Applied Thermal Engineering 29 (2009) 1991–1997

Contents lists available at ScienceDirect

Applied Thermal Engineering

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

Determination of the compressor optimal working conditions

C. Aprea [a], R. Mastrullo [b], C. Renno [a,*]

[a] Department of Mechanical Engineering, University of Salerno, Via Ponte Don Melillo, 84084 Fisciano (Salerno), Italy
[b] DETEC, University of Naples Federico II, P.le Tecchio 80, 80125 Naples, Italy

ARTICLE INFO

Article history:
Received 8 February 2008
Accepted 5 October 2008
Available online 17 October 2008

Keywords:
Scroll and reciprocating compressor
Optimum frequency
Energy saving
Exergy and economy aspects

ABSTRACT

The replacement of environmentally unfriendly refrigerants and the energy saving demand have recently caused changes of the components and operation of the vapour compression plants; in particular, the compressors have been experiencing upgrades and modifications. The compression systems are usually designed for working under maximum load conditions, but most of the time these plants work under partial load conditions with compressor on–off cycles regulated by a thermostat. As for the variable speed compressor, the speed is continuously controlled to match the compressor capacity to the load required; this allows to save energy when compared to the thermostatic control. The aim of this paper is to identify the compressor current frequency that optimizes the energy, exergy and economy aspects. The determination of the optimum frequency for each working condition is key to build a control algorithm that allows the compressor speed to be continuously regulated by an inverter. This analysis has been applied to the reciprocating and scroll compressors and high energy savings have been achieved.

1. Introduction

The compressor, that is the principal component of a vapour compression plant, has recently undergone optimizations and changes [1–3]. Scroll and reciprocating compressors for small and medium size plants are used both in residential and commercial applications. Scroll compressors are used in supermarkets, bulk milk cooling, marine containers and refrigerator trucks and in air conditioning applications where they allow both to reduce noise and vibration levels, and as well as to optimize the energy performances. Reciprocating compressors are highly efficient; scroll compressors require a further improvement for it to be as efficient as reciprocating compressors. When compression ratio are high, reciprocating compressor is more efficient than the scroll without injection; however, it is possible to increase the scroll compressor efficiency by injecting some refrigerant in liquid or vapour phase [3]. Though designed to satisfy the maximum load, the vapour compression plants usually work most of the time under partial load conditions, and are regulated with on–off cycles which result in high energy consumptions. A less energy consumption with respect to the on–off control [4] can be obtained with a continuous control of the compressor speed. As for scroll compressors the frequency of the motor supply current can be about 15 Hz; while for reciprocating compressors, it cannot be lower than 30 Hz, because of the increase in the compressor vibrations, the noise and the lubrication problems due to the splash system. For small-medium size plants it is not clear if a decrease of the compressor speed could result in an energy saving. Hence, it is important to determine the optimum frequency, that corresponds to a definite cooling/heating load, for the compressor motor in order to obtain the highest energy saving as compared to the thermostatic control. The optimum frequency can be experimentally or theoretically obtained [5]. The first step is to get experimentally the compressor performance parameters in terms of refrigerant mass flow rate, input power and cooling capacity when the frequency varies. The second step is to get, for each type of compressor, curve fitted polynomials that allow the optimum frequency to be evaluated in terms of compressor data at the basic frequency [6], in order to come out with a control algorithm. This paper aims at optimizing the energy, exergy and economy aspects of variable speed reciprocating and scroll compressors.

2. Experimental plants

The first experimental plant, working both as refrigeration system and heat pump, is made up of a hermetic scroll compressor, a finned tube air heat exchanger, two thermostatic expansion valves and a plate-type water heat exchanger inserted in a water tank (Fig. 1). Additionally, the refrigerant circuit has check valves, a cycle inversion valve, a non return valve and a liquid receiver. The cycle inversion valve controls the working of the plate-type water heat exchanger that works as a condenser when the plant operates as heat pump, and as an evaporator when the plant works as a water chiller. The refrigerant fluid experimentally tested is R407C

* Corresponding author. Fax: +39 089 964037.
E-mail address: crenno@unisa.it (C. Renno).

1359-4311/$ - see front matter © 2008 Elsevier Ltd. All rights reserved.
doi:10.1016/j.applthermaleng.2008.10.002

Nomenclature

a	discounting back rate
CBS	structural bond coefficient
C_E	electric energy unit cost (€/kW h)
DPB	discount pay-back (year)
$Ex_{des,k}$	exergy destruction rate (kW)
F	cash flow (€)
f	frequency (Hz)
I_k	component cost (€)
k	parameters ratio
M	refrigerant mass flow rate (kg/s)
Q	cooling capacity (W)
SC	extra cost (€)
T	Temperature (°C)
W	compressor input power (W)
$y_{i,k}$	optimization variable
Greeks	
γ	amortization coefficient (year^{-1})
Σ_{tot}	total cost (€/year)
τ	working time (h/year)
Superscripts	
$'$	parameters at basic frequency
Subscripts	
des	destroyed
E	electric energy
k	component
tot	total

(R32/R125/R134a 23/25/52% in mass). When the condenser inlet air temperature is about 30 °C and the water temperature is 7 °C, the cooling capacity and the electric power of the compressor, at the basic frequency, are respectively about 8 and 3 kW. The unit chiller–heat pump water circuit is linked to an air heat exchanger. In the water pumping circuit there are: a circulation pump, a tank where the plate-type heat exchanger is inserted, an automatic air vent valve and two manual air vent valves, an expansion tank, a safety valve, an effluent valve for the water and an automatic filling group. The second experimental vapour compression refrigeration plant, linked to a cold store and shown in Fig. 2, is made up of a reciprocating compressor, an air condenser followed by a liquid receiver, a manifold with three expansion valves (electronic, thermostatic, manual) to feed an air cooling evaporator inside the cold store. The fluid refrigerant used is the R407C. For an evaporation temperature of 0 °C and a condensation temperature of 30 °C, the cooling capacity and the compressor electric power at the basic frequency of 50 Hz are, respectively, equal to about 2.4 kW and 850 W. The cooling load in the cold store is simulated by means of some electric heaters linked to a regulator. For both plants the compressor speed is regulated by means of a PWM inverter. It consists of a rectifier that converts the compressor motor three-phase main voltage (380 V, 50 Hz) to DC voltage and then reverses this to a three phase AC supply-voltage by an inverter; at the output of the inverter the voltage is adjustable in magnitude and frequency. In Table 1, for both the plants used, the transducers specifications (Coriolis effect flowmeter, RTD 100 4 wires thermoresistance, piezoelectric absolute pressure gauges, wattmeter, electric energy meter) are reported.

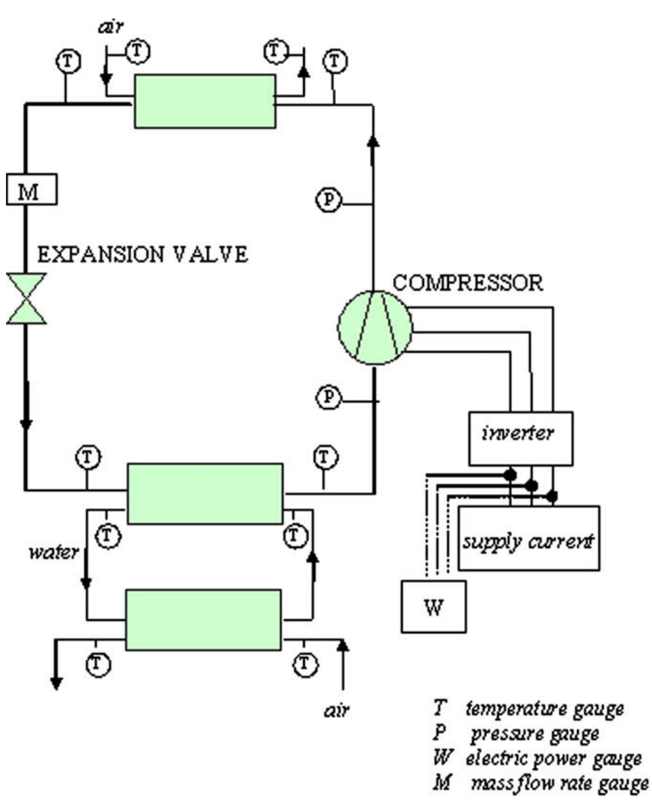

Fig. 1. Vapour compression plant working both as water chiller and heat pump.

3. Optimization of the compressor exergy, economy, energy aspects

The compressor performances are represented by means of diagrams in terms of cooling capacity, input power, refrigeration mass flow rate, when the condensation and evaporation temperatures vary. These diagrams are generally supplied only at the basic frequency; the manufacturers seldom provide the working curves for different frequency values of the compressor motor supply current. Hence, it is important to determine the compressor performances mapping when the frequency varies. In particular, the real working of a variable speed compressor can be considered as a set of infinite compressor performances at constant speed. In order to study the compressor performances when the frequency varies, the trend of refrigerant mass flow rate, input power and cooling capacity can be experimentally obtained and expressed by means of the second-order functions of the evaporation and condensation temperatures. For example, in Fig. 3 the cooling capacity trend of a reciprocating compressor is reported in terms of evaporation temperature, for a condensation temperature of 30 °C and for fixed frequencies (30, 40, 50 and 60 Hz). Additionally, in Table 2, referring to the reciprocating compressor, it is possible to note that, for different condensation and evaporation temperatures, the ratios of refrigerant mass flow rate, input power and cooling capacity values at any frequency with respect to the values at the basic frequency, remain constant. In Table 2 the environment temperature, the superheating and subcooling degrees of the refrigerant fluid are fixed too. In other words, the above-mentioned ratios are independent from condensation and evaporation temperatures and depend only on the compressor frequency, and

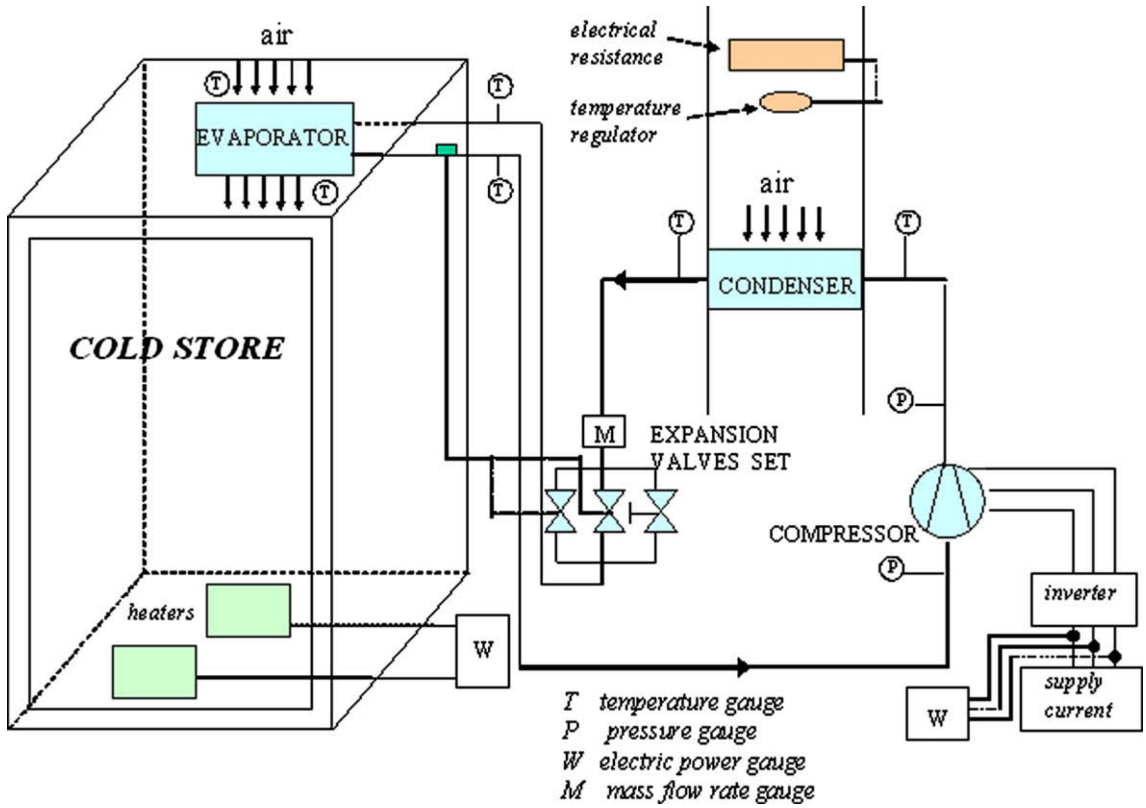

Fig. 2. Experimental plant subjected to a cold store.

Table 1
Transducers specifications.

Transducer	Range	Accuracy
Coriolis effect mass flow rate meter	$0 \div 2$ kg/m	±0.2%
RTD 100 4 wires	$-100 \div 500$ °C	±0.15 °C
Piezoelectric absolute pressure gauge	$1 \div 10$ bar; $1 \div 30$ bar	±0.2; ±0.5 F.S.
Wattmeter	$0 \div 3$ kW	±0.2%
Electric energy meter	$0 \div 1000$ MW h	±1.0%

hence can be expressed by means of the second-order functions of the frequency [6]:

$$k = \frac{G}{G'} = a_1(f - f')^2 + a_2(f - f') + a_3, \tag{1}$$

where G could be the compressor input power (W), the refrigerant mass flow rate (M) or the cooling capacity (Q); a_i are constant coefficients; f the general frequency (Hz) and f' the basic frequency (Hz).

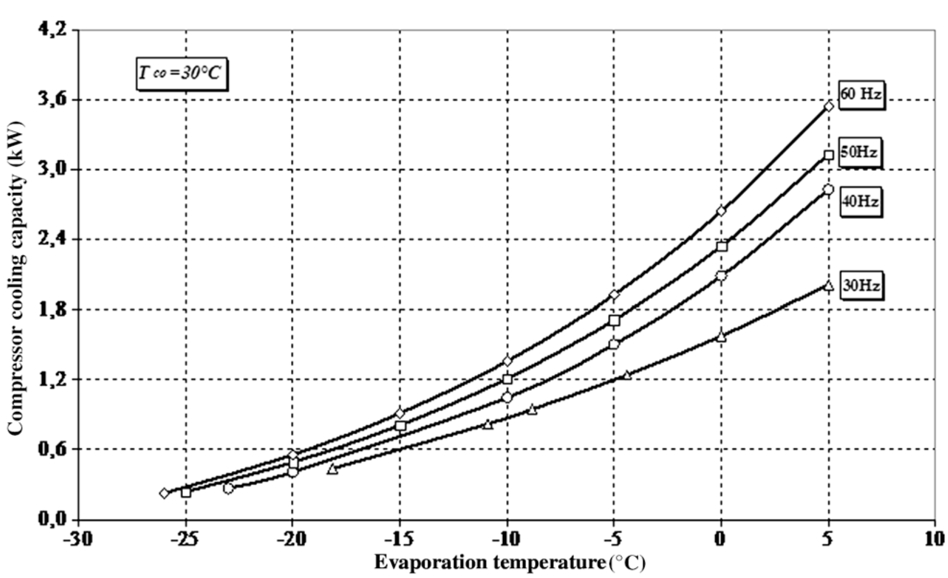

Fig. 3. Cooling capacity when the compressor speed and evaporation temperature vary.

Table 2
Cooling capacity, input power and refrigeration mass flow rate ratios values referring to the reciprocating compressor.

	Refrigerant mass flow race (k_M)				Input power (k_W)				Cooling capacity (k_Q)			
Frequency (Hz)												
T_{ev} (°C)	30 Hz	30 Hz	40 Hz	60 Hz	30 Hz	30 Hz	40 Hz	60 Hz	30 Hz	30 Hz	40 Hz	60 Hz
−12.0	0.3406	0.3398	0.6122	1.723	0.6721	0.6726	0.8583	1.076	0.6289	0.6145	0.8851	1.139
−11.0	0.3453	0.3471	0.6121	1.746	0.6671	Q.6699	0.8561	1.065	0.6256	0.6222	0.8934	1.126
−10.0	0.3488	0.3490	0.6123	1.731	0.6512	0.6587	0.8584	1.052	0.6263	0.6298	0.8729	1.116
−9.0	0.3513	0.3521	0.6227	1.753	0.6534	0.6545	0.8643	1.056	0.6273	0.6179	0.8712	1.127
−8.0	0.3528	0.3526	0.6319	1.733	0.6438	0.6478	0.8672	1.057	0.6123	0.6155	0.8654	1.123
−7.0	0.3537	0.3541	0.6218	1.717	0.6412	0.6462	0.8671	1.046	0.6191	0.6198	0.8928	1.134
−6.0	0.3541	0.3489	0.6153	1.746	0.6539	Q.6599	0.8674	1.073	0.6183	0.6134	0.8763	1.127
−5.0	0.353?	0.3439	0.6259	1.733	0.6548	0.6589	0.8539	1.068	0.6138	0.6166	0.8792	1.114
−4.0	0.3534	0.3527	0.6134	1.768	0.6537	0.6588	0.8562	1.053	0.6129	0.6156	0.8974	1.129
−3.0	0.3526	0.3529	0.6316	1.731	0.6642	0.6678	0.8572	1.062	0.6239	0.6199	0.8942	1.113
−2.0	0.3516	0.3489	0.6112	1.746	Q.6659	0.6688	0.8482	1.073	0.6127	0.6234	0.8934	1.124
−1.0	0.3503	0.3525	0.6426	1.724	0.6621	0.6671	0.8438	1.093	0.6153	0.6206	0.8956	1.137
0.0	0.349	0.3567	0.6328	1.730	0.6682	0.6692	0.8482	1.096	0.6153	0.6267	0.8743	1.134
1.0	0.3475	0.3543	0.6011	1.723	0.6649	Q.6655	0.8495	1.092	0.6316	0.6278	0.8942	1.113
2.0	0.3459	0.3533	0.6404	1.731	0.6782	0.6788	0.8482	1.083	0.6122	0.6222	0.8712	1.236
3.0	0.3442	0.3487	0.6303	1.718	0.6749	0.6689	0.8573	1.077	0.6010	0.6124	0.8934	1.162
4.0	0.3425	0.3501	0.6205	1.748	0.6781	Q.6695	0.8579	1.055	0.6231	0.6299	0.8541	1.139
5.0	0.3408	0.3489	0.6153	1.733	0.6746	0.6734	0.8548	1.047	0.6143	0.6223	0.8572	1.126
	30 °C	40 °C	30 °C	30 °C	30 °C	40 °C	30 °C	30 °C	30 °C	40 °C	30 °C	30 °C

These equations are experimentally obtained for fixed superheating and subcooling degrees of the refrigerant, but when the compressor works at variable speed, both the superheating degree and the specific volume vary influencing the input power and the cooling capacity; hence, it is necessary to make some corrections [7]. In particular, Eq. (1) gives the optimum frequency for each working condition. This is possible once fixed the evaporation and condensation temperatures, and then the cooling/heating load (Q^*), and when the value of Q' at the basic frequency, provided by the manufacturers for the same working conditions, is known (Fig. 4). Hence, Eq. (1) and the typical coefficients of the compressor allow to determine the frequency corresponding to Q^*, to be imposed on the compressor in order to obtain the greatest energy saving in comparison with the compressor on/off control. The optimum frequency determination, for each working condition, is fundamental to carry out a control algorithm which continuously regulates the compressor speed by an inverter.

The performances of a plant with a variable speed compressor must be optimized in terms of energy and costs saving. Hence, once determined the optimum frequency, that allows an energy saving as compared to the working at the basic frequency under the same conditions of cooling load, it is necessary to compare both of them simultaneously in terms of energy and costs saving. To this extent, it is possible to use the following Eq. [8]:

$$\frac{\partial \Sigma_{tot}}{\partial y_{i,k}} = CBS_{i,k} C_E \left(\frac{\partial Ex_{des,k}}{\partial y_{i,k}} \right) \tau + \gamma \left(\frac{\partial I_k}{\partial y_{i,k}} \right), \tag{2}$$

where Σ_{tot} is the total cost (€/year), $y_{i,k}$ is the optimization variable (frequency), *CBS* is the structural bond coefficient determined as ratio between the plant exergy destruction rate derivative with respect to the optimization variable and the component exergy destruction rate derivative, $Ex_{des,k}$ is the component exergy destruction rate (kW), C_E is the electric energy unit cost (€/kW h), I_k component cost (€), τ is working time (h/year), γ amortization coefficient (year^{-1}). Eq. (2) allows the exergy destruction rates of the compressor and plant to relate to the frequency, that in this analysis is the optimization variable. The determination of the optimum frequency by means of Eq. (1), allows the reduction in the exergy destruction and then the costs (Eq. (2)). From Eq. (2), it can be easily obtained the equation

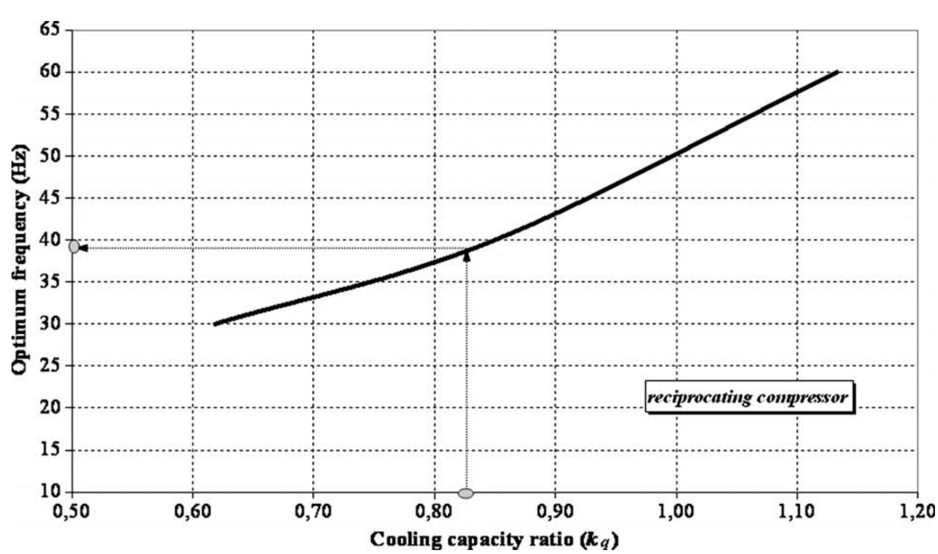

Fig. 4. Optimum frequency determination referring to the reciprocating compressor under each working condition.

$$\Delta\Sigma_{tot} = (CBS \cdot C_E \cdot \Delta\dot{E}x_{dis,k} \cdot \tau) + (\gamma\Delta I_K) \tag{3}$$

that represents a finite difference form of Eq. (2). In particular, Eq. (3) allows to compare in terms of total cost, the compressor working at the optimum frequency and at the basic frequency, and then to calculate the discount pay-back (DPB):

$$DPB = \frac{SC}{\sum_{K=1}^{N} \frac{F_K}{(1+a)^K}}, \tag{4}$$

where SC, a and F represent, respectively, the extra cost (€), the discounting back rate and the cash flow (€).

4. Results

The knowledge of the optimum frequency, for each type of cooling/heating load, allows to optimize the energy saving obtained when a continuous compressor speed control is used instead of the classical thermostat. The optimum frequency can be determined either experimentally or by means of Eq. (1) with the same results. As for the reciprocating compressor, the optimum frequency has been experimentally determined for different working conditions. This frequency allows an energy saving when compared with that working under the basic frequency of 50 Hz. The cooling load and the cold store air temperature have been changed by means of electrical resistances connected to a voltage converter able to regulate the thermal power. In particular, the optimum frequency has been experimentally determined to match the cooling load for each working condition. Hence, considering the cooling loads matched for frequencies between 30 and 45 Hz, corresponding to evaporation temperatures between −18 and −8 °C, a comparison in terms of energy saving for a compressor working both at the basic frequency and at the above-mentioned frequencies has been realized. In Fig. 5, the optimum frequency is reported for the above-mentioned working conditions. In particular, considering a cooling load of about 900 W (Fig. 6), to keep a 0 °C cold store air temperature and then an evaporation temperature of about −8 °C, the optimum frequency is 30 Hz. This frequency al-

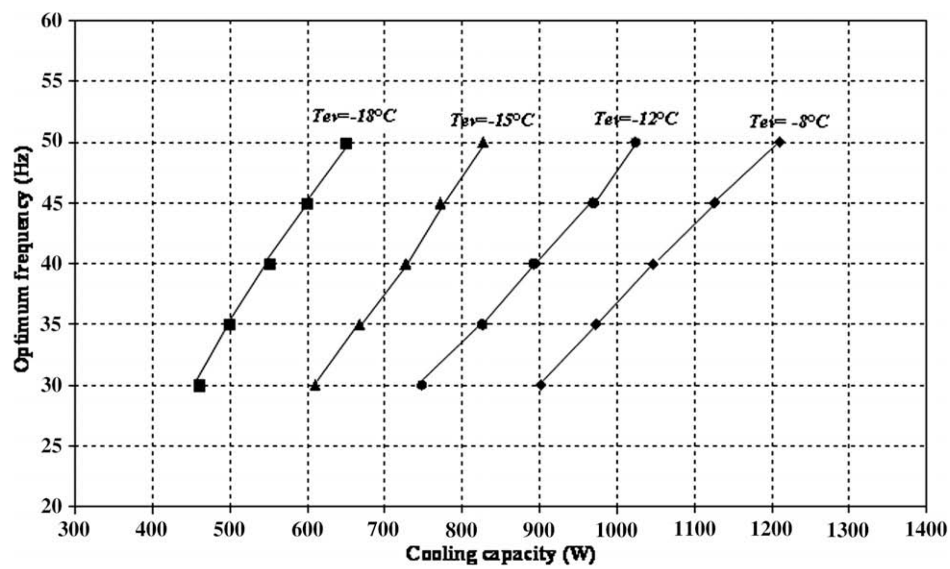

Fig. 5. Optimum frequency referring to the reciprocating compressor for evaporation temperatures between −18 and −8 °C.

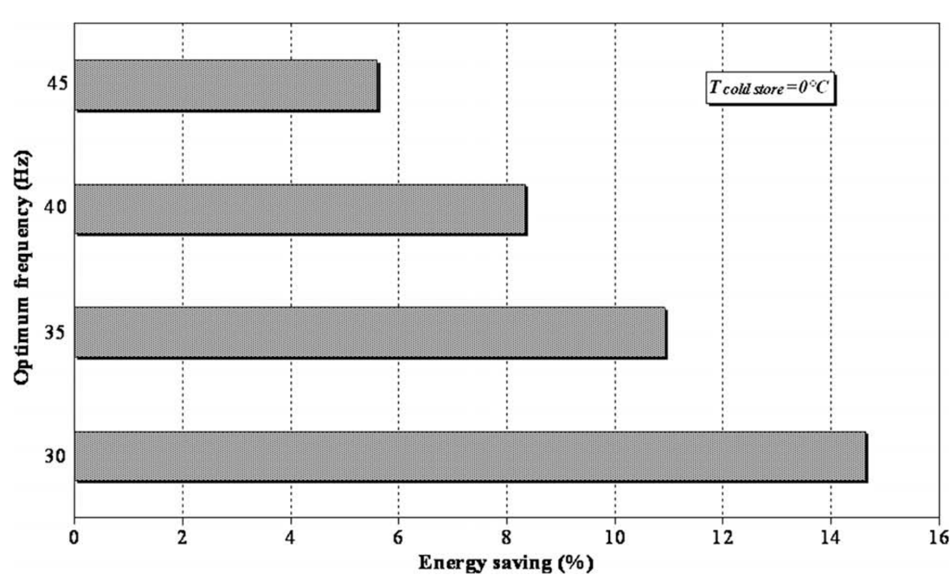

Fig. 6. Energy saving corresponding to optimum frequency (reciprocating compressor).

lows an energy saving of about 15% when compared with that working at the basic frequency and at the same cooling load. In Fig. 6, some comparisons in terms of energy consumption have been achieved for different optimum working frequencies. It is necessary to observe that the comparisons have been obtained under the same cooling load in the cold store and for fixed set-point temperature. The optimum frequency determination is important because it is not clear if the compressor speed reduction can result in an energy saving in comparison with the thermostatic control. In fact, referring to the reciprocating compressor, it has been experimentally observed that it is convenient to match the cooling load by varying the compressor speed with frequencies to get at least 40% of the cooling capacity at 50 Hz. As for the cold store, as pointed out in Eqs. (2) and (3), it is possible to compare from the thermoeconomic point of view, the plant working at the frequencies 30 and 50 Hz under the same conditions of cooling load. Considering about 6200 h/year the working time of the component, 100 € the inverter cost, 0.20 €/kW h the electric energy medium cost, 0.20 year^{-1} the amortization coefficient (γ), 0.05 the discounting back rate (a) and 10 years the plant life (N), the DPB is equal to 2.8. Fig. 7 shows how for a reciprocating compressor the plant exergy destruction rate, the component exergy destruction rate and the discount pay back change when the optimum frequency varies.

As for the water chiller, for a water set-point temperature of 7 °C, the optimum frequency of the scroll compressor and the corresponding energy saving have been obtained for different cooling loads (Fig. 8). In particular, for a cooling load of 3.5 kW the

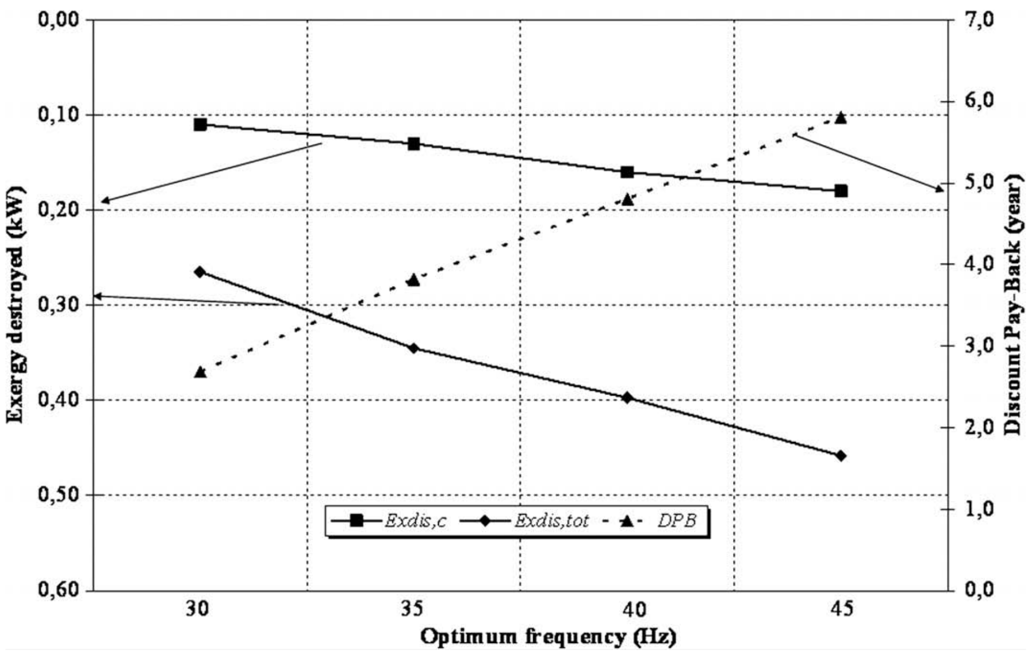

Fig. 7. Exergy destruction rates and discount pay-back referring to the reciprocating compressor.

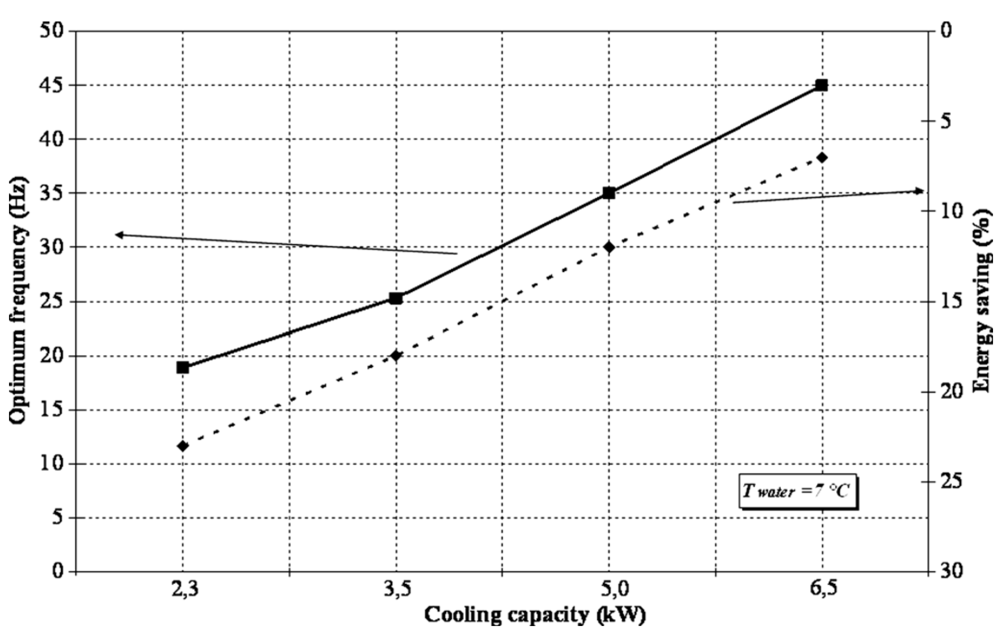

Fig. 8. Energy saving corresponding to optimum frequency (water chiller).

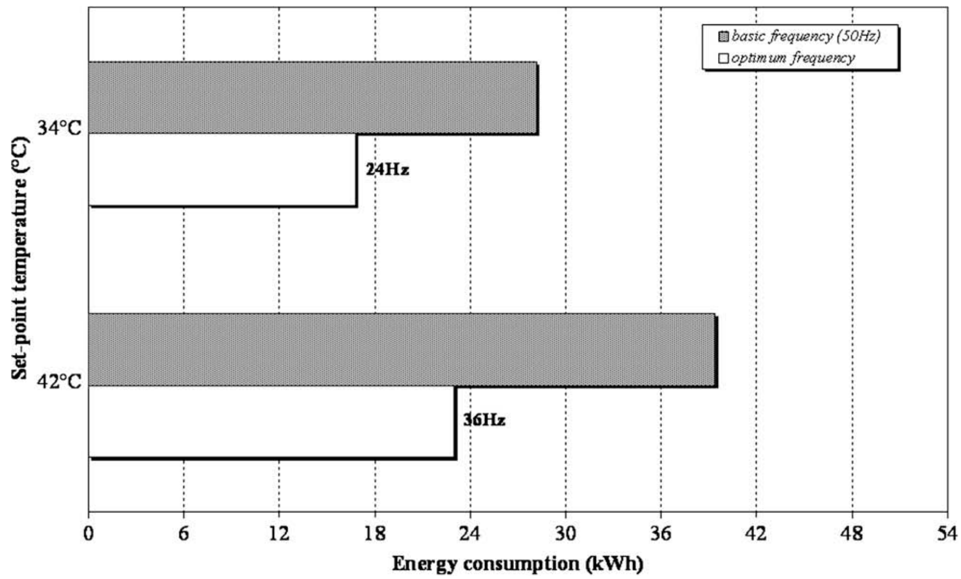

Fig. 9. Energy saving corresponding to optimum frequency (heat pump).

optimum frequency is 25 Hz and the corresponding energy saving is about 18%. As for the heat pump working, the values of 34 and 42 °C have been fixed as set-point temperatures of the water heated. When the set-point is 42 °C, it has been observed that for a frequency of 36 Hz the heating load can be matched; while when the set-point is 34 °C, the scroll compressor frequency can also reach 24 Hz to match the heating load. In particular, in Fig. 9 two comparisons in terms of energy consumption between the plant working at 24 and 50 Hz with a temperature of set-point 34 °C, and between the plant working at 36 and 50 Hz with a temperature of set-point 42 °C, are reported. The energy saving, in comparison with the thermostatic control when the plant works at the basic frequency of 50 Hz, is in both cases of about 30%. Finally, to use an inverter to regulate the compressor speed is advantageous in refrigeration systems, particularly for small size plants with one compressor. The advantages derive from a good control of the air/water output temperature and from the isentropic efficiency increase, linked to its smaller compression ratio, even though electric and seal losses take place. In fact, when the compressor speed decreases, both the refrigerant mass flow rate and the cooling capacity decrease. In these conditions the temperature difference in the heat exchangers decreases and then the evaporation pressure increases, while the condensation pressure decreases [4]. Sometimes, as for the heat pumps, it is useful to use the inverter over summer, while in winter its use is not very reliable due to the decrease in the compressor performances because of higher compression ratios; in this situation it could be useful to use a digital scroll [9].

5. Conclusions

The main objective of this paper is to identify for variable speed compressors the current frequency that optimizes the energy, exergy and economy aspects. It is important to identify it because it is not sure that the compressor speed decrease results in an energy saving when compared to the thermostatic control. The optimum frequency corresponds to a defined cooling/heating load has been determined. In order to analyze the variable speed compressor working conditions, experimental tests have been performed using two different types of compressors: scroll and reciprocating. In particular, as for the reciprocating compressor, an exergy and economy analysis has also been carried out in terms of exergy destruction rates and discount pay-back; the energy saving obtained fixing the frequency at 30 Hz has been about 15% when compared to the working at basic frequency. As for the scroll compressor a 25% medium energy saving has been obtained. In future, it will be interesting to get a compressor theoretical model in order to determine the optimum frequency when the compressor speed varies.

References

[1] K.T. Ooi, Design optimization of a rolling piston compressor for refrigerators, Appl. Therm. Eng. 25 (2005) 813–829.
[2] M. Cui Michael, Comparative study of the impact of the dummy port in a scroll compressor, Int. J. Refriger. 30 (2007) 912–925.
[3] H. Cho, J.T. Chung, Y. Kim, Influence of liquid refrigerant injection on the performance of an inverter-driven scroll compressor, Int. J. Refriger. 26 (2003) 87–94.
[4] C. Aprea, R. Mastrullo, C. Renno, Fuzzy control of the compressor speed in a refrigeration plant, Int. J. Refriger. 27 (6) (2004) 639–648.
[5] R.N.N.N. Koury, L. Nachado, K.A. R Ismail, Numerical simulation of a variable speed refrigeration system, Int. J. Refriger. 24 (2000) 192–200.
[6] S. Shao, W. Shi, X. Li, H. Chen, Performance representation of variable-speed compressor for inverter air conditioners based on experimental data, Int. J. Refriger. 27 (2004) 805–815.
[7] A.E. Dabiri, C.K. Rice, A compressor simulation method with corrections for the level of suction gas superheat, ASHRAE Trans. 87 (2) (1981) 771–782.
[8] T.J. Kotas, The Exergy Method of Thermal Plant analysis, Krieger, Florida, 1995.
[9] H. Shih Cheng, Y. Rong Hwa, Development and testing of a multi-type air conditioner without using AC inverters, Energy Convers. Manage. 46 (2005) 373–383.