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# Fuzzy control of the compressor speed in a refrigeration plant

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

[a]*Department of Mechanical Engineering, University of Salerno, Via Ponte Don Melillo 1, 84084 Fisciano (Salerno), Italy*
[b]*DETEC, University of Naples Federico II, P.le Tecchio 80, 80125 Naples, Italy*

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**Abstract**

In this paper, referring to a vapor compression refrigeration plant subjected to a commercially available cold store, a control algorithm, based on the fuzzy logic and able to select the most suitable compressor speed in function of the cold store air temperature, is presented. The main aim is to evaluate the energy saving obtainable when the fuzzy algorithm, which continuously regulates the compressor speed by an inverter, is employed to control the compressor refrigeration capacity instead of the classical thermostatic control, which imposes on/off cycles on the compressor that works at the nominal frequency of 50 Hz. The variation of the reciprocating compressor speed is obtained by controlling the compressor electric motor supply current frequency in the range 30–50 Hz, as it is not possible to consider values smaller than 30 Hz because of the lubrication troubles due to the splash system. In this range, two among the most suitable working fluids proposed for the R22 substitution, such as the R407C (R32/R125/R134a 23/25/52% in mass) and the R507 (R125/R143A 50/50% in mass) are tested. Comparing the compressor speed fuzzy control with the classical thermostatic control, frequently used in the cold stores and in other refrigeration systems, the experimental results show a meaningful energy saving equal even to about 13% when the R407C is used as a working fluid. In particular, to explain from the energy saving point of view the best performances of the refrigeration plant when the compressor speed varies, an exergetic analysis is realized. Besides, with regard to the inverter cost, the pay-back period determined is more than acceptable for the plant size examined.


*Keywords:* Compression system; Cold room; Piston compressor; Variable speed; Regulation; Fuzzy logic; R407C; R507

# Régulation à logique floue de la vitesse d'un compresseur d'une installation frigorifique

*Mots-clés:* Système à compression; Chambre froide; Compresseur à piston; Vitesse variable; Régulation; Logique floue; R407C; R507

## 1. Introduction

The vapor compression refrigeration plants, though designed to satisfy the maximum load, work at part-load for much of their life generally regulated by on/off cycles of the compressor, working at the nominal frequency of 50 Hz, imposed by a thermostatic control which determines a high energy consumption. Moreover, the inefficient use of electricity to supply the refrigeration and air-conditioning compressors is considered as an indirect contribution to the greenhouse gases emitted in the atmosphere; these emissions can be reduced by improving the energy conversion efficiency of the above mentioned systems. A theoretical comparison of various refrigeration capacity control methods in full and part-load conditions shows that the compressor speed variation is the most efficient technique

* Corresponding author. Tel.: +39-089-964327; fax: +39-089-964037.
*E-mail address:* crenno@unisa.it (C. Renno).

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**Nomenclature**

| | |
|---|---|
| COP | coefficient of performance |
| $\dot{Ex}$ | exergy (W) |
| $ex$ | specific exergy (J/kg) |
| $f$ | compressor electric motor supply frequency (Hz) |
| $\dot{L}_{cp}$ | compressor power input (W) |
| $\dot{m}$ | mass flow rate (kg/s) |
| $\dot{Q}$ | thermal power (W) |
| $T$ | temperature (°C) |
| $T_0$ | environmental temperature (°C, K) |
| $t$ | time (s) |
| $T_{air,cs}$ | cold store air temperature (°C) |
| $T_{set}$ | set-point temperature (°C) |

Greek symbols

| | |
|---|---|
| $\delta$ | efficiency defect |
| $\eta$ | efficiency |
| $\Delta h$ | enthalpy variation (J/kg) |
| $\Delta T$ | temperature difference (°C) |
| $\tau$ | dimensionless exergetic temperature |

Subscripts

| | |
|---|---|
| co | condenser |
| cp | compressor |
| des | destroyed |
| ev | evaporator |
| ex | exergetic |
| in | inlet |
| is | isentropic |
| mt | mean thermodynamic |
| out | outlet |
| ref | refrigerant |

[1,2]. This method of refrigeration capacity control, which consists in varying the compressor speed to continuously match the compressor refrigeration capacity to the load, has been analyzed during the last years [3–10]. An inverter can be used to regulate the compressor speed. 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. The application of this type of control of the refrigeration capacity to a commercial compressor, though presents some advantages in terms of energy saving, determines some disadvantages such as the inverter cost and some troubles linked to the compressor lubrication and reliability [11,12] and to the correct working of the expansion devices. This last problem is negligible when the secondary fluids at the heat exchangers are in gas-phase, as in the plant examined, but it seems to be remarkable when the secondary fluids are in the liquid-phase [13]. So, the primary aim of this paper is to setup a controller capable of regulating continuously the speed of a reciprocating compressor frequently used in cold stores and in other small size refrigeration systems whose this type of compressor generally has no oil pump. This kind of control allows us to match the compressor refrigeration capacity to the cooling load at any time, so that the compressor can also work at other frequencies smaller than 50 Hz. It is to be considered that with the classical thermostatic control frequently used in the cold stores and in other small size refrigeration systems, the compressor works only at 50 Hz. In particular, referring to a vapor compression refrigeration plant subjected to a commercially available cold store, a control algorithm based on the fuzzy logic, and able to select the most suitable compressor speed in function of the cold store air temperature, is presented in this paper. Apart from the fuzzy logic, the compressor speed control might also be obtained by means of other techniques such as the traditional proportional-integral and derivative control (PID) [14–18]. In particular, the fuzzy control logic, compared with PID, allows both to use better the experimental knowledge related to the trend of the variables which characterize the working of the refrigeration plant and to adopt a control logic based on a non-mathematical model, and hence to avoid the determination of the specific models of the refrigeration plant components [19–21]. Moreover, a fuzzy controller with respect to PID control generally might allow obtaining performances comparable or sometimes better in terms of precision of the set point required. Besides referring to a fuzzy controller the overshoot of the variables is small and the settling time fast as regards the dynamic response during the sudden variations in the cooling load; all this generally results in a robust control [22–24]. So, experimental tests have been conducted to compare the plant performances obtainable using as compressor refrigeration capacity control systems, both the fuzzy algorithm and the classical thermostat that determines on-off cycles of the compressor that works at a frequency of 50 Hz. The working fluids tested, the R407C (R32/R125/R134a 23/25/52% in mass) and the R507 (R125/R143A 50/50% in mass), are among the most diffuse substitutes of R22.

## 2. Experimental plant

The vapor compression experimental plant, subjected to a commercially available cold store and shown in Fig. 1, is made up of a semi-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 lubricated with polyester oil and its speed is regulated by means of a PWM inverter. It is formed by a rectifier that converts the three-phase main 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. The manifold with both valves has been mounted to solve possible troubles, because the expansion valves behavior, when the compressor speed varies, is unknown [13]. The expansion valves used are specifically designed for the R407C and R507. In the evaporation temperature range −20 to 10 °C at a 30 °C condensing temperature, working with the R407C at the nominal frequency of 50 Hz, the compressor refrigeration capacity varies in the range 1.4–4.8 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 exactly obtain the same temperature fixed for the air, a regulator is used to control the electrical resistances supply. In some experimental tests, the cooling load in the cold store is simulated by means of some electric heaters linked to a regulator and the electric power is measured by means of a Wattmeter. Table 1 lists the specifications of the transducers used. 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 and the R407C and R507 thermodynamic properties have been evaluated using a dedicated software that has also been used to determine the energy and exergy balances.

## 3. Experimental procedure description

To evaluate the plant performances when an inverter is used, it is necessary to compare the plant energy consumption when the refrigeration capacity is regulated by on/off cycles of the compressor that works at a supply current frequency of 50 Hz, and when the refrigeration capacity is controlled by the fuzzy algorithm. In the experimental tests, different types of cooling loads have been considered. First of all some experimental tests when the cooling load is due both to the periodic opening of the cold store door and to the inevitable heat exchanges with outdoor air, even when the cold store door is closed, have been realized. These tests have been performed at various temperature levels for the air in the cold-store and, precisely, at 5, 0 and −5 °C, opening the cold store door every 20 min for about 5 min with an outdoor air temperature of about 18 °C. Moreover, in some tests the cooling load has been obtained by means of controllable electrical heaters located in the cold store, while in other tests a real cooling load has been considered represented by 200 kg of fruit and vegetables for whose preservation the temperature has been fixed at 5 °C in the cold store. In these last two

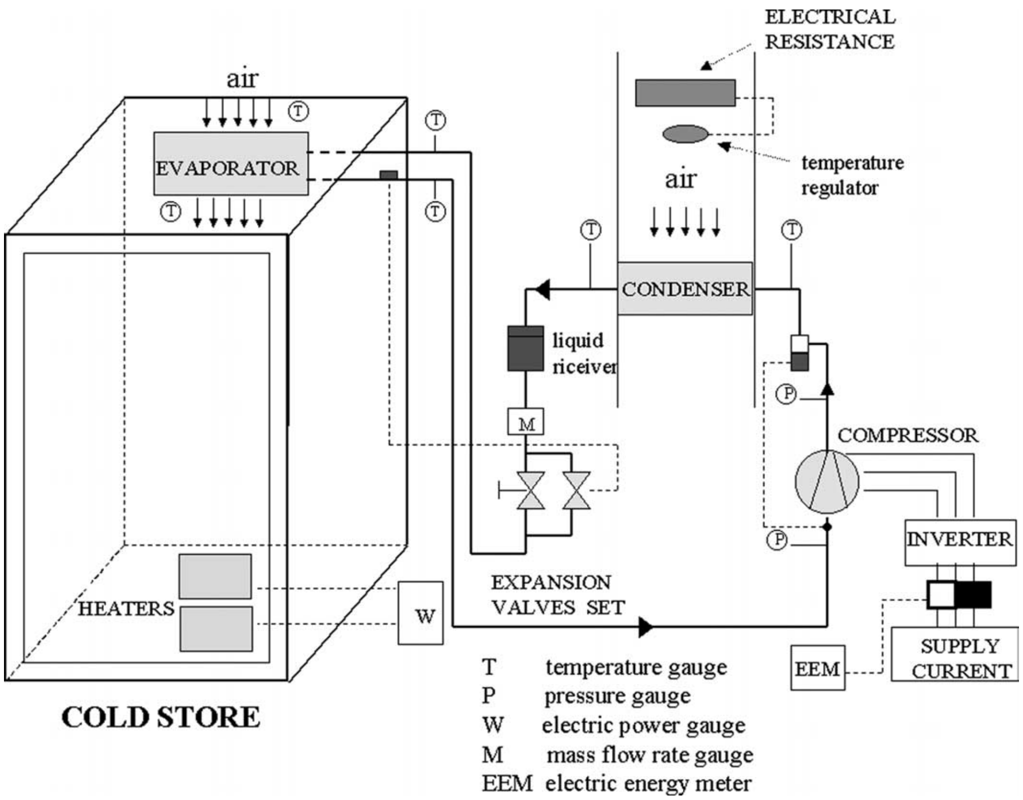


Fig. 1. Sketch of the experimental plant.

Table 1
Transducers specifications

| Transducer | Range | Accuracy |
|---|---|---|
| Coriolis effect mass flow rate meter | 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 | 360–420 V; 0–16 A | ± 0.5% F.S.; ± 0.5% F.S. |

situations, the cold store door has been opened every 10 min to simulate a real working condition; moreover, the tests have been performed both in the winter and in the summer season. As for the summer tests the outdoor air temperature at the condenser has been kept at about 32 °C thanks to a channel where the air is heated by means of an electric heater, while in winter the outdoor air temperature has been kept at 10 °C. The experimental results are mostly presented in terms of electrical energy consumption, measured by means of an opportune electric energy meter, evaluating the energy saving obtained when a compressor speed control is used. The tests, which lasted 2 days, have been realized for the R407C and the R507.

## 4. Fuzzy logic in the compressor speed control

The fuzzy logic represents a methodology that allows us to obtain defined solutions from vague, ambiguous or uncertain information. For this the fuzzy process is very similar to that of the human mind capable of finding defined conclusions starting from approximated information and data. In contrast to the classic logic approach, that requires an exact definition of the mathematical model equations characterizing the phenomenon, the fuzzy logic allows us to solve problems not well defined and for which it is difficult, or even impossible, to determine an exact mathematical model. Therefore, the human experience and knowledge is necessary for this type of modelling. In particular, the fuzzy logic is a valid alternative for the solution of non-linear control problems. In fact the non-linearity is treated by means of rules, membership functions and inferential process, that ensure simpler implementations and minor design costs. On the other side the linear approximation of a non-linear model is simple enough, but it has the disadvantage to limit the control performances and can result, in some situations, expensive. Moreover, the fuzzy controllers are robust and allow us to realize improvements or changes in a very simple way by means of the use of the other rules or the membership functions. Many examples of fuzzy control can be found in some recent applications. In particular, in the heating ventilation and air-conditioning industry there are various fuzzy control applications of the air temperature and humidity [25–28]. The design of a fuzzy controller requires three essential phases. The first is to establish the input and output variables. The second is to define the membership functions for the input and output variables. The last is to select or formulate the control rules. The main goal of this paper is to determine a fuzzy controller capable of regulating the compressor electric motor supply current frequency. In Fig. 2 a block diagram of the fuzzy control process of the commercially avalaible cold store air temperature is reported. In particular, the figure shows a two-input one-output fuzzy controller. The input variables are the temperature difference between the set-point temperature and the real temperature of the air in the cold store ($\Delta T$), and the derivative of this temperature difference with time ($d(\Delta T)/dt$); the fuzzy output variable is the frequency of the supply current of the compressor electric motor ($f$). The fuzzy logic is based on the determination of the fuzzy-set that represents the possible values of the variables. The fuzzy theory with respect to the traditional logic theory, according to which an element can belong or not to a particular set, allows the partial membership of an element to a set. Each value of the variables is characterized by a membership value which changes with continuity from zero to one. Thus, it is possible to define a membership function for each variable that establishes the membership rate of a variable at a certain set. From an operative point of view, a controller fuzzy receives the values of the input variables, performs some operations and determines an output value. This process is characterized by three principal phases: fuzzification, inference mechanism and defuzzification. The fuzzification process allows to transform a value defined into a fuzzy value; the inference process determines the output fuzzy by means of the rules fixed according to the experimental reality; the defuzzification process permits to transform the fuzzy output into a defined value. The main difficulty of the fuzzy logic is connected with the necessity of a good specific experience in the design and the building of a fuzzy controller. So, as for the regulating parameters some experimental considerations have allowed us to set the control variables of the reciprocating compressor speed. It is clear that the choices of the rules and membership functions of the controller can be properly changed. However, it is to be considered that it is certainly convenient to control from the energy saving point of view the compressor speed because it works at lower frequencies, but in this situation the time required to obtain the set-point temperature will be

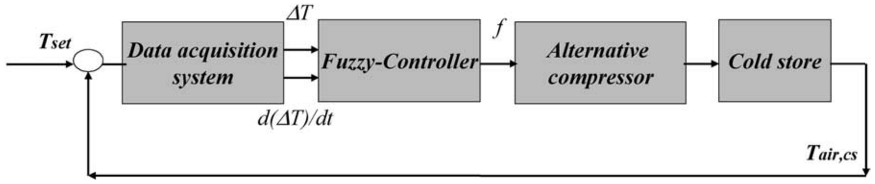


Fig. 2. Fuzzy control algorithm block diagram.

major in comparison with the time necessary when the compressor works at a 50 Hz nominal frequency. So, it may happen sometimes that even when the compressor works for frequencies lower than that nominal ones the energy saving exepected may be partially obtained because the compressor has worked at lower frequencies indeed but for too much time. So, in order to regulate the working time of the compressor at lower frequencies it is important that, when the fuzzy algorithm input and output variables membership functions are to be defined, the choice of the subset number and of its wideness has to be proper and guided by the experimental knowledge. As for the the choice of the rules it is necessary to do similar consideration. For this reason the algorithm membership functions and rules suggested from the authors have been experimentally verified. Table 2 shows the rules fixed to set the algorithm and the five fuzzy subsets used to characterize the input and output linguistic variables marked with the following labels: very low (VL), low (L), medium size (MS), high (H) and very high (VH). As to tune the membership function is much easier than to tune the control rules, the attention is focused here on the former, in order to realize a robust fuzzy controller for the compressor speed control. After some experimental considerations to understand the control characteristics of the alternative compressor, the membership functions in Figs. 3–5 have been defined for the temperature difference between the set-point temperature and the real temperature of the air in the cold store, the derivative of this temperature difference with the time and the frequency of the compressor motor supply current. The triangular membership function, with one center and two limits, has been adopted here. As for the temperature difference the range covered is located between 0 and 13 °C (Fig. 3). In order to increase the sensitivity of the fuzzy controller as the cold store air temperature approaches the setpoint, the membership function is tuned more narrowly to the VL, L and MS temperature differences. As for the derivative with the time of the temperature difference (Fig. 4), a range comprised between 0.001 and 0.013 K/s has been covered. It has been considered as a variable input with the derivative also taking into account mainly the fast variations when the cooling load varies suddenly; this happens when the cold store door is open. To increase the sensitivity of the controller with respect to the rate of the change of the derivative with time of the temperature difference, the fuzzy subset might have a smaller definition, perhaps ranging from 0.004 to 0.008. However, satisfactory results can be obtained with the previous definition of the fuzzy subsets. The values of the compressor motor supply frequency considered in the output fuzzy subset membership function (Fig. 5), are located in the range 30–50 Hz. It was not possible to consider values under 30 Hz because the compressor vibrations and the noise increase considerably together with the lubrication troubles due to the splash system increase. The inference mechanism employed has been the product inferencing method whereby the minimum operator for the ‘and’ is replaced by the product operation [29–31]. This mechanism allows a better interpolative reasoning among the input and output variables because the effect of these variables on each other is obtained more effectively. The adopted defuzzification method is based on the determination of the mass center of a compound set; so the fuzzy output is turned into a well-defined analogic signal [32,33]. The control algorithm, based on the fuzzy logic, has been built in a Labview environment. In particular, this algorithm provides as an output variable a voltage signal which can be continuously used by an inverter to control the compressor speed.

Table 2
Fuzzy algorithm rules

| $f$ | | $\Delta T$ | | | | |
|---|---|---|---|---|---|---|
| | | vl | l | ms | h | vh |
| | l | vl | l | ms | h | h |
| $d(\Delta T)/dt$ | ms | l | l | ms | h | vh |
| | h | ms | ms | h | h | vh |

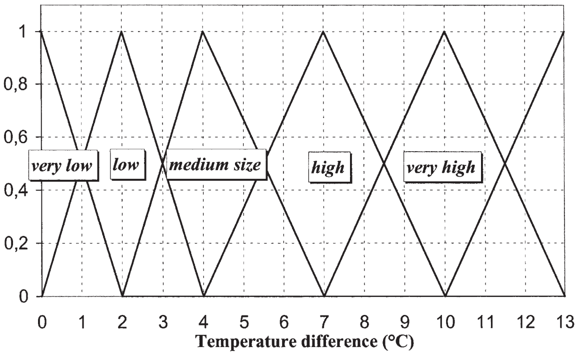


Fig. 3. Membership function of the temperature difference between the set-point temperature and the real temperature of the air in the cold store.

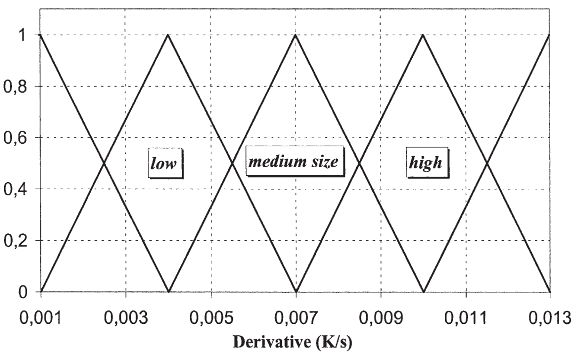


Fig. 4. Membership function of the derivative of temperature difference in the time.

## 5. Test results and discussion

Several experimental tests have been conducted to explain the energy saving obtainable with the fuzzy algorithm in comparison with the classical thermostatic control, that determines the on/off cycles of the compressor that works at a nominal frequency of 50 Hz. To simulate better the real working conditions of the cold store, various types of cooling loads have been considered. In particular, in the experimental tests either the electric heaters or the fruits and vegetables have been adopted as cooling load. Moreover, a further load results to be both due to the periodic opening of the cold store door and also due to the inevitable heat exchanges with outdoor air when the cold store door is closed. In Fig. 6 a comparison in terms of electric energy consumption, measured by means of a proper electric energy meter, between the control on–off realized by the classical thermostat and the compressor speed continuous control obtained by the fuzzy algorithm, is reported when the cooling load is due only to the periodic opening of the cold store door. The experimental tests have been realized for cold store air temperatures fixed at −5, 0 and 5 °C and for a constant cooling load obtained opening the cold store door every 20 min for about 5 min with an outdoor air temperature of about 18 °C. One can clearly observe that the energy consumption increases when the cold store air temperature decreases. This is due to the fact that a constant cooling load has been considered for all the cold store air temperatures and so the time necessary to reach the temperature of −5 °C will be greater and will determine a higher electric consumption. Moreover, it is possible to observe that the energy saving obtained with the algorithm with respect to the thermostat is on an average of about 10%, even if it clearly diminishes slowly when the cold store air temperature decreases because under this circumstance the working time of the compressor increases.

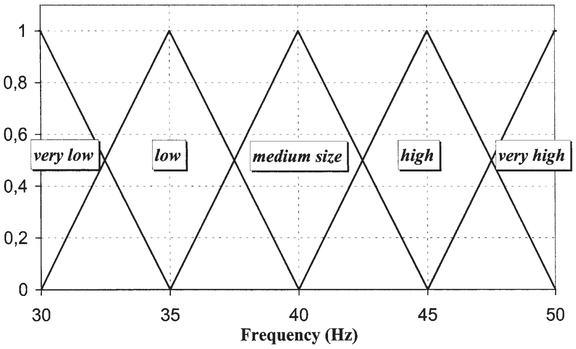


Fig. 5. Membership function of the compressor electric motor supply current frequency.

In Fig. 7 the electric energy consumption of the compressor obtainable with two control systems, related to the R507 and R407C, is reported both for the summer and for the winter, when the cooling load is due both to the periodic opening of the cold store door and to the presence of the electric heaters. In these experimental tests related to the electric heaters it has been considered an electric power constant of about 200 W. It has been observed that the best performances are related to the R407C that allows, with a continuous control of the compressor speed, an medium energy saving of about 13% in comparison with the thermostatic control for both the outdoor air temperatures considered. In particular, the absolute electric energy consumption in the summer season is about 5% higher than that of the winter season even if the energy saving in the two seasons is practically the same. In Fig. 8 the energy consumption related to a real cooling load, represented by 200 kg of fruit and vegetables and by the periodic opening of the cold store door, is considered when both the fuzzy control and the thermostatic control are used. In addition, in this situation the highest energy saving is obtainable by means of the fuzzy control with the R407C and is of about 13% with respect to the one obtained by means of the thermostatic control. Selecting a value of 10 °C for the outdoor air temperature, the results in terms of energy saving are practically the same. It is important to note that the fuzzy control system allows to reach the temperature of the air needed in the cold store and to maintain its oscillation

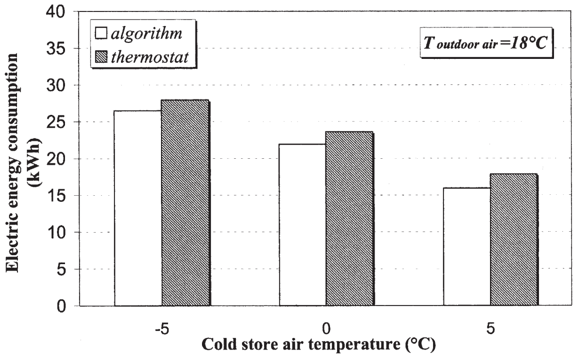


Fig. 6. Electric energy consumption for R507 using both the fuzzy control and the thermostatic control (cooling load → periodic opening cold store door).

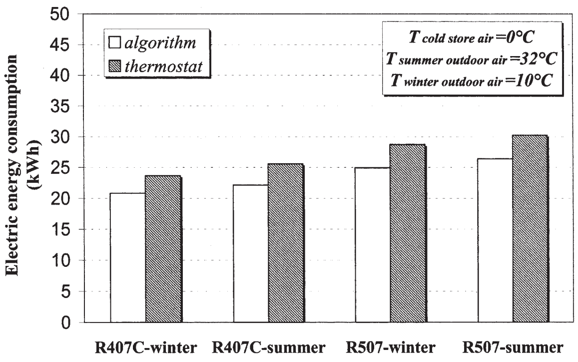


Fig. 7. Electric energy consumption for R507 and R407C using both the fuzzy control and the thermostatic control (cooling load → electric heaters).

in the range ±1 °C; this last value is acceptable as it corresponds to the differential band of a thermostat.

To explain the reason for the energy saving obtainable with a fuzzy control algorithm that allows to regulate continuously the compressor velocity in comparison with a thermostatic control, an exergetic analysis of the components of the refrigeration plant has been realized on varying the compressor speed. For this purpose, it results more correct to realize the exergetic analysis in the steady-state conditions instead of the transient conditions, which derive from the compressor speed fuzzy control, selecting again the same conditions in terms of compressor refrigeration capacity. In particular, the refrigeration power corresponding to each frequency of 30, 35, 40, 45, 50 Hz under the transient conditions is considered again under the steady-state conditions to allow a correct measurement process. This experimental analysis has been performed related to the summer season, with the outdoor temperature at the condenser kept at about 32 °C, but similar results have been obtained also in other working conditions. In particular, the tests realized in the winter season verify the

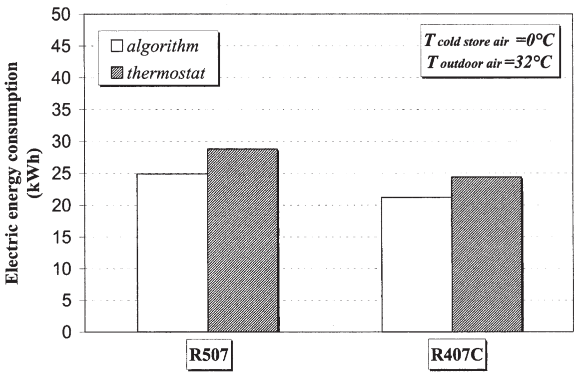


Fig. 8. Summer electric energy consumption for R507 and R407C versus cold store air temperature using both the fuzzy control and the thermostatic control (cooling load → fruits and vegetables).

correct refrigerant lamination when, varying the compressor speed, the compression ratio across the valve is low.

The exergetic analysis allows to obtain important information about the plant total irreversibility distribution among the components. The overall plant exergetic efficiency has been evaluated as the ratio between the exergy output and the exergy input and can be expressed as:

$$\eta_{ex} = \frac{\sum \dot{E}x_{out}}{\sum \dot{E}x_{in}} = 1 - \frac{\dot{E}x_{des}}{\sum \dot{E}x_{in}} \quad (1)$$

An accurate analysis can be realized evaluating the exergy destroyed for each single component of the plant. In particular, the exergy flow destroyed in the condenser is evaluated as:

$$\dot{E}x_{des,co} = \dot{m}_{ref}(ex_{in,co} - ex_{out,co}) - \dot{Q}_{co}\tau_{co} \quad (2)$$

and the exergy flow destroyed in the evaporator is evaluated as:

$$\dot{E}x_{des,ev} = \dot{m}_{ref}(ex_{in,ev} - ex_{out,ev}) - \dot{Q}_{ev}|\tau_{ev}| \quad (3)$$

where the dimensionless exergetic temperature can be defined as:

$$\tau = 1 - \frac{T_0}{T_{mt,air}} \quad (4)$$

where $T_0$ is the environmental temperature while $T_{mt,air}$ is properly evaluated for the evaporator and the condenser. The exergy flow destroyed in the compressor, neglecting the heat transfer with the environment, is evaluated as:

$$\dot{E}x_{des,cp} = \dot{m}_{ref}(ex_{in,cp} - ex_{out,cp}) + \dot{L}_{cp} \quad (5)$$

The exergy flow destroyed in the valve is evaluated as:

$$\dot{E}x_{des,va} = \dot{m}_{ref}(ex_{in,va} - ex_{out,va}) \quad (6)$$

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:

$$\delta_i = \frac{\dot{E}x_{des,i}}{\dot{L}_{cp}} \quad (7)$$

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

$$\eta_{ex} = 1 - \sum_i \delta_i \quad (8)$$

In Fig. 9 a comparison in terms of the exergetic efficiency of the whole plant, when the R407C and the R507 are used, is reported versus the frequency of the current feeding the compressor; moreover, in Fig. 9 the values of the evaporation power and the trend of the electric energy consumption are also reported. The experimental tests have been carried out for an air temperature settled in the cold store equal to 0 °C and for each frequency of 30, 35, 40, 45, 50 Hz of the compressor electric motor supply current. The exact refrigeration capacity that the compressor can supply at the selected frequency has been considered as cooling load by means of some controllable electric heaters located in the cold store. The exergetic efficiency of the whole plant is linked to the actual COP and to the reversible COP of the plant: $\eta_{ex} = COP/COP_{rev}$; as the $COP_{rev}$ is fixed, referring to the temperatures shown in Fig. 9, the exergetic efficiency follows the trend of the COP. It is to be noted that when the compressor speed decreases the COP increases. In particular, it is necessary to observe that, 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 temperature differences in the condenser and in the evaporator will be lower and thus consequently a lower condensation pressure and a higher evaporation pressure, as shown in Fig. 10 related to the winter season, will be obtained. In these conditions the compression ratio and the mean temperature difference between the air and the refrigerant in the evaporator and in the condenser diminish; so that the exergy destroyed in the compressor [13,34,35] diminishes and also the exergy destroyed in the heat exchangers, while the COP and the exergetic efficiency increase. 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. As for the improvement shown in Fig. 9 for R407C in comparison with R507 in terms of exergetic efficiency and electric energy consumption, it is necessary to observe that it is due above all to the greater refrigeration capacity of R407C and to its smaller compression ratio when the compressor speed decreases (Fig. 10). However, a more detailed analysis is reported in Refs. [36,37].

To understand to what degree the exergy destroyed in the components of the refrigeration plant affects the energy consumption, in Fig. 11 a comparison is reported, referring to R507, between the efficiency defects in each component of the plant when the compressor works at the nominal frequency of 50 Hz and the efficiency defects in each component when the refrigeration plant operates at a frequency of 30 Hz. Besides, referring to Fig. 11 it is possible to observe the influence of the single components on the total exergy destroyed in the plant. In particular, it is interesting to note that, when the compressor operates at a frequency of 30 Hz, the efficiency defect of the compressor is equal about to 30%; in the heat exchangers the values of the efficiency defects are near (19% condenser; 17% evaporator); in the valve the efficiency defect is lower (10%). In order to increase the overall plant performance, the compressor and both the heat exchangers must be optimized because of their higher efficiency defects, while the contribution to the irreversibility of the valve is marginal.

To get a further confirmation of the energy saving obtainable when a continuous control of the compressor speed is used and to determine the influence of the exergy destroyed in the refrigeration plant on the energy consumption, it is possible to compare the exergy destroyed in time in the plant when the compressor works at the nominal frequency of 50 Hz with the exergy destroyed in time when the refrigeration plant operates at a frequency of 30 Hz, in the hypothesis that the evaporation power is the same both for the tests with the thermostatic control and for those with the algorithm as shown in the experimental tests related to Figs. 6–8. So, considering the efficiency defect definition in terms of exergetic efficiency ($\delta = 1 - \eta_{ex}$), it is possible to obtain the following equation:

$$\frac{E\dot{x}_{des(50\ Hz)}t_{50} - E\dot{x}_{des(30\ Hz)}t_{30}}{E\dot{x}_{des(50\ Hz)}t_{50}} = \frac{(\dot{L}_{50\ Hz} - \dot{Q}_{ev}/COP_{rev})t_{50} - (\dot{L}_{50\ Hz} - \dot{Q}_{ev}/COP_{rev})t_{30}}{(\dot{L}_{50\ Hz} - \dot{Q}_{ev}/COP_{rev})t_{50}}$$

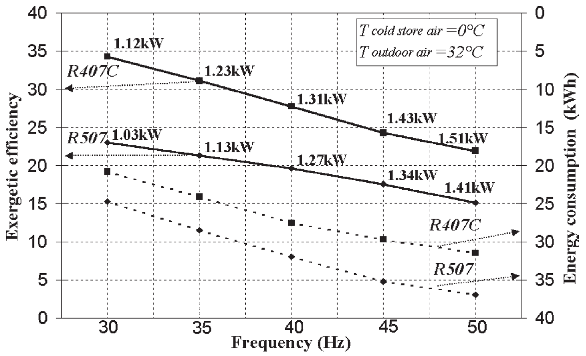


Fig. 9. Exergetic efficiency related to R507 and R407C versus compressor electric motor supply current frequency.

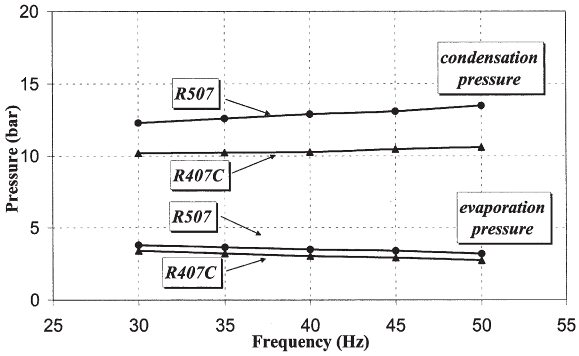


Fig. 10. Evaporation and condensation pressures trend on varying the compressor speed.

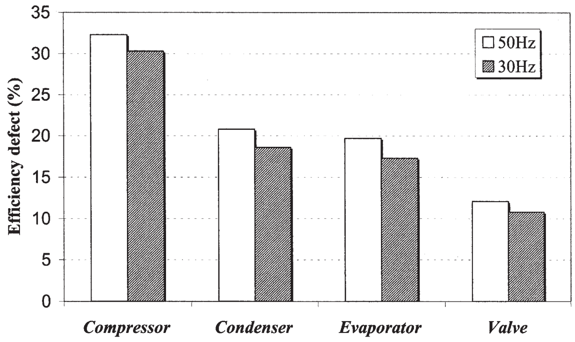


Fig. 11. Efficiency defect of the components of the refrigeration plant related to R507 refrigerant fluid.

where $t_{50}$ and $t_{30}$ represent the times when the plant works respectively at 50 Hz (when the thermostat works) and at 30 Hz (possible working frequency when the fuzzy control is used). From this equation it is possible to observe that the percentage deviation of the exergy destroyed in time in the plant is linked to the electric energy consumptions at 30 and 50 Hz. It is important to observe that for the hypothesis considered the term $\dot{Q}_{ev}/COP_{rev}$ is constant and that generally in the working situation at 50 Hz, when the thermostat is on, the plant works for about 70% of the total working time ($t_{30}$). So, considering the values of the exergy destroyed in the components of the refrigeration plant subjected to the cold store determined experimentally for the R507 related to an outdoor air temperature of about 32 °C, the percentage deviation of the global exergy destroyed in time and of the energy consumption are equal to about 13%. This value is very near to the values of the energy saving obtained in the experimental analysis above reported comparing the thermostatic control with the fuzzy control.

Finally, it is important both to observe that the inverter efficiency is of the order of 95% [38] and 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 on an average of 10% in the whole range of the frequencies (30–50 Hz) considered. Moreover, considering a real working situation of a cold store the compressor is stopped on an average for about 7 h a day when it operates at 50 Hz. Under these circumstances the energy saving corresponds to about 500 kW h per year; an evaluation of the inverter cost for the compressor electric power considered and of the further additional costs linked to the application of the compressor speed control, allows to know that the pay-back period is of about 3 years.

## 6. Conclusions

In this paper, referring to a vapor compression refrigeration plant subjected to a commercially available cold store, the performances of the classical thermostatic control, that imposes on/off cycles at the compressor working at the nominal frequency of 50 Hz, are compared with that of a control algorithm based on the fuzzy logic and built in Labview environment. This algorithm is able to select the most suitable compressor speed in function of the cold store air temperature. The variable speed compressor has been gained thanks to a PWM. The fluids tested, R407C and R507, are among the most suitable substitutes of the R22. The experimental comparison between the compressor speed control by means of a control algorithm and by the classical thermostatic control has been conducted under various experimental conditions. A significant energy saving on an average equal to about 13% has been obtained using the compressor speed control algorithm based on the fuzzy logic in comparison with the thermostatic control. The major energy saving among the substitutive fluids of the R22 is related to the R407C in comparison with the R507. The reason of the energy saving and of the best performance obtainable by the fuzzy control have been analyzed by means of an exergetic analysis determining the efficiency defects of the components and the exergetic efficiency of the plant on varying the compressor electric motor supply current frequency. Moreover, an analysis of the inverter costs for the compressor electric power considered and of the further additional costs linked to the application of the compressor speed control, has resulted in a pay-back period of about 3 years.

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