INTERNATIONAL JOURNAL OF ENERGY RESEARCH
Int. J. Energy Res. 2003; **27**:653–669 (DOI: 10.1002/er.903)

Refrigeration plant exergetic analysis varying the compressor capacity

C. Aprea[1], F. de Rossi[2], A. Greco[3] and C. Renno[1,*,†]

[1] *Department of Mechanical Engineering, University of Salerno, Via Ponte Don Melillo 1, Fisciano 84084, Salerno, Italy*
[2] *Departement of Engineering, University of Sannio, Palazzo Bosco Lucarelli, C.so Garibaldi, Benevento 82100, Italy*
[3] *DETEC, University of Naples Federico II, P.le Tecchio 80, Naples 80125, Italy*

SUMMARY

The paper presents an exergetic analysis of a vapour compressor refrigeration plant when the refrigeration capacity is controlled by varying the compressor speed. The aim is performance evaluation of both the whole plant and its individual components. The analysis of the exergy flow destroyed in each device of the plant varying the compressor speed has been carried out in order to determine the relative irreversibility of the plant components. The vapour compression plant is subjected to a commercially available cold store. The compressor working with R22, R407C and R507 and designed for a revolution speed corresponding to 50 Hz supply current frequency, has been used varying the frequency in the range 30–50 Hz. In this range, the most suitable working fluids proposed as substitutes of R22, as R407C (R32/R125/R134a 23/25/52% in mass), R507 (R125/R143A 50/50% in mass) and R417A (R125/R134a/R600 46.6/50/3.4% in mass), have been tested. The variable-speed compressor is fitted with a pulse-width modulated source inverter (PWM) predominantly used in medium power applications due to its relatively low cost and high efficiency. The basic difference between variable speed refrigeration and conventional refrigeration systems is in the control of the system capacity at part-load conditions. The conventional refrigeration systems are characterized by compressor on/off cycles arising from by the thermostatic control. On the contrary when the inverter is used the capacity of the refrigeration system is matched to the load regulating the compressor motor speed. When the control of the compressor capacity is obtained by varying its speed there is an energy saving with respect to the thermostatic control. The best results of the exergetic analysis have been obtained using R22 followed by the non-azeotropic mixture designed as R407C that confirms, among the fluid candidates R22 substitution a better performance, shown also at the compressor nominal speed.

KEY WORDS: refrigeration system; exergetic analysis; variable speed compressor; R22 substitutes

1. INTRODUCTION

In this paper the results of the research work carried out to determine the exergetic performance of a vapour compression refrigeration plant linked to a typical cold store under variable loading condition are reported, using an inverter to vary the compressor speed and to match the

*Correspondence to: Dr. Carlo Renno, Department of Mechanical Engineering, University of Salerno, Via Pante Don Melillo 1, 84084 Fisciano (Salerno) Italy.
† E-mail address: crenno@unisa.it

Received 23 August 2002
Accepted 16 October 2002

refrigeration power required. The second law of thermodynamic and the exergy concept allow to establish the impact of each component performance on the overall irreversibility of the other components (Kotas, 1985; Bejan, 1988) when the compressor speed is reduced. The exergy is the maximum theoretical work obtainable when a system is brought in equilibrium with the environment. The maximum theoretical work obtainable from a rotating shaft is equal to the work transmitted by the shaft, while the maximum theoretical work obtainable from the heat flow is always smaller than the energy it contains. The exergy balances allow to determine for each component of the plant the exergy destroyed that, during the design stage, represents an important information. In fact it is useful to identify the components with the greatest losses so that further improvement can be studied to increase the plant performance. Considering the local exergy losses, an exergetic efficiency can be derived. In particular, the exergy destroyed due to the flow friction in the heat exchangers is negligible while the main reason of the energy losses is linked to the heat transfer between the fluids. The exergy destroyed in the compressor is caused by the refrigerant vapour warming at the inlet of the cylinder, the imperfection of the stuffing between the piston and the cylinder and the losses in the suction and discharge valves. These losses can rise or diminish when the speed is varied (McGovern and Harte, 1995; Stecco, 1986) The refrigeration power control obtained varying the compressor speed related to the refrigeration systems, has been studied during the last 20 years (Qureshi and Tassou, 1996; Pedersen *et al*., 1999; Jang and Jeong, 1999; Binneberg *et al*., 1999; Qureshi and Tassou, 1998). To modulate the compressor speed an inverter can be used. An inverter consists of a rectifier which converts the three-phase main voltage i.e., 380 V, 50 Hz to DC voltage and an inverter which inverts the DC voltage to AC supply voltage to the compressor motor which is adjustable in magnitude and frequency. There are different types of electronic variable-speed drives but the pulse-width modulated source inverter (PWM) is the most suitable for its low cost and high efficiency. Despite the potential advantages of this control of the refrigeration power, application to the commercial compressor has made little progress because some disadvantages seem to be present. The disadvantages are related to the inverter cost, some troubles linked to the lubrication of the compressor mechanism, its reliability and to some problems related to the correct working of the expansion devices. Referring to this aspect when the secondary fluids in the heat exchangers are in the gas-phase, as in the plant examined, this problem is negligible but it seems to be relevant when the secondary fluids are in the liquid-phase (Qureshi and Tassou, 1998). The most suitable working fluids such as R407C, R507 and R417A have been considered for the substitution of R22 which presents an ODP (ozone depletion potential) different from zero and therefore is destined to be phased out (Regulation [EC]; Aprea and Greco, 1998).

2. EXPERIMENTAL PLANT

The working fluids tested are the most common substitutes of R22; in particular they are R407C (R32/R125/R134a 23/25/52% in mass), R507 (R125/R143A 50/50% in mass) and R417A (R125/R134a/R600 46.6/50/3.4% in mass). The experimental vapour compression plant, subjected to a commercially available cold store and reported in Figure 1, is made up of a semihermetic reciprocating compressor, an air condenser followed by a liquid receiver, a manifold with two expansion valves, a thermostatic one and a manual one mounted in parallel, to feed an air cooling evaporator inside the cold store. The compressor, as declared by the

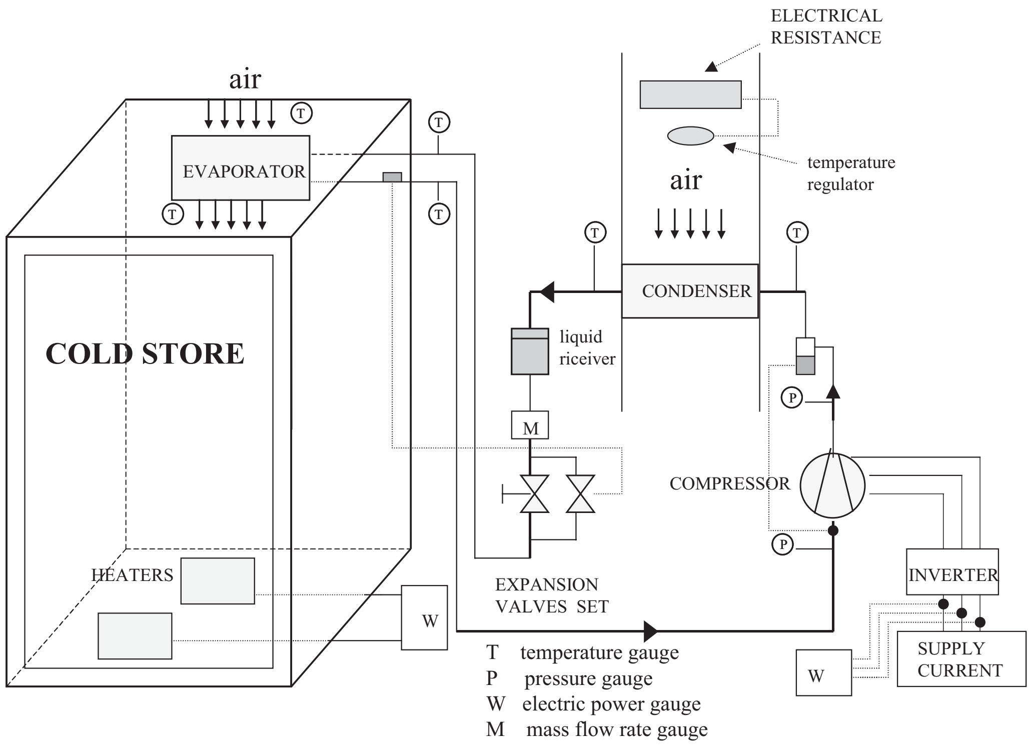

Figure 1. Sketch of the experimental plant.

manufacturer, can work with the fluids R22, R507 and R407C, it is filled with polyester oil and its speed is regulated by means of a PWM inverter. The manifold with both valves has been mounted to solve possible troubles because the expansion valves behaviour, when the compressor speed varies, is unknown (Qureshi and Tassou, 1998). The expansion valves used are specifically designed for R22, R407C and R507. No specific valve has been found for R417A for which the valve designed for R407C has been used, this because the R417A is a non-azeotropic mixture like the R407C. In the evaporation temperature range -20–10°C at a 35°C condensing temperature, working with R22 at the nominal frequency, the compressor refrigerating capacity varies in the range 1.4–4.4 kW. To fix the air temperature on the condenser and to simulate the external conditions, air is made to flow under the influence of a blower in thermally insulated channel where some electrical resistances are located. To obtain just the same temperature settled for the air, a regulator is used to control the electrical resistances supply. The refrigeration duty in the cold store is simulated by means of some electrical resistances linked to a regulator and the electric power is measured by means of a wattmeter. In Table I are reported the transducers specifications used; other specifications for the experimental plant are reported in a previous work (Aprea *et al.*, 2001). The test apparatus is equipped with 16 bit A/D converter acquisition cards linked to a personal computer that allows a high sampling rate monitoring all the measures carried-out by means of the transducers. The data acquisition software has been realized in a Labview environment.

Table I. Transducers specifications.

Transducers	Range	Uncertainty
Coriolis effect flowmeter	0–2 kg min^{-1}	$\pm$ 0.2%
RTD 100 4 wires	−100–500°C	$\pm$ 0.15°C
	1–10 bar	$\pm$ 0.2%
Piezoelectric absolute pressure gauge	1–30 bar	$\pm$ 0.5% F.S
Wattmeter	0–3 kW	$\pm$ 0.2%

3. DESCRIPTION OF THE TEST PROCEDURE

To evaluate the energetic performances when an inverter is used, it is necessary, before determining the most suitable control logic, to compare the compressor energetic consumption when the refrigeration power is obtained with on/off cycles of the compressor working at the nominal frequency of 50 Hz and when the current frequency selected at the compressor is such that the refrigerant mass flow rate and its enthalpy increase match exactly the refrigeration duty required. This step is essential before finding the best control logic that assigns for each cold store air temperature requested the most suitable frequency of the compressor supply current. The refrigeration duty, as above reported, is obtained by means of controllable electrical resistances located in the cold store. Referring to the outdoor conditions reported in the tests, to simulate summer conditions the air temperature at the condenser has been kept at 32°C thanks to a channel where the air is heated by means of an electrical resistance. The tests in the winter season have been performed fixing the outdoor air temperature at 10°C.The comparison tests have been carried out for air temperatures settled in the cold store by the thermostatic device equal to 5, 0, −5, −10 and −15°C and for each frequency of 30, 35, 40, 45, 50 Hz of the compressor supply electrical current. The exact refrigeration power that the compressor can supply at the selected frequency has been fixed for the electric heaters located in the cold store. The 30 Hz lower limit for the frequency has been selected to avoid troubles for the compressor lubricating by splash. Each refrigeration power selected has been proposed, for each value of the cold store air temperature considered, for the compressor supply current frequency of 50 Hz with the thermostat operating. The comparison is presented in terms of the plant exergetic efficiency considering, then, the exergy flow destroyed in each device. Each test has been conducted for two days and has been proposed for each of the working fluids considered. By means of Matlab the experimental values have been interpolated and the integral medium has been obtained.

4. EXERGETIC ANALYSIS

The exergetic analysis is a useful tool to establish the way to obtain the best performance, evaluating for each component of the plant the exergy destroyed and providing important information about the plant total irreversibility distribution among the plant components, determining what component makes it possible to raise the plant efficiency.

The overall plant exergetic efficiency has been evaluated as the ratio between the exergy output and the exergy input and can be expressed as

$$\eta_{ex} = \frac{\Sigma \dot{E}x_{out}}{\Sigma \dot{E}x_{in}} = 1 - \frac{\dot{E}x_{des}}{\Sigma Ex_{in}} \tag{1}$$

An accurate analysis can be realized evaluating the destroyed exergy for every single component of the plant. The thermodynamic properties of fluids tested have been evaluated using both a known computer program (Mc Linden *et al*., 1998) and a software created on purpose that has also been used to determine the energy, entropy and exergy balances and the compressor efficiencies.

The exergy flow destroyed in the evaporator is evaluated as:

$$\dot{\mathrm{E}}\mathrm{x}_{\mathrm{des,ev}} = \dot{m}_{\mathrm{ref}}\left(\mathrm{ex}_{\mathrm{in,ev}} - \mathrm{ex}_{\mathrm{out,ev}}\right) - \dot{Q}_{\mathrm{ev}}|\tau| \tag{2}$$

and the exergy flow destroyed in the condenser is evaluated as

$$\dot{\mathrm{E}}\mathrm{x}_{\mathrm{des,co}} = \dot{m}_{\mathrm{ref}}\left(ex_{\mathrm{in,co}} - ex_{\mathrm{co}}\right) - \dot{Q}_{\mathrm{co}}|\tau| \tag{3}$$

where the dimensionless exergetic temperature can be defined as

$$\tau = 1 - \frac{T_{\mathrm{o}}}{T_{\mathrm{MT_a}}} \tag{4}$$

where T_{o} is the environmental state and represents the dead state (Kotas, 1985).

The exergy flow destroyed in the compressor, neglecting the heat transfer with the environment, is evaluated as

$$\dot{\mathrm{E}}\mathrm{x}_{\mathrm{des,cp}} = \dot{m}_{\mathrm{ref}}\left(\mathrm{ex}_{\mathrm{in,cp}} - \mathrm{ex}_{\mathrm{out,cp}}\right) + \dot{L}_{\mathrm{cp}} \tag{5}$$

The exergy flow destroyed in the valve is evaluated as

$$\dot{\mathrm{E}}\mathrm{x}_{\mathrm{des,v}} = \dot{m}_{\mathrm{ref}}\left(ex_{\mathrm{in,v}} - ex_{\mathrm{out,v}}\right) \tag{6}$$

The efficiency defect has been evaluated for each device of the plant, considering the ratio between the exergy flow destroyed in each component and the exergy flow required to sustain the process, i.e. the electrical power supplied to the compressor:

$$\delta_i = \frac{\dot{\mathrm{E}}\mathrm{x}_{\mathrm{des,i}}}{\dot{L}_{\mathrm{cp}}} \tag{7}$$

The efficiency defects of the components are linked to the exergetic efficiency of the whole plant by means of the following relation:

$$\eta_{\mathrm{ex}} = 1 - \sum_i \delta_i \tag{8}$$

To characterize the behaviour of the two heat exchangers the efficiency has been evaluated. The latter is defined for the evaporator, considering the superheating negligible, as

$$\varepsilon_{\mathrm{ev}} = \frac{\dot{Q}_{\mathrm{ev}}}{Q_{\mathrm{max}}} = \frac{T_{\mathrm{in\ air,ev}} - T_{\mathrm{out\ air,ev}}}{T_{\mathrm{in\ air,ev}} - T_{\mathrm{in\ ref,ev}}} \tag{9}$$

The condenser efficiency is evaluated referring only to the condensing zone of this exchanger:

$$\varepsilon_{\mathrm{co}} = \frac{\dot{Q}_{\mathrm{co}}}{Q_{\mathrm{max}}} = \frac{T_{\mathrm{out\ air,co}} - T_{\mathrm{in\ air,co}}}{T_{\mathrm{in\ ref,co}} - T_{\mathrm{in\ air,co}}} \tag{10}$$

5. TEST RESULTS AND DISCUSSION

As the results in terms of energy saving both in winter and in summer are very similar, only the experimental tests performed in the summer season, with the air temperature at the condenser kept at 32°C and the cold store air temperature at 0°C, will be presented. However the tests in the winter season verify the correct refrigerant lamination when, varying the compressor speed, the compression ratio across the valve is low. Referring to the other cold store temperatures selected, the results are similar changing in only the absolute value of the exergy destroyed but remaining practically constant the relative exergetic performances presented by the working fluids considered. In Figure 2 a comparison of the exergetic efficiency of the whole plant, when the substitutes of R22 are used, is reported versus the current frequency feeding the compressor. The exergetic efficiency is particularly suitable to characterize the degree of thermodynamic perfection of the process. The exergetic efficiency of the whole plant is linked to the actual COP and to the reversible COP of the plant: $\eta_{ex} = COP/COP_{rev}$; because the COP_{rev} is fixed, referring to the temperatures reported in Figure 2, the exergetic efficiency following the trend of COP. It should be noted that when the compressor speed decreases the COP increases. When the compressor speed decreases, the refrigerant mass flow-rate as well as the condensation pressure diminish; the evaporation pressure presents a small increase when the compressor speed decreases. So the difference among the COP values, similar to those of the exergetic efficiency, for each fluid at the same compressor speed is mainly linked to the compression work and then to the compression ratio. It can be observed that the overall exergetic performance of the R22 is

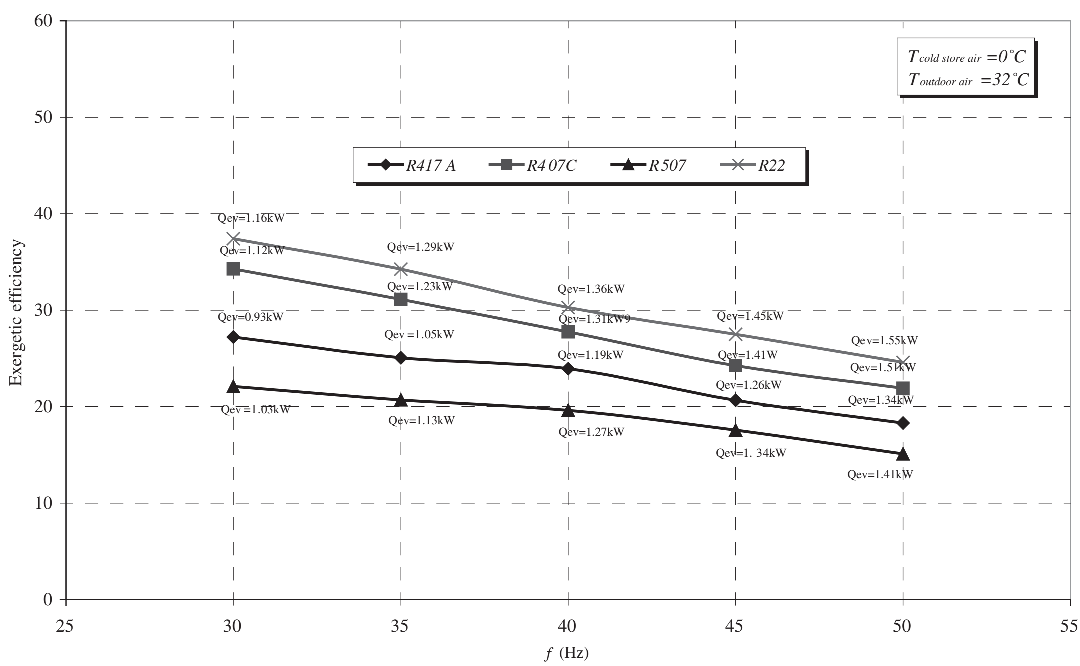

Figure 2. Exergetic efficiency related to R22 and its substitutes versus compressor motor supply current frequency.

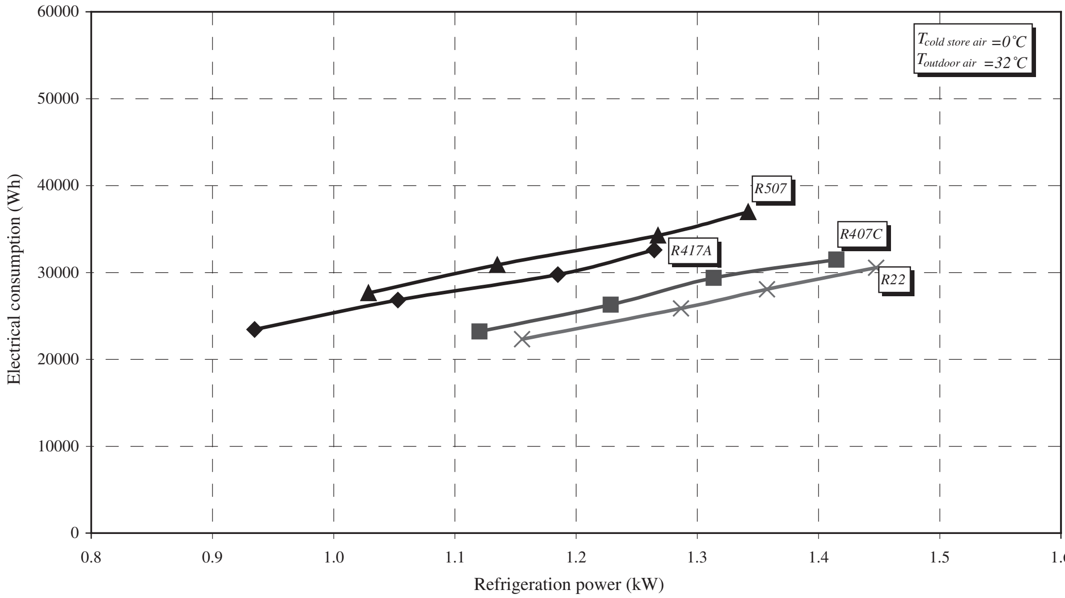

Figure 3. Electric energy consumption related to R22 and its substitutes at 50 Hz versus the refrigeration power.

significantly better than that of its substitutes decreasing the compressor speed as it happens at the compressor nominal speed. The non azeotropic mixture R407C seems to be the most suitable substitute for R22 also at lower frequencies. Referring to experimental tests it is clear that using the substitutes of R22 the exergy flow destroyed in the overall plant is greater than that related to the R22. It is interesting to note the more marked increase of the overall exergetic performance for R407C at a frequency lower than 40 Hz. Considering the plant running in this time, the maximum percentage decrease of the energy consumption is related to R407C among the R22 substitutes and is of about 12% at 30 Hz compared with working, at the same refrigeration duty, at 50 Hz with the thermostat working. In Figure 3 is shown the compressor electric energy consumption evaluated at the compressor supply current nominal frequency of 50 Hz and referring to the refrigeration power reported in Figure 2. Moreover, the energy saving medium, when the compressor works in the range 30–50 Hz, related to R407C, is of about 10%. Also referring to energy saving the best performance is related to the R22 followed by R407C.

As for the exergetic analysis in the Figures 4 and 5 the exergy flows destroyed in the compressor and in the evaporator are reported for all the fluids considered versus the current frequency feeding the compressor. To explain the trend of the exergy flow destroyed at the compressor, it is necessary to introduce its volumetric and isoentropic efficiencies. The factors that influence the volumetric efficiency are the pressure drop across the suction and discharge valves, the leakage past the rings of the piston and the leakage back through the suction and discharge valves. Moreover, it is necessary to observe that the suction gas entering the cylinder warms and expands increasing the specific volume which will hence be higher than the gas specific volume at the compressor inlet which represents the position considered when the volumetric efficiency is calculated (Stoecker and Jones, 1982). In addition, the motor electrical efficiency also plays a role in the refrigerant heating at the compressor suction because the

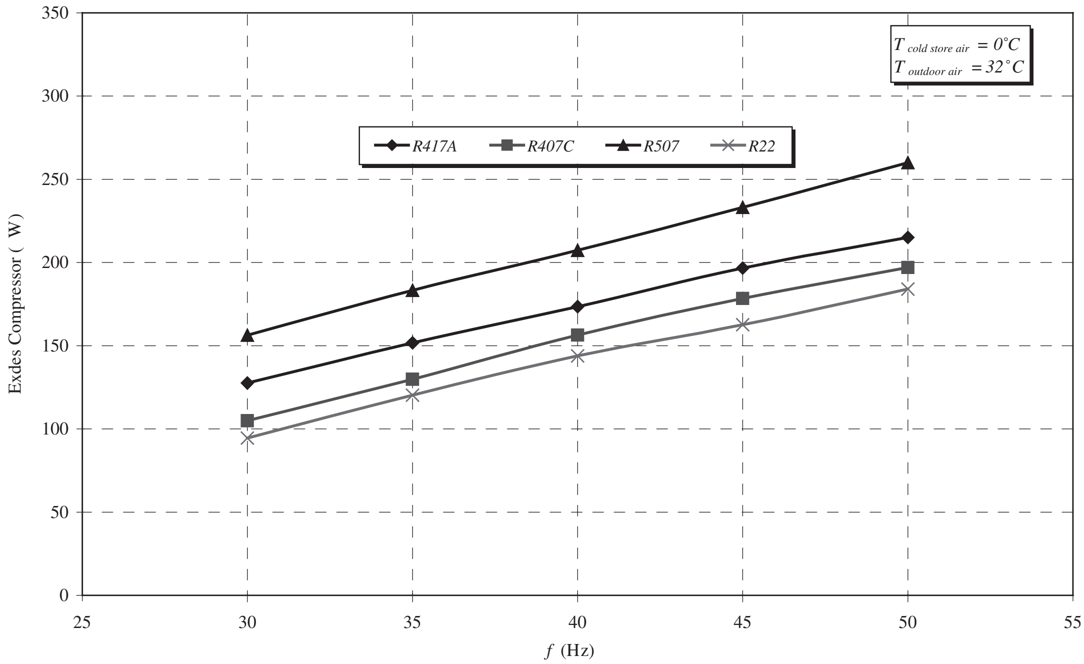

Figure 4. Exergy flow destroyed in the compressor related to R22 and its versus compressor motor supply current frequency.

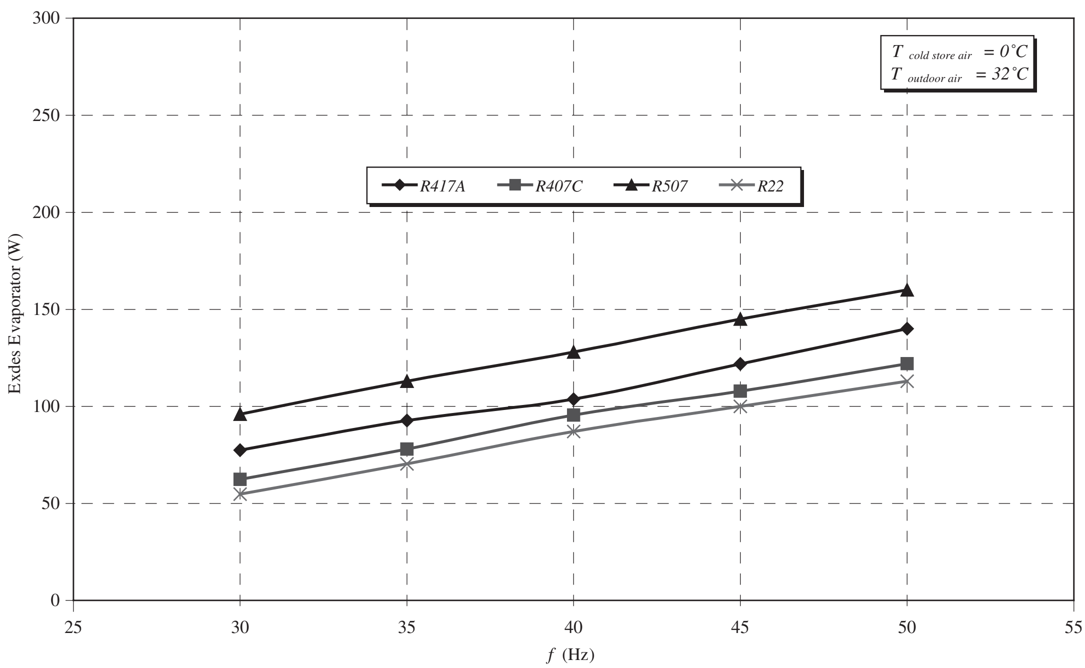

Figure 5. Exergy flow destroyed in the evaporator related to R22 and its substitutes versus compressor motor supply current frequency.

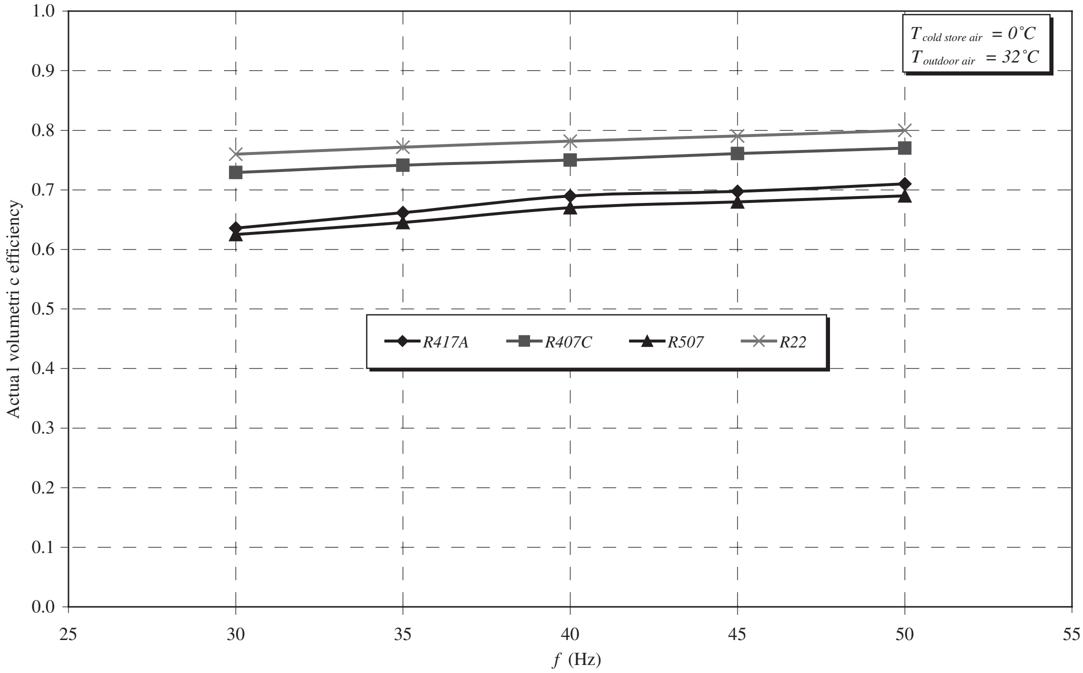

Figure 6. Actual volumetric efficiency related to R22 and its substitutes versus compressor motor supply current frequency.

semihermetic compressor electric motor is cooled by means of the refrigerant fluid. Figure 6 shows the actual volumetric efficiency for all the fluids versus the current frequency feeding the compressor. The volumetric efficiency decreases lightly when the compressor speed diminishes. One might expect a volumetric efficiency increase when the speed is reduced due to the smaller pressure drop in the valves and due to the reduced cylinder heating of the suction fluid as represented by some compressor indexes (Fornasieri, 1983; Stouff *et al.*, 2001). Moreover, no influences on the refrigerant superheating entering in the cylinder are related to the smaller efficiency of the electric motor as the speed is reduced (Krueger and Schwarz, 1994); this happens because, as evidenced by a specific index (Fornasieri, 1983), the cooling capacity of the fluids equalizes, practically, most of the heat is removed from the electric motor due to its smaller efficiency, which diminishes until 5% (Boyde, 2000), when the compressor speed decreases. Really the small decrease of the volumetric efficiency seems to be due to the leakage which takes place between the vanes and the cylinder wall arising from the reduced hydrodynamic sealing at lower speeds (Qureshi and Tassou, 1998). It should be noted that the lower mixture actual volumetric efficiency is also due to the smaller superficial tension of the polyester oil. Actually the latter does not result in such an effective seal between the cylinder and the piston as the one produced by the mineral oil (Boyde *et al.*, 2000). This drawback is confirmed by the higher lubricant oil consumption in the compressor associated with the use of mixtures without chlorine. For this reason an oil separator has been mounted on the plant. The decrease of the compressor volumetric efficiency allows a smaller refrigerant mass flow-rate and, then, a decrease of the exergy destroyed when the compressor speed decreases. In Figure 7 the

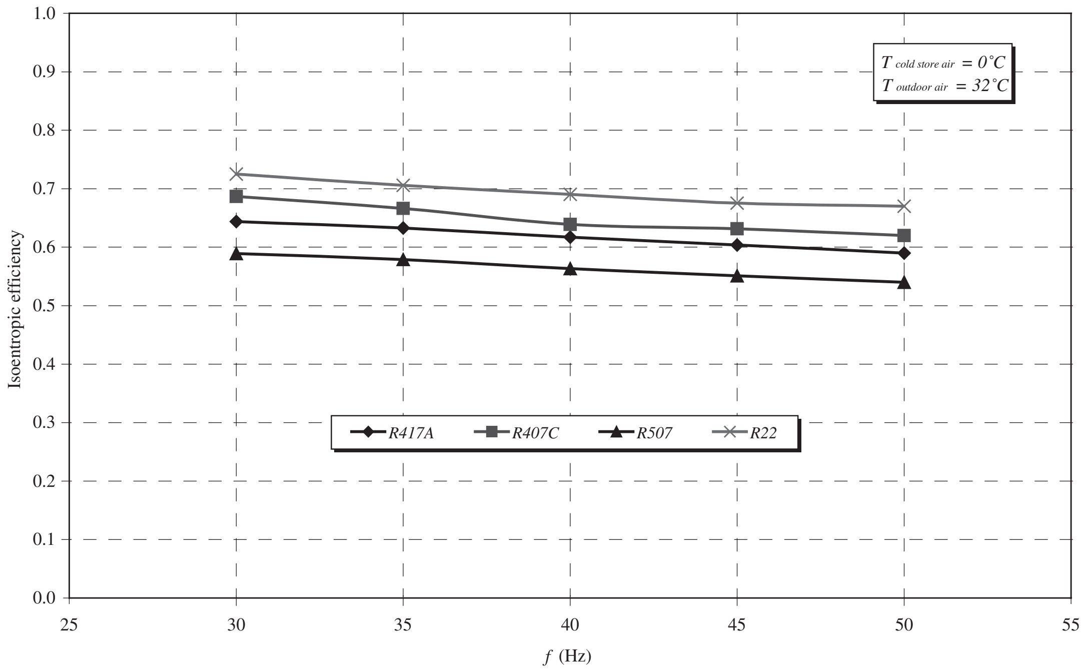

Figure 7. Isoentropic efficiency related to R22 and its substitutes versus compressor motor supply current frequency.

compressor isoentropic efficiency is reported for all the fluids versus the current frequency feeding the compressor. To evaluate the isoentropic efficiency the compressor has been insulated by means of one cm layer of cellular insulate. The isoentropic efficiency increases when the speed decreases because the discharge superheat reduces diminishing the compressor speed. Besides, no increase of the refrigerant superheating entering in the compressor cylinder is evidenced because, as the speed decreases, the valve loss is smaller as the cylinder heating of the suction gas and the fluids exhibit an increase in the cooling of the electric motor. The trend of the compressor efficiency is an additional contribution to the decrease of the exergy destroyed due to a smaller entropy change in the compression process. It should be noted that, when the compressor speed decreases, the compression ratio decreases too. The volumetric and isoentropic efficiencies can be provided with uncertainties respectively equal to $\pm 0.9\%$ and $\pm 1.1\%$ (ISO, 1999).

After all, considering the rising of the mechanical efficiency at a low revolution speed due to the smaller effect of the friction, the trend of the volumetric and isoentropic efficiencies and the smaller decrease of the electric motor efficiency (until 5%) when the speed decreases, the global compressor efficiency increases (Benamer and Clodic, 1998; Cohen *et al*., 1974). This increase represents a meaningful contribution to the energy saving presented above.

Comparing the behaviour of the R22 substitutes, as shown in Figures 6 and 7, for a frequency lower than 40 Hz, the decrease of the volumetric efficiency for the R407C is less marked. On the contrary, the isoentropic efficiency increase is more evident compared with the R507 and the R417A because the compressor discharge temperature and pressure are lower. The consequent

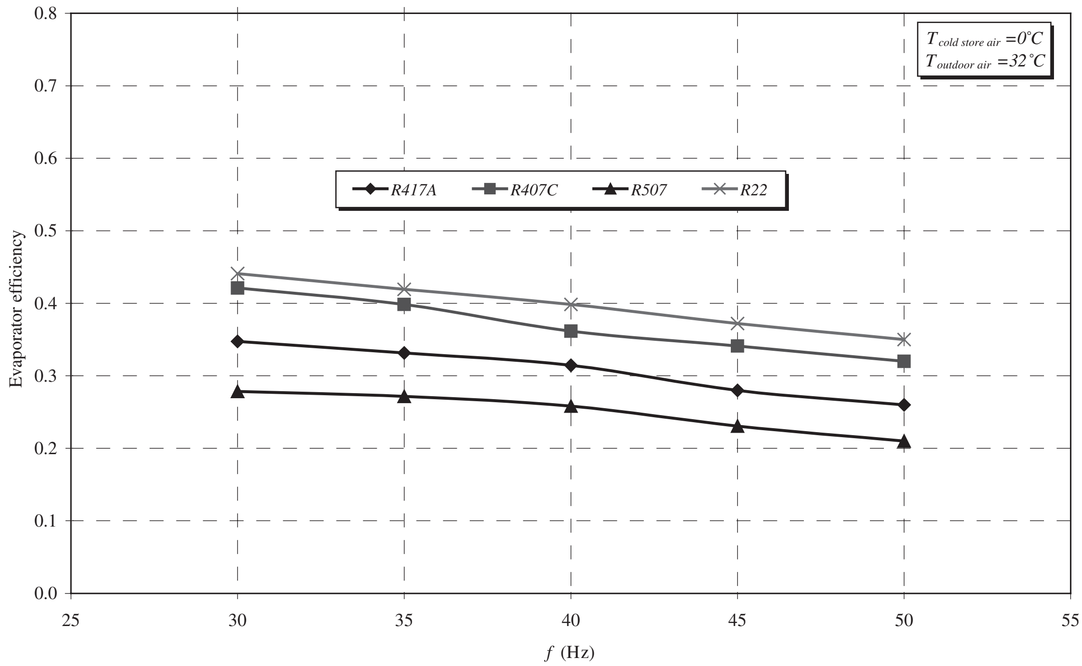

Figure 8. Evaporator efficiency related to R22 and its substitutes versus compressor motor supply current frequency.

rise of the refrigerant mass flow-rate and of the enthalpy increase at the evaporator, allow a major refrigeration power and a smaller compression work; for this reason the exergy flow destroyed for R407C decreases more than the other fluids at frequencies lower than 40 Hz.

As for the evaporator the destroyed exergy diminishes when the compressor speed decreases. This is due to the decrease of the compressor volumetric efficiency that allows a diminishing of the refrigerant mass flow-rate. Besides, decreasing the compressor speed, the exergy destroyed decreases because, when the refrigerant evaporation temperature rises, the mean temperature difference between the air and the refrigerant decreases in the evaporator. The best performance is related to R22 followed by R407C; for this mixture the decrease of the exergy destroyed is more marked than for the other two substitutes at a frequency lower than 40 Hz. Really, as the refrigerant mass flow-rate diminishes less than the other fluids, the evaporation pressure rises more than the other two fluids, determining a more meaningful decrease of the mean temperature difference between the air and the refrigerant at the evaporator. To characterise the evaporator in Figure 8 the efficiency is evaluated for R22 and for the other fluids considered. The shape of the curves in Figure 8 is a direct consequence of the destroyed exergy in this component. In fact decreasing the compressor speed, the evaporator irreversibility decreases diminishing the medium temperature difference between the air and the refrigerant at the evaporator, owing to the rise of the evaporation temperature; as a consequence the efficiency rises. It is possible to note the more marked increase of the evaporator efficiency, for R407C, at frequency lower than 40 Hz. In Figures 9 and 10 the exergy flow destroyed in the expansion valve and in the condenser is reported, respectively, for all the fluids considered versus the

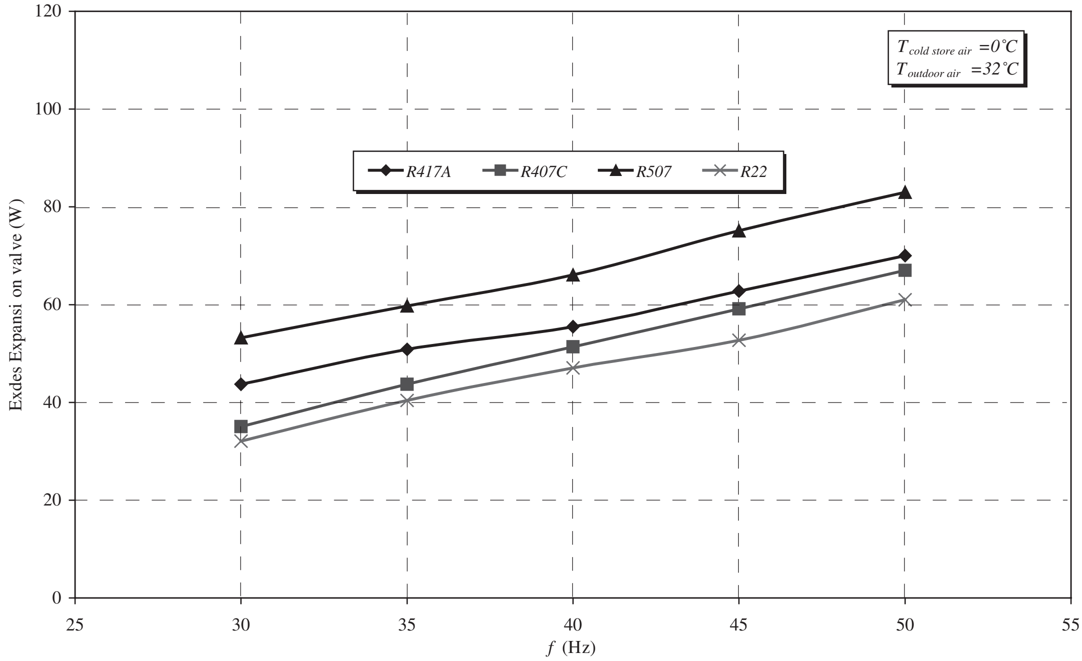

Figure 9. Exergy flow destroyed in the expansion valve related to R22 and its substitutes versus compressor motor supply current frequency.

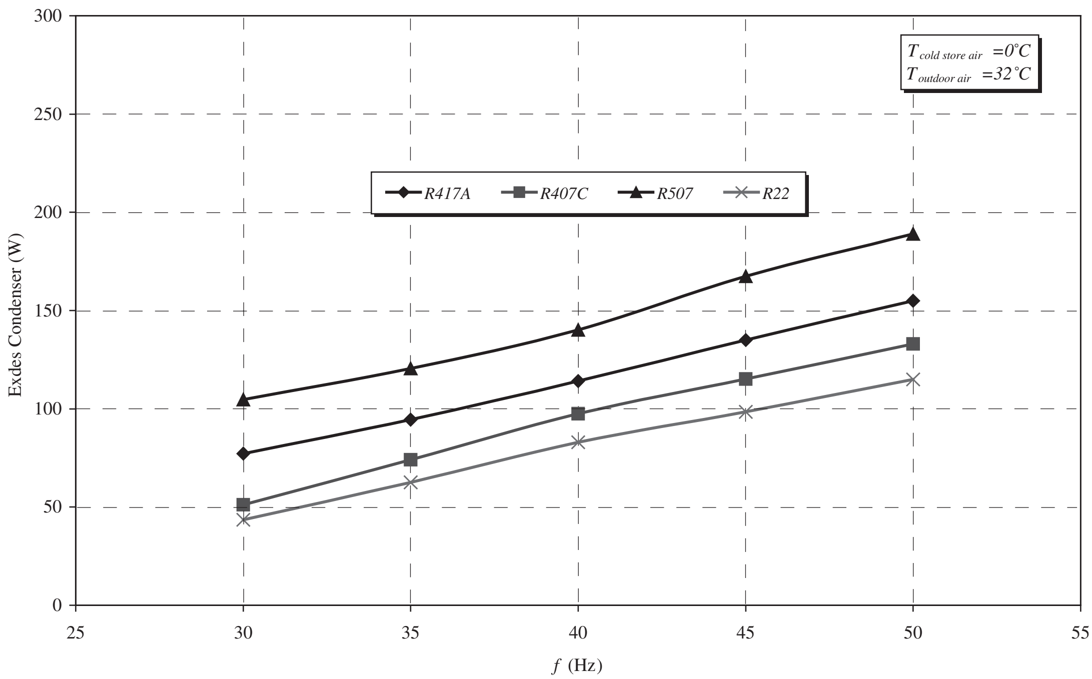

Figure 10. Exergy flow destroyed in the condenser related to R22 and its substitutes versus compressor motor supply current frequency.

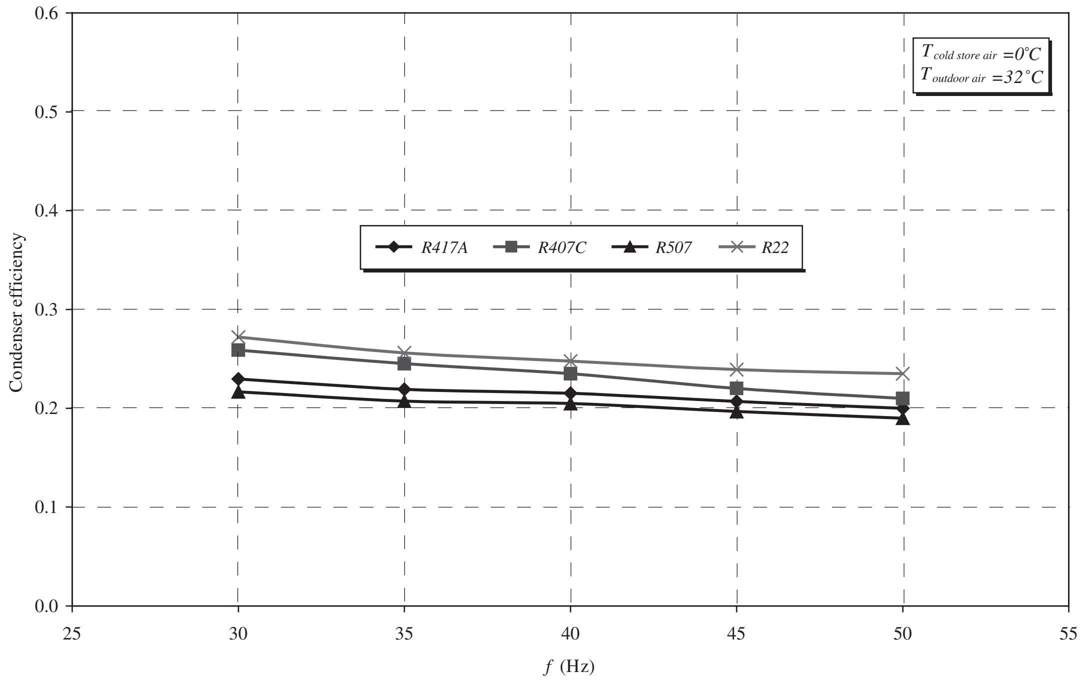

Figure 11. Condenser efficiency related to R22 and its substitutes versus compressor motor supply current frequency.

compressor speed. The irreversibility of the expansion valve decreases when the compressor speed decreases. This is linked to the compression ratio decrease due to the evaporator pressure increase and to the condensation pressure decrease when the compressor speed diminishes. The more marked decrease showed by R407C at a frequency lower than 40 Hz, is due to a more meaningful compression ratio decrease. Referring to the condenser, the destroyed exergy, at a fixed flow rate and air temperature decrease when the compressor speed is reduced. This happens because decreasing the compressor speed the refrigerant mass flow-rate and then the condensing pressure decrease; the mean temperature between the air and the refrigerant decreases and as the exergy destroyed is smaller. For R407C at a frequency lower than 40 Hz similar considerations presented for the evaporator can be carried out showing a condensation pressure lower than the other two substitutes. In Figure 11 the condenser efficiency is evaluated for all the fluids examined with reference to the condensing zone of the heat exchanger. The trend is explainable with the same observation reported for the exergy destroyed in the component. A similar trend is shown by R407C for a frequency lower than 40 Hz. To compare the performance of the fluids, the plant component efficiency defects are reported in Figures 12 and 13. In Figure 12 the compressor and the evaporator are compared, while in Figure 13 the efficiency defects of the condenser and of the valve are shown. The general behaviour can be explained, since the mechanical power input to the compressor decreases less than the exergy destroyed in the component when the compressor speed decreases; the smaller compression work is due to a smaller compression ratio when the compressor speed diminishes. Obviously the best performance is related to R22 followed by R407C that, as noted, at frequency lower

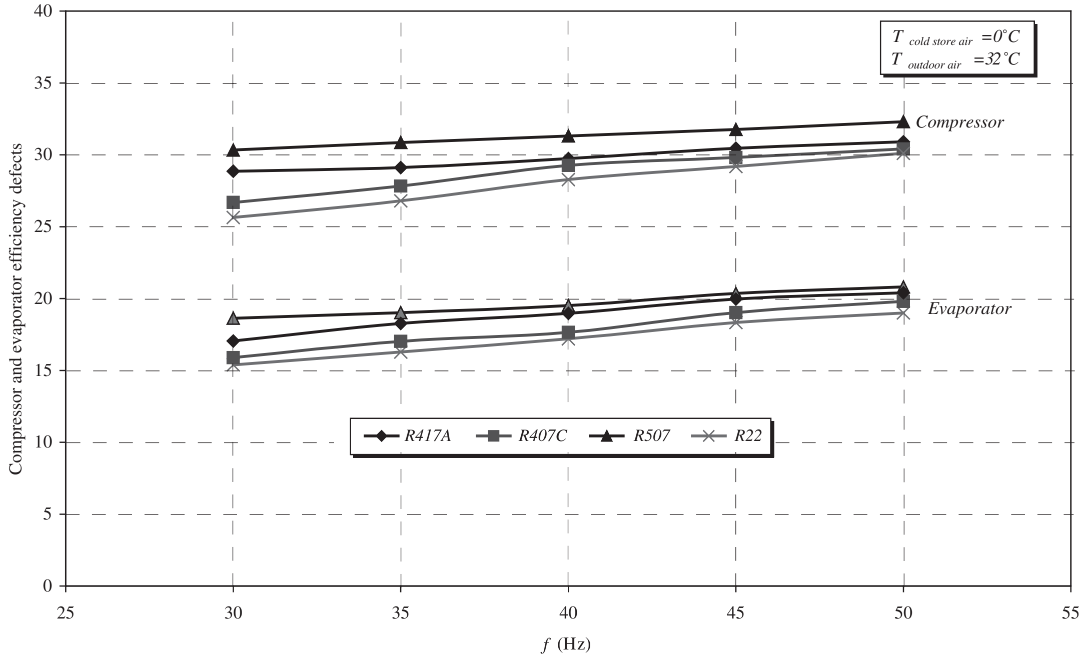

Figure 12. Compressor and evaporator efficiency defects related to R22 and its substitutes versus compressor motor supply current frequency.

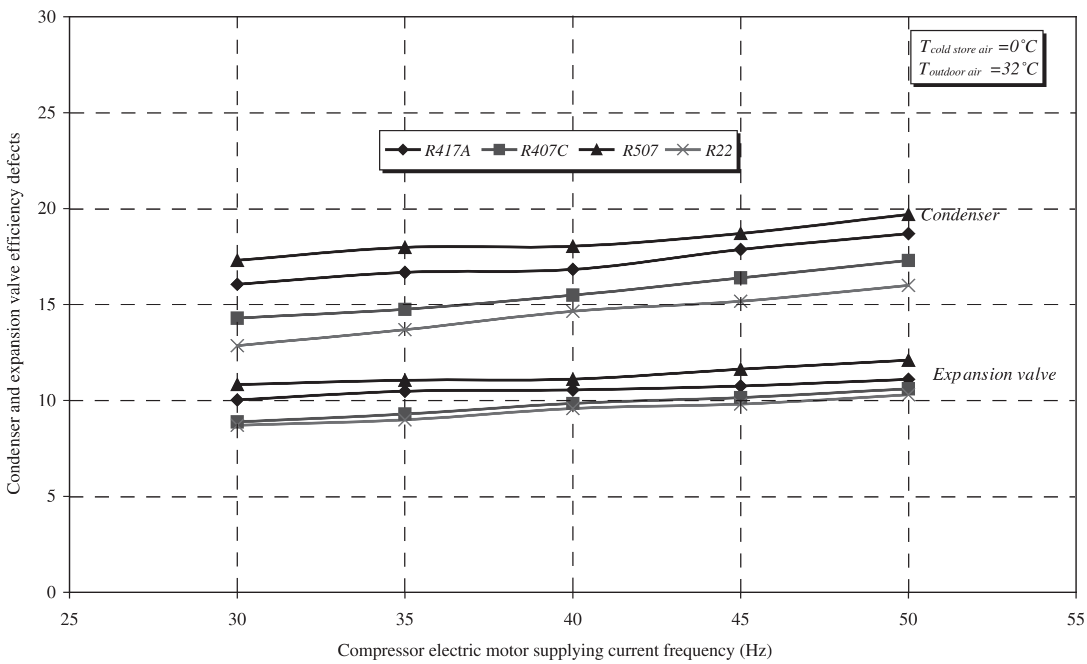

Figure 13. Condenser and valve efficiency defects related to R22 and its substitutes versus compressor motor supply current frequency.

than 40 Hz shows a more marked decrease in the efficiency defect than the other two substitutes. In order to increase the overall plant performance, the compressor and both the heat exchangers must be optimised because of their higher efficiency defects while the contribution to the irreversibility of the valve is marginal.

6. CONCLUSIONS

This paper deals with the substitution of R22 in a vapour refrigeration compression plant in the presence of a variable speed compressor. The substitutes of R22 considered are R407C, R507 and the R417A. The analysis carried out in this paper follows an exergetic approach. The performance both of whole plant and of its individual components has been presented. The analysis of the exergy flow destroyed in each device of the plant varying the compressor speed has been carried out, in order to pinpoint the greatest contribution to the exergetic performance decrease. The results obtained allow to establish that R407C is the most suitable substitute of R22 also in variable speed applications. However the best performance is related to R22. The best performance of R407C, compared with R507 and R417A, obtainable with the compressor at its nominal speed corresponding to a 50 Hz supply current frequency, has been observed also at lower frequencies; besides, at frequencies lower than 40 Hz, R407C improves its relative performance compared with R507 and R417A. The results obtained allow the following remarks:

1. The overall exergetic performance of the plant working with R22 is consistently better than that of its substitutes; among the candidate substitutes, also in variable speed applications, the best performance is related to the non-azeotropic mixture R407C.
2. The contribution of the compressor to the overall irreversibility is the most relevant. Its exergetic performance has been correlated with the variation of its volumetric and isoentropic efficiencies versus the compressor speed. In this component the best performance is related to R22; among the substitutive fluids the best is R407C.
3. For the evaporator and the condenser the lower exergetic performance of all the substitutes of R22 is a direct consequence of the higher temperature difference between the refrigerant fluid and the secondary fluid in both the heat exchangers. The lower difference between the refrigerant and the secondary fluid is presented by R407C that shows a favourable trend at a mass flow-rate corresponding to the compressor speeds obtained with current frequencies lower than 40 Hz.
4. The contribution of the expansion valve to the overall irreversibility is marginal. In this case the differences among the fluids are less marked.

NOMENCLATURE

COP = coefficient of performance
ex = exergy
f = compressor electric motor supply frequency (Hz)

$\dot{Q}$ = refrigeration power (W)
T = temperature (°C)

Greek symbols

δ = efficiency defect
η = efficiency

Subscripts

a = air
co = condenser
cp = compressor
des = destroyed
ev = evaporator
ex = exergetic
in = inlet
out = outlet
ref = refrigerant
v = valve

REFERENCES

Aprea C, Greco A. 1998. An experimental evaluation of the greenhouse effect in R22 substitution. *Energy Conversion and Management* **39**(9):877–887.

Aprea C, Mastrullo R, Renno C. 2001. Refrigeration power control in variable-speed compressor applications. *Proceedings of the 7th World Congress CLIMA 2000*, Naples, Italy.

Bejan A. 1988. *Advanced Engineering Thermodynamics*. Wiley: New York.

Benamer A, Clodic D. 1998. AFF, ELEC; 53–66.

Binneberg P, Philipp J, Krauss WE. 1999. Variable-speed hermetic compressor in a household refrigerator. *Proceedings of the 20th International Congress of Refrigeration*, Sydney, Australia, paper code 539.

Boyde S, Randles S, Gibb P *et al.* 2000. Effect of lubricant properties on efficiency of refrigeration compressors, *Proceedings of the 2000 International Compressor Engineering Conference*, Purdue University, US, vol.1, 311–317.

Cohen R, Hamilton JF, Pearson JT. 1974. Possible energy conservation through the use of variable capacity compressor. *Proceedings Purdue Compressor Technology Conference*, Purdue, US; 50–55.

Fornasieri E. 1983. Analisi del processo di compressione dei fluidi frigorigeni mediante compressori alternativi. *Il Freddo* **37**(3):163.

ISO, Guidance to the expression of uncertainty in measurement, UNI CEI ENV 13005, 1999.

Jang K, Jeong S. 1999. Temperature heat flux measurement inside variable-speed scroll compressor. *Proceedings of the 20th International Congress of Refrigeration*, Sydney, Australia, paper code 109.

Kotas KS. 1985. *The Exergy Method of Thermal Plant Analysis*. Butterworths: London.

Krueger M, Schwarz M. 1994. Experimental analysis of a variable-speed compressor. *Proceedings of the International Computer Engineering Conference*, vol. 2, Purdue University, US; 599–604.

Mc Linden M *et al.*, 1998. NIST Standard Reference Database 23: Refprop 6, computer software, U.S. Department of Commerce, Technology Administration, National Institute of Standard and Technology, Gaithersburg.

McGovern JA, Harte S. 1995. An exergy method for compressor performance analysis. *International Journal of Refrigeration* **18**(6):421–433.

Pedersen PH, Poulsen CS, Gundtoft S. 1999. Refrigerators and freezers with variable speed compressors. *Proceedings of the 20th International Congress of Refrigeration*, Sydney, Australia, paper code 153.

Quereshi TQ, Tassaou SA. 1998. Comparative performance evaluation of positive displacement compressors in variable-speed refrigeration applications. *International Journal of Refrigeration* **21**(1):29–41.

Qureshi TQ, Tassou SA. 1996.Variable-speed capacity control in refrigeration systems. *Applied Thermal Engineering* **16**(2):103–113.

Regulation [EC] No 2037/2000 of the European Parliament and of the Council of 29 June 2000 on substances that deplete the ozone layer, vol. 43. Off. J. Eur. Communities, Legis, LU, 29 September 2000, L244; 4-25 (7 append.)

Stecco SS. 1986. Exergy analysis of compression and expansion process. *Energy* **11**(6):573–577.

Stoecker WF, Jones JW. 1982. *Refrigeration and Air Conditioning*. Mc Graw-Hill Book: New York.

Stouffs P, Tazerout M, Wauters P. 2001. Thermodynamic analysis of reciprocating compressors. *International Journal of Thermal Sciences* **40**:52–66.

