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PERFORMANCE IMPROVEMENT OF AN INDUSTRIAL CHILLER THROUGH THE OPTIMIZATION OF THE CONTROL LOGIC

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ABSTRACT
According to the tasks agreed within the Kyoto protocol, in terms of CO_2 emissions reduction, the European Community is promoting energy saving. The introduction of energetic classification for electrical household appliances, makes consumption as a parameter of choice for consumers and thus represents a relevant tool for the promotion of energy saving. Therefore the interest in the evaluation and minimization of electric consumption for industrial appliances is also evident.

The aim of the present study is to investigate alternative control solutions to improve the performance and diminish the energy consumption of an industrial chiller. This was carried out through software modeling of the system and experimental validation of the best performing solutions.

Simulated trends show good agreement with experimental data and allow the definition of a suitable PID controller that performs good temperature regulation while slightly reducing energy demand.

Further ameliorations were considered and preliminary results are shown both for optimal control and neural modeling of the chiller.

Results obtained encourage to continue in the development of better performing regulators, also through neuro – fuzzy logic control.

NOMENCLATURE

T	Temperature
N	Compressor speed (Hz)
C	Inner thermal load
P	Compressor electric power
W	Thermal power
m	Flow rate
c	Specific heat
a	Coefficient of transfer function denominator
b	Coefficient of transfer function numerator
u	Input vector of the system
y	Output vector of the system
x	State vector of the system
RMS	Root Mean Squared Error

SUBSCRIPTS

a	Air
amb	Ambient conditions
i	Inlet
in	Conditions on the lowest shelf of the cooled vane
m	Average conditions inside the cooled vane
pa	Air at constant pressure
o	Outlet
1	Relative to one previous instant
2	Relative to two previous instants

CHILLER CHARACTERIZATION
A refrigerator cabinet for pastry shop is the chiller under study. Setup of the computerized system for data acquisition was carried out in order to acquire pressure and temperature data of the cooled vane and of main components of the refrigeration plant. A dedicated system was used to measure compressor electric consumption whose speed was varied through the inverter.

All the tests were carried out in an automatic way by use of dedicated software developed in Labview environment. The characterization of the system response was carried out through tests in a climatic chamber, imposing known variations of the main inputs to the system that were identified in:

- compressor speed;
- ambient temperature;
- inner thermal load.

During the tests the variation of the external thermal load was simulated as the variation of the temperature T_{amb} at the climatic chamber, while the transients occurring during opening of the cooled vane and with the introduction of warm pastries were simulated with an electric resistance inside the vane.

The system was assumed linear therefore the correlation existing between T_{in} , assumed as the controlled variable, and the single inputs was studied separately [1].

<u>Temperature pull – downs at different compressor speed and ambient temperature</u>
In the 40 - 65 Hz compressor operational range of frequencies, 6 levels of speed corresponding to 65, 60, 55, 50, 45 and 40 Hz named respective u1 (maximum speed), u2, u3, u4, u5 and u6 (minimal speed), have been fixed.

The system response to different ambient conditions was determined repeating the tests at different thermal levels considering 20°C and 30°C in the climatic chamber.

The behaviour of the system at different compressor speed was determined through temperature pull – downs, starting from internal condition equal to T_{amb} , at a particular speed level and running until the attainment of the regimen condition, assumed when the variation endured in the last 5 minutes from the average temperature inside the cooled vane remains inferior to 0,5°C. Such condition must then be maintained for further 2 minutes.

Tests at variable compressor speed

To verify the system thermal inertia, compressor speed was variated at constant steps: from the minimal speed the compressor accelerates to a greater speed level until the temperature stabilization of T_{in} is reached. It proceeds then in this way until the attainment of the maximum speed.

A same outline was followed from the maximum speed until the minimal one. This last test achieves a duration greater than 6 hours. Therefore the compressor was shut off until the attainment of 6°C measured on evaporator fins. Such condition is sufficient to guarantee a correct defrosting of the evaporator. The defrosting is shown with the characteristic peak in figure 2.

All tests have been carried out maintaining the cooled vane closed.

Tests at variable inner thermal load

In order to evaluate the refrigerator response to the variation of inner thermal load, an appropriate test at constant compressor speed and ambient temperature of 30°C was planned.

After a temperature pull - down, a series of 2 electric resistances (power of series equal to 795,5 W) located in the cooled vane was switched on until the attainment of a new equilibrium of the system correspondent to a value of T_{in} greater than that previous one (figure 3). At the same temperature and compressor speed conditions, a further test, similar to the previous one, was carried out to characterize system response to the opening of the cooled vane.

Such tests, in the successive phase of system modeling, have allowed to estimate the C step corresponding to the opening and to simulate chiller operation also in such conditions.

Figure 1 shows the trends of compressor electric power and average internal temperature (Tm) relative to both 20°C and 30°C pull – downs and for different compressor speeds.

In particular the first diagram shows the Tm trends relative to na and nb speed levels (with nb greater than na) at 30°C and that one obtained for the nb compressor speed and 20°C in the climatic chamber.

The second diagram, instead, shows the trends of compressor electric power in relation to nc and nd speed levels at 30°C and that one relative to nd speed at 20°C (nd > nc).

It is to be noted that in the tests carried out with climatic chamber at 20°C, the extremely low temperatures achieved are due to compressor over - dimensioning with respect to the considered chiller. Consequently, to make the compressor run at all speed levels, the operational set point of the refrigerator was lowered from T_{in} of 1°C to -12°C.

Moreover, observing the diagrams in figure 1 and 2, arises that all acquired signals are characterized from one oscillation of equal frequency due to the presence of the mechanical valve of expansion, located at the evaporator inlet, that adjusts the coolant flow in relation to the temperature found at the outlet of evaporator.

As it can be seen, oscillations are strongest at the start up due to the fact that the valve operates continuous corrections moving the point of end - evaporation of the coolant to mount and valley of the evaporator outlet. The oscillations then go out during the stabilization of the system. The variation of coolant flow produce correspondent oscillations of P and T_{in} trends.

Smoothing of the oscillation through valve calibration resulted in an excessive reduction in the chiller efficiency. Therefore no modification was implemented.

In order to develop an algorithm for the compressor control, the continuous variations of the controlled variable T_{in} will be held in to account both for the regulator and in the model.

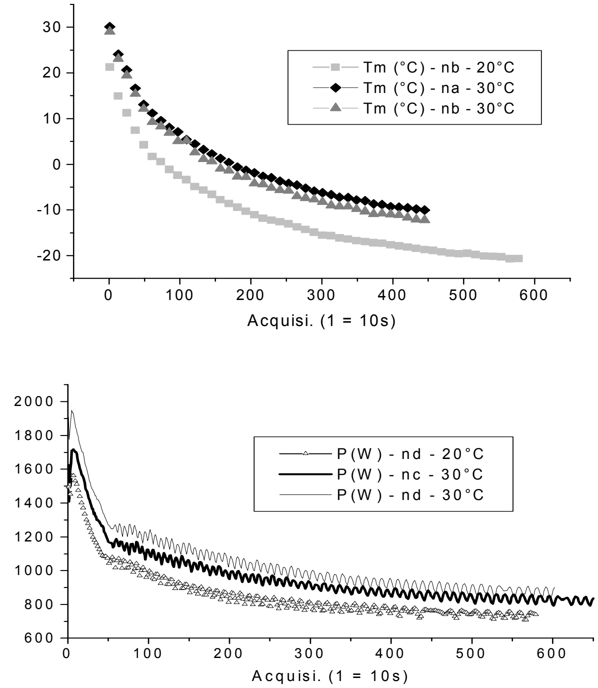


Figure 1

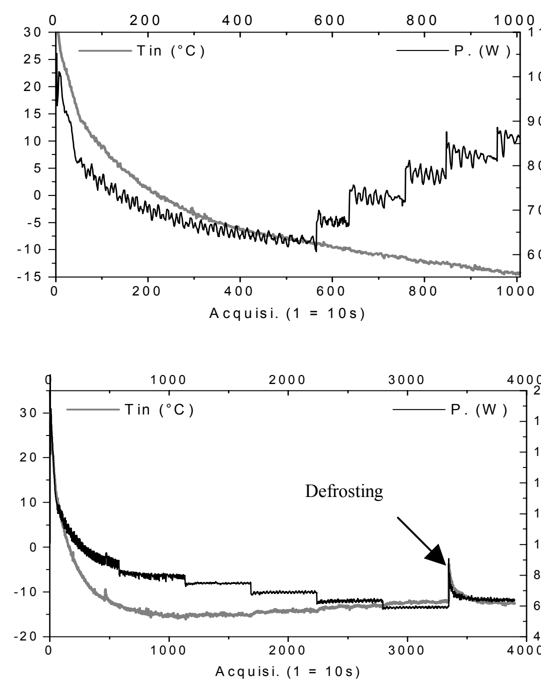


Figure 2

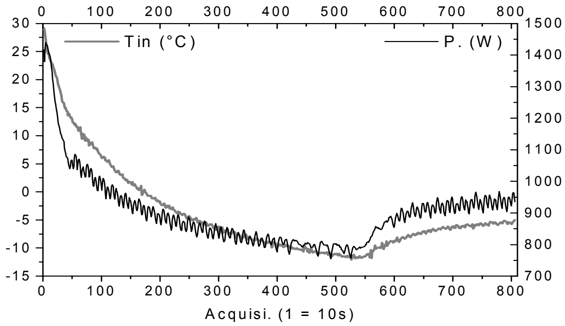


Figure 3

CHILLER MODELING

Observing the T_{in} response to variable compressor speed and thermal loads (fig. 1, 2 and 3) it can be noted that trends are neither divergent nor oscillating, therefore the chiller may be assimilable to a system with real negative poles and eventually zeros. An approximate model of a this system may be described with a transfer function with a real pole and a delay that in this case can be neglected.

A similar conclusion was reached through an energy balance of the cooled vane. From such balance a three inputs differential model of the first order is obtained. The thermal balance is in this case of the form:

$$W_i + C - W_o = m_a * c_{pa} * \frac{dT_{in}}{dt} \qquad (1)$$

Wo is the refrigerating power extracted from the vane proportional to the compressor speed n (Wo = a*n), while Wi is the thermal power entering the cooled vane and proportional to the difference between ambient and inner temperature (Wi = b*((Tamb - Tin)). Moreover, an eventual inner contribution of thermal load was indicated with the term C, while the heat capacity of the air inner the cooled vane, equal to the product m_a*c_{pa}, with the constant c. It follows that:

$$\frac{dT_{in}}{dt} = \frac{b}{c} * (T_{amb} - T_{in}) + \frac{C}{c} - \frac{a}{c} * n \qquad (2)$$

In the Laplace domain a three input transfer function is then obtained:

$$T_{in}(s) = \frac{b/c}{s + b/c} * T_{amb}(s) + \frac{1/c}{s + b/c} * C(s) - \frac{a/c}{s + b/c} * n(s) \qquad (3)$$

The coefficients of expression (3) were determined applying the area - method to the experimental response of the system to compressor speed and inner thermal load step variations. This method evaluates the gain and time constant of a system with one negative real pole from experimental data.

In figure 4 the experimental and simulated trends relative to pull – down tests are shown relatively to T_{amb} value of 30°C. Similar results were obtained with T_{amb} value of 20°C.

The models shown are in acceptable agreement with experimental data and will therefore be used in the simulation.

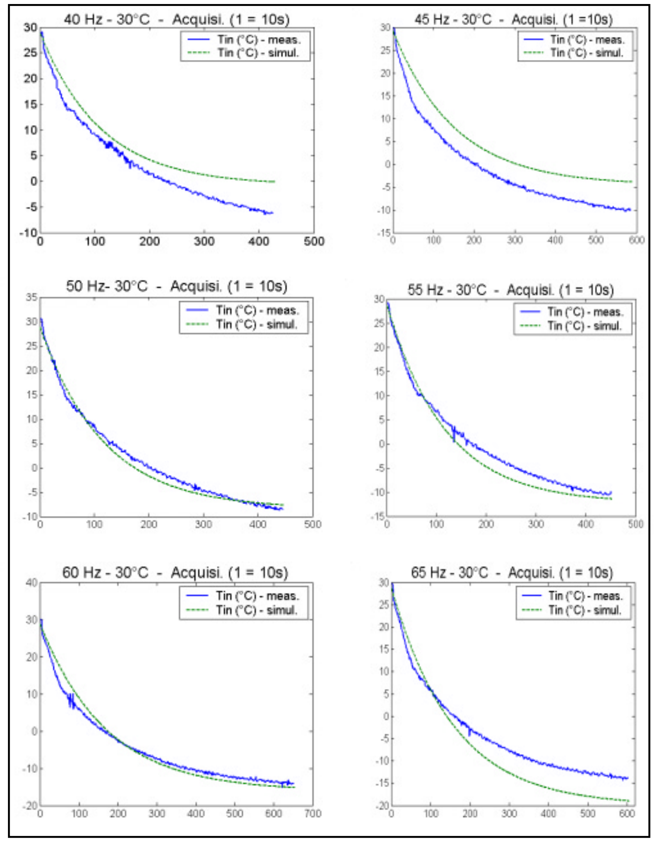


Figure 4: measured and simulated T_{in} data

Moreover, known the gain and the time constant of the system response to C variation, the entity of the C step occurring at the opening of the cooled vane was evaluated, utilising the data shown in figure 3. That allows to simulate the operation of the refrigerator also in the conditions of opened cooled vane.

Finally, in order to test different control algorithms, was necessary to simulate compressor electric absorption P as a function of the speed profile imposed by the regulator.

For this purpose a differential discrete time model is chosen where the system output y(t) at the instant t is calculated as a linear combination of the inputs and the outputs relative to previous instant. Between many possible configurations, the better results were obtained, considering from experimental data, when y(t) is a function of the inputs and outputs of two previous instants.

This is equivalent to consider the discrete time transfer function, as follows:

$$y(z) = \frac{b_1 z + b_2}{z^2 + a_1 z + a_2} * u(z) \qquad (4)$$

In figure 5 the P trend in the acceleration shown in figure 2 at the T_{amb} value of 30°C is compared to the one obtained in simulation. Data obtained are in acceptable agreement with measured ones and can be used in simulation.

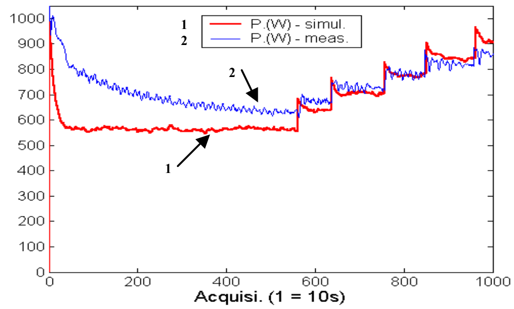


Figure 5: measured and simulated P data

CHILLER CONTROL OPTIMIZATION

The control problem is to bring inner temperature to set point value and its derivative to zero, in presence of disturbances such as the variation of ambient temperature and inner thermal load. In the lower possible time and the minim energy expanse the controller must also take the refrigerator to set conditions respecting the technical specifications of the chiller.

A PID controller [2, 3, 4] was designed through simulations using Simulink Matlab package. In the feedback loop of figure 6 the regulator was set between chiller output variance from set value and chiller input. Moreover in the feedback arrangement a disturbance, similar to the noise produced from the valve of expansion and regulation, was introduced. Noise introduction carried a strong reduction of the proportional constant of the PID to avoid continuous and strong corrections of compressor speed. The controller is moreover planned to obtain better chiller performance by reducing the transient time at the initial start up and those following the defrosting procedure.

Inner temperature, compressor speed and power absorption trends obtained in simulation (simul. 1) similar to figure 3, are shown in figure 7. In particular the first diagram on the left shows the temperature trend obtained with PID logic compared to the constant speed one. The comparison shows that the PID regulator is able to bring back the temperature to the set point value also in presence of a strong increase of the inner thermal load.

When controlling the chiller with a PID control the system behaves as shown in figure 8 where the inner temperature trends are obtained for a pull – down test at 30°C ambient temperature and in absence of disturbance. In particular the first diagram shows the simulated (simul. 2) and measured inner temperature trend obtained with PID control together with the measured data for constant speed.

It can be noticed that the degree of approximation of the system's model causes different trends of the inner temperature for the simulation and real test case. Anyway PID control produces a considerably performance improvement when compared to on – off regulation in which the compressor is maintained at 50Hz constant speed.

In the second and third diagram both simulated and measured trends of inner temperature and compressor speed are represented. Observing these diagrams it can be noticed that the difference between simulated and real speed data are a direct consequence of the difference between temperature data.

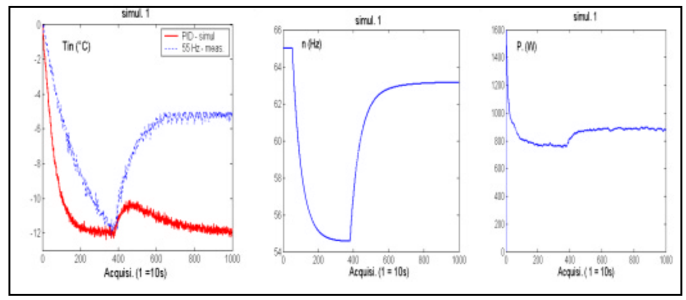


Figure 7: T_{in}, n and P simulated trends relative to the chiller response to the C step of 795,5 W

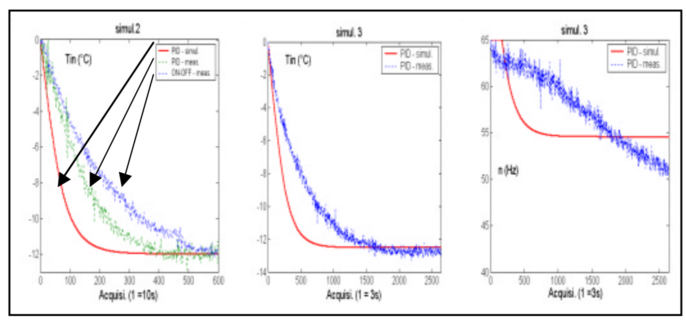


Figure 8: simulated and measured T_{in} trends obtained with PID control and Tin data relative to on – off regulation

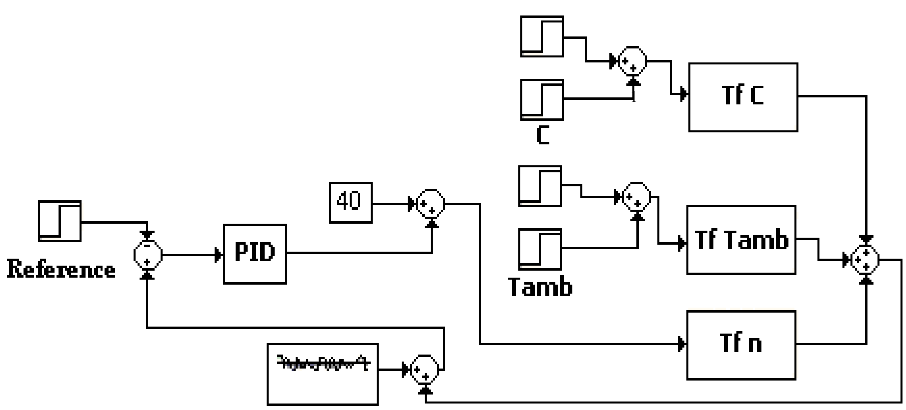


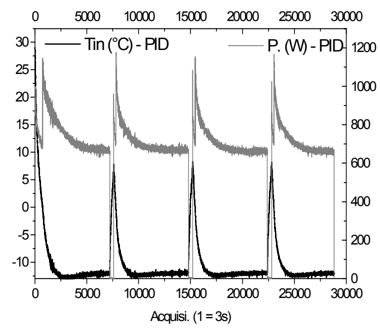
Figure 6: feedback arrangement with PID controller

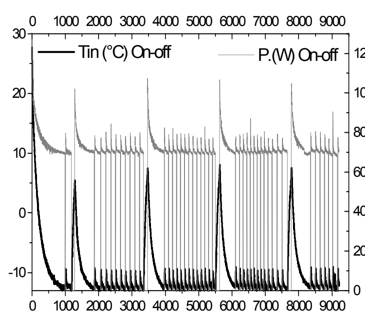
On – off versus PID

The results obtained in simulation show a better performance of PID control compared to on – off running of the same compressor. These results were confirmed when applying the controller to the real refrigerator in a test in the climatic chamber at constant ambient temperature and in absence of disturbances. In figure 9 T_{in} and P trends are shown both for the logic PID (I diagram) and on – off regulation (II diagram). Moreover, in the same figure the profile of compressor speed imposed by the controller is also shown (III diagram). Defrosting was carried out commanding by software the suitable resistances.

Similar improvements were not obtained in terms of energy consumption that remained in the 24 hours almost unchanged compared to the one of on – off test, although the absorption level of the compressor is lower. This because that the shutoffs of the compressor in a test without variations of ambient temperature or inner thermal load, gives to a lower average power value with respect to the one obtained with the PID regulation. This is influenced by the strong peaks of absorption that also happen at every sturt up when the regulator carries the compressor to the maximum speed.

At compressor start up and generally at every variation of speed, the chiller plant experiences a transient until new condition of stabilization is obtained corresponding to new speed of the compressor. This transient is as more important as larger is the speed variation.





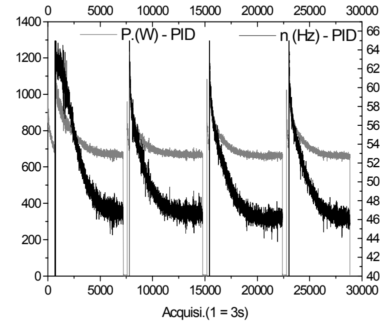


Figure 9: I) Tin and P trends of PID. II) Tin and P trends of on- off. III) P and n trends of PID.

Further conditions were therefore introduced in the control algorithm such as:

- at every start up that happens with T_{in} values greater than 0°C, the speed is set up to the minimum value until the attainment of 0°C (losses in terms of efficiency are not experienced, as it can be noticed from fig.1, until such value because Tin divergent profiles, obtained in the different pull – down tests, are not sensible). Subsequently speed is raised according to an acceleration ramp in which every speed value is maintained for 5 minutes before the activation of the PID controller.
- Once the set point conditions are achieved and maintained T_{in} at values minor or equal to set for at least 10 minutes, the compressor is turned - off and then started up with PID controlling when the same temperature endures an increment of 1.5 °C. Such variation is approximately the half of the one recorded in on – off working. That means that also during maintening of the regimen conditions the logic control of compressor guarantees temperatures near to the set point.

Such regulator was implemented on the real chiller producing, in terms of performance improvement, the same results obtained with the previous test and a reduction of energy consumption of 6% in 24 hours of operation, with respect to the on - off regulation.

The results of the test are shown in figure 10 and confirm what above said. From the first diagram it can be seen like the planned controller yields T_{in} more fastly to the set value after every refrigerator start up.

In the second diagram compressor electric absorption trend is shown relative to the initial chiller start up and to the first period of regimen working.

It can be noticed how P attests to a lower value than that one characterized by on – off running, producing the above said energy consumption reduction.

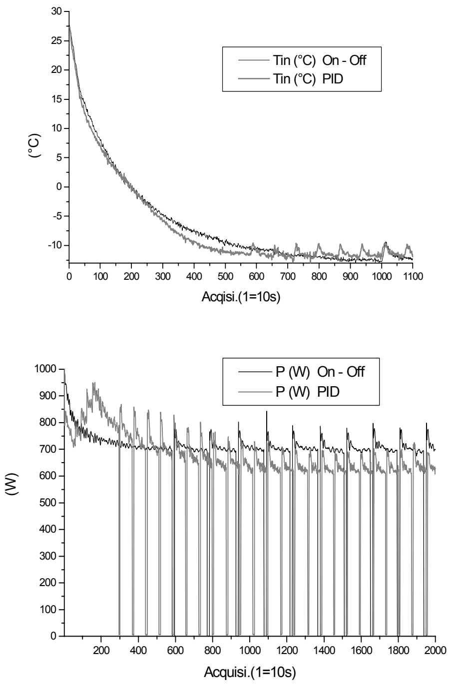


Figure 10: comparison between variable speed control and on – off regulation.

UPGRADING

To maximise the benefits of the simulation tool and to introduce the minimization of energy consumption into the control logic some further developments were considered. Preliminary results are described below.

OPTIMAL LINEAR CONTROL

With optimal linear controlling it is possible to regulate a system taking into account the minimization of control effort function and consequently energy consumption also.

In the optimal linear control a linear controller, that can however contain the integral term, is applied to the refrigerator supposed as a linear system. This controller represents the control law u(t) that minimizes the quadratic performance index defined, for a infinite – time regulator problem applied to a time – invariant system, like:

$$V\left(x(t_0),u(.),t_0\right)=\int_{t_0}^{\infty}\left(u'(t)Ru(t)+x'(t)Qx(t)\right)dt \tag{5}$$

where u and x represent inputs and plant states's variations from set point conditions and R and Q are constant matrices positive and not negative respectively. Defined with this assumption the optimal control law, obtained as the solution of the Riccati equation [5], is of the form u(t) = K*x(t), for some constant matrix K of appropriate dimension

Controller input are the states of the system, assumed that they are measure and available for feedback, while controller output are system input. If the plant states are not available for measurement the state estimator supplies the state vector starting from the system input and output vectors as shown in the scheme of figure 11.

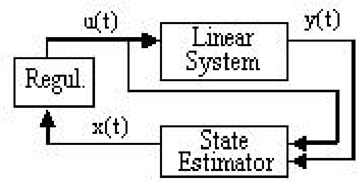


Figure 11: feedback loop

In order to insert the integral term in the control law, a second plant state, in adding to Tin temperature state variable, equal to the integral of Tin, was introduced. The state – space model of Tin - n correlation is in this case of the form:

$$\begin{aligned}\dot{x} &= Ax+Bn\\ y &= Cx\end{aligned} \tag{6}$$

where:

$$x=\left[T_{in};\int T_{in}\right]$$

$$A=\begin{bmatrix}a & 0;1 & 0\end{bmatrix}$$

$$B=[b;0]$$

$$C=\begin{bmatrix}1 & 0\end{bmatrix}$$

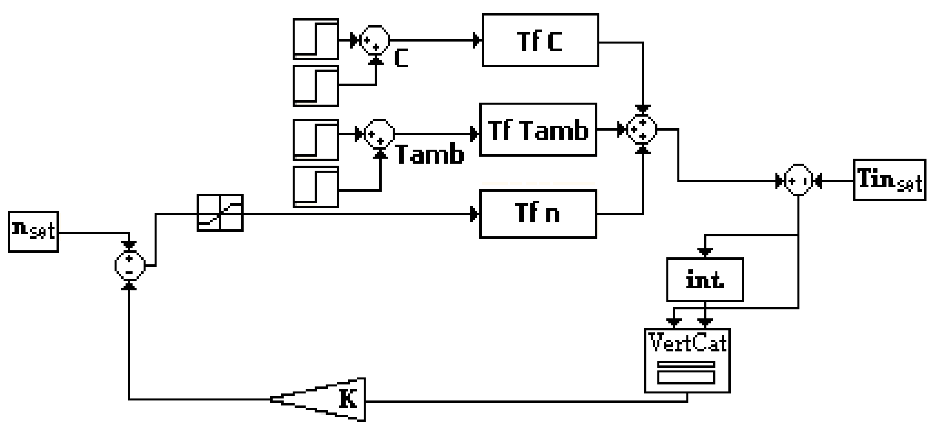


Figure 12: feedback arrangement with optimal

Starting from this state - space model, different optimal regulators were developed and then tested through simulation using the control loop described in figure 12, in which the state estimator is absent because the Tin state is the same as the system's output.

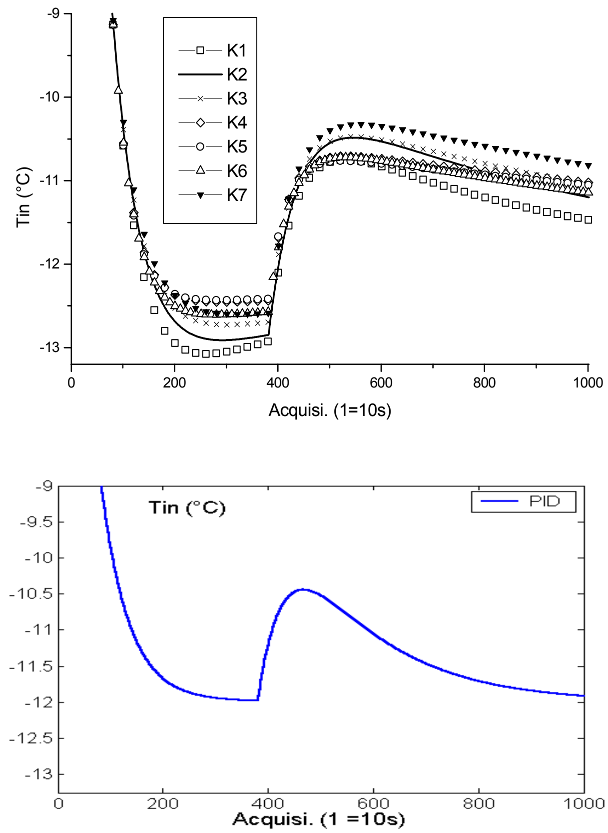


Figure 13: T_{in} trends obtained by simulation of both K1 – K7 optimal controllers and PID regulator

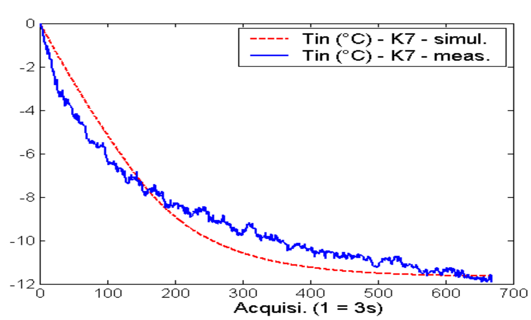


Figure 14: simulated and measured T_{in} trends obtained with K7 controller

In order to test the different regulators a test was carried out, similar to the one shown in figure 3, in which after the stabilization of the system at an ambient temperature of 30°C, an inner thermal load step of 795,5 W is imposed and maintained until the end of the test. The total duration is of approximately 167 minutes.

Such test was chosen because it demands a response of the system to a disturbance of considerably greater entity with respect to those that characterize the normal operation of the refrigerator. A good performance of the regulator in this test, can therefore be endured to normal operation.

Figure 13 shows the result obtained for the regulators from K1 to K7 where the differences are obtained by weighing the variations of the plant states in minor way with respect to the variation of the inputs, in the quadratic performance index. The performances obtained with the controller K7 (marked with down triangle in the I diagram) can be thought acceptable in relation to the chiller technical specifications, even if inferior to those of PID regulator (II diagram), involving an energetic saving approximately equal to 1%.

Figure 14 shows T_{in} simulated and measured trends obtained with K7 controller for a pull - down test.

Subsequently the K7 regulator was used to control the real chiller with the same procedure above described for the PID control algorithm. On a 24 hours operation, a reduction in energy consumption, with respect to the on - off regulation, greater than the one relative to PID control was achieved.

Finally figure 15 shows P and T_{in} trends both for PID regulator and K7 controller. The comparison confirms the results of the previous simulations: K7 controller yields T_{in} close to the set point in a greater time (comparable to that one recorded during on – off operation) than PID, while obtaining a lower absorption level of the compressor.

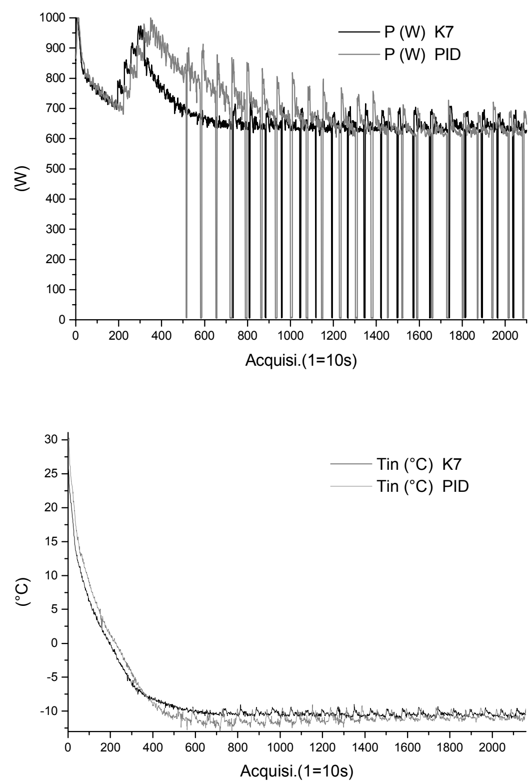


Figure 15: comparison between PID regulation and linear optimal control

NEURAL MODELING

To improve the performance of the chiller simulation model a neural – network approach seems suitable because better approximates a not linear and complex system [6, 7, 8].

To such purpose a neurale model of the refrigerator, based on the Back - Error propagation algorithm, was developed. In such model, valid in order to simulate the chiller operation at constante T_{amb} equal to 20°C, the value of T_{in} at the instant t is obtained from the previous values of compressor speed n, T_{in} and evaporator temperature and the current value of n.

The neural model was trained and tested on data sets obtained from 6 pull – down curves, 1 acceleration and 1 deceleration curve. The training set consists of 8164 values while the testing set consists of 3632. Different network architectures were considered, obtaining the best results, characterized by an error equal to 1.38% committed in the test, for a network with a 2 hidden layers and 15 neurons in each of them architecture.

Test and training RMS trends and comparison between measured and simulated data for two pull – down tests are shown in figure 16 demonstrating a better understanding of the trends with respect to data simulated with the classical approach. The encouraging results obtained with the neural simulation call for a wider effort in this direction. This will be carried out as future work after a more comprehensive data harvesting campaign will be completed. A definitive neural model is expected both for optimal software simulation and on board diagnostics.

Finally the introduction of fuzzy – logic [9, 10, 11, 12, 13, 14] control may define the ultimate solution to intelligent control for the optimization of performance and the minimization of energy consumption.

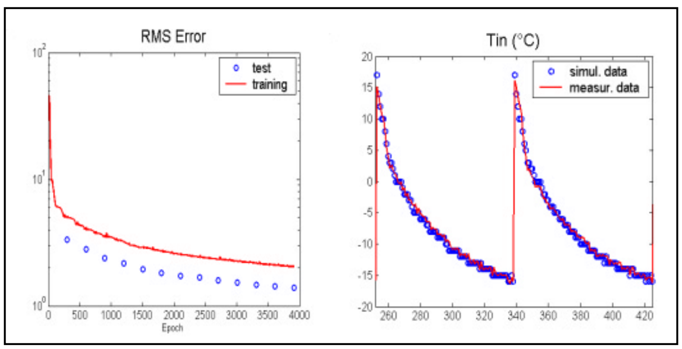


Figure 16: performances of the neural model

CONCLUSIONS

An industrial chiller was characterized in its behaviour at different working conditions by varying the compressor speed through machine modeling and control system simulation. Alternative techniques were validated on the real chiller yielding interesting results.

The results obtained through PID controller implementing, confirm the validity of the compressor variable speed regulation yielding to new solutions planning and application also in the optic of reducing energy consumption.

Preliminary testing on optimal linear controlling and neural modeling gave encouraging results both for energy consumption reduction and performance improving. An integrated neural – linear control loop simulator will be realized in the next future to define the best architecture to test on the machine.

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