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SIMULATION OF HVAC PLANTS IN 2 BRAZILIAN CITIES USING MATLAB/SIMULINK

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ABSTRACT

In this paper, the mathematical model of a HVAC (Heating, Ventilation and Air-Conditioning) equipment is described. This system is inserted into a computer code developed in Matlab/Simulink platform which is devoted to the analysis of building thermal behavior and HVAC closed-loop control performance. The model is presented in terms of non-linear state-space equations that represent the energy and mass balances for each HVAC component. Two control loop structures are described and analyzed. The IMC structure is used for the PID controllers tuning procedure. Finally, comparative results are provided in terms of energy efficiency and indoor air temperature and humidity control performance by using data of two Brazilian cities.

INTRODUCTION

Initially, the main concern of HVAC systems manufacturers was to maintain the building conditions in predefined values related to occupants' thermal comfort. However, in the early 70's, the high-cost of energy boosted interest in energy efficient equipment. In this sense, HVAC systems parameter optimization is necessary in order to reduce their energy costs, since they are responsible for a great part of energy consumption, especially in comercial buildings. For such optimization, a mathematical model of each component is required. However, the modeling procedure for each HVAC component is a difficult task due to the complexity of the system.

In this context, some research has been focused on HVAC modeling. For instance, Wang et. al (2002) have used heat transfer and energy balance principles to identify a linear model to represent a non-linear model of a cooling coil, which is one of the most difficult components to mathematically describe. Clarke (2001) has presented a transient model for a secondary system for each component such as humidifier, re-heater and mixing box, but no specific model for cooling coil was given. Several authors, such as Barbosa and Mendes (2003), have analyzed those models by testing a fan-coil performance for humid and dry conditions. In their work, a simple model for the cooling coil that releases a constant heat flux to the primary system has been used. Brandemuel et al. (1993) presented also models for simulating secondary HVAC systems. Based on both mentioned models, Nassif et al. (2003) presented a validation of the cooling coil, mixture box and a fan for a VAV system (Variable Air Volume).

Moreover, many other authors have also conducted research on HVAC system modeling, focusing on cooling coil modeling, such as Chow *et al.* (1997), Strand *et al.* (2002), Henderson *et al.* (2003).

An issue related to thermal systems modeling, such as HVAC systems, is the use of an appropriate simulation tool. Thus, Lebrun (2001) presented a simulation of a HVAC system by using the so-called EES (Engineering Equation Solver). Therefore, by defining simplified models of each HVAC component, a group of polynomial equations can be generated and easily connected to the building. Yu and Paassen (2004) presented a comparison between two modeling approaches of an air-conditioned room using Simulink and Bond graph.

Recently, some Matlab/Simulink-based simulation tools have been developed to performance simulation of HVAC systems integrated to a building thermal model. Riederer et al. (2001) presented the SIMBAD Building and HVAC Toolbox, which contains a large number of ready-to-use HVAC models and related utilities to perform dynamic simulation of HVAC plants. For simulation involving automatic control, this toolbox must be connected to other existing toolboxes. Mendes et al. (2003) described a computational environment, developed also in Matlab/Simulink platform, for thermal simulation of buildings and control strategies analysis applied to HVAC systems (the ASTECCA toolkit). This computational tool contains a library of equipment and transport phenomena models that have influence on the thermal performance of the building. Their library contains the following components: i) building; ii) HVAC; iii) sensor; iv) controller; v) weather data and perturbation.

The present paper is focused on the simulation and control of a HVAC integrated to a building zone. In

this way, a model of an air-conditioning system, which is one of the available tools found in the ASTECCA program library, is described. The environment allows testing different kinds of control loops and this paper compares two possibilities. An example of an economizer performance is also presented. The provided comparative results are in terms of energy efficiency and indoor air temperature and humidity control performance for two Brazilian cities.

HVAC SYSTEM

Figure 1 shows schematically a HVAC system such as the one presented by Brandemuel *et al.* (1993). In order to build the whole-system model, a mathematical description of each component of the primary and secondary systems is provided by applying the energy and mass conservation principles.
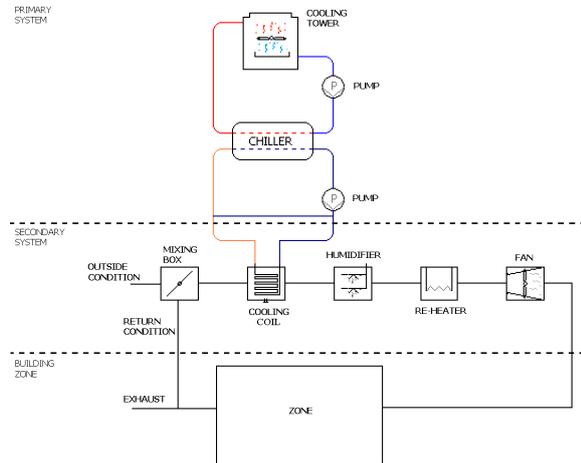

Figure 1. Schematic representation of the primary and secondary systems components.

SECONDARY SYSTEM

The equations that represent each component of the secondary system in steady-state are given by Eqs.1-10.

Component 1 – Mixing box

$$\dot{m}_0 h_0(t) + \dot{m}_r h_r(t) - \dot{m}_1 h_1(t) + q_{e1} = 0 , \qquad (1)$$

$$\dot{m}_0 w_o(t) + \dot{m}_r w_r(t) - \dot{m}_1 w_1(t) = 0 , \qquad (2)$$

Component 2 – Cooling coil

The cooling coil output values of enthalpy (h_2) and humidity ratio (w_2) are obtained by using the model described in Brandemuehl et al. (1993), which presents a detailed model of coil (CCDET), based on the concept of enthalpy effectiveness. The model calculates the convection heat transfer coefficients for wet, dry and partially wet coil, using geometric data of the coil. The calculation of the condensed

vapor mass flow is influenced for conditions of dry, wet and partially wet coil.

Component 3 – Humidifier

$$\dot{m}_2 h_2(t) - \dot{m}_3 h_3(t) + q_{e3} = 0 , \qquad (3)$$

$$\dot{m}_2 w_2(t) - \dot{m}_3 w_3(t) + \dot{m}_u = 0 , \qquad (4)$$

Component 4 – Re-heater

$$\dot{m}_3 h_3(t) - \dot{m}_4 h_4(t) + q_{e4} + q_{x4} = 0 , \qquad (5)$$

$$\dot{m}_3 w_3(t) - \dot{m}_4 w_4(t) = 0 , \qquad (6)$$

Component 5 – Fan

$$\dot{m}_4 h_4(t) - \dot{m}_5 h_5(t) + q_{e5} = 0 , \qquad (7)$$

$$\dot{m}_4 w_4(t) - \dot{m}_5 w_5(t) = 0 , \qquad (8)$$

where i is the corresponding index of each component $(i = 1,\ldots,5)$, $\dot{m}_i$ is the mass flow rate of the air/vapor mixture, h_i is the mixture specific enthalpy, q_{ei} is the heat exchanged with the surrounding components, q_{x2} is the cooling coil total heat transfer, q_{x4} is the re-heater total heat transfer, w_i is the humidity ratio, $\dot{m}_c$ is the cooler vapor extraction rate and $\dot{m}_u$ is the humidifier vapor addition rate. The index r and o are respectively related to the internal room psychometric state and external air ventilation. In steady-state situations, the dry air mass flow rates in components 1 through 5 are identically the same.

CHILLER

For the cold water supply a model of water chiller is presented as:

$$Q_{chiller} = \dot{m}_{H_2O,chiller} c_{p,H_2O} (T_{in,chiller} - T_{out,chiller}) \quad (9)$$

where $Q_{chiller}$ is the chiller capacity, $T_{in,chiller}$ is the entering temperature, obtained from an energy balance at the cooling coil return valve, which is function of cooling coil water flow rate, cooling coil leaving temperature, coil by-pass water flow rate and coil by-pass leaving temperature, $T_{out,chiller}$ is the chiller leaving water temperature (set-point), $\dot{m}_{H_2O,chiller}$ is the water mass flow rate and c_{p,H_2O} is the water specific heat.

The Part Load Ratio (PLR) is calculated from the value obtained from Eq. 9 and from the heat absorbed by the cooling coil ($Q_{cooling\_coil}$) so that:

$$PLR = \frac{Q_{cooling\_coil}}{Q_{chiller}} \qquad (10)$$

The chiller power ($\dot{W}$) is determined from regression models as described by Zmeureanu et al.

(2002) and Brandemuel (1993) in terms of $Q_{chiller,nom}$ (nominal chiller capacity), EIR (Energy Input Ratio) and CAP_FT (chiller capacity in terms of entering temperatures at condenser and evaporator):

$$\dot{W} = Q_{chiller,nom} \cdot EIR \cdot EIR(T_{in,cond}, T_{out,evap}) \cdot$$
$$EIR(PLR) \cdot CAP\_FT(T_{in,cond}, T_{in,evap}) \qquad (11)$$

The condenser leaving water temperature can be calculated as:

$$T_{out,cond} =$$
$$\frac{\dot{m}_{H_2O,chiller} \cdot c_{p,H_2O} \cdot T_{in,cond,chiller} + Q_{cond,chiller}}{\dot{m}_{H_2O,chiller} \cdot c_{p,H_2O}} \qquad (12)$$

where $T_{in,cond,chiller}$ is the condenser entering water temperature and $Q_{cond,chiller}$ is the condenser thermal power calculated by applying the first law of thermodynamics at the whole chiller, i.e., using $Q_{chiller}$ and $\dot{W}$.

PUMPS

The pump energy consumption calculation is base on the model described in Brandemuehl et al. (1993), which assumes an adiabatic pump:

$$\dot{m}c_p\Delta T = f_{m,loss}(1-\eta_m)\dot{W}_t + (1-\eta_p)\eta_m\dot{W}_t \qquad (13)$$

where, $f_{m,loss}$ is the fraction of motor losses that are transferred to the fluid stream, η_m is the motor efficiency, $\dot{W}_t$ is the corrected motor power and η_p is the pump efficiency.

TOWER

The cooling tower model is based on tower model included in ASHRAE's HVAC1 Toolkit for primary HVAC system energy calculations (Lebrun (2001)).

The fraction of time that the tower fan must operate is calculated based on the following equation (Energyplus, 2004):

$$\omega = \frac{T_{set} - T_{wout,off}}{T_{wout,on} - T_{wout,off}} \qquad (14)$$

Where, T_{set} is the exiting water set point temperature (°C), $T_{wout,off}$ is the exiting water temperature with tower fan off (°C) and $T_{wout,on}$ is the exiting water temperature with tower fan on (°C). The average fan power for the simulation time step is calculated by multiplying ω by the steady-state fan power specified by the user.

HVAC CONTROL STRUCTURE

In this paper, two control loop structures for controlling the building thermal conditions are analyzed. Both are based on two single monovariable control loops for the internal temperature and relative humidity signals.

CONTROL LOOP 1 (CL1)

In this case, the temperature control loop is characterized by adjusting the cooling coil water flow rate based on the internal temperature signal and set-point. The control strategy is the "on-off" algorithm. Moreover, when the temperature signal is below the set-point and the relative humidity signal is above the set-point, the re-heater is turned on to improve the system temperature control performance. The relative humidity control loop is characterized by acting on the humidifier based on the relative humidity signal and set-point. The control law is also based on the "on-off" algorithm.

CONTROL LOOP 2 (CL2)

In this case, the temperature control loop is characterized by adjusting the cooling coil water flow rate based on the internal temperature signal and set-point. An "on-off" control law is applied. The relative humidity control loop is characterized by acting, alternatively, on the humidifier and on the re-heater, based on the internal relative humidity signal and set-point. The control law can be either the "on-off" or the "PID" algorithm.

ENTHALPY BASED ECONOMIZER

The enthalpy based economizer is an energy-efficient strategy, which is based on changing the rate of outdoor air in the mixing box when its enthalpy is lower than the return air enthalpy. This economizer strategy is applied together with a temperature on-off based control loop that acts on the cooling coil water flow.

PID TUNING PROCEDURE: THE IMC STRATEGY

The PID controller is composed by three parameters: proportional (P), integral (I) and derivative (D); and represents an alternative for the simple and reliable on-off scheme.

It is well known that PID algorithms can provide better performance than on-off algorithms in terms of set-point following and variance of the control signal. At the same time, the robustness and disturbance rejection properties of the closed loop system are equivalent with or even better than the on-off scheme ones. However, such good characteristics can be easily degenerated if the controller gains are not well tuned.

In order to keep the desirable properties of PID controller in the HVAC context, two steps must be conducted.

The first one is to implement the algorithm with an anti-windup scheme (Franklin et al.,1994). This is due to the fact that constraints in the control signal of such a control loops are always present. These constraints are related with the actuator properties, which can only vary between the levels minimum (turn off) and maximum (100% of the available power). The anti-windup scheme provides low overshoot in the output signal by disabling the integral action when the constraints are reached.

The second step is to perform an adequate tuning procedure for HVAC control system parameters. The PID parameters tuning is not trivial and, in practice, is usually made by a trial-and-error procedure. Auto-tuning methods are available in the literature and in commercial products. An example is the Ziegler Nichols method, which can compute the PID gains based on the system open loop or closed loop shape, provide that these behavior is within a certain class of shape. On the other hand, Internal Model Control (IMC) structure proposed by Rivera et al. (1986) can be viewed as a strategy for PID parameter determination. Such strategy is summarized in following and applied to HVAC control system in this document.
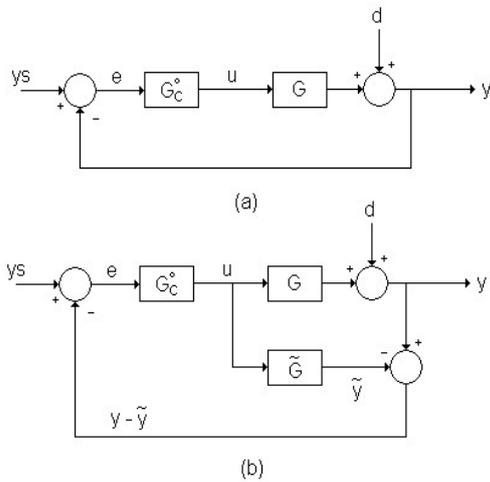

Figure 2 – (a) Conventional and (b) IMC closed-loop control structure.

The main characteristic of the IMC is that the control law is derived from the system model and a quite interesting advantage is that performance and robustness of the closed loop system are adjusted by using a single parameter, named ε. If the system can be approximated by a low order model, it can be additionally shown that the IMC law is closed related with the PID control law, in such a way that the three PID parameters can be computed as a function of the IMC ε tuning parameter. This procedure is summarized in following.

The IMC (Internal Model Control) structure is based on the block diagram shown in Figure 2b. In this Figure 2, $G(s)$ represents the actual system transfer function, $\widetilde{G}(s)$ is the system model, y is the system output signal, $\widetilde{y}$ is the model output signal, d is the disturbance signal and u is the control signal.

In this work, IMC ideas are used to tune the PID of the humidification control loop since the open loop model can be approximated by a first order transfer function.

The PI parameter selection is presented as follows. Thus, the model transfer function is given by:

$$\widetilde{G}(s) = \frac{k}{\tau.s + 1} , \qquad (15)$$

The controller transfer function of the IMC law (see Figure 2b) is given by the following equation:

$$G_C^o = \widetilde{G}_-^{-1} F(s) \qquad (16)$$

where $\widetilde{G}_-$ is a factor of the model transfer function that contains no delays and no zeros in the right side of the s plane. Since the system is described by a minimum phase model, this factor can be given by:

$$\widetilde{G}_- = \frac{k}{\tau s + 1} . \qquad (17)$$

$F(s)$ is a filter, selected to guarantee the causality of the control law, given by:

$$F(s) = \frac{1}{(\varepsilon s + 1)} \qquad (18)$$

$F(s)$ also provides the tuning parameter for the control system.

The control law in a conventional structure feedback (see Figure 2a) is computed by using $\widetilde{G}(s)$ and G_C^o. It is given by:

$$G_c(s) = \frac{(\tau s + 1)}{k(\varepsilon + \theta)s} . \qquad (19)$$

The PI controller has the following structure:

$$G_c(s) = P\left(1 + \frac{1}{I.s}\right), \qquad (20)$$

By comparing Equations (19) and (20), it can be viewed that the parameters P and I of the PI controller have the following relation with the ones in the IMC

$$P = \frac{\tau}{k(\varepsilon + \theta)} ; \qquad (21)$$

$$I = \tau .$$

Therefore, the tuning method can be sumarized as follows: (a) identify the system gain and time

constant. (b) compute the P and I parameters as shown in Eq. 21. (c) change the filter time constant ε to adjust the closed-loop performance and robustness.

SIMULATION ENVIRONMENT

The analysis of the described HVAC system is done by implementing the model equations in a Simulink structure and by linking it to the ASTECCA toolkit. By using the available tools found in the program library, a simulation using the proposed HVAC system integrated to a building environment can be carried out. The building mathematical model is described in details in Mendes et al. (2003).

The HVAC system model is composed of sub-blocks called "TOWER", which has a cooling tower model, "CHILLER", which has water Chiller model, "PUMP", which has pump model, "VALVE", which has the valve model and "FAN-COIL", which has the mixing box, cooling coil, humidifier, re-heater and fan models. The inputs of this last block - fan-coil - are: the chiller heat flux (q_{x2}), supplied by "CHILLER" block, the re-heater heat flux (q_{x4}), the humidifier vapor addition rate ($\dot{m}_u$), the external conditions data and the zone air hygrothermal conditions. The outputs are: the heat flux (Q_{HVAC}) and the moisture flow (J_{HVAC}).

RESULTS AND DISCUSSIONS

In this section, comparative results are provided in terms of energy efficiency and indoor air temperature and RH (relative humidity) control performance by using TRY data of two Brazilian cities. In addition, an example of the system performance with economizer is also presented.

First of all, the PI controllers tuning procedure using the IMC principle is presented. The PI algorithm is implemented with antiwindup, which is a strategy to compensate actuator amplitude constraints (Franklin et al.,1994).

The first step for IMC-PI tuning is the model identification. Here, it is assumed that the system can be approximated by a first order model, which is characterized by a gain and time constant. To compute these parameters for the RH control loop in the CL2 structure, two step signals are applied, that is, a step in the humidifier and a step in the re-heater. These two steps characterize the system behavior for positive input signals (to increase the internal RH) and for negative input signals (to decrease the RH). The hygrothermal initial conditions are equal to 16.6ºC and 59.2% for Brasília, Brazil, and 27.7ºC and 85.7% for Rio de Janeiro, Brazil.

Relative humidity step responses (not shown) using alternatively a re-heater and a humidifier presented a

non linear behavior, since the gain and transitory phase are different. By means of those response curves, two first-order linear models can be identified and used to approximate the system behavior, which parameters are depicted in Table 1.

Table 1 – Model and controller parameters.

| | k * | T** | ε = 100 | |
			P	I
Brasilia				
Humidifier	39,03	5796	$3,7 \times 10^{-1}$	$3,85 \times 10^{-4}$
Re-heater	25,97	972	1,485	$2,56 \times 10^{-4}$
Mean value	32,5	3384	$9,2 \times 10^{-3}$	$3,2 \times 10^{-4}$
Rio de Janeiro				
Humidifier	15,0	5940	3,96	$6,66 \times 10^{-4}$
Re-heater	51,0	18360	3,60	$1,96 \times 10^{-4}$
Mean value	33	12150	3,78	$4,31 \times 10^{-4}$

*Gain, **Time Constant (seconds).

The second step for IMC-PI tuning is the gain calculations (equation 21). The results are presented in Table 1, for ε equal to 100. Since two models are identified, two sets of PI parameters are computed. The gain mean values of these two controllers are used in the PI control law of the CL2.

The following examples show the system behaviors when the two previously mentioned control structures (CL1 and CL2) are used.

Table 2 – Simulation parameters

	Brasília	Rio de Janeiro
Building		
People	200 (from 8:00 to 20:00)	200 (from 8:00 to 20:00)
Volume	25x25x7	25x25x7
Fan-coil		
Fan coil Capacity	150kW	150kW
Air flow rate	8.08 kg/s	8.08 kg/s
Outdoor air flow rate	20%	20%
Re-heater power	90kW*/ 55kW**	90kW
Humidifier Capacity	0.01 kg/s	0.01 kg/s
Chiller		
Chiller Capacity	153kW	170kW
Water temperature	7.2°C	7.2°C

*CL1 , **CL2

The external temperature and RH data are provided by TRY weather data files for Brasília, during winter

time – July, and for Rio de Janeiro, during summer – January. The set-point signal for temperature controllers is equal to 24ºC (±1ºC) and, for RH controllers, is equal to 60% (±5%, in the "on-off" case).
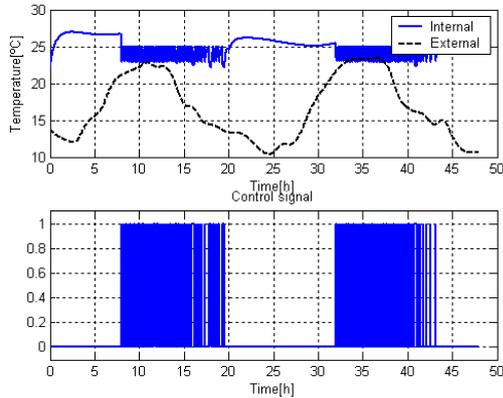

Figure 3 - Closed loop behavior (temperature) for CL1 structure - Brasilia.

The simulation time is 48 hours and, the maximum step time of the solver is 10 seconds. Table 2 contains the simulation parameters. Both control loops are turned on only during working hours (from 8:00 to 20:00 hours).
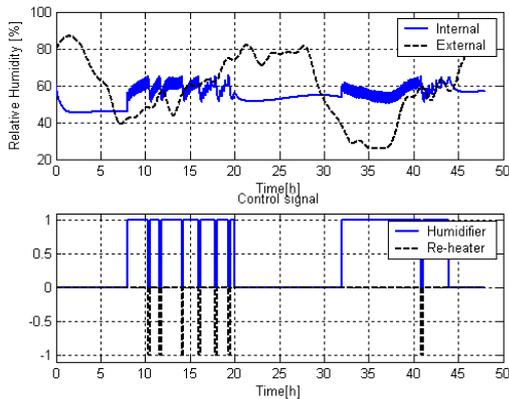

Figure 4 - Closed loop behavior (RH) for CL1 structure - Brasilia.

The temperature and RH signals for Brasilia by using the CL1 structure are presented in Figures 3 and 4. It can be notice that the closed loop behavior is satisfactory, the temperature and RH signals are inside the desired band during almost all the time. However, the temperature signals is slightly below the desired band close to 16, 17.9, 19.4 and 40 hours. This is due to the fact that, at this moment, the internal RH is above the set-point and the cooling coil is turned on. In Figure 4, it can be notice that the humidifier is turned on during 22.5 hours (by summing all the small amount of time during the 2 days) to keep the RH signal inside the desired band, however, during 5.8 hours, the RH is below the set-point band since the temperature control is acting on

cooling coil (which tends to dehumidifier the environment).

The temperature and RH signals for Brasilia by using the CL2 structure are presented in Figures 5 and 6. It can be notice that the closed loop behavior is better than the one provided by CL1, since the temperature and RH signals are inside the desired band during all the time. This is due to the fact that the temperature loop is not directly related with the RH loop.In Figure 6, it can be notice that, during 4.7 hours, the RH is below the set-point band. This is due to external conditions that interact with the internal environment by means of ventilation and infiltration.
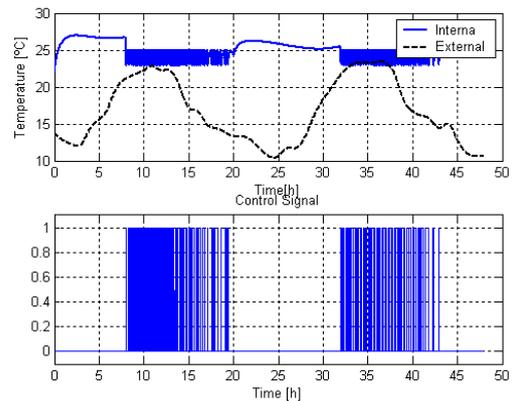

Figure 5 - Closed loop behavior (temperature) for CL2 structure - Brasilia.

The RH signals for Brasilia by using the CL2 structure (with IMC-PI control law) is presented in Figure 7. The rise time of the RH signal is close to 0.7 hours and its deviation around the set-point is smaller than the one provided by the "on-off" law. This deviation can be further reduced by selecting the filter time constant smaller than the one actually used, e.g., $\varepsilon = 0,5$ e $\varepsilon = 10$.
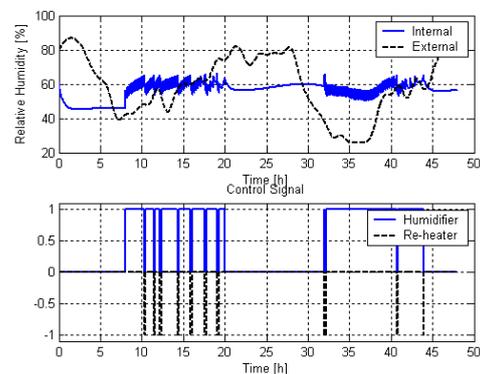

Figure 6 - Closed loop behavior (RH) for CL2 structure and "on-off" control law – Brasilia.

Figure 8 shows the HVAC energy consumption for Brasília for a 48-h period. A PI-IMC based control strategy presents an energy consumption reduction of 99.9 kWh (30.75%) when compared to the control

loop 1 and of 43.2 kWh (16.11%) when compared to the control loop 2 for the simulated period. It can also be observed that the control loop 1 demands more energy than the control loop 2, due to a higher required re-heater power in order to keep the temperature within the pre-defined temperature range.
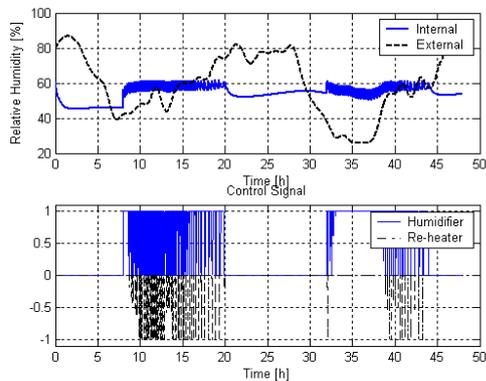

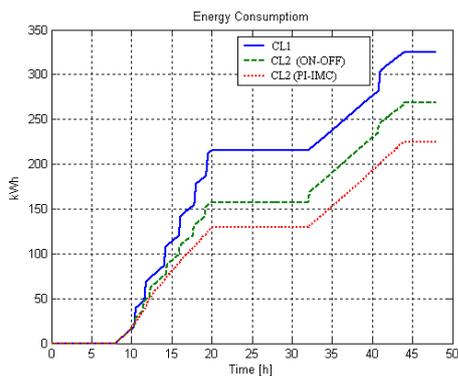*Figure 7 - Closed loop behavior (RH) for CL2 structure and IMC-PI control law – Brasilia.*

Figure 8 – Energy consumption

Figure 9 shows the room air temperature evolution controlled by the water flow rate at the cooling coil for Rio de Janeiro using CL1. For high external air RH values, it becomes harder to reach the temperature set-point due to the cooling coil demand to dehumidify the air that comes from the mixing box. Similar results have been observed when CL2 has been with an on-off control.

Figure 10 presents the RH evolution using the IMC based control, showing that it took 0.2h to be within the RH set-point with a ±4% fluctuation. It has been also noted (not shown) that RH has been kept within the pre-defined range using CL1 and CL2 with on-off control.

It has been observed (not shown) for Rio de Janeiro using the two control strategies that the CL2 with a PI-IMC based control strategy provided an energy consumption reduction of 42.52 kWh (4.12 %) when compared to the same control structure, but with an on-off based controller and of 13.4 kWh (1.33%) when compared to CL1. An interesting point

observed for the HVAC system under Rio climate conditions is that the control of both relative humidity and temperature may raise the energy consumption in 71.9% when compared to a simple temperature control system.
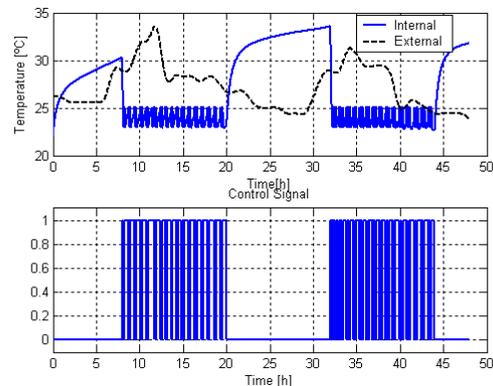

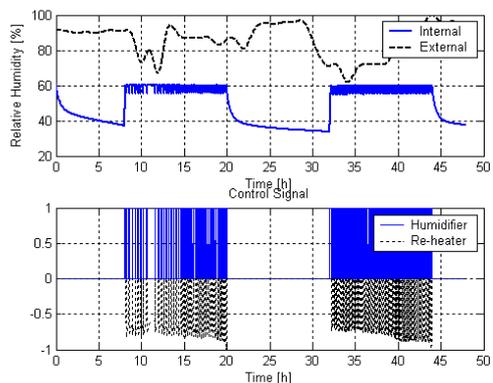*Figure 9 - Temperature evolution using CL1.*

Figure 10 – Relative Humidity evolution using CL2 and a PI-IMC based controller.

It has been also analyzed the energy consumption using an enthalpy based economizer for both cities. In Brasilia, an energy consumption reduction of 1.4% has been reached, while no reduction has been realized in Rio de Janeiro due to the high outdoor air enthalpy.

CONCLUSION

This paper has presented a modeling approach for primary and secondary HVAC systems based on the Matlab/Simulink platform, applying two control structures and two control strategies (on-off e PI-IMC) for room air temperature and relative humidity.

Simulations have been performed in two Brazilian cities (Rio de Janeiro and Brasilia) to show the integration among the different ASTECCA toolkit components (building, HVAC, sensor, controller, weather data and perturbation) and to present results in terms of room air temperature and relative humidity, control signal and energy consumption.

Results have shown that the strategy followed in this paper to implement the PID algorithm (anti-windup

and IMC based tuning schemes) generates a closed loop performance compatible with the expectations, that is, a PID performance better than the on-off performance in term of set-point tracking. Moreover, the well-tuned PID presented low energy consumption than the on-off technique.

Moreover, simulations performed with enthalpy-based economizers have demonstrated that they may not be a good option in Brazil due to the high levels of humidity in big cities on the coast like Rio de Janeiro. However, in a dry climate such as the one found in the Brazilian capital - Brasilia –, the use of economizer led to a energy consumption reduction of 1.4% in the month of July.

To conclude, results have also shown that the simultaneous control of temperature and relative humidity using cooling coils and re-heaters consumes much more energy than the simple temperature control, which stimulates further investigation on dehumidifier equipment and on control strategies using the simulation platform presented in this work.

ACKNOWLEDGMENT

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