Energy Conversion and Management 51 (2010) 2195–2201

Contents lists available at ScienceDirect

Energy Conversion and Management

journal homepage: www.elsevier.com/locate/enconman

Exergy analysis of R413A as replacement of R12 in a domestic refrigeration system

Miguel Padilla [a,*], Rémi Revellin [b], Jocelyn Bonjour [b]

[a] *School of Mechanical Engineering, Central University of Venezuela, Caracas 1051, Venezuela*
[b] *Université de Lyon, CNRS*
INSA-Lyon, CETHIL, UMR5008, F-69621, Villeurbanne, France
Université Lyon 1, F-69622, France

ARTICLE INFO

Article history:
Received 28 July 2009
Accepted 21 March 2010
Available online 18 April 2010

Keywords:
R12
R413A
Exergy
Energy saving
Domestic refrigerator
Retrofitting

ABSTRACT

This paper deals with an exergy analysis of the impact of direct replacement (retrofit) of R12 with the zeotropic mixture R413A on the performance of a domestic vapour-compression refrigeration system originally designed to work with R12. Parameters and factors affecting the performance of both refrigerants are evaluated using an exergy analysis. In the literature, no experimental data for exergy efficiency are reported, so far, for R413A. Twelve tests (six for each refrigerant), are carried out in a controlled environment during the selected cooling process from evaporator outlet temperature from 15 °C to −10 °C. The evaporator and condenser air-flows are modified to simulate different evaporator cooling loads and condensers ventilation loads. The overall energy and exergy performance of the system working with R413A is consistently better than that of R12.

1. Introduction

The refrigerant chlorofluorocarbon (CFC) dichlorodifluoromethane (CCl_2F_2), known from its ASHRAE classification as R12, is a refrigerant that has been widely used in refrigeration systems and air conditioning since its invention in the 1930s. However, because of its high ozone depleting potential (ODP) and global warming potential (GWP), it cannot be used since 1998. The issue of the use of substances that deplete the ozone layer, such as R12, has led to a search for environmentally-friendly alternatives. Some R12 refrigerant substitutes that meet this requirement are a key process in this investigation.

1.1. Experimental studies with R12 substitutes in refrigerating systems

In response to the international protocol agreements, there has been carried out worldwide a variety of studies which have reported alternative substitutes for R12. The hydrofluorocarbon (HFC) refrigerants, such as R134a with zero ODP, have emerged as candidates for R12 substitution in refrigeration systems. But even with its ODP equal to zero, R134a has a relatively high global warming potential (1300 times that of CO_2). Spauschus [1] discussed the compressor and refrigeration system requirements and information gaps for R134a application as a R12 substitute. Havelský [2] studied the influence of R12 working fluid replacements such as R134a, R401A, R409A, R22 and R134a/R12 mixture on energy efficiency and global warming by using parameters such as the coefficient of performance (COP) and the total equivalent warming impact (TEWI). It is shown that the use of R134a, R401A and R409A refrigerants enables the increase of COP and reduces the value of TEWI in comparison with R12 application.

Other possibilities of R12 substitutes are hydrocarbon refrigerants (HC) which have favorable characteristics such as zero ODP and very low GWP. Nevertheless, the main disadvantage is their flammability [3] and their limitation in the charge quantity due to safety regulations. Several experimental studies with HC mixtures have been developed, in particular, the propane/isobutane mixtures. Richardson and Butterworth [4] conducted experiments to investigate the performance of propane/isobutane mixtures in a hermetic vapour-compression system. It is shown that propane and propane/isobutane mixtures may be used in an unmodified R12 system, giving better COP than R12 under the same operating conditions. Camporese et al. [5] evaluated 16 refrigerant mixtures as potential substitutes for R12 considering pure components R22, R32, R125, R134a, R143a, R290 and R270. They evaluated the influence of some of these components on the performance of refrigerating units and on the refrigerant miscibility with the lubricant oils. Herbe and Lundqvist [6] studied the influence of levels of contaminants such as acid, moisture and residual mineral oil in converted refrigeration systems. In this work, the authors presented a database of results from laboratory analysis of oil samples from converted systems using the retrofit practice.

* Corresponding author. Tel.: +58 212 605 17 96; fax: +58 212 605 31 56.
E-mail address: miguel.padilla@ucv.ve (M. Padilla).

0196-8904/$ - see front matter © 2010 Elsevier Ltd. All rights reserved.
doi:10.1016/j.enconman.2010.03.013

Nomenclature

CFC	chlorofluorocarbon
COP	coefficient of performance (W/W)
GWP	global warming potential
HFC	hydrofluorocarbon
HC	hydrocarbon
h	specific enthalpy (J/kg)
$\dot{m}$	mass-flow rate (kg/s)
ODP	ozone depleting potential
p	pressure (kPa)
$\dot{Q}$	heat transfer rate (W)
$\dot{S}$	entropic flow (W/K)
s	specific entropy (J/kg K)
TEWI	total equivalent warming impact
T	temperature (K)
t	time (s)
V	volume (m^3)
$\dot{W}$	power (W)
Greeks	
χ	exergy (J)
η	efficiency
ψ	flow exergy (J/kg)
Sub and superscripts	
ex	exergy
CV	control volume
0	ambient properties
in	inlet
out	outlet
f	control surface interactions
gen	generated
comp	compressor
evap	evaporator
cond	condenser
exp	expansion device
des	destroyed
sup	supplied

In relation to the investigations developed for domestic refrigerators, [7] carried out experiments for evaluating the performance parameters such as the evaporator capacity, the compressor power consumption, the COP and the cooling rate characteristics of a domestic refrigerator. They used different ratios of propane/isobutane mixtures as possible alternative replacements to R12. It was found that the hydrocarbon mixture with 50% propane/38.3% butane/11.7% isobutane was the most suitable alternative refrigerant with the best performance among all others hydrocarbon mixtures investigated. Jung et al. [8] evaluated the performance of a propane/isobutane (R290/R600a) mixture in domestic refrigerators in the composition range of 0.2–0.6 mass fraction of R290, which yielded an increase in the COP of up to 2.3% as compared to R12. Lee and Su [9] carried out an experimental study on the performance of a domestic vapour-compression refrigeration system with isobutane (R600a) as the refrigerant. The coefficients of performance of the system were between 0.8 and 4.5, which is comparable with those of the system with R12 as the refrigerant. Tashtoush et al. [10] tested new hydrocarbon/hydrofluorocarbon refrigerant mixtures for the replacement of R12 in domestic refrigerators by comparing parameters and factors such as the COP, the compressor power consumption, the volumetric efficiency, the condenser duty and the compressor discharge pressure and temperature to those of R12. The results showed that butane/propane/R134a mixtures provide performance characteristics very close to those of R12. The COP of the hydrocarbon/hydrofluorocarbon refrigerant mixture evaluated at a 100 W evaporator duty is 5.4% less than that for R12 and 0.8% at a 350 W duty, and the volumetric efficiency of the compressor was better than that when using R12. Although HC mixtures may be able to replace the R12 in several applications, their disadvantages related to the flammability and solubility in oil remain unsolved.

In the context of commercial refrigeration equipments, comparative studies of the behaviour of R134a and R12 have shown that R134a has the worst performance when comparing with R290 and R401A at the same originally working conditions in a system designed for R12. Halimic et al. [11] observed that R134a operates at conditions similar to R12. With some improvements in devices for expansion, it was even possible to further improve this performance. Recently, Mani and Selladurai [12] carried out a study of the vapour-compression refrigeration system performance with the new R290/R600 a refrigerant mixture as drop-in replacement compared with R12 and R134a. The experimental results showed that the R290/R600a mixture had 19.9–50.1% higher refrigerating capacity than R12 and 28.6–87.2% than R134a. The refrigerant R134a showed slightly lower COP than R12.

Based on the above observations, it is shown that R134a was considered to be the preferred HFC replacement working fluid in refrigeration systems and air conditioning operating with R12, but there are some limitations in the retrofit practice between these two refrigerants. R134a is not miscible with mineral oil; therefore it is necessary to use synthetic oil instead of mineral oil. This type of synthetic oil is highly hygroscopic and expensive and it is harmful to health as it causes irritation when in contact with skin and mucous membranes. Furthermore, the inappropriate use of synthetic oil in converted systems which were initially designed for operate with mineral oil, have brought serious problems in cooling systems that have resulted in the equipment damage by internal failures in the compressor and other devices. Janssen and Engels [13] referred to the problem of miscibility of the HFC refrigerant. This problem can be eliminated if we consider mixtures of HFCs with hydrocarbons (HC). Sekhar and Lal [14] carried out experimental investigations with R134a and HC blends as replacements of R12 in low temperature systems. In order to study the oil miscibility of the refrigerant mixture with mineral oil, they found that the R134a/HC blend used exhibited better performance, resulting in 5–15% less energy consumption than R12 in low temperature systems.

Refrigerant R600a has been considered as a viable additive because of its availability and favourable thermo physical properties. However, at the same operating pressure, R600a is evaporated at a higher temperature than R134a. As it is the most volatile, the vapour phase is enriched in that component and oil could not return to the compressor as expected. This was the background idea for including another component more volatile than R600a and R134a to form the R413A mixture, namely R218 (Octafluoropropane).

The base composition of the zeotropic mixture R413A is 88% of R134a, 9% of R218 and 3% of R600a. In this study, we consider the direct retrofit of R413A in a R12 system, and evaluate the thermodynamic behaviour of both fluids using an exergy analysis. Fig. 1 shows the variation of vapour pressure with respect to the

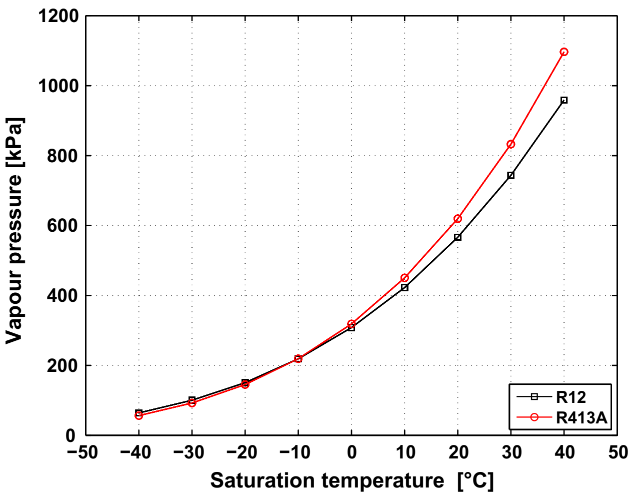

Fig. 1. Vapour pressure variation of R12 and R413A with respect to saturation temperature.

saturation temperature for both refrigerants. The evaporation and condensation temperatures typical for the cooling system studied, i.e. domestic refrigeration, are from −15 °C to 40 °C in an unmodified R12 domestic refrigerating system.

1.2. Exergy analysis of refrigerants substitution in refrigerating systems

In the open literature, there exist several works related to thermodynamic analysis, which deal with different well-known thermodynamic efficiencies used to characterize refrigerating systems. Usually, the attention focuses on the coefficient of performance ($COP = \dot{Q}_{evap}/\dot{W}_{comp}$), the volumetric efficiency, the compressor power consumption and the evaporator duty. In this paper, compared to other existing works, we adopt a different approach for comparing the performance of new refrigerant mixtures with R12 at the same working conditions in a converted system. A comparative exergy analysis based on experimental tests was developed for the R413A as drop-in substitute for R12 in domestic refrigerators.

There are few articles in the open literature that deal with exergy analysis of refrigerant substitutions. Aprea and Greco [15] performed an exergy analysis of R407C as substitute of R22 in an experimental vapour-compression air-conditioning system. They presented their results related to the secondarily fluids (water and air). They did not consider the exergy supplied to the evaporator fan, neither the condenser water pump and condenser heater and also there are no remarks in relation of refrigerant charge savings. The results obtained allowed for remarking that the overall exergy performance of the plant working with R22 is consistently better than when working with R407C and the compressor contribution to the overall exergy destruction is the most important.

The analysis carried out in this article follows an exergy approach in order to evaluate the thermodynamic performance of R413A in an unmodified R12 domestic refrigerating system. These results cannot be obtained by a traditional energy analysis, because this leads to an incomplete thermodynamic analysis. For this particular reason, this article includes an exergy analysis based on experimental data, which demonstrates the complementarity between the classical thermodynamic approach based on the first law and the exergy approach (second law) for the evaluation of a refrigeration system.

2. Experimental setup and procedure

2.1. Description of the experimental apparatus

The domestic refrigerator used in the present work was a refrigerating test bench unit originally designed to work with R12. Fig. 2 shows the schematic diagram of the experimental setup. The main loop of the system under investigation was composed of four basic components, i.e., a compressor, an evaporator, a condenser and a thermostatic valve as expansion device. The test bench has the ability to control the velocity of the evaporator and condenser fans. The compressor was an hermetic type reciprocating compressor with 330 W nominal input power at 115 V (60 Hz) and a displacement volume of 10.09 cm^3. The evaporator cooling capacity was between 2.5 and 3 kW and the temperature range was from −10 °C to 15 °C. The refrigerant charge was 0.49 kg (R12). The unit also had other devices such as filters, liquid flow indicators, a liquid receiver, a suction accumulator and the compressor protection devices. The refrigerant mass flow was measured by a coriolis type flow meter with ±0.15% of accuracy. The power consumed by the compressor was measured using a power meter with 0.01 kW h of accuracy. The temperature was measured by using thermocouples with an accuracy of ±0.1 °C. The inlet mass flow at evaporator and condenser fans was measured with a thermo-anemometer with an accuracy of ±0.1 m/s. Aiming to compare the domestic refrigerator performance under the same environmental conditions, the tests were carried out in a calorimetric chamber with controlled temperature environment.

2.2. Experimental procedure

Twelve tests (six for each refrigerant) have been carried out in a controlled environment (calorimetric chamber) with an environment temperature of 18 °C and evaporator and condenser air-flows discharge conditions as shown in Table 1.

The pressure of refrigerant in the condenser and the evaporator, the temperatures in the refrigeration loop and the compressor power consumption data for each of the 12 tests, were recorded with a period of 10 s per measurement in the dynamic cooling process from 15 °C to −10 °C as measured as the outlet of the evaporator. The experiment was started with R12 to set up the base reference for comparison with R413A. The thermodynamic properties of the refrigerants were obtained from the NIST thermodynamic properties of refrigerants and refrigerant mixtures database [16].

3. Exergy analysis

With the experimental data obtained from the tests using both R12 and R413A, we proceed to analyze the system by the evaluation of exergy loss of the domestic refrigerator in order to obtain

Table 1
Test conditions at T_0 18 °C.

Percentage of max. fan velocity at condenser	Percentage of max. fan velocity at evaporator			
	0%	100%	0%	100%
0%	Test 1	Test 4	Test 7	Test 10
50%	Test 2	Test 5	Test 8	Test 11
100%	Test 3	Test 6	Test 9	Test 12
Refrigerant	R12		R413A	

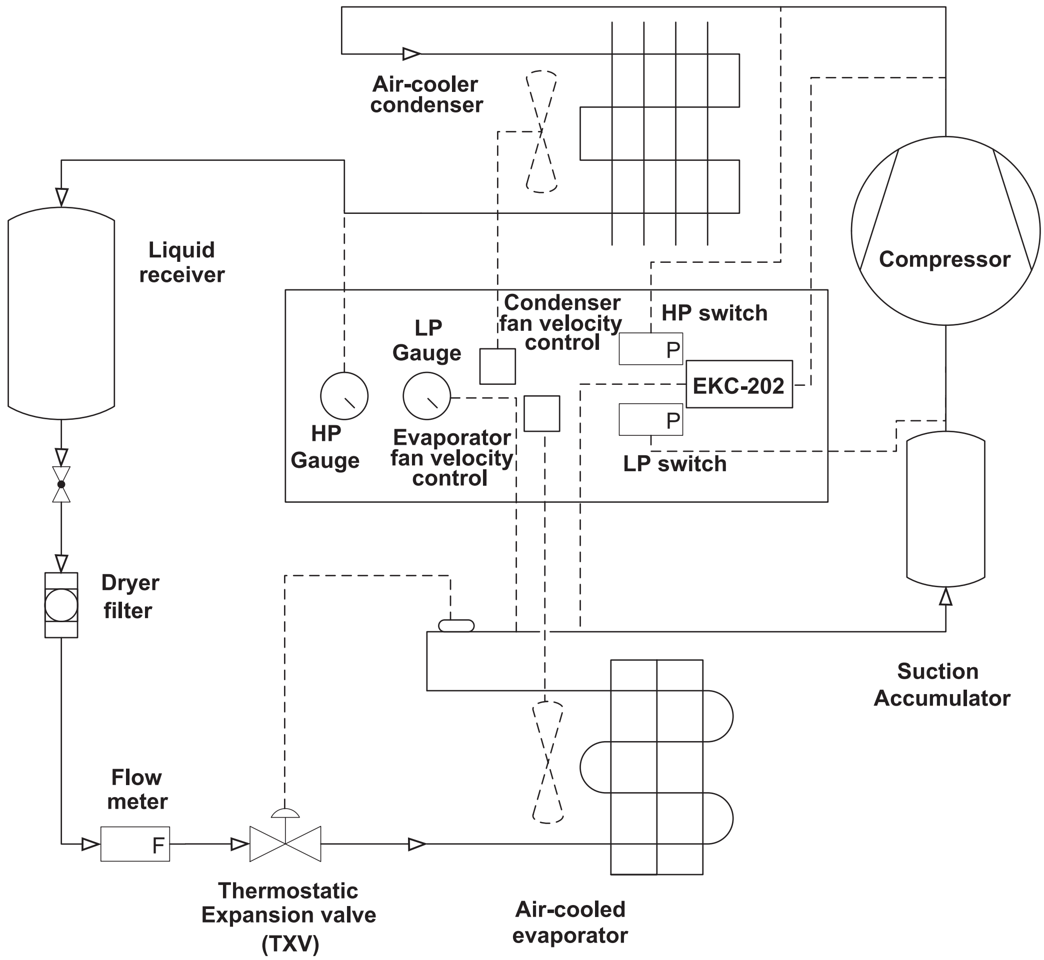

Fig. 2. The experimental apparatus.

a quantitative measurement of the process inefficiency. It can be recalled that the exergy of the working fluid in a control volume represents the maximum work that can provide this fluid through a reversible process until it reaches the thermodynamic equilibrium with the environment. Under the assumption that the change of kinetic and potential energy is negligible, exergy is:

$$\chi = h - T_0 s \tag{1}$$

For Eq. (1), the enthalpy and the entropy are considered as equal to 0 in the equilibrium state with the environment. Furthermore, the exergy can be expressed as the sum of all contributions of rate exergy with respect to time during the time that lasts the test.

$$\chi = \int_{t_0}^{t} \dot{\chi}(t)\,\mathrm{d}t \tag{2}$$

The exergy balance general expression in a control volume considered is:

$$\left(\frac{\mathrm{d}\chi}{\mathrm{d}t}\right)_{\mathrm{CV}} = \sum\left(1-\frac{T_0}{T_\mathrm{f}}\right)\delta\dot{Q}_\mathrm{f} - \left(\delta\dot{W}_\mathrm{f} - p_0\frac{\mathrm{d}V_{\mathrm{CV}}}{\mathrm{d}t}\right) + \sum\dot{m}\psi_{\mathrm{in}} - \sum\dot{m}\psi_{\mathrm{out}} + T_0\dot{S}_{\mathrm{gen}} \tag{3}$$

where $\mathrm{d}\chi/\mathrm{d}t$ is the exergy variation with respect to the time in the control volume, T_0 is the ambient temperature, T_f is the control surface temperature, $\dot{Q}_\mathrm{f}$ and $\dot{W}_\mathrm{f}$ corresponds to the heat and work interactions between the system boundaries and its surrounding, ψ is the flow (or stream) exergy and $\dot{S}_{\mathrm{gen}}$ is the entropy generation due to the irreversibilities.

When the compression is non-adiabatic in the compressor, it transfers heat to the environment, making it difficult to directly calculate the entropy generated. In that sense, through a steady-flow exergy balance in the compressor, we can calculate the exergy destroyed in the compressor by the following expression:

$$(T_0\dot{S}_{\mathrm{gen}})_{\mathrm{comp}} = \left(1-\frac{T_0}{T_\mathrm{f}}\right)\dot{Q}_{\mathrm{comp}} - \dot{W}_{\mathrm{comp}} + \dot{\psi}_{\mathrm{comp,in}} - \dot{\psi}_{\mathrm{comp,out}} \tag{4}$$

where $\dot{Q}_{\mathrm{comp}}$ and $\dot{W}_{\mathrm{comp}}$ are the heat rate transferred by the compressor to the environment and the compressor power supplied respectively. In this case, the heat transfer rate $\dot{Q}_{\mathrm{comp}}$ was obtained using an energy balance on the compressor.

Similar expressions can be written for the other components of the system such as the evaporator, condenser and expansion device. The total exergy flow destroyed in the system can be expressed as follow:

$$\dot{\chi}_{\mathrm{des}} = (T_0\dot{S}_{\mathrm{gen}})_{\mathrm{comp}} + (T_0\dot{S}_{\mathrm{gen}})_{\mathrm{evap}} + (T_0\dot{S}_{\mathrm{gen}})_{\mathrm{cond}} + (T_0\dot{S}_{\mathrm{gen}})_{\mathrm{exp}} \tag{5}$$

where

$$(T_0\dot{S}_{\mathrm{gen}})_{\mathrm{evap}} = \left(1-\frac{T_0}{T_\mathrm{f}}\right)\dot{Q}_{\mathrm{evap}} + \dot{W}_{\mathrm{evap}} + \dot{\psi}_{\mathrm{evap,in}} - \dot{\psi}_{\mathrm{evap,out}} \tag{6}$$

$$(T_0\dot{S}_{\mathrm{gen}})_{\mathrm{cond}} = \left(1-\frac{T_0}{T_\mathrm{f}}\right)\dot{Q}_{\mathrm{cond}} + \dot{W}_{\mathrm{cond}} + \dot{\psi}_{\mathrm{cond,in}} - \dot{\psi}_{\mathrm{cond,out}} \tag{7}$$

$$(T_0\dot{S}_{\mathrm{gen}})_{\mathrm{exp}} = \dot{\psi}_{\mathrm{exp,in}} - \dot{\psi}_{\mathrm{exp,out}} \tag{8}$$

where $\dot{W}_{\mathrm{evap}}$ and $\dot{W}_{\mathrm{cond}}$ in Eqs. (6) and (7), corresponds to the power supplied to fans in the evaporator and condenser, respectively.

These values come from the experimental measurements of the power supplied to the compressor, condenser fan and evaporator fan. Then, the exergy supplied was obtained by an exergy balance in the compressor, evaporator and condenser (Eq. (3)). The exergy efficiency (or second law efficiency) of the system is given by:

$$\eta_{ex} = 1 - \frac{\dot{\chi}_{des}}{\dot{\chi}_{sup}} \tag{9}$$

where $\dot{\chi}_{sup}$ corresponds to the exergy supplied to the system.

4. Results

Fig. 3 shows the results of the power consumed by the compressor for each test run in the system with refrigerant R12. It is considered only the cooling process of the refrigerant temperature at the outlet of the evaporator from 15 °C to −10 °C. The shortest time in which the system achieved the target temperature of −10 °C was 107 s for test 3, on the contrary, test 4 failed to reach its target because of the condenser high pressure which activated the pressure switch and safety shut down of the compressor. Finally, the test for which the system took the longest time to achieve its target temperature was test 5, which needed 296 s. As a matter of fact, test 3 was done considering 100% condenser fan velocity, which permitted a proper heat transfer between the refrigerant and the environment in the condenser. On the other hand, considering 0% evaporator fan velocity, it can be emulated a situation of no cooling load, which permits decrease rapidly the refrigerant temperature at the evaporator outlet and the system can quickly reach the temperature target. In the case of test 4, with the evaporator and condenser fans working at 100% and 0% respectively, the cooling capacity required demands higher pressure levels in the condenser. These values increase until exceeding the limit prescribed for these tests, which was 1448 kPa. Finally, test 6, with both evaporator and condenser fans working at 100%, the system took longer time to reach the target temperature.

Similarly, Fig. 4 shows the values of power consumption of the compressor, but in the case where the working fluid is R413A. The shortest time for which the system achieved the target temperature of −10 °C was 134 s for test 9. The longest time was achieved by test 12 with 430 s . The system was unable to reach the objective in tests 7, 8, 10 and 11, because the compressor shut down by the action of the high pressure switch. From these results it can be remarked that the number of tests not completed with R413A is higher than when working with R12, which makes the R413A a less versatile refrigerant than R12 and also highly dependent on an adequate air-flow in the condenser.

Only tests 1, 2, 3, 5, 6, 9 and 12 could reach the target temperature of −10 °C. In this sense, only those seven runs were considered for the comparison by exergy analysis. Regarding the values of exergy, with Eq. (4) we obtained the values of the exergy destroyed in the compressor during the cooling process for tests described above. The average values of mass-flow rate in the tests were from 0.0135 kg/s to 0.0133 kg/s. It should be mentioned that tests 3 and 9 and also tests 6 and 12 were carried out at the same conditions. The only difference is the type of refrigerant. With the purpose of comparing the system performance with both refrigerants, tests 3 and 9 were selected.

Fig. 5 shows the values of exergy supplied to the system. Integrating the values of exergy supplied for each test with respect to time, were obtained 0.0133 kW h for test 3 and 0.021 kW h for test 9. The cooling process with R413A (test 9) was the one that required more energy. Fig. 6 shows the exergy destroyed in the system due to irreversibilities. Respect to the exergy destroyed values, integrating this values for each test with respect to time, were obtained 0.01 kW h for test 3 and 0.0144 kW h for test 9. Comparing this values, at the cooling process with R413A (test 9) was the one that destroyed more exergy to carry out the process. This is directly attributed to the duration of the tests.

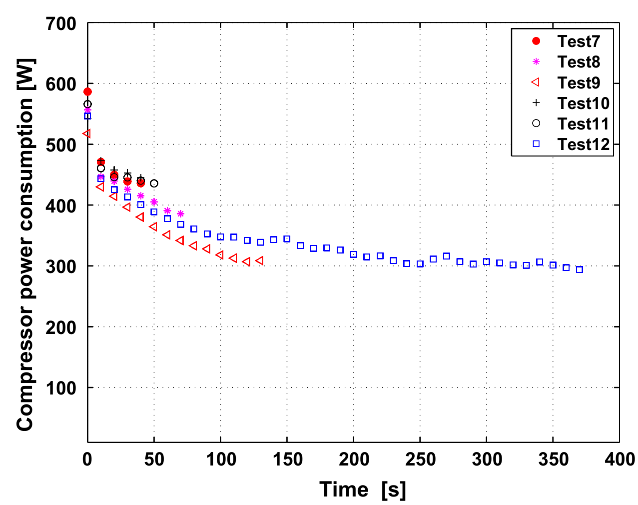

Fig. 4. Compressor power consumption working with R413A in the cooling process.

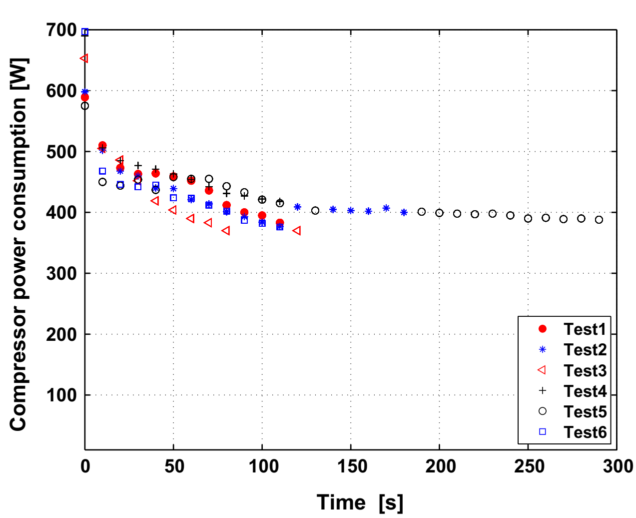

Fig. 3. Compressor power consumption working with R12 in the cooling process.

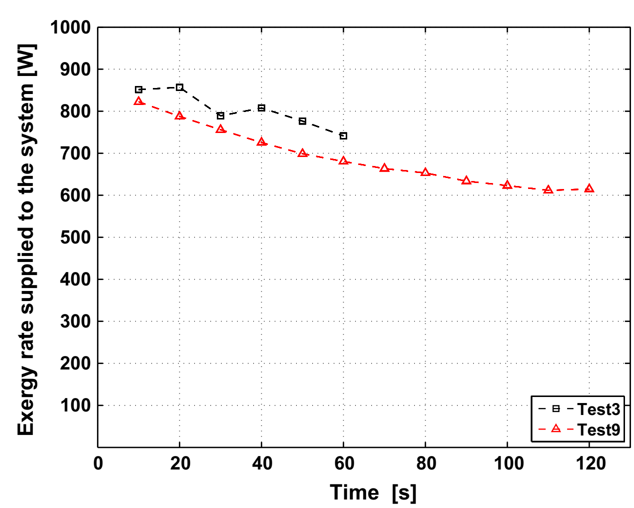

Fig. 5. Exergy rate supplied to the system in the cooling process.

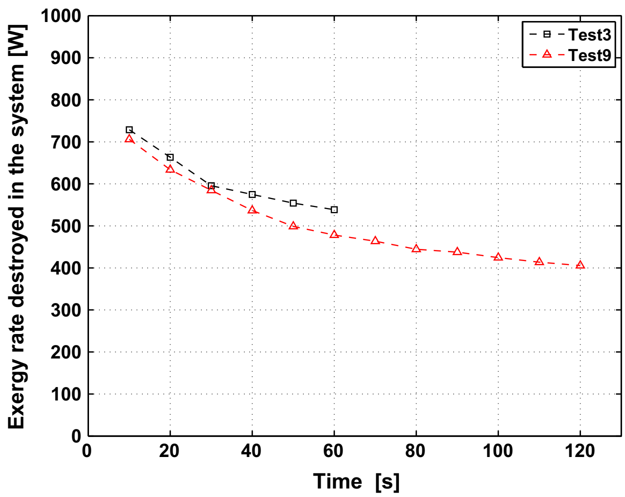

Fig. 6. System exergy rate destroyed in the cooling process.

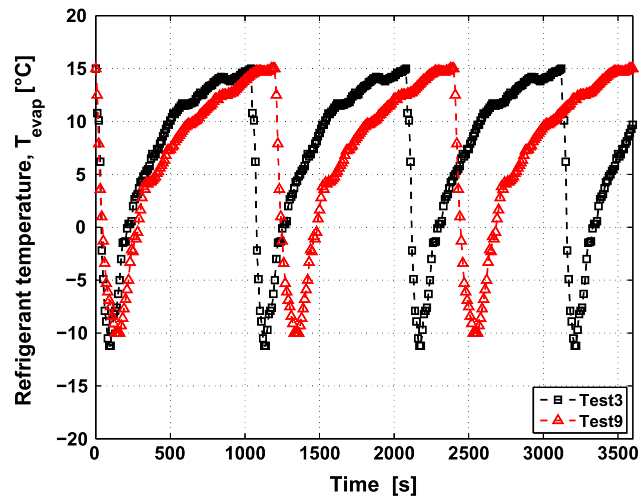

Fig. 8. Comparison of the hourly system operation cycle using R12 and R413A.

To characterize energy efficiency of this domestic refrigeration system, the value of the system COP was used. The values of COP were obtained integrating with respect to time, the evaporator duty heat transfer (W) and the compressor power consumption (W). The values of COP obtained from an energy analysis to the system are 3.53 for test 3 (R12) and 3.72 for test 9 (R413A). It can be observed that while the values of energy consumption are higher in test 9, the efficiency ratio of the amount of cooling provided by the evaporator to the energy consumed by the compressor in test 9 is higher than in test 3.

The values of exergy efficiency for tests 3 and 9 have been obtained by applying an exergy analysis to the system. Fig. 7 reports the exergy efficiency performance of the entire system. The overall exergy performance of R413A is better than of R12. This can be attributed to two causes: the first one is that when working with R413A, the system requires less power supply than working with R12, which is reflected in the values of supplied exergy for, with respect to test 3. The second one is that the exergy destruction values in test 9 are lower than those for test 3. The values of exergy destroyed in the evaporator for test 3 are up to 56.6% higher than the values obtained in test 9. The same was observed at the condenser, where exergy destroyed in test 3 reached up to 32.0% higher than the values obtained in test 9.

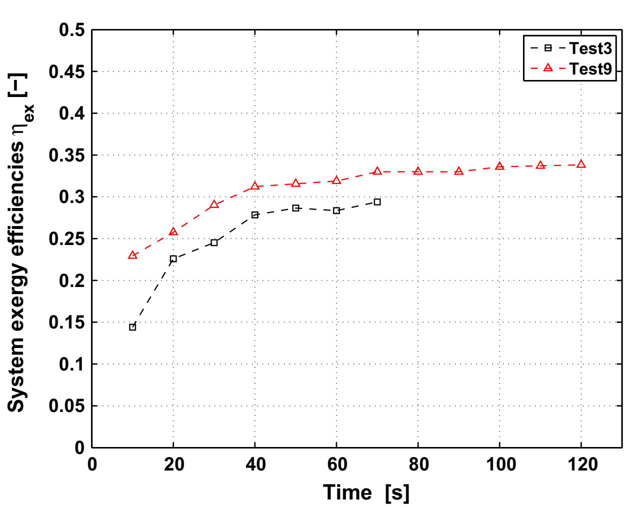

Fig. 7. Comparison of the system exergy efficiencies in the cooling process.

With the aim of knowing the system's annual performance, the terms of exergy destroyed and supplied for both refrigerants were summarized in a given period of time equivalent to one year. For this, it was considered the cooling process from 15 °C to −10 °C and the heating process from −10 °C to the initial temperature. For test 3, average values of 97 s and 940 s were measured for the cooling and heating process respectively. Respect to R413A, the average time recorded was 128 s and 1037 s. Fig. 8 shows the system operation cycle during 1 h. It can be seen that when the time arrives to 3600 s, the R12 system has performed 4 times the cooling process, while the R413A system only 3 times. Furthermore, when the time arrives to 24 h, the system working with R12 has run 84 times the cooling process, while working with R413A system has done 72. If this behaviour is extrapolated to a year, it can be obtained that the system working with R12 consumes 406 kW h per year, while the same working with R413A consumes 551 kW h per year. Under the same conditions, the exergy destroyed by the system working with R12 is around 302 kW h/year whereas that working with R413A destroys 380 kW h/year of exergy. If we primarily analyze these quantities, it can be concluded that R413A consumes more energy and destroys more exergy, nevertheless this comparison must be made using the values of exergy efficiency (Eq. (9)): 25.6% for the system working with R12 and 31.1% for that working with R413A. The conclusions are thus different and yield a better overall performance of the system working with R413A.

In addition, for the system using R12, the refrigerant charge was 0.49 kg while using R413A, the refrigerant charge was 0.46 kg. In summary, this corresponds to a save of 5% of refrigerant mass in the system. Refrigerant charge reduction being an important issue presently, this aspect has a real positive environmental impact.

5. Conclusions

An exergy analysis was conducted for a single evaporator domestic refrigerator between the evaporation and condensation temperatures range from −15 °C to 40 °C by using R12 and R413A refrigerants. Based on both experimental study and exergy analysis on the performance of R12 retrofit with R413A in domestic refrigerator, the following conclusions are drawn:

(1) Test 9 (R413A) had the shortest time in which the system achieved the target temperature of −10 °C when working with 0% of maximum fan velocity at evaporator and 100% of maximum fan velocity at the condenser. The system

was unable to reach their goal in four tests working with R413A which is higher than when working with R12, making the R413A a less versatile refrigerant than R12 and also highly dependent on an adequate ventilation air-flow in the condenser.

(2) The rate of exergy supplied to the compressor in the cooling process for R413A is always lower than that for R12, requiring less power supplied and improving its exergy performance. Regarding the exergy destroyed in the system due to the irreversibilities, for tests 3 and 6 we remark that the exergy destroyed performance through the cooling process was affected by the action of the evaporator electro-fan, which increases the heat transfer and also the exergy destroyed becoming less availability to produce useful work in the compressor.

(3) The average values of the system exergy efficiency for test 3 and 9 are 0.25 and 0.31 respectively. Those values show that the overall exergy performance of R413A is better than working with R12. It can be concluded that the system working with R413A requires less power consumption and it produces less irreversibilities. It also improves its overall performance.

Thus, it can be concluded that R413A could be an ozone friendly, exergy efficient and safe viable alternative to R12 for domestic and small commercial refrigeration systems with the main advantage that it can be replaced directly without the need to replace or modify any system component.

Acknowledgements

The authors acknowledge the support provided by the fundation Fondo Venezolano de Reconversión Industrial y Tecnológica (FONDOIN).

References

[1] Spauschus HO. HFC 134a as a substitute refrigerant for CFC 12. Int J Refrig 1988;11:389–92.
[2] Havelský V. Investigation of refrigerating system with R12 refrigerant replacements. Appl Therm Eng 2000;20:133–40.
[3] Richard RG, Shankland IR. Flammability of alternative refrigerants. ASHRAE J 1992;34:20–4.
[4] Richardson RN, Butterworth JS. The performance of propane/isobutane mixtures in a vapour-compression refrigeration system. Int J Refrig 1995;18:58–62.
[5] Camporese R, Bigolaro G, Bobbo S. Experimental evaluation of refrigerant mixtures as substitutes for CFC12 and R502. Int J Refrig 1997;20:22–31.
[6] Herbe L, Lundqvist P. CFC and HCFC refrigerants retrofit – experiences and results. Int J Refrig 1997;20:49–54.
[7] Hammad MA, Alsaad MA. The use of hydrocarbon mixtures as refrigerants in domestic refrigerators. Appl Therm Eng 1999;19:1181–9.
[8] Jung Dongsoo, Kim Chong-BO, Song Kilhong, Park Byoungjin. Testing of propane/isobutane mixture in domestic refrigerators. Int J Refrig 2000;23:517–27.
[9] Lee YS, Su CC. Experimental studies of isobutane (R600a) as the refrigerant in domestic refrigeration system. Appl Therm Eng 2002;22:507–19.
[10] Tashtoush B, Tahat B, Shudeifat MA. Experimental study of new refrigerant mixtures to replace R12 in domestic refrigerators. Appl Therm Eng 2002;22:495–506.
[11] Halimic E, Ross D, Agnew B, Anderson A, Potts I. A comparison of the operating performance of alternatives refrigerants. Appl Therm Eng 2003;23:1441–51.
[12] Mani K, Selladurai V. Experimental analysis of new refrigerant mixture as drop-in replacement for CFC12 and HFC134a. Int J Therm Sci 2008;47:1490–5.
[13] Janssen M, Engels F. The use of HFC134a with mineral oil in hermetic cooling equipment. ASHRAE J, Report 95403/NO 07: presented in the 19th international congress of refrigeration, The Hague; 1995.
[14] Sekhar SJ, Lal DM. HFC134a/HC600a/HC290 mixture a retrofit for CFC12 systems. Int J Refrig 2005;28:735–43.
[15] Aprea C, Greco A. An exergetic analysis of R22 substitution. Appl Therm Eng 2002;22:1455–69.
[16] McLinden MO, Klein SA, Lemmon EW, Peskin AP. NIST standard reference database 23, NIST thermodynamic and transport properties of refrigerants and refrigerant mixture. National Institute of Standards and Technology, REFPROP version 8.0; 2007.