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Developing a new cost-efficient control strategy for an actual confectionery plant through the combined exploitation of experimental and numerical analysis

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SUMMARY

Achieving energy absorption reductions while improving indoor air quality is a major task when designing new air conditioning systems. A cost-effective way to improve energy efficiency without compromising the thermal comfort consists of developing better control. In the present work, an extensive experimental campaign has been coupled with a theoretical analysis with an effective approach. A simulation tool has been implemented and, through its predictions, an efficient control strategy has been developed in a system that resulted in significant energy savings and environmental benefits. Copyright © 2003 John Wiley & Sons, Ltd.

KEY WORDS: control; energy saving; hybrid approach

1. INTRODUCTION

Ventilation and air-conditioning (VAC) system efficiency in buildings means not only saving money for the user but also smaller refrigerating plants and a reduced use of electric energy.

The first effect has, as a direct consequence, a reduction in the amount of greenhouse and ozone depleting gases being released in the atmosphere; the second leads to a decrease in the fossil resources usage and in the polluting emissions connected to energy production and conversion.

While in the past, significant attention had been paid by entrepreneurs to energy savings in air conditioning, the lack of scientific knowledge had led to drastic actions resulting in poor indoor air quality (IAQ) which determined, consequently, the widespread belief that energy saving is in direct conflict with users' thermal comfort.

The recent policies of sustainable growth and improvement of the quality of life constitute an important step in the progressive abandonment of this trend. This implies important stimulus for a renewed research activity with the aim of finding energy saving measures that donot compromise IAQ (Parent *et al.*, 1998).

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A cost-effective way to improve the energy efficiency of VAC systems, without compromising IAQ, consists of implementing a better control. It is important to notice that the requirement of a new control system does not directly imply an incorrect design of the original system. A factory is, in fact a living essence, with continuous changes in the production lines thus bringing about not only increased complexity of the systems but also changes both in the thermal loads and in the required thermohygrometric set points. Introduction of a more advanced control system also adds greater flexibility to the VAC system due to its ability to self-adapt its working parameters to different conditions.

While, on the basis of these considerations, the benefits of this action can be easily understood, the resulting changes in energy consumption and internal comfort are often difficult to predict (Mathews *et al*., 2000). To achieve these predictions, due to the enormous complexity of the system and to the large number of factors that affect it, a simulation tool that can efficiently and accurately reproduce the building with its integrated VAC and control system is required (Gadi, 2000).

One of the most important aspects of producing an accurate object with an affordable monetary and time investment, is the modelling approach. The modelling approach used consists of singling out three macro steps: identification, estimation and validation as in Andersen *et al*. (2000). The most efficient way to reach the mentioned target appeared to be an integrated hybrid (experimental and theoretical) analysis in all three macro steps.

The work resulted in a new control strategy that improves both energy savings and thermal comfort. These necessities prompted the plant management toward implementing a new control system.

2. TEST ZONE AND VAC SYSTEM DESCRIPTION

The plant under analysis belongs to a confectionery factory. The main production building of the factory is divided in different areas each with its peculiar temperature and humidity requirements. The building's main air-conditioning system consists of eight air treatment stations.

A part of the production building has been chosen to be used as a test cases. This part is constituted by the 'chocolate zone', where chocolate is produced from the milled cocoa beans. Since this zone is not physically separated by walls from the 'toasting plant zone' and air is free to flow from one zone to the other, both zones are shown in Figure 1.

The present set point temperature is 29°C, while the minimum required temperature is about 26°C and it is linked to the problem of freezing the fluid chocolate in the pipes.

Two air treatment stations serve the two zones. The situation of the 'toasting plant zone' is in reality more complex since it is located on two floors and also utilises some autonomous cooling systems. The stations are placed inside box girders just under the shed roof. Their location is described in Figure 1 (shaded areas). Each air treatment station is divided in two parts: the north and the south. Due to the similar nature of the stations, only the south part of the 'chocolate' will be analysed. The main components of each station are filter, electric fan and heat exchange battery. The heat exchange battery was originally designed to provide heating requirements; afterwards, due to the increased internal thermal load of the zone, it was used as a cold battery for the zone cooling. The dimensions of the station are $5 \times 4 \times 21\,\mathrm{m}^3$. From it, longitudinal box girders (vertical lines in Figure 1) bring the return air in its top end part and the outlet air in its

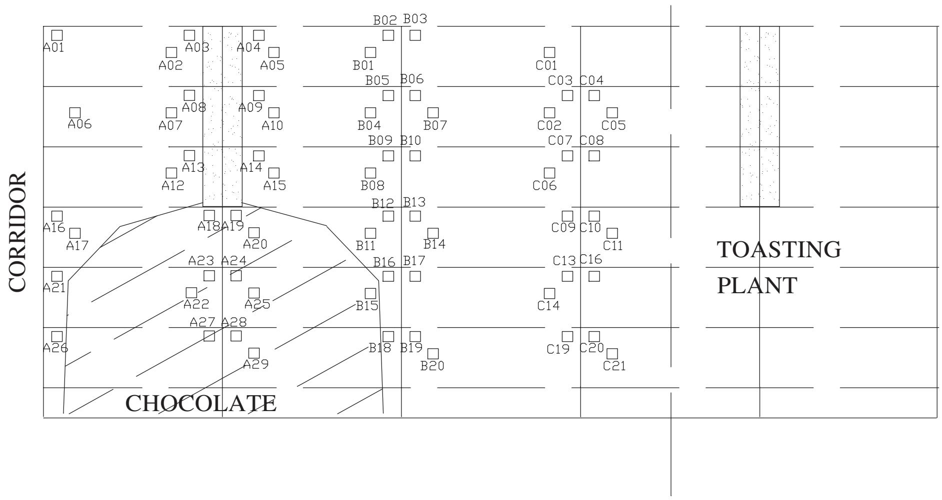


Figure 1. Chocolate zone air-conditioning system, and south influence area.

lower end part. For every 7 m there is a transversal duct (horizontal lines in Figure 1) in which the inlet draw holes are placed. Over the roof, there is a box with lateral openings where manual regulating registers allow the entry of external air so that the return air can be mixed with it. At present the station works with 100% recirculated air. On the roof of the zone, a large number of extraction electric fans are present. Some of them were positioned to compensate for the entry of external air. In the following years, their number increased since it was believed that they could improve the IAQ and the sum of their nominal flow rates is now greatly superior to the nominal flow rate of the station fan.

Due to the absence of partitioning walls and necessity of simplifying the analysis, identification of a pertinence zone of the south part of the 'chocolate' air treatment station has been performed assuming that no heat exchange between this and the neighbouring zones takes place. This zone is represented by the hatched area in Figure 1.

3. THE MODEL DEFINITION

One of the most important aspects when researching is to identify the most suitable approach to the problem. Its importance is fundamental since it must guarantee sufficient accuracy and save time and financial resources. The classical approaches to such problems are experimental and numerical.

In the present case, while the huge amount of complex realities entailed the impossibility of constructing a physical model, the absence of a complete set of data over the entire range and the huge amount of time and resources required to obtain them, especially due to the necessity to vary each individual parameter to investigate its influence better, did not allow to simply utilize experimental analysis. The selected approach was hybrid one that attempted to integrate theoretical and experimental analysis. An alternative method of conducting such complex

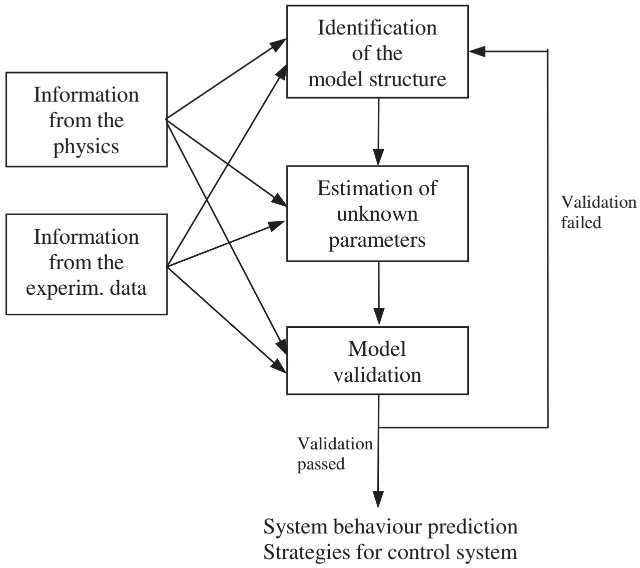


Figure 2. Representation of the modelling approach.

investigations is to resort to blackbox analysis (Grimaldi and Mariani, 1996). The approach scheme used in the present study is reported in Figure 2. While hybrid analysis is not new to the trained reader, the novel aspect of this work consists of the complete integration of the two aspects (theoretical and experimental) in all three key stages of the model construction process: identification of the model structure, estimation of unknown parameters and model validation.

A complete formulation of the model will be given in section 5; in any case, at this point it is important to focus and examine some methodological aspects.

In the complex plant analysed, there are many external and operating parameters that may affect the system behaviour. Trying to put all parameters in a model would have been time consuming and would have probably led to an excessive complex model with limited accuracy.

The preliminary theoretical analysis of the first stage has helped to reduce the number of input variables excluding some of them due to their limited relevance and influence; only the experimental analysis was able to give more ad hoc information required to identify the model structure with its equations and relations.

The function of the experimental analysis in the second stage was not limited to the understanding of input parameters range and fluctuations; in fact, while some physics equations are able to track the trend of the outputs, as a function of the relevant inputs, they are less qualified to yield their absolute values (this is due to the lack of a complete knowledge of the real system characteristics). Simplifying the problem, it can be said that, while a law of proportionality is generally known, it is almost impossible to derive from theory the proportionality constant when dealing with real systems.

In the validation stage the relevance of the experimental data was underlined and some macro checks based on consolidated physics equations were also performed. The just described process was iterated until good model accuracy was achieved.

4. EXPERIMENTAL ANALYSIS

Due to the vastness of number of measured variables, differences in techniques used and to variety of sampling rates and acquisition periods, it is not possible to report an extensive documentation on the data acquisition parameters.

The experimental campaign covered the period between July 1999 and October 1999. The measured variables were the following: internal and external temperature and humidity (T_{ea}, X_{ea}, $T_{ia(i)}$, $X_{ia(i)}$,), temperature of water at the inlet and outlet of the heat exchange battery ($T_{w,in}$, $T_{w,out}$), air temperature and humidity at the inlet and outlet of the air treatment station ($T_{a,in}$, $X_{a,in}$, $T_{a,out}$, $X_{a,out}$), air flow rate across the air treatment station (m_{ac}) and its electric absorption (W) and extraction fans air flow rates ($m_{ef(i)}$). The values of the internal temperatures and humidities use the subscript "$_i$" since they have been monitored in seven different points of the zone and in the proximity of different kinds of production machines.

In this paper, only the figures of the temperatures are reported. This is a consequence of the fact that the humidity analysis is not as fundamental as the temperature one: the high internal temperature set point, in fact, causes the absolute external humidity content to be generally inferior to the bounds set for the internal one even when the external relative humidity is high.

Since an extensive analysis of the diagrams is not feasible in this context, only some macroscopic considerations are here reported. Some results extrapolated in Figures 3–5 are shown in section 5.

A quick glance at Figure 3 shows how, even in summer, due to the high set point temperature of the 'chocolate' zone, the external temperature is often lower than the internal one. As a consequence a choice was made to examine the possibility of exploiting external air for air conditioning since its lower enthalpy content can determine energy savings. At the end of

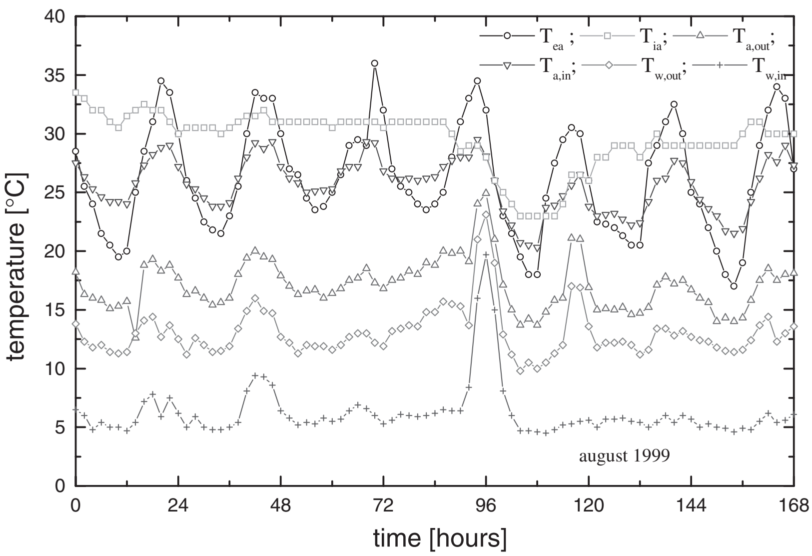


Figure 3. Temperatures of external, internal and conditioning air; temperatures of cooling water in August 1999.

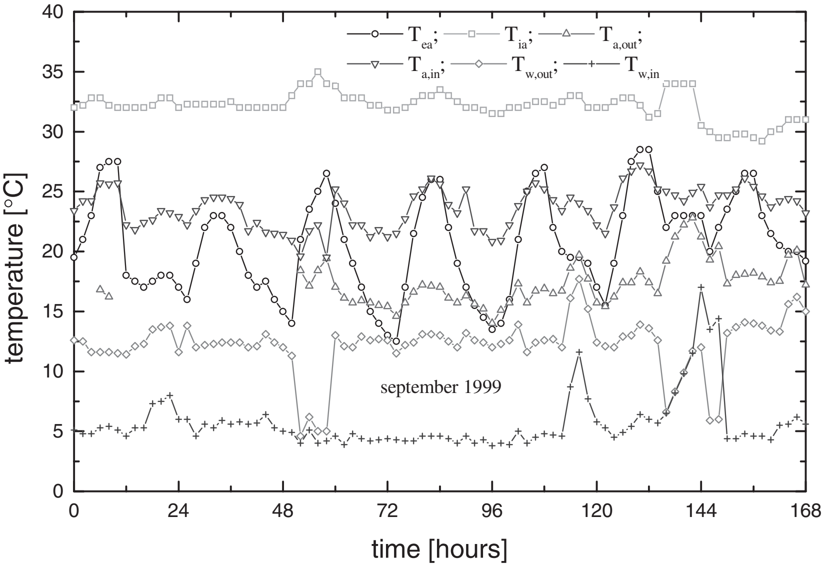


Figure 4. Temperatures of external, internal and conditioning air; temperatures of cooling water in September 1999.

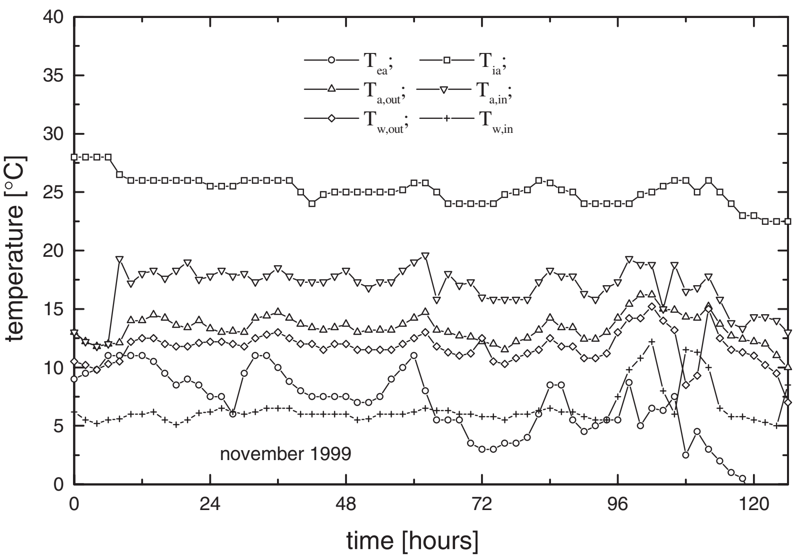


Figure 5. Temperatures of external, internal and conditioning air; temperatures of cooling water in November 1999.

summer, the outdoor temperature falls even below the air outlet temperature of the treatment station. The exploitation of this air would consequently allow the shut-up of the cooling machines thus demonstrating the possibility of free cooling and validating the considerations contained in Grimaldi *et al*. (2000). In winter, when the outdoor temperature is below that of the water at the inlet in the heat exchange battery, it is even possible to exploit external air to contribute in producing the cold water necessary in the plant: the air treatment station becomes a refrigerating machine (where the water gets cold and the air gets hot), allowing energy savings in the winter season when electricity prices are the highest (referring to the Italian average electric energy prices).

Apart from providing all this invaluable information, the same experimental campaign was fundamental in pointing out the main mass and heat fluxes and in providing real values for the energy balances giving reference points for the following modelization phase.

Figure 6 schematically describes the main fluxes coming out of and going into the zone under investigation.

In stationary conditions, the mass balance equation derived from Figure 6 is

$$\rho_{ea} m_{ea} + \Sigma \rho_{ef} + \Sigma \rho_{op} m_{op} = 0 \tag{1}$$

where $\Sigma\rho_{op}\, m_{op}$ represents the mass flow rate coming from outside, from the adjoining zones through various openings and through infiltrations; m_{ea} represents the eventual air introduced from the outside by the air-conditioning system in a 'free cooling' strategy and $\Sigma\rho_{ef}\, m_{ef}$ represents the air mass flow rate extracted by the air extraction fans.

The air flow rates in the air treatment stations ($\rho_{ea}\, m_{ea}$) and those in the extraction fans ($\Sigma\, \rho_{ef}\, m_{ef}$) have been measured while the value of the mass coming from the openings ($\Sigma\, \rho_{op}\, m_{op}$) has been derived.

Once the mass balance equation has been solved, the attention was put on the energy balance. Since an accurate evaluation of all the ambient thermal loads ($\Sigma\, q_{s,tl} + \Sigma\, q_{l,tl}$) was not conceivable, the experimental analysis has been employed to give an estimation by means of an indirect approach. In stationary conditions (this is the aim of a good control system), the thermal balance equation can be written as follows:

$$\begin{aligned}\Sigma q_{s,tl} &+ \Sigma q_{l,tl} + q_{s,ac} + q_{l,ac} + \Sigma h_{s,ea}\rho_{ea} m_{ea} + \Sigma h_{l,ea}\rho_{ea} m_{ea} + \Sigma h_{s,ef}\rho_{ef} m_{ef} \\ &+ \Sigma h_{l,ef}\rho_{ef} m_{ef} + \Sigma h_{s,op}\rho_{op} m_{op} + \Sigma h_{l,op}\rho_{op} m_{op} = 0\end{aligned} \tag{2}$$

where h represents the specific enthalpy, ρ is the density and m the volume flow rate.

$\Sigma q_{s,tl}$ and $\Sigma q_{l,tl}$ represent the sum of all the ambient thermal loads due to: presence of human beings, production machines, auxiliary and service systems, hot products, heat coming from the outside through the irradiation from the windows and through the convection from the external

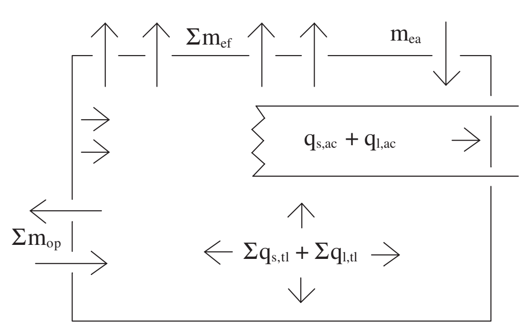


Figure 6. Major heat and mass flows rate in the investigated zone.

Table I. Sample heat flow rates estimated through the experimental analysis (year 2000).

	13th July			23rd September			17th November		
Time	6:00h	16:00h	22:00h	6:00h	16:00h	22:00h	6:00h	16:00h	22:00h
Thermal power ($J\,s^{-1}$)									
$q_{s,ac}$	−200	−250	−250	−170	−250	−200	−120	−140	−125
$q_{l,ac}$	−35	−35	−35	−15	−35	−15	−15	−15	−80
$q_{s,ea}$	−35	20	−5	−50	−20	−50	−60	−50	−70
$q_{l,ea}$	10	−20	−30	−10	−15	−10	−15	−25	−20
$q_{s,op}$	−10	−15	−10	−30	−25	−25	0	0	0
$q_{l,op}$	15	0	0	−5	−5	−10	0	0	0
q_{tl}	255	300	330	280	350	310	210	230	295

and internal walls and from other minor sources. $\Sigma q_{s,ac}$ and $\Sigma q_{l,ac}$ represent the latent and sensible heat added to the air by the air-conditioning system. The measurements of the external air velocities and the following evaluation of the air flow rates, coming from the openings, were used together with the measured temperatures and humidity to estimate the main heat flow rates (Chou and chang, 1997).

The heat exchange in the air treatment station $q_{s,ac}$ and $q_{l,ac}$ has been evaluated by measuring the air and water mass flow rates and temperatures and then using the following equation:

$$q_{s,ac} + q_{l,ac} = m_w c_{pw} \Delta T_w = m_a \Delta h_a \tag{3}$$

The measurements were also used to give an estimation of the water mass flow rate according to Equation (2). To give an idea of the order of the thermal power involved some results are reported in Table I for some representative days. All these results have been usefully exploited in implementing the model.

5. FORMULATION OF THE MODEL

Since the final aim of the work is to define a control strategy that allows energy savings, once the factory requirements were defined, the following step consisted in evaluating the sets of the air conditioning system operating parameters for each set of thermal loads, for every external climatic condition and for every target internal condition like electric fan rotational speed, cold water flow rate and free cooling gate opening that minimizes electrical absorption.

In particular, the aim of the model is to allow a simulation in all the possible and potential conditions of the selected zone in terms of input parameters, in order to predict the feasibility of establishing the desired target conditions to examine the performance of the control system and its resulting potential savings.

The main input variables are: the external temperature and humidity, and the ambient thermal loads. The manipulating variables are: air treatment station electric fan speed, cooling water mass flow rate and ratio between the external and re-circulating air in the air treatment station. The boundary conditions are the internal temperature and humidity set point. The target is to minimize the total electric energy consumption.

In each condition within the inputs variations, the tool must show if the variation of the selected air-conditioning system variables inside the desired range can guarantee that the target is achieved; among all the admissible solutions the optimization tool must point out the best solution in terms of total electric energy consumption.

Rewriting Equation (3) it can be pointed out that, in order to maintain the desired conditions, the heat exchanger must subtract the following amount of heat:

$$q_{\mathrm{req}} = \Sigma q_{\mathrm{s,ac}} + \Sigma q_{\mathrm{l,ac}} + \Sigma h_{\mathrm{s,ea}}\rho_{\mathrm{ea}}m_{\mathrm{ea}} + \Sigma h_{\mathrm{l,ea}}\rho_{\mathrm{ea}}m_{\mathrm{ea}} + \Sigma h_{\mathrm{s,ef}}\rho_{\mathrm{ef}}m_{\mathrm{ef}} + \Sigma h_{\mathrm{l,ef}}\rho_{\mathrm{ef}}m_{\mathrm{ef}} + \Sigma h_{\mathrm{s,op}}\rho_{\mathrm{op}}m_{\mathrm{op}} + \Sigma h_{\mathrm{l,op}}\rho_{\mathrm{op}}m_{\mathrm{op}} \tag{4}$$

Among other things, the actual subtracted heat is a function of the hot and cold fluid properties, their mass flow rates and of the heat exchanger surface. It can be evaluated using the number of transfer unit method (ε-NTU):[‡]

$$q_{\mathrm{act}} = \varepsilon q_{\mathrm{max}} \tag{5}$$

where ε is the effectiveness and q_{max} the maximum amount of heat ideally exchanged, expressed by

$$q_{\mathrm{max}} = C_{\mathrm{min}}(T_{\mathrm{hf,in}} - T_{\mathrm{cf,in}}) \tag{6}$$

where $T_{\mathrm{hf,in}}$ and $T_{\mathrm{cf,in}}$ are the input temperature of the hot and cold fluids, respectively.

To evaluate q_{max}, the flow with the minimum heat capacity must be chosen:

$$C_{\mathrm{min}} = \min(\rho m c_p) \tag{7}$$

Kays and London (1964) showed that the effectiveness is, for a given heat exchanger, a function of the ratio of the heat capacity of the two fluids and of the adimensional parameter NTU defined by

$$\mathrm{NTU} = UA/C_{\mathrm{min}} \tag{8}$$

where U is the global heat exchange coefficient and A is the total surface. The global heat exchange coefficient has been determined using the results of experimental analysis.

For each given set of input parameters, the operational variables are changed until the actual heat exchange matches the required one. It is important to notice that the ambient thermal loads are a function of values of operational variables; e.g. the amount of heat released in the ambient by the electric fan is a function of the electric power absorption which depends on the fan speed.

Of course some assumptions have been made to determine the laws of variation of all the quantities in response to the change of operational variables. This has been aided by the theoretical and experimental analysis. To express the variation of the conditioned air quantity versus the fan speed, for instance, both a circuit characteristic law and a fan curve dependence with n have been employed; the experimental analysis has determined proportionality constants. The final task of the tool is to select the set of operational variables that minimize electric consumption. The main energy absorption has been found to be the one due to air treatment station electric fan, extraction fans, cooling water pumps and refrigerating machines. The consumption of all the electric machines has been determined as a function of variables and implemented in the model; e.g. for the air treatment electric fan consumption law a cubic dependence with n has been chosen, starting from theoretical considerations. A great research

[‡] $q_{\mathrm{act}} = C_{\mathrm{hf}}(T_{\mathrm{hf,in}} - T_{\mathrm{hf,out}}) = C_{\mathrm{cf}}(T_{\mathrm{cf,out}} - T_{\mathrm{cf,in}})$.

effort has been dedicated to estimate the specific electric absorption of the refrigerating machines, i.e. the absorption due to a unitary heat subtraction. This has implied both determination of machines COPs laws and quantification of the global system performance (piping and heat exchangers heat losses): a global average efficiency of 3.3 has been experimentally evaluated.

6. RESULTS DISCUSSION

The first simulations were performed for validation purposes; in this step the results from the final model were compared with those coming from the experimental ones. Finally, through the use of the simulation tool, the feasibility of the proposed control strategy was investigated. With reference to this, with the aim of the minimum electric energy consumption research, different simulations, for different external air conditions and for different thermal loads, were executed; for the sake of brevity the corresponding curves havenot been reported.

In Figure 7 three of the optimized manipulating variables are reported, as a function of the external temperature, for a target internal temperature of 27°C. As can be seen, this target temperature is 2° below the present situation reinforcing the strategy of increasing system performances and improving thermal comfort at the same time.

In this figure, the results relative to a rather high factory production rate are presented. A quick glance at the figure shows three important macroscopic results. First of all, in the lower values of the temperature range, it is possible to exploit the beneficial cooling effect of the external air that, having an enthalpy content lower than the internal one, results in a negative

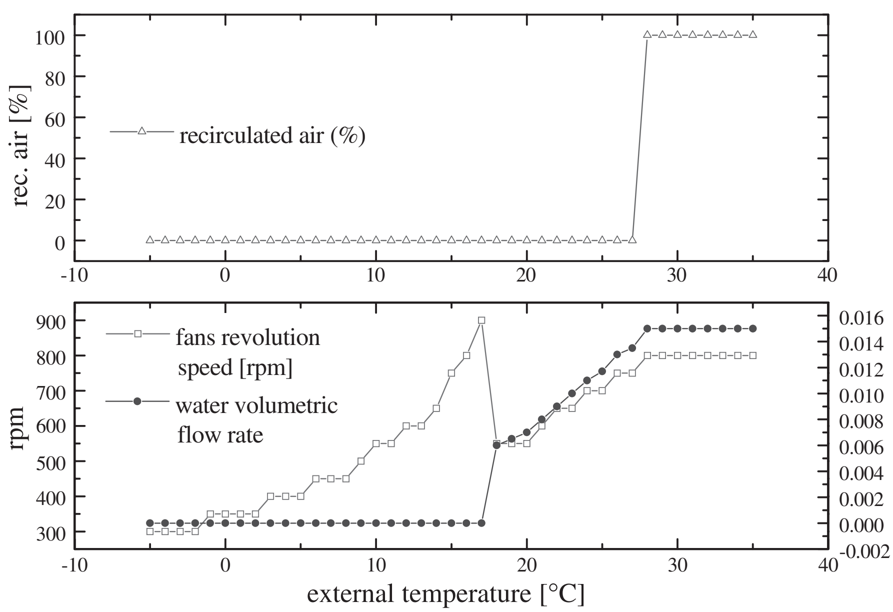


Figure 7. Trends of the principal manipulated variables as a function of the external temperature.

thermal loads. This results in a decreased energy consumption. The second result is the possibility of shutting off the water cooling system (refrigerating machines, circulating pumps, etc.): when the external temperature is below 17°C, m_w is set to zero. This means that a complete free cooling, without the aid of the refrigerating machines, is possible. The third effect is the possibility to use the heat exchange battery as a cooling system for the refrigerated water. This is possible when the external temperature is below the temperature of the water coming in the air treatment station (5–6°C): this allows shutting off further refrigerating machines, if not put off they can work for other factory cooling requirements.

The analysis of the diagram shows that optimization of electric absorption implies fully exploitation of the external air (%rec = 0) each time its temperature is lower than that of the internal one. Moreover, it can be deduced that the fan electric engine absorption is less intensive than the one due to the refrigerating machines. This brings to $m_w = 0$ until the fan reaches its maximum rotation speed ($T_e = 17$°C). At this point it is necessary to switch on the refrigerating

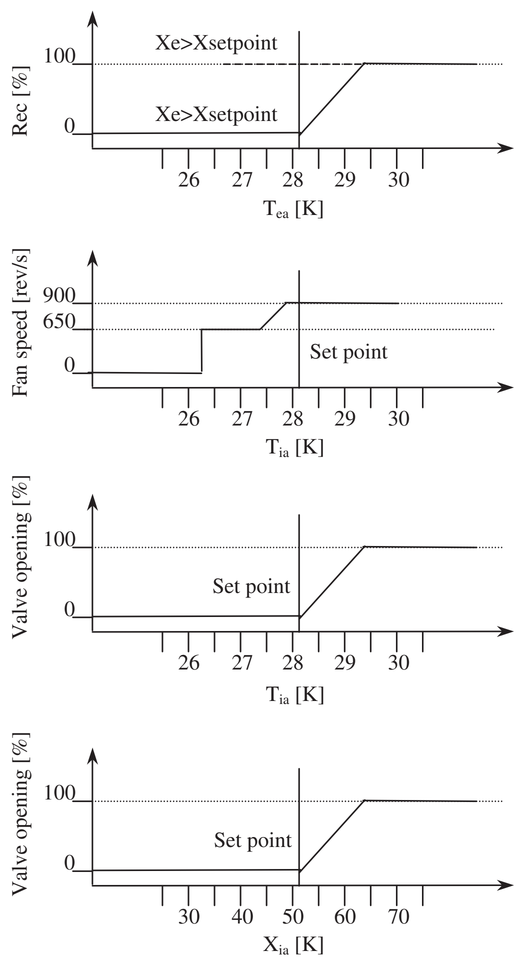


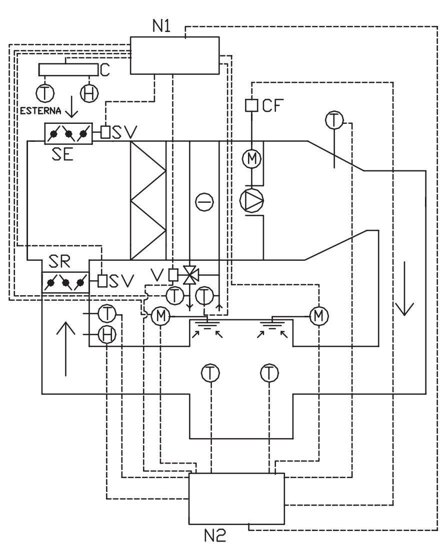
Figure 8. Air conditioning new control strategy.

machines and it is convenient to partially reduce the electric fan speed. When the external temperature is about 19°C, the electric fan works at about 65% of its maximum speed. After this point the electric fan speed increases again and reaches its maximum, together with the water mass flow rate, when the external temperature equals the internal one. From this point all the manipulated variables are constant.

The implementation of the control strategy accurately follows the rules of the optimization analysis except for one point. The electric fan speed does not regulate starting from zero to its maximum value: due to instability problems at low rotation speeds, a minimum value has been fixed. The logic of control is reported in Figure 8, where it can be easily seen that the electric fan starts to work at a lower temperature compared to the set point one. This allows the refrigerating machines, which are the most onerous, to switch on only when the fan is at maximum speed and is not enough to cool the system. At the set point temperature the valve of the cooling water opens guaranteeing that the zone is cooled under all conditions.

The recirculating percentage is also reported. The exploitation of the external air is employed every time its temperature is lower than the internal one, with the only exception of the rare case where its humidity content is too high (dashed line).

The scheme implementing this strategy is represented in Figure 9 where, for simplicity reasons, only the south part of the air treatment station is represented. The control system is governed by the two central units N1 and N2. The central unit N1 is mostly dedicated to opening and closing the recirculating gates and the external gates. It receives the external temperature and humidity and after a comparison with the internal ones it decides which is the most convenient action to perform on the gates engines. Another function carried out by the central unit N1 is to act on the three-way valve of the cooling water system in response to the external temperature and to the outlet water temperature to avoid water freezing. To avoid this last problem, a partial closing of the external gates is also employed.



CF	fan speed control
H	humiditysensor
M	fan engine
N1	first central unit
N2	second central unit
SE	gates for external air
SR	gates for recirculated air
SV	gate control
T	temperature sensor
V	three way valve control

Figure 9. Schematic representation of the air treatment station with the new control system.

The central unit N2 acts mostly on the air treatment station electric fan speed and on the cooling water three-way valve. It compares the internal thermohygrometric conditions with the target ones and then it acts on the mentioned variables giving priority to the fan speeding up.

The simulation of the zone cooling using this control strategy gave satisfactory results. The savings estimated, by the simulation tool, for the south part of the zone was about 180 000 kWh per year, leading to an estimated savings of more than 1 GWh for the chocolate and toaster zones.

7. CONCLUSIONS

In this work an example, how a full integration of a hybrid experimental–theoretical analysis can lead to model accuracy with reduced waste of time and resources is given. The experimental analysis has proved to be precious in order to suggest optimization strategies (like the choice of the air with minimal enthalpy content), to estimate the most relevant thermal and mass flow rates in some working conditions, and to point out the most influent variables. This has prompted a model with reduced parameters and improved accuracy due to the introduction of optimized proportionality constants in the physical laws. The tool constitutes an important instrument in the development of the control strategy, thanks to the global energy minimization subroutine. The manipulated variables which resulted are air treatment station electric fan speed, cooling water mass flow rate, rate of recirculated air flow rate to total air flow rate and the number of active extraction fans. The resulting control system was described and the potential energy savings were quantified. The system is now in the testing phase.

NOMENCLATURE

A	= total exchange surface (m^2)
C	= flow stream heat capacity rate ($J\,K^{-1}s^{-1}$)
COP	= Coefficient of performance
c_p	= specific heat of fluid at constant pressure ($J\,K^{-1}\,kg^{-1}$)
h	= specific enthalpy ($J\,kg^{-1}$)
m	= air volumetric flow rate ($m^3\,s^{-1}$)
NTU	= number of heat transfer unit
q	= heat transfer rate (W)
rpm	= revolutions per minute
T	= temperature (K)
U	= global heat exchange coefficient ($J\,K^{-1}\,m^{-2}\,s^{-1}$)
W	= electric power (kW)
X	= humidity content (g of water/kg of dry air)

Subscripts

a	= air
ac	= air conditioning

act = actual
cf = cold fluid
ea = external air
ef = extraction fans
hf = hot fluid
ia = internal air
in = input
l = latent
op = openings
out = output
req = required
s = sensible
w = water
tl = thermal loads

Greek letters

ε = exchanger effectiveness
ρ = density ($\mathrm{kg\,m^{-3}}$)
Σ = sum

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