenser outlet.

Taking into account both ideal cycles shown in Figure 1, some differences can be highlighted:

- R152a has a higher latent heat of vaporization and condensation than R134a.
- Discharge temperature is higher for R152a than R134a.
- The degree of desuperheating at the discharge line is larger for R152a than R134a.
- Working pressures at the same evaporating and condensing temperatures are slightly lower in R152a than R134a but compression ratios are very similar in both refrigerants.
- Isentropic compression lines are less sloped in R134a than R152a, which means higher isentropic specific compression work (w_S) in R152a than R134a.

To highlight the influence of the refrigerant on the isentropic specific compression work, Figure 2 shows the evolution of w_S in both refrigerants with an evaporating temperature at a fixed condensation level of 50ºC with and without a total superheating of 10K. Accordingly, it can be observed that the isentropic specific compression work is always higher for R152a than R134a regardless of the evaporating level (up to 59.05%). In the same way, the influence of superheating in the specific compression work is higher for R152a than R134a. This fact makes R152a more sensitive to superheating in terms of the compression process.

Figure 3 presents how the specific volume of R152a and R134a varies with evaporating temperature with and without a total superheating of 10K. Results in Figure 3 show that the specific volume is larger in R152a than R134a, which reduces the mass flow rate driven by the compressor and consequently its power consumption. Moreover, the effect of superheating on specific volume is more pronounced using R152a as a refrigerant. As a result, if a direct "drop-in" is carried out in a refrigeration facility designed for R134a, a reduction in refrigerant mass flow rate will be expected and consequently pressure drop will decrease as well.

3. Experimental plant and tests

In this section, the experimental refrigeration plant and the experimental methodology used are described. Measurement devices installed in the experimental facility are also presented.

3.1 Experimental facility

The refrigeration facility used to carry out the experimental analysis corresponds to a single-stage vapour compression plant designed for R134a, since the working pressures of R152a and R134a are very similar (Figure 1), and according to Uemura *et al.* (1992) the materials used with R134a are fully compatible with those employed with R152a.

Figure 4 presents a diagram showing the refrigeration facility driven by a single-stage reciprocating hermetic compressor with a cubic capacity of 12.11 cm^3 running at 2900 rpm (number 1). The hermetic compressor is designed to work with R134a using 350 ml of POE ISO22 as lubricating oil. Additionally, the test ring has two brazed plate heat exchangers working as a condenser and evaporator with heat exchange areas of 0.576 m^2 and 0.216 m^2, respectively (numbers 2 and 3); an internal heat exchanger (IHX) with a corrugated tube-in-tube arranged in counter-current layout (number 4); an electronic expansion valve working as a thermostatic (TEV) (number 5); and finally, a coalescing oil separator installed in the discharge line and connected to the compressor service port through a ball valve (number 6).

To avoid heat transfer with the environment all heat exchangers and pipe lines were insulated with an elastomeric foam with a thermal conductivity of 0.037 $W \cdot m^{-1} \cdot K^{-1}$. The IHX was installed with two by-passes in order to be able to connect or disconnect it from the refrigerating plant according to the test.

To maintain operating conditions during tests, two secondary fluid loops are used. The first is connected to the condenser and uses water as the secondary fluid. The second is connected to the evaporator and uses a water/propylene-glycol mixture, 70/30% by mass, to prevent it from freezing at low evaporating temperatures.

3.2 Measurement elements

All measurement devices used in the refrigeration plant are summarized in Table 2.

The states of the refrigerant and the secondary fluids (water and water + propylene-glycol mixture) were obtained using the RefProp v.9.1 database (Lemmon *et al.*, 2013) and ASHRAE Handbook (2001) through pressure and temperature measurements.

3.3 Methodology and experimental data tests

To evaluate the energy performance of the refrigeration facility, 36 steady-state tests were conducted for each refrigerant under a wide range of operating conditions, as is detailed in Table 3.

The evaporating temperature range was selected in accordance with the range stated by the manufacturer of the R134a compressor, covering air conditioning and refrigeration applications.

Operating conditions shown in Table 3 were performed working with and without an IHX with subcooling at the condenser outlet rated between 1.5 and 6K, and superheating at the evaporator outlet rated between 11 and 13K. Each trial had a minimum steady-state period of 20 minutes with a sampling rate of 10 s. To consider steady-state conditions a maximum variability of 0.2 K has been considered in temperature; 0.005 MPa in pressure; 0.4 $kg \cdot h^{-1}$ in the refrigerant mass flow rate, and finally, a maximum deviation of 0.02 $m^3 \cdot h^{-1}$ in the secondary flow rate. After each test, oil separator (number 6) is slowly connected to the compressor crankcase for an oil return.

Table 4 summarizes all the tests that were performed with the range of variations of the different parameters. Variables marked with an asterisk represent the average value obtained during the entire test. Evaporating and condensing pressures shown in Table 4 are obtained as an average value between the inlet and outlet pressure for each element. Evaporating and condensing temperatures are calculated in saturated conditions with these averaged pressures. Superheating at the evaporator (SH_{ev}) and subcooling at the condenser (SC_k) are obtained through Expressions (1) and (2). Similarly, IHX thermal effectiveness (ε_{ihx}), cooling capacity ($\dot{Q}_{ev}$) and the heat rejected by the condenser ($\dot{Q}_k$), are obtained by using the Expressions (3), (4) and (5) respectively.

$$SH_{ev} = T_{ev,o} - T_{ev,sat} \tag{1}$$

$$SC_k = T_{k,sat} - T_{k,o} \tag{2}$$

$$\varepsilon_{ihx} \approx 100 \cdot \frac{T_{ihx,o}\big|_{lp} - T_{ihx,i}\big|_{lp}}{T_{ihx,i}\big|_{hp} - T_{ihx,i}\big|_{lp}} \tag{3}$$

$$\dot{Q}_{ev} = \dot{m}_{ref} \cdot \left(h_{ev,o} - h_{ev,i}\right) \tag{4}$$

$$\dot{Q}_k = \dot{m}_{ref} \cdot \left(h_{k,i} - h_{k,o}\right) \tag{5}$$

The refrigeration facility was first tested with R134a with and without an IHX. Afterwards, the same procedure was used for R152a, in both cases a manual oil return to the crankcase being carried out after running for 2 hours.

3.4 Validation

Figures 5 and 6 compare the heat rejected by the secondary fluid and refrigerant at the condenser (Figure 5) and the evaporator (Figure 6). The maximum deviation registered by the measurements at the condenser is 6.1%, with a deviation lower than 6% for 91.7% of the data measured. The maximum deviation obtained in the evaporator is 10.88%, with a deviation lower than 7.5% for 67.8% of the data registered.

4. Results and discussion

This section is devoted to presenting and analysing the results obtained from the experimental refrigeration plant described above.

4.1 Refrigerant mass flow rate

The refrigerant mass flow rate is a parameter that depends on several variables in accordance with Equation (1).

$$\dot{m_{ref}} = \frac{\eta_V \cdot \dot{V}_G}{v_{C,i}} = \frac{\eta_V \cdot V_G \cdot N}{v_{C,i} \cdot 60} \qquad (6)$$

The geometrical volume (V_G) is a variable that only depends on the geometrical dimensions of the compressor: stroke, bore and number of pistons. According to the data from the compressor manufacturer the volumetric capacity is 12.11 cm^3. The rotation speed (N) is a parameter that depends on the electrical frequency and the mechanical operating conditions of the compressor. Despite the possible variation of this parameter and the solutions presented by Demay *et al.* (2011) about using internal and externally methods to measure the rotation speed of the hermetic compressor, we decided to use its nominal value as the work presented by Pisano *et al.* (2015). According with the manufacturer datasheet, the rotation speed is rated in 2900 rpm for an electrical frequency of 50 Hz

The volumetric efficiency (η_V) is a variable that can be obtained from experimental data using Equation (6), since the mass flow rate can be measured directly from the Coriolis mass flow meter installed in the setup (see Figure 4) and the geometrical flow rate ($\dot{V}_G$) has been considered unvarying. Assuming the specific volume measured at the compressor inlet ($v_{C,i}$) (Da Riva *et al.*, 2011), the experimental values obtained for volumetric efficiency are presented in Figure 7.

From Figure 7, it can be observed that the compressor volumetric efficiency is slightly higher working with R152a than with R134a, especially at low compression ratios. However, this difference is difficult to quantify because of the assumptions about rotation speed and specific volume considered previously. The parametric adjustment of the volumetric efficiency is has been analysed by Lawson and Miller (1986), suggesting a linear adjustment for volumetric efficiency against pressure ratio. An alternative of this adjustment is presented in Equation (7) where the presence of specific volume at the inlet compressor increases its accuracy. The coefficients from Equation 7 are presented in Table 5.

$$\eta_V = a_0 + a_1 \cdot \frac{P_{dis}}{P_{suc}} + a_2 \cdot v_{C,i} \qquad (7)$$

Regarding specific volume ($v_{C,i}$), from Figure 3 it is clear that the specific volume of R152a is up to 73.59% higher than that of R134a thereby reducing the mass flow rate driven by the compressor in accordance with Equation 6.

The measured refrigerant mass flow rate is shown in Figure 8, where a decrease of up to 41.5% is registered by using R152a with an average value of 39.8%. This reduction is more pronounced when an IHX is installed since it introduces additional superheating that increases the specific volume. Studies by Ghodbane (1999) and Kim *et al.* (2008) confirmed the reduction in mass flow rate using R152a in automotive open-type compressors with variable rotation speed. The reduction values registered by these authors were about 40 to 46% theoretically and 5 to 22% experimentally, respectively.

Despite the enthalpy variation, a mass flow rate reduction with R152a will have a negative impact in cooling capacity according with Expression 4. The influence over the power consumption of the compressor will be positive as be discussed below.

4.2 Compressor power consumption

Values for compressor power consumption are directly measured with the aid of a digital wattmeter as is shown in Table 2. Figure 9 presents the experimental data obtained by using both refrigerants with and without IHX.

According with Figure 9, it is evident that at the same working conditions R152a yields lower compressor power consumption than R134a, which means that the mass flow rate depletion registered in Figure 8 compensates the isentropic compression work increment shown in Figure 2. The reduction in power consumption is assessed in a range between 5.52 and 16.03% which corresponds with simulations obtained by Ghodbane (1999) where depletions varied between 5.72 and 10.8% using an open-type compressor and including mechanical losses related with drive belt.

The presence of an IHX increases temperature at the suction port reducing the mass flow rate and enlarging the specific compression work. The combined effect of both reveals the low influence of IHX on the electrical power consumption according with the results presented by Domanski (1995) and Navarro *et al.* (2005).

Electrical power consumption of a hermetic compressor (P_c) does not only depend on the mass flow rate ($\dot{m}_{ref}$) but also on the isentropic specific compression work (w_s) and its global efficiency, which are related with the working pressures and the suction temperature (Equation 8).

$$P_c \approx \dot{m}_{ref} \cdot \frac{w_s}{\eta_G} = \dot{m}_{ref} \cdot \frac{h_{dis,s} - h_{suc}}{\eta_G} \tag{8}$$

The global efficiency (η_G) is a variable that includes the mechanical and electrical efficiencies of the compressor as well as the isentropic efficiency related with the compression process in accordance with Da Riva *et al.* (2011) and Sánchez *et al.* (2010). Using experimental data summarized in Table 4 and Equation 8, the global efficiency obtained from the tested hermetic compressor with both refrigerants is assessed in a range of [0.23 ÷ 0.43] depending on operating conditions. Comparing the results from both refrigerants, only a slightly improvement using R152a has been noticed (up to 2%).

4.3 Discharge Temperature

In accordance with Da Riva *et al.* (2011), discharge temperature is an important parameter to be considered in any refrigeration facility since its higher value is detrimental to the durability of the compressor and the stability of the lubricating oil. This parameter is affected by several variables such as the total superheating degree, the compressor-type, the global efficiency and the shape of isentropic lines. As it has been explained in Figure 1, isentropic lines of R152a are less sloped than R134a ones, that means larger increments in discharge temperature with similar superheating at suction line. On the other hand, the low refrigerant mass flow rate driven by the compressor using R152a increases superheating in suction line because. This superheating depends on the vapour specific isobaric heat, the heat transfer coefficient in suction line, the temperature difference between the refrigerant at evaporator outlet and the environment, and the mass flow rate. Since the specific isobaric heat and the environment temperature is quite similar in both cases, and the heat transfer coefficient can be considered the same for both refrigerants, the mass flow rate is the main cause to increase the heat transfer between environment and suction pipe line as can be seen in Table 6.

The combined effect of these variables is presented in Figure 10 where it can be noticed that R152a has always a discharge temperature higher than R134a even using IHX.

The maximum difference registered between R152a and R134a was 4.49K without IHX and 5.49K working with IHX. This behaviour has been supported by other authors with very different values according the compressor used in their investigations: Ghodbane (1999), Kim *et al.* (2008) and Bolaji *et al.* (2011). As example, Uemura *et al.* (1992) adverts a theoretical and experimental difference of 7.6K working with an open-type compressor with evaporating level of 5ºC, condensing temperature of 45ºC, total superheating of 9K and subcooling at the condenser of 10K.

Despite the difference registered in the discharge temperature, all measured values do not affect the proper operation of the compressor.

4.4 Cooling capacity

The cooling capacity is obtained by Equation 4 as a product between mass flow rate ($\dot{m}_{ref}$) and specific cooling capacity (Δh_{ev}). The specific cooling capacity depends on the refrigerant latent heat which is defined by the pressure level. As explained in Section 2, the latent heat for R152a is approximately 52.1% higher than R134a at the same pressure level that means a specific cooling capacity higher for R152a. However, R152a mass flow rate is up to 41.5% lower than R134a, and consequently the combined effect could provide an improvement or a worsening in cooling capacity. Figure 11 shows the results obtained from experimental tests.

From the results presented in Figure 11, it can be concluded that at the same working conditions the cooling capacity using R134a is higher than using R152a especially at high evaporating temperatures. The difference registered is ranged between 1.13 and 9.75% that is in accordance with the results presented by Uemura *et al.* (1992) or Ghodbane (1999).

The influence of internal heat exchanger in the cooling capacity provides a slightly improvement especially at low evaporating levels. The maximum improvement reached is 7.15% for R134a and 4.49% for R152a at evaporation temperature of -10ºC.

4.5 Coefficient of performance (COP)

COP in a refrigerating plant is defined by the ratio between the cooling capacity ($\dot{Q}_{ev}$) and the power consumed by the plant (P_c) as expressed by Equation (5).

$$COP = \frac{\dot{Q}_{ev}}{P_C} = \frac{\dot{m}_{ref} \cdot \Delta h_{ev}}{\dot{m}_{ref} \cdot \frac{w_s}{\eta_G}} = \frac{\eta_G \cdot \Delta h_{ev}}{w_s} \quad (9)$$

Taking into account the results showed in Figure 8 and 11 for electrical power consumption and cooling capacity, respectively, an increment in COP using R152a would be expected since the decrement in electrical power consumption with respect R134a (Figure 6) is higher than the reduction in cooling capacity. The combined effect of both is presented in Figure 12 for evaporating levels of 10 and -10ºC to make easier the understanding.

Considering the experimental results presented in Figure 12, it can be highlighted that the use of R152a instead of R134a improves the COP of the refrigeration facility whatever the evaporating level is. Registered increments are bigger working with high evaporating temperatures than low evaporating levels with a maximum improvement of 13.20%. On the other hand, the effect of IHX in COP is reduced specially at high evaporation temperatures. Only when the evaporating temperature is -10ºC the maximum increment registered is 7.15% for R134a and 4.49% for R152a working at high condenser temperatures (45ºC).

4.6 Sankey diagram of the vapour compression cycle for both refrigerants

As a summary of all the analysis presented above, Figure 13 represents two Sankey diagram for each refrigerant at the similar operating conditions depicted in Table 6.

The heat transfer rate from discharge line ($\dot{Q}_{dis}$), liquid line ($\dot{Q}_{liq}$), and suction line ($\dot{Q}_{suc}$), are obtained by enthalpy difference between the beginning and the end of the line multiplied by the refrigerant mass flow rate. The heat rejected per time unit by the compressor surface to the environment ($\dot{Q}_{comp}$), is obtained from the global energy balance.

From Figure 13 is evidenced that the vapour compression cycle performed by R152a absorbs approximately 3.07% lower cooling load in the evaporator than R134a but needs 6.24% less electrical power at the compressor. A combination of both factors reports better energy efficiency (COP) in R152a than R134a (3.39% higher).

The heat rejected from the discharge line and the compressor surface to the environment is higher for R152a than R134a because of its higher discharge temperature. As a consequence, the heat rejected by R152a at condenser is lower than R134a. In the same way, the rejected heat from the liquid line to environment is also higher in R152a due to its low mass flow rate.

5. Conclusions

In this paper, an experimental investigation was carried out to compare the energy performance of two HFC refrigerants: R134a and R152a, working in the same experimental facility using a hermetic-type compressor and an IHX. For both refrigerants three different evaporating level conditions were tested: 10, 0 and -10ºC, at three different condensation temperatures: 25, 35 and 45ºC. All tests were run at steady state conditions keeping the value of the superheating very similar. Based upon the experimental results, the following conclusions were drawn:

a) Aside safety issues R152a has been used successfully as a “drop-in” refrigerant in a refrigeration test bench developed for R134a. No problems were found with the compressor (hermetic reciprocating one), lubricant (POE type) or the expansion device (electronic).

b) The experimental results show that the mass flow rate driven by the compressor is up to 41.5% lower in R152a than R134a. This is mainly due to the high specific volume of R152a.
c) According to the mass flow reduction, R152a presents lower electrical power consumption than R134a (up to 16.03%) despite its higher specific compression work.
d) Regarding cooling capacity, R152a has lower cooling capacity than R134a despite its higher latent heat. The reduction registered is ranged between 1.13 and 9.75% with hardly improvement working with IHX.
e) COP obtained with R152a is up to 11.70% better than R134a working without IHX, and up to 13.20% working with IHX.
f) Discharge temperature is always higher in R152a since its isentropic lines are less sharp than R134a. The maximum differences noticed are 4.49K operating without IHX and 5.49K with IHX.

Acknowledgements

The authors acknowledge Jaume I University of Spain, who financed partially the present study through the research project P1·B2013-10.

References

ASHRAE Handbook CD, 2001 Edition, American Society of Heating, Refrigerating and Air Conditioning Engineers, Englewood, Colorado (USA), 2001.

ASHRAE Standard 34-2013. Designation and Safety Classification of Refrigerants. ISSN:1041-2336. 2013

Pisano A., Martínez-Ballester S., Corberán J.M., William Mauro A., Optimal design of a light commercial freezer through the analysis of the combined effects of capillary tube diameter and refrigerant charge on the performance. International J. of Refrigeration 2015; 52; 1 – 10.

Bolaji B. O., Experimental study of R152a and R32 to replace R134a in a domestic refrigerator. Energy 2010; 35 (9); 3793 – 3798.

Bolaji B. O., Akintunde M. A., Falade T. O., Comparative analysis of performance of three ozone-friends HFC refrigerants in a vapour compression refrigerator. Journal of Sustainable Energy & Environment 2011; 2; 61 – 64.

Bryson M., Dixon C., StHill S., Testing of HFO-1234yf and R152a as mobile air conditioning refrigerant replacements. AIRAH. Ecolibrium May 2011, 30 - 38. (http://www.airah.org.au/imis15_prod/Content_Files/EcoLibrium/2011/May2011/2011_05_01.pdf)
B. Demay M., A. Flesh C., P. Rosa A., Indirect measurement of hermetic compressor speed through externally-measurable quantities. International J. of Refrigeration 2011; 34; 1268 - 1275.

Calm J. M., Refrigerant transitions... again. Moving towards sustainability. In: ASHRAE/NIST Conference, 29 – 30 October, Gaithersburg, MD, USA, 2012 (http://nccashrae.org/images/meeting/031214/calm_jm__2012._refrigerant_transitions_..._again._ashrae_nist_refrigerants_conference.pdf)

Da Riva E., Del Col D., Performance of a semi-hermetic reciprocating compressor with propane and mineral oil. International J. of Refrigeration 2011; 34; 752 – 763.

Directive 2006/40/EC of the European Parliament and of the Council of 17 May 2006 relating to emissions from air-conditioning systems in motor vehicles and amending Council Directive 70/156/EEC (http://eur-lex.europa.eu/LexUriServ/LexUriServ.do?uri=OJ:L:2006:161:0012:0018:en:PDF)

Domanski P. A., Theoretical evaluation of the vapour compression cycle with a liquid-line/suction-line heat exchanger, economizer, and ejector. NISTIR 1995; 5606.

Ghodbane M., An Investigation of R152a and Hydrocarbon Refrigerants in Mobile Air Conditioning. In: International congress and exposition, 1 – 4 March, Detroit, Michigan, 1999. SAE Technical Paper Series, 1999-01-0874.

IPCC, 2013: Climate Change 2013: The Physical Science Basis. Contribution of Working *Group I to the Fifth Assessment Report of the Intergovernmental Panel on Climate Change*. 1st ed. New York; Cambridge University Press; 2003.

Lawson S., Millet H., Rating thechnique for reciprocating refrigeration compressors, In: International Compressor Engineering Conference, 1986. Paper 526. (http://docs.lib.purdue.edu/icec/526)

Maratou A., Skačanová K., Vanaga G., GUIDE+: HFC taxes & fiscal incentives for natural refrigerants in Europe, Shecco Publications 2013; 17 December 2013.

Kim M. H., Shin J. S., Park W. G., Lee S. Y., The Test Results of Refrigerant R152a in an Automotive Air-Conditioning System. In: SAE 9th Alternative Refrigerant Systems Symposium, 10 – 12 June, 2008. (www.sae.org/events/aars/presentations/2008/manhoekim.pdf)

Lemmon E. W., Huber M. L., McLinden M. O., Reference fluid thermodynamic and transport properties (REFPROP), NIST Standard Reference Database 23, v.9.1. National Institute of Standards 2013. Gaithersburg MD, USA.

Navarro-Esbrí J., Cabello R., Torrella E., Experimental evaluation of the internal heat exchanger influence on a vapour compression energy efficiency working with R22, R134a and R407C. Energy 2005; 30 (5); 621 – 636.

Regulation (EU) No 517/2014 of the European Parliament and of the Council of 16 April 2014 on fluorinated greenhouse gases and repealing Regulation (EC) No 842/2006 (eur-lex.europa.eu/legal-content/EN/TXT/PDF/?uri=CELEX:32014R0517&from=EN)

Sánchez D., Torrella E., Cabello R., Llopis R., Influence of the superheat associated to a semihermetic compressor of a transcritical CO_2 refrigeration plant. Applied Thermal Engineering 2010; 30 (4); 302 – 309.

Uemura S., Inagaki S., Kobayashi N., Teraoka T., Noguchi M., Characteristics of HFC refrigerants. In: International Refrigeration and Air Conditioning Conference, 1992. Paper 177. (http://docs.lib.purdue.edu//iracc/177)

Wisniak J., Frederick Thomas Trouton: The man, the rule, and the ratio. The Chemical Educator 2001; 6 (1); 55 – 61.

Figure 1 – R134a and R152a ideal vapour compression cycle (T_{ev}: -10ºC, T_k: 40ºC)

Figure 2 – Isentropic compression work with and without superheating (SH: 10K) (T_k: 50ºC)

Figure 3 – Specific volume with and without superheating (SH: 10K)

Figure 4 – Refrigeration facility diagram

Figure 5 – Heat transfer rate validation at the condenser

Figure 6 – Heat transfer rate validation at the evaporator

Figure 7 – Compressor volumetric efficiency (η_V) vs. Pressure ratio

Figure 8 – Mass flow rate vs. Pressure ratio

Figure 9 – Electrical power consumption vs. Pressure ratio

Figure 10 – Discharge Temperature vs. Pressure ratio

Figure 11 – Cooling capacity vs. Pressure ratio

Figure 12 – COP vs. Pressure ratio

Figure 13 – Sankey diagram for R134a and R152a

Fluid	Chemical formula	P_{crit} (MPa)	T_{crit} (ºC)	MW ($kg \cdot kmol^{-1}$)	NBP (ºC)	$v_{sat,v}$ (NBP) ($m^3 \cdot kg^{-1}$)	λ (NBP) ($kJ \cdot kg^{-1}$)	qv (NBP) ($kJ \cdot m^{-3}$)	Safety Group	RCL ($gr \cdot m^{-3}$)	HOC ($MJ \cdot kg^{-1}$)	GWP [1] WGI-AR5[1]
R152a	$C_2H_4F_2$	4.52	113.26	66.051	-24.02	0.296	329.91	1113.66	A2	32	17,4	138
R134a	$C_2H_2F_4$	4.06	101.06	102.032	-26.07	0.190	216.97	1140.81	A1	210	4.2	1300

[1]Working Group I contribution to the IPCC Fifth Assessment Report

Table 1 – Main thermodynamic, safety and environmental properties of R134a and R152a (ASHRAE, 2001) (Lemmon *et al.*, 2013)

Sensors	Measured Variable	Measurement device	Calibration Range	Calibrated accuracy
16	Temperature	T-type thermocouple	-40.0 to 145.0 ºC	± 0.5 K
3	Pressure	Pressure gauge	0.0 to 4.0 MPa	± 0.012 MPa
3	Pressure	Pressure gauge	0.0 to 0.9 MPa	± 0.002 MPa
1	Refrigerant mass flow rate	Coriolis mass flow meter	0.0 to 40.0 $kg \cdot h^{-1}$	± 0.1 % of reading
2	Secondary fluid volume rates	Magnetic flow meter	0 to 4 $m^3 \cdot h^{-1}$	± 0.25 % of reading
1	Power consumption	Digital wattmeter	0 to 1250 W	± 0.1 % of reading

Table 2 – Accuracies and calibration range of the transducers

Refrigerant	IHX	T_{ev} (ºC)	T_k (ºC)
R134a and R152a	with and without IHX	10	25
			35
			45
		0	25
			35
			45
		-10	25
			35
			45

Table 3 – Operating conditions

	EVAPORATOR				CONDENSER				COMPRESSOR					IHX		
Refrigerant	T_{ev}^* (ºC)	P_{ev}^* (MPa)	SH_{ev}^* (K)	$\dot{Q}_{ev}$ (W)	T_k^* (ºC)	P_k^* (MPa)	SC_k^* (K)	$\dot{Q}_k$ (W)	P_c (W)	T_{dis} (ºC)	T_{suc} (ºC)	T_{suf} (ºC)	$\dot{m}_{ref}$ ($kg \cdot s^{-1}$)	ε_{ihx} (%)	COP (-)	T_{env} (ºC)
Without IHX																
R134a	9.78	0.41	3.41	1484.30	25.52	0.68	5.09	1818.12	433.76	65.08	15.24	35.46	29.88	-	3.41	24.69
	10.29	0.42	12.46	1431.17	35.69	0.90	6.02	1721.34	467.69	75.40	22.31	39.25	29.38	-	3.07	26.92

	9.88	0.41	12.55	1273.89	44.76	1.15	5.44	1569.77	496.42	82.46	22.17	41.87	28.31	-	2.60	24.43
	0.09	0.29	12.87	1033.99	25.35	0.67	4.51	1255.34	358.86	69.17	14.26	36.56	20.59	-	2.88	23.27
	0.57	0.30	12.87	956.94	34.99	0.88	4.53	1179.31	390.35	75.99	14.84	39.11	20.40	-	2.47	24.36
	0.12	0.29	12.58	838.93	45.34	1.17	4.83	1049.49	419.41	83.18	14.09	40.20	19.36	-	2.04	24.16
	-10.23	0.20	11.39	621.99	24.66	0.66	3.81	797.95	297.88	72.60	8.31	40.93	13.09	-	2.08	26.66
	-9.67	0.20	11.95	580.81	34.71	0.88	3.13	743.70	318.73	77.52	8.19	41.08	13.08	-	1.85	26.47
	-9.93	0.20	11.94	505.25	44.50	1.15	3.85	659.31	337.43	85.84	9.46	46.39	12.25	-	1.54	28.24
With IHX																
	10.21	0.42	3.62	1496.68	25.56	0.68	4.99	1822.92	434.51	66.57	17.61	36.17	29.82	48.05	3.44	24.60
	10.39	0.42	12.41	1422.79	35.62	0.90	5.90	1711.25	464.51	77.58	24.87	40.96	28.90	41.34	3.06	27.78
	9.91	0.41	12.50	1286.67	45.09	1.16	5.74	1556.42	493.16	86.29	27.88	44.42	27.63	37.61	2.61	25.60
	0.26	0.30	12.86	1027.34	25.28	0.67	4.84	1256.44	358.19	72.85	18.04	40.08	20.24	53.55	2.87	25.60
R134a	0.18	0.30	12.94	948.44	34.80	0.88	3.98	1146.38	384.29	79.36	21.12	41.53	19.64	42.14	2.47	25.72
	-0.02	0.29	12.75	851.60	45.02	1.16	4.58	1035.30	414.19	89.55	24.72	45.84	18.59	41.72	2.06	27.16
	-10.34	0.20	11.48	627.84	24.38	0.65	3.42	786.27	292.92	76.35	16.32	43.52	12.78	53.46	2.14	26.40
	-9.76	0.20	12.00	592.40	35.05	0.89	2.94	725.12	318.35	82.29	19.51	44.29	12.61	47.78	1.86	25.78
	-9.52	0.20	11.70	545.65	45.01	1.16	3.61	655.61	340.08	89.45	22.62	47.54	12.15	46.00	1.60	26.76
Without IHX																
	10.23	0.38	6.08	1417.65	25.68	0.61	1.46	1653.96	365.98	67.72	17.79	38.24	18.33	-	3.88	25.32
	10.75	0.38	9.64	1341.41	35.30	0.80	2.26	1588.25	404.33	78.44	21.16	44.33	18.06	-	3.34	30.06
	10.28	0.38	9.59	1192.00	45.16	1.04	1.98	1425.49	441.02	85.83	20.64	46.27	17.14	-	2.75	27.13
	0.24	0.27	9.98	948.45	25.11	0.60	1.86	1128.97	316.36	72.56	13.54	40.85	12.36	-	3.00	25.95
R152a	0.15	0.27	9.86	863.60	35.33	0.80	2.00	1040.95	349.46	81.47	13.79	44.69	11.93	-	2.50	27.17
	0.14	0.27	10.01	775.27	44.79	1.03	1.60	934.19	376.65	88.12	13.73	47.09	11.37	-	2.11	28.04
	-9.82	0.18	11.66	606.09	25.24	0.60	2.68	736.50	278.55	76.93	10.66	45.18	8.06	-	2.17	26.93
	-9.45	0.19	11.35	566.51	35.00	0.79	2.84	685.37	296.09	82.32	10.26	46.06	7.92	-	1.94	26.92
	-10.09	0.18	12.27	489.75	44.68	1.03	2.83	589.94	316.36	90.58	11.89	51.83	7.21	-	1.58	28.42
With IHX																
	9.92	0.37	7.08	1403.71	24.91	0.60	3.18	1649.96	364.84	70.59	20.43	40.73	17.82	68.69	3.85	26.65
	10.22	0.38	9.68	1312.98	35.10	0.80	1.77	1536.03	398.05	80.79	25.61	46.10	17.37	41.87	3.30	28.10
	10.36	0.38	9.84	1224.54	44.90	1.03	2.20	1420.51	438.20	89.43	28.53	47.44	16.83	39.76	2.79	26.49
	0.21	0.27	9.90	946.01	25.69	0.61	2.47	1116.90	316.48	75.99	18.95	43.25	12.12	52.11	2.99	26.49
R152a	0.07	0.27	9.77	875.15	35.12	0.80	2.37	1023.04	344.49	83.84	21.71	45.75	11.61	45.84	2.54	26.36
	-0.32	0.26	9.86	789.97	44.83	1.03	1.79	904.45	372.53	92.27	24.89	48.03	10.89	44.57	2.12	25.30
	-9.73	0.18	11.58	614.47	24.90	0.60	2.17	729.94	271.56	79.11	18.77	46.71	7.96	58.31	2.26	26.54
	-9.25	0.20	11.39	585.69	35.12	0.80	2.94	676.55	297.91	84.69	20.38	46.92	7.79	50.32	1.97	25.03
	-9.58	0.20	10.36	519.77	45.15	1.04	1.12	586.40	321.31	93.19	24.40	52.97	7.24	48.80	1.62	27.17

Table 4 – Test summary

	a_0	a_1	a_2	e_{Max}
η_V (R134a)	0.81883057	-0.02193881	-0.11200566	2.30 %
η_V (R152a)	0.83589098	-0.03040696	0.10290412	3.89 %

Table 5 – Coefficients and maximum estimation error for Equation (2)

	T_{ev} (ºC)					
	10.20		0.22		-9.87	
T_k (ºC)	R134a	R152a	R134a	R152a	R134a	R152a
25.26	0.60	1.72	1.57	3.55	7.45	9.00
35.17	0.05	0.97	1.66	3.99	6.21	8.55
44.87	0.05	0.95	1.66	3.76	7.73	9.89

Table 6 – Superheating at suction line for R134a and R152a without IHX

	Operating conditions				
Refrigerant	T_{ev}^* (ºC)	SH_{ev}^* (K)	T_k^* (ºC)	SC_k^* (K)	T_{env}^* (ºC)
R134a	-9.93	11.94	44.50	3.85	28.24
R152a	-10.09	12.27	44.68	2.83	28.42

Table 6 – Operating conditions by the refrigeration facility