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An evaluation of R22 substitutes performances regulating continuously the compressor refrigeration capacity

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Abstract

This paper presents the results of an experimental analysis which compares in terms of energetic performances the refrigeration capacity control obtained by means of a variable-speed compressor with the on/off control deriving from a classical thermostatic device. The compressor considered is semi-hermetic reciprocating and is a component of a vapour compression refrigeration plant subjected to a commercially available cold store. The compressor working with the fluids R22, R507 and R407C and designed for a revolution speed corresponding to the compressor supply current nominal frequency of 50 Hz, has been tested varying the frequency in the range 30–50 Hz. In this range, the most suitable working fluids proposed as substitutes of the R22 as the R407C (R32/R125/R134a 23/25/52% in mass), the R507 (R125/R143A 50/50% in mass) and the R417A (R125/R134a/R600 46.6/50/3.4% in mass) have been tested. The results show that, using the R407C, it is possible an average an electric energy consumption about 12% smaller when an inverter is employed to control the compressor refrigeration capacity instead of the thermostatic control which imposes on/off cycles on the compressor, working at the nominal frequency of 50 Hz. So the R407C confirms its superiority in comparison with the R417A and R507; only the R22 shows a better performance.

Keywords: Refrigerating system; Variable speed compressor; Energy saving; R22 substitutes

1. Introduction

The vapour compression refrigeration systems, though designed to supply the maximum load, operate at part-load for much of their life, regulated by on/off compressor cycles imposed by a

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Nomenclature

COP	coefficient of performance
ev	evaporator
f	compressor electric motor supply frequency (Hz)
$\dot{Q}$	refrigeration capacity (W)
T	temperature (°C)

thermostatic control which determines a high energetic consumption. Theoretically the most efficient method of refrigeration capacity control is the variable speed compressor which continuously matches the compressor refrigeration capacity to the load.

The refrigeration capacity control obtained varying the compressor speed with reference to refrigeration systems, has been studied during the last 20 years [1–8]. To regulate the compressor speed an inverter can be used. It is formed by a rectifier that converts the three-phase mains voltage, i.e. 380 V, 50 Hz to DC voltage and by an inverter that inverts the DC voltage to a three phase AC supply voltage to the compressor motor. At the output of the inverter the voltage is adjustable in frequency and magnitude. 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. Though the application of this type of control of the refrigeration capacity to a commercial compressor presents some potential advantages, it has made little progress because some disadvantages seem to be present. In particular, the disadvantages are relative to the inverter cost, to some troubles linked to the compressor lubrication and reliability and besides to some problems related to the correct working of the expansion devices [9]. Referring to this aspect if the secondary fluids in the heat exchangers are in 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 [10].

In this paper are reported the results of the research work aimed to evaluate the energetic performances of a vapour compression refrigeration plant, subjected to a typical cold store, under variable loading conditions using an inverter to match the refrigeration capacity required. The most suitable working fluids as R407C, R507 and R417A have been considered for the substitution of the R22 which presents an ozone depletion potential (ODP) different from zero and therefore is destined to be phased out [11]. In order to verify if the substitutive fluids maintain their peculiar performances, the same fluids have been experimentally tested when the compressor electric motor supply current frequency varies in the range 30–50 Hz, and compared with the results obtained when the frequency is the nominal 50 Hz one.

2. Experimental plant

The working fluids tested are the most common substitutes of R22; in particular they are the R407C (R32/R125/R134a 23/25/52% in mass), the R507 (R125/R143A 50/50% in mass) and the 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 Fig. 1, is made up of a semi-

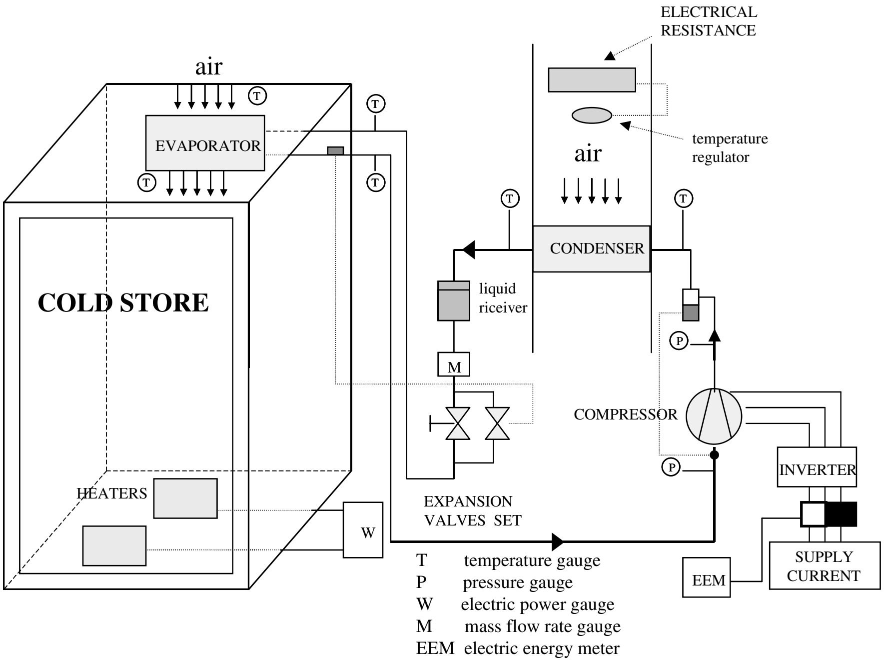

Fig. 1. Sketch of the experimental plant.

hermetic 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 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 [10]. The expansion valves used are specifically designed for the R22, R407C and R507. No specific valve has been found for the R417A for which the valve designed for the R407C has been used, this because the R417A is a non azeotropic mixture like the R407C. In the evaporation temperature range −20 to 10 °C at a 35 °C condensing temperature, working with the R22 at the nominal frequency, the compressor refrigeration capacity varies in the range 1.4–4.4 kW. To fix the air temperature on the condenser and to simulate the external conditions, the air flows under the influence of a blower in a 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 cooling load in the cold store is simulated by means of some electrical heaters linked to a regulator and the electric power is measured by means of a wattmeter. In Table 1 are reported the transducers specifications used (Coriolis effect flowmeter, RTD 100 4 wires thermoresistances, electric energy meter, piezoelectric absolute pressure gauge, wattmeter). The test apparatus is equipped with 32 bit A/D acquisition cards linked to a personal computer allowing a high sampling rate and a monitoring of all the measures carried out by means of the transducers. The data acquisition software has been realized in a Labview environment. The thermodynamic properties of the fluids tested have been evaluated using a software created on

Table 1
Transducers specifications

Transducers	Range	Uncertainty
Coriolis effect flowmeter	0÷2 kg/min	±0.2%
RTD 100 4 wires	−100 to 500 °C	±0.15 °C
Piezoelectric absolute pressure gauge	1÷10 bar, 1÷30 bar	±0.2%, ±0.5% F.S
Wattmeter	0÷3 kW	±0.2%
Electric energy meter	5÷40A	±2%

purpose that has been also used to determine the energy balances and the quantities presented below.

3. Experimental procedure description

To evaluate the energetic performances when an inverter is used, it is necessary to compare the compressor energetic consumption when the cooling load is matched with on/off cycles of the compressor, working at the nominal frequency of 50 Hz, and when the cooling load is matched varying the compressor speed by means of an inverter. 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 cooling load, as above reported, is obtained by means of controllable electrical heaters 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, and 50 Hz of the compressor motor supply electrical current. The 30 Hz lower limit for the frequency has been selected to avoid troubles for the compressor lubricating by splash. The exact refrigeration capacity that the compressor can supply at the frequency selected by means of an inverter has been fixed for the electric heaters located in the cold store and has been considered as cooling load in the comparison between the compressor working at the nominal frequency of 50 Hz and the compressor working at the frequency selected by means of the inverter. The comparison is presented in terms of electric energy consumption and the most significant values are reported to understand how the plant operates. Each test has been conducted for two days and has been proposed for each of the working fluids considered.

4. 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 only the energy saving absolute value but remaining practically constant the relative performances presented by the working fluids considered. In Fig. 2 the COP versus the compressor supply current frequency for the working fluids considered is reported when the cold store air temperature is 0 °C and the air temperature at the condenser is 32 °C. It should be noted that on decreasing the compressor speed the COP increases. On decreasing the compressor speed the refrigerant mass flow rate diminishes and the condensation pressure too while the evaporation pressure presents a small increase; it follows that the compression ratio diminishes on decreasing the compressor speed. So the difference among the COP values for each fluid at the same compressor speed is mainly linked to the compression work and then to the compression ratio as shown in Fig. 3. In these tests for each frequency and working fluid, the cooling load, selected by means of the electrical heaters in the cold store, matches exactly the refrigeration capacity supplied by the compressor. The COP values calculated should be considered with an uncertainty smaller than ±0.5% when the refrigerating power is evaluated as the product between the refrigerant enthalpy increase and its mass flow rate and when the compressor electric power is contemplated. The compression ratio can be provided with uncertainties equal to ±0.7% [12].

In Fig. 4 the refrigeration power, used to evaluate the COP, is reported. In Fig. 5 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 Fig. 4; in this

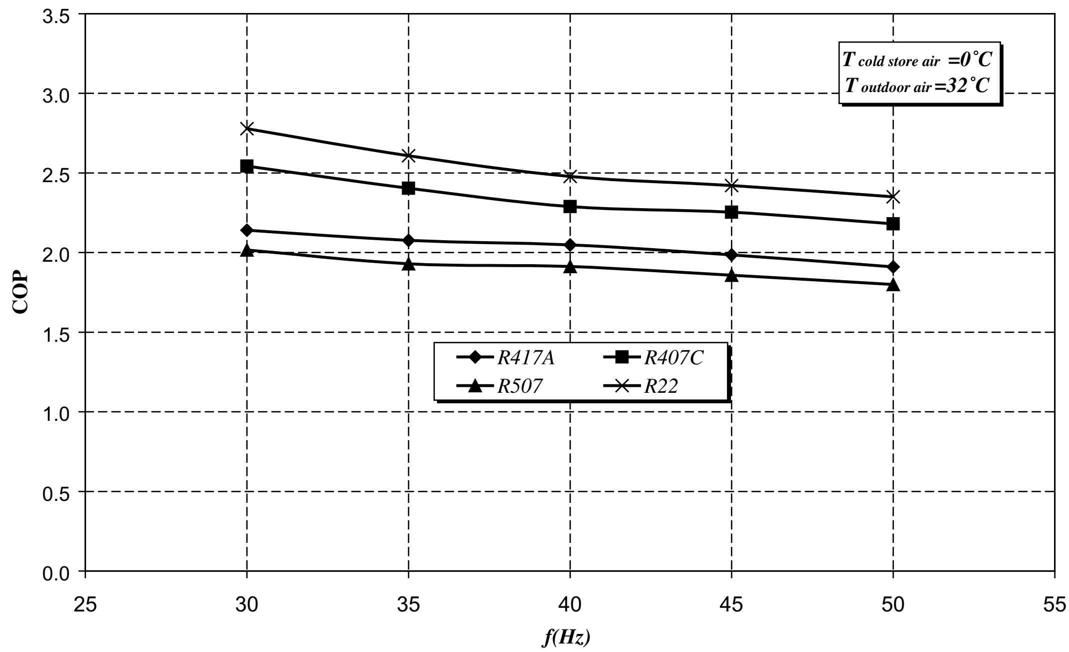

Fig. 2. Plant COP related to R22 and its substitutes versus compressor motor supply current frequency.

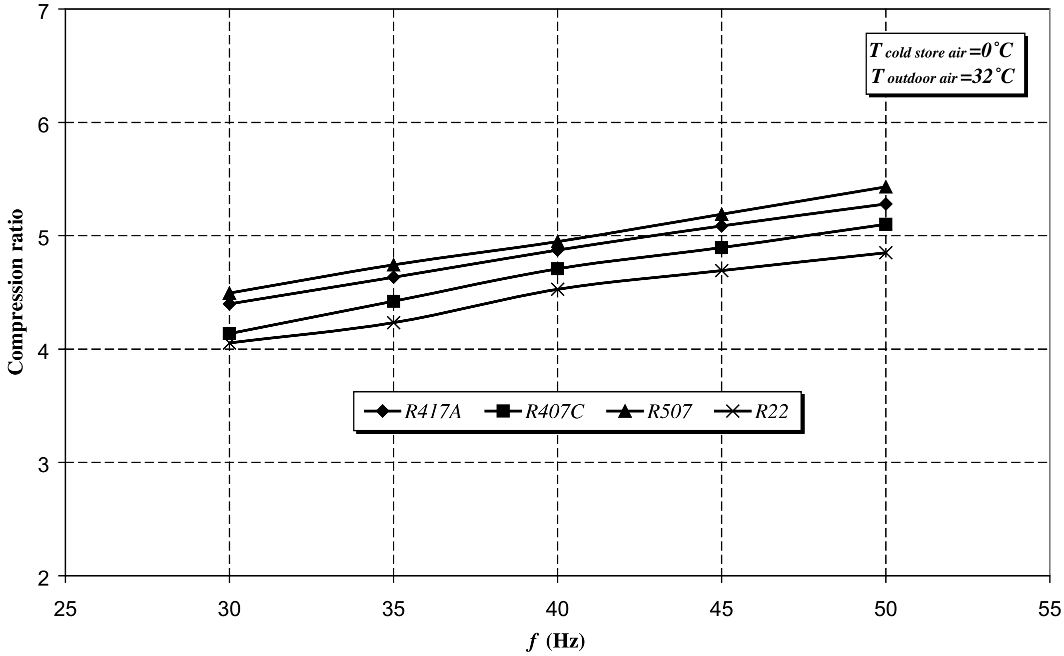

Fig. 3. Compression ratio related to R22 and its substitutes versus compressor motor supply current frequency.

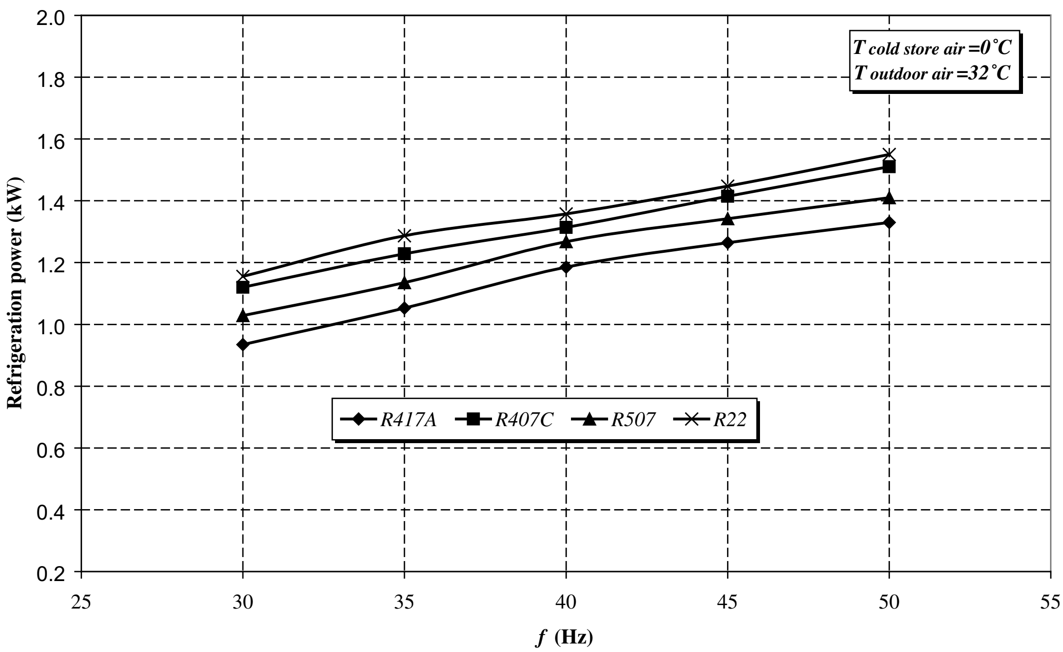

Fig. 4. Refrigeration power related to R22 and its substitutes versus compressor motor supply current frequency.

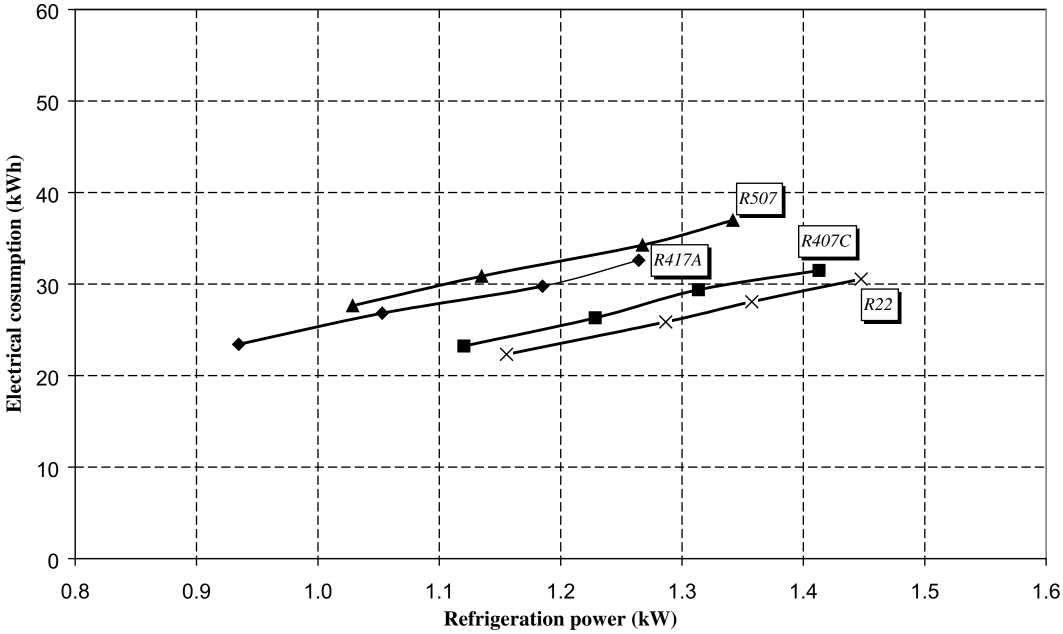

Fig. 5. Electric energy consumption related to R22 and its substitutes at 50 Hz versus the refrigeration capacity.

situation the cold store air temperature at 0 °C is maintained by the thermostat. Similar results have been obtained with different cold store air temperatures simulating the summer and winter seasons. It is interesting to note that, in all conditions, the best performance is related to the R22 followed by the non azeotropic mixture R407C and then by R417A and R507. In particular, the COP of R407C on decreasing the compressor speed rises much more than the other two substitutive fluids of R22. This is due to its greater refrigeration capacity and to a smaller compression work. So a comparison, at the same cooling load, is possible between the electric energy consumption when the compressor works with a nominal frequency of 50 Hz and when is supplied with current at lower frequencies (30, 35, 40, and 45 Hz). In Fig. 6 the percentage gain is reported, in terms of electric energy consumption, obtained at frequencies lower than the nominal ones. 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 the working, at the same cooling load, at 50 Hz with the thermostat working. 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 the energy saving the best performance is related to the R22 followed by the R407C.

To explain the reasons of this energy saving it should be noted that the best performance found when the compressor speed decreases is due to the increase of the compressor global efficiency. The compressor global efficiency depends, mainly, on the trend of the volumetric and isentropic efficiencies. The factors that influence the volumetric efficiency are the pressure drop across the suction and discharge valves and the leakage past the rings of the piston. Moreover, it

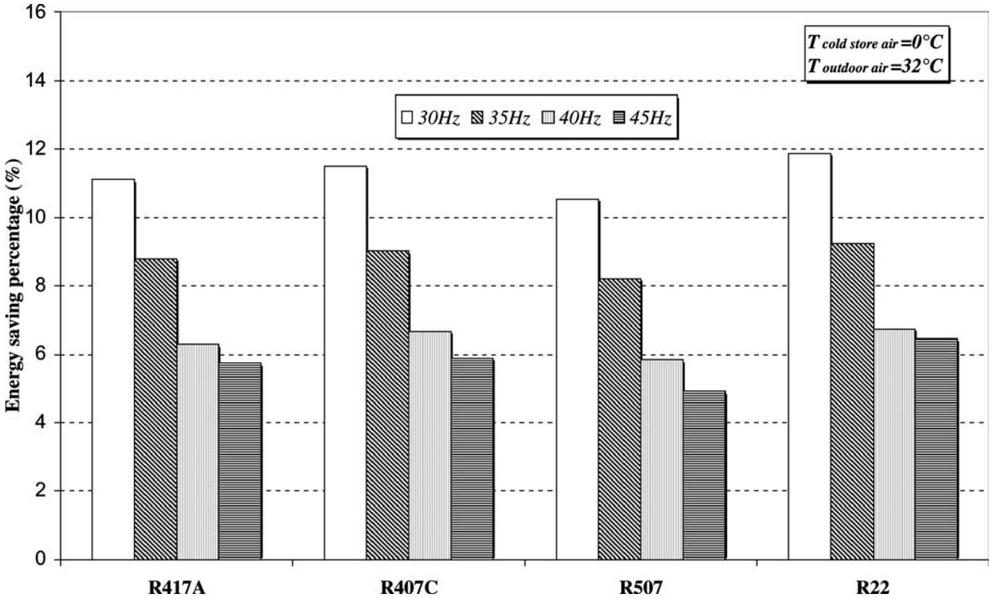

Fig. 6. Percentage gain in terms of electric energy consumption comparing the R22 with its substitutes for supply current frequency included in the range 30–45 Hz.

is necessary to observe that the suction gas entering the cylinder warms and expands on increasing the specific volume that so will be higher than the gas specific volume at the compressor inlet which represents the position considered when the volumetric efficiency is calculated [13]. Besides also the motor electrical efficiency plays a role in the refrigerant heating at the compressor suction because the semi-hermetic compressor electric motor is cooled by means of the refrigerant fluid. Fig. 7 shows the actual volumetric efficiency for all 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 to the reduced cylinder heating of the suction fluid as represented by some compressor indexes [14,15]. Moreover, no influences on the refrigerant superheating entering in the cylinder are related to the smaller efficiency of the electric motor when the speed is reduced [16]; this happens because, as evidenced by a specific index [14], the cooling capacity of the fluids equalizes, practically, most of the heat to remove from the electric motor due to its smaller efficiency, which diminishes until 5% [16], 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 hydrodynamic sealing reduced at lower speeds [10]. It should be noted that the lower mixture actual volumetric efficiency is also due to the smaller superficial tension of the polyester oil. Really the latter does not result in such an effective seal between the cylinder and the piston as that produced by the mineral oil [17]. So, higher quantity of lubricant oil circulates in the plant when the mixture without chlorine is used in comparison with the R22 use; this circumstance is evidenced by the lowest oil level shown by the compressor glass when the plant works. For this reason an oil separator has been mounted on the plant.

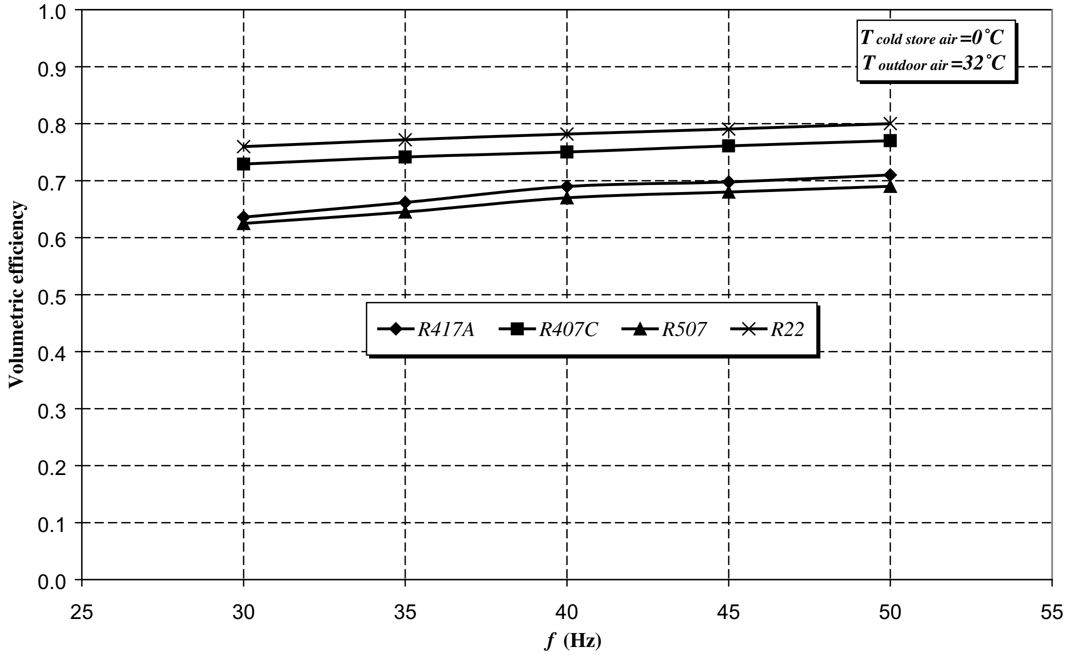

Fig. 7. Volumetric efficiency related to R22 and its substitutes versus compressor motor supply current frequency.

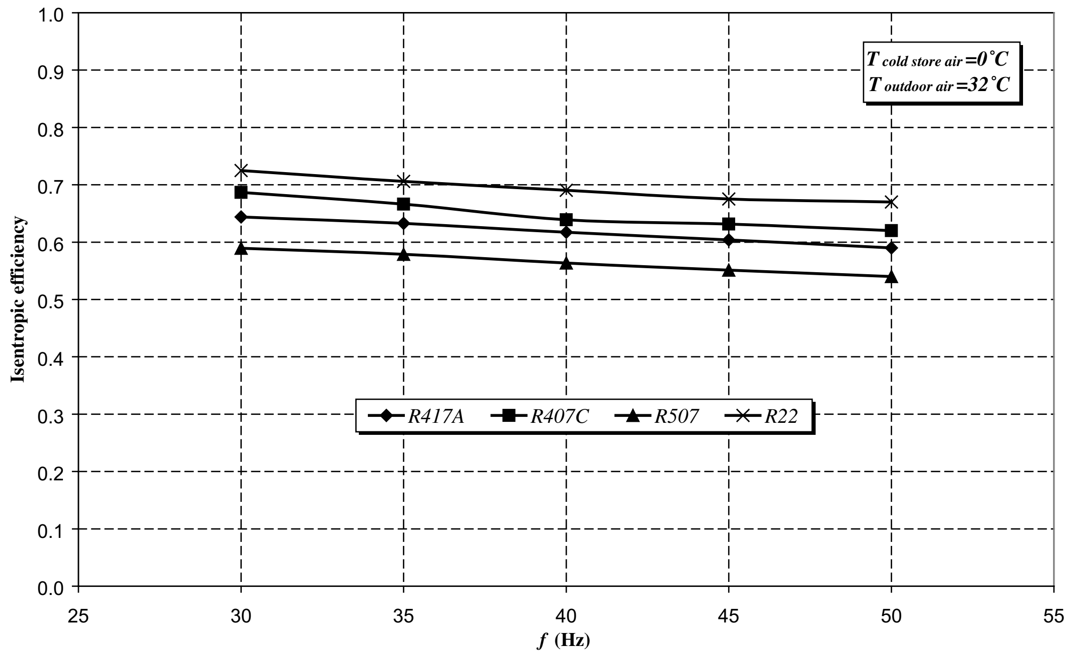

Fig. 8. Isentropic compression efficiency related to R22 and its substitutes versus compressor motor supply current frequency.

In Fig. 8 the compressor isentropic efficiency is reported for all the fluids versus the current frequency feeding the compressor. To evaluate the isentropic efficiency the compressor has been insulated by means of a cm layer cellular insulate. The isentropic efficiency increases when the speed decreases because the discharge superheat reduces on 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 show an increase in the cooling of the electric motor. After all, considering the rising of the mechanical efficiency at low revolution speed due to the smaller effect of the friction, the trend of the volumetric and isentropic efficiencies and the smaller decrease of the electric motor efficiency (until 5%) as the speed decreases, the global compressor efficiency increases [18,19]. This rise represents a meaningful contribution to the energy saving presented above. Moreover, when the compressor speed decreases the global heat transfer coefficient of the heat exchangers is practically constant because the variation of the air temperature at the evaporator is negligible, the air mass flow rate is constant and the influence of the refrigerant heat transfer coefficient on the heat transfer global coefficient is small. It follows that on decreasing the compressor speed and consequently the refrigerant mass flow rate, the area available for the condensation and the evaporation of the refrigerant will be greater leading to lower temperature differences in the condenser and in the evaporator and thus lower condensation pressure and higher evaporation pressure. So in these conditions the compression ratio and the temperature differences in the heat exchangers, and consequently the inefficiencies, decrease when the compressor speed diminishes. As for the expansion valve in all tests a correct working with an acceptable superheating has been verified.

Comparing the behaviour of the R22 substitutes, as showed in Figs. 7 and 8, for a frequency lower than 40 Hz the decrease of the volumetric efficiency for the R407C is less marked as the compression ratio shows in Fig. 3. On the contrary the isentropic efficiency increase is more evident compared with the R507 and the R417A because the compressor discharge temperature and pressure are lower. The consequent rise of the refrigerant mass flow rate and of the enthalpy increase at the evaporator allow a greater refrigeration capacity together with a smaller compression work; for this reason the COP of R407C increases more than the other fluids at frequencies lower than 40 Hz. The energy consumption due to the compressor start current has been evaluated and it is not meaningful for the plant size considered. On the contrary an increase of the energy consumption has been found when the current frequency feeding the compressor is lower than 30 Hz because of the poor performance of the induction motors of the compressor; for these frequencies the inverter generates harmonics which lead to an increase in the losses in the motor compared to direct main operations [20,21]. These frequencies are not interesting for the small size semi-hermetic compressor examined because of its splash lubricant system. The electric energy consumption presented for the compressor includes the power losses caused by the inverters and the motor. The efficiency of the inverter is of the order of 95% and does not significantly change with load or frequency [22]. Because of the difficulty to measure directly the shaft power of the compressor used not trial was made. In fact, the total electric energy consumed by the system is more interesting for the user of variable speed refrigeration systems than the shaft power input to the compressor. As noted the R407C shows the best comparative performance because the compressor volumetric and isentropic efficiencies are better than the other substitutes of R22 when the compressor speed decreases. Besides the compressor electrical efficiency for R407C is better than the other fluids thanks to its constant pressure specific heat and enthalpy change across the compressor as showed calculating the specific index [14], while the mechanical compressor efficiency is, practically, similar for all the fluids tested. The volumetric and isentropic efficiencies can be provided with uncertainties respectively equal to ±0.9% and ±1.1% [12].

It is interesting to note that for all the fluids and for all the compressor speeds examined the requested temperature for the air in the cold store is reached and maintained oscillating in the range ±1 °C; this last value is acceptable since it corresponds to the differential band of a thermostat. In particular, it is possible to observe that only the R507 and the R22 hold the cold store air temperature at −15 °C when the outdoor temperature is 32 °C but this happens when the supply current frequency is located between 40 and 45 Hz. When the frequency of the current feeding the compressor is 30 or 35 Hz it is not possible to hold the cold store air temperature at −15 °C because of the too high evaporation temperature.

Finally, it is important to make some economic considerations about the convenience in adopting a control logic based on the use of the inverter. Referring to the R407C the percentage gain of energy saving is of about 10% in all the range of the frequencies considered 30–50 Hz. It is necessary to observe that, even if to a lesser degree, the thermostatic control determines on/off cycles of the compressor also when the speed compressor is varied because, when the cooling load is due only to the heat exchanged trough the cold-store walls, the refrigeration capacity is too high also at 30 Hz. Moreover, considering the real working situation of a cold store the compressor is stopped on average for about seven hours a day when it operates at 50 Hz. In these circumstances the energy saving corresponds to about 500 kW h/year and then considering the inverter cost for this compressor electric power and the further additional costs linked to the application of the compressor speed control, the pay-back period is of about three years. The pay-back period might further decrease considering plants of bigger size.

5. Conclusions

A research work on an experimental vapour compression plant linked to a commercial cold store has been presented to evaluate the energetic performances when the compressor refrigeration capacity is controlled varying its speed. These energetic performances have been compared with that obtained using the classical thermostatic control considering the most suitable substitutes of the R22. Different temperatures for the air in the cold store and for the air at the condenser have been experimented. The R22 allows the highest energy saving; among the R22 substitutive fluids tested as R407C, R507 and R417A the greater energy saving is shown by the non azeotropic mixture designed as R407C in all the working conditions. This mixture continues to maintain its best performance also at a revolution speed lower than the compressor nominal speed examined corresponding to the frequency of 50 Hz. So among the R22 substitutes the maximum percentage decrease of the electric energy consumption, obtainable comparing the compressor working at 30 Hz with that at 50 Hz at the same cooling load, is the one related to the R407C and is equal about to 12%. The percentage decrease of the energy consumption for each fluid remains practically constant varying the operative conditions. A short analysis of the compressor efficiency to explain the energy saving has been presented. No troubles have regarded the thermostatic expansion valves used that are designed, when available, for each fluid tested. From the economical point of view it has been observed that, using a refrigeration capacity control logic that allows to regulate the compressor speed, the pay-back period results very satisfactory. The usage of bigger size plants can allow to obtain a pay-back period even more favourable taking into account that in the last years the inverter prices have been constantly decreasing.

References

[1] P. Binneberg, J. Philipp, W.E. Krauss, Variable-speed hermetic compressor in a household refrigerator, Proceedings 20th International Congress of Refrigeration, Sydney, Australia, paper code 539, 1999.
[2] J.P. Fredsted, J. de Bernardi, Multi-variable control of vapour compression processes using a variable speed control compressor with integrated inverter, Proceedings 20th International Congress of Refrigeration, Sydney, Australia, paper code 551, 1999.
[3] K. Jang, S. Jeong, Temperature heat flux measurement inside variable-speed scroll compressor, Proceedings 20th International Congress of Refrigeration, Sydney, Australia, paper code 109, 1999.
[4] A. Lundeberg, Speed controlled screw compressor units, The Institute of Refrigeration, 1996.
[5] B.D. Rasmussen, Modelling of a variable speed hermetic compressor, 19th Int. Congress of Refrigeration, vol. IIIA, The Hague, The Netherlands, 1995.
[6] T.Q. Qureshi, S.A. Tassou, Variable-speed capacity control in refrigeration systems, Applied Thermal Engineering 16 (2) (1996) 103–113.
[7] P.H. Pedersen, C.S. Poulsen, S. Gundtoft, Refrigerators and freezers with variable speed compressors, Proceedings 20th International Congress of Refrigeration, Sydney, Australia, paper code 153, 1999.
[8] C. Aprea, R. Mastrullo, C. Renno, Refrigeration capacity control in variable-speed compressor applications, 7th World Congress CLIMA 2000, Napoli (Italia), 15–18 September 2001.
[9] Z. Tamainot-Telto, A. Puttagarts, P. Haberschill, M. Lallelmand, Comportament dynamique de détendeurs thermostatiques de machines frigorifiques, International Journal of Refrigeration 19 (2) (1996) 124–131.
[10] S.A. Tassaou, T.Q. Quereshi, Comparative performance evaluation of positive displacement compressors in variable-speed refrigeration applications, International Journal of Refrigeration 21 (1) (1998) 29–41.
[11] 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.
[12] ISO, Guidance to the expression of uncertainty in measurement, (UNI CEI ENV 13005) 1999.
[13] E. Fornasieri, Analisi del processo di compressione dei fluidi frigorigeni mediante compressori alternativi, Il Freddo 37 (3) (1983) 163–171.
[14] P. Stouffs, M. Tazerout, P. Wauters, Thermodynamic analysis of reciprocating compressors, International Journal of Thermal Science 40 (2001) 52–66.
[15] M. Krueger, M. Schwarz, Experimental analysis of a variable-speed compressor, Proceedings Int. Comp. Eng. Conf., vol. 2, Purdue Univ., US, 1994, pp. 599–604.
[16] S. Boyde, S. Randles, P. Gibb, et al., Effect of lubricant properties on efficiency of refrigeration compressors, Proceedings 2000 Int. Compressor Eng. Conf., vol. 1, Purdue Univ., US, 2000, pp. 311–317.
[17] A. Benamer, D. Clodic, AFF, ELEC, 1998, pp. 53–66.
[18] R. Cohen, J.F. Hamilton, J.T. Pearson, Possible energy conservation through the use of variable capacity compressor, Proceedings Purdue Compressor Technology Conf., Purdue, US, 1974, pp. 50–55.
[19] T.Q. Qureshi, S.A. Tassou, Investigation of the effect of inverters on the power supply and the performances of variable speed refrigeration systems, Proceedings Int. Refrig. Conf., Purdue, US, 1994, pp. 267–274.

[20] S.R. Colby, D.L. Flora, Measured efficiency of high efficiency and standard induction motors, IEEE Trans. (May) (1990) 18–23.
[21] S.A. Tassou, T.Q. Quereshi, Performance of a variable-speed inverter/motor drive for refrigeration applications, IEE Computing and Control Engineering Journal (August) (1994) 193–199.