THE POWERDOMUS ENVIRONMENT FOR SIMULATING HVAC SYSTEMS

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ABSTRACT

In this paper, the new capabilities of PowerDomus to simulate central HVAC systems combined to its whole-building hygrothermal model is presented. First, models for the primary (chiller, cooling tower, primary pumps and condensation pumps) and secondary (cooling and dehumidifying coil, humidifier, fan and mixing box) systems are presented. Those mathematical models have been integrated into the whole-building PowerDomus program. The program simulation environment is then presented and results show the usability aspects of the program in terms of energy consumption comparison between air- and water-cooled chillers. Influence of the moisture effects on the energy consumption calculation is also illustrated, showing that disregarding moisture may underestimate the cooling energy consumption.

KEYWORDS

Moisture, HVAC systems, hygrothermal simulation

INTRODUCTION

HVAC systems (Heating, Ventilation and Air Conditioning) are of paramount importance on the energy consumption of buildings so that since the energy world crisis in the seventies, many research works have been focused on building energy efficiency, aiming to simulate whole buildings and their HVAC systems.

Thus, there is a need to develop accurate models to evaluate the integral performance of the whole building, considering the presence of moisture within the porous elements so that, in late 2003, a new international research project (Annex 41) in the framework of the International Energy Agency (IEA/ECBCS) entitled "Whole Building Heat, Air and Moisture Response (MOIST-ENG)" was started (Hens 2003). In the same context, recent attempts to simulate the hygrothermal performance of non-conditioned buildings were presented by Rode *et al.* (2004), Mendes *et al.* (2003), Holm *et al.* (2002) and Simonson *et al.* (2002).

Despite the importance of the subject concerning moisture and its impact on the HVAC system performance, just a very few publications can be found in the literature such as in Knabe and Le (2001), Cherem-Pereira (2003) and Cherem-Pereira

and Mendes (2003). However, no work has mentioned the combined simulation of a central HVAC system with a detailed hygrothermal model for the building envelope. However, steady-state models have been largely presented such as in Knabe and Le (2001), Hensen (1991), Chow *et al.* (1997), Lam *et al.* (1997), Nassif *et al.* (2003) and other works in literature for modeling central HVAC systems. On the other hand, unsteady-state mathematical models have been presented by Barbosa and Mendes (2003) and Novak *et al.* (2004) to simulate secondary HVAC systems.

In this way, this paper presents the work that has been carried out in order to implement HVAC systems models into the PowerDomus platform (Mendes *et al.*, 2003). so that combined hygrothermal buildings could be simulated in a fully integrated way with their HVAC systems so that moisture effects could be investigated even at a funicular state of porous building element. Models.

Results of coupling are presented for a multizone building to show the usability of the proposed algorithm where it is presented a energy consumption comparison between HVAC systems based on air-cooled and water-cooled chillers, illustrating the capabilities to perform simulations using different HVAC systems. Then, the effect of building envelope moisture is presented in terms of building energy consumption and indoor air temperature and relative humidity.

This work presents first the models, then the software interface and a simulation procedure to generate results to compare the performance of air- and water-cooled chillers. Then, the analysis of the influence of the moisture effects on the energy consumption calculation is illustrated, showing the impact of moisture on cooling energy consumption for the case presented.

HVAC COMPONENTS MODELS

Chiller Model

For the Chiller, a model based on regression functions that describe how the cooling capacity and the energy efficiency vary with operation conditions, as well as the power consumption varies with part-load-ratio (Pacific Gas and Electric 1996) is employed.

The chiller model can be described by the following curves:

$$CAPFT = a_1 + b_1 T_{ev,out} + c_1 T_{ev,out}^2 + d_1 T_{cd,in} + e_1 T_{cd,in}^2 + f_1 T_{ev,out} T_{cd,in}; \quad (1)$$

$$EIRFT = a_2 + b_2 T_{ev,out} + c_2 T_{ev,out}^2 + d_2 T_{cd,in} + e_2 T_{cd,in}^2 + f_2 T_{ev,out} T_{cd,in}; \quad (2)$$

$$EIRPFPLR = a_3 + b_3 PLR + c_3 PLR^2, \text{ with } PLR = \frac{Q_{ev}}{Q_{ev,available(T_{ev,out},T_{cd,in})}}, \quad (3)$$

where,

- $T_{ev,out}$, outlet evaporator water temperature (ºC);

- $T_{cd,in}$, inlet condenser temperature (external air or condensing water) (ºC);

- PLR, the part-load ratio;

- $Q_{ev,available}$, the chiller cooling capacity available in full load conditions (W);

- a, b, c, ..., f, the curve coefficients, which can be obtained experimentally or from catalogs.

Equation (1) is a biquadratic function that describes how the chiller cooling capacity varies at different outlet evaporator water temperatures and inlet condenser water temperatures. Equation (2) is also a biquadratic function that describes how the full load efficiency varies with water temperatures, while Eq. (3) is a quadratic function that describes how the efficiency varies with part-load-ratio.

Combining all three equations (Eqs. 1-3), the power at specified operating conditions can be evaluated by:

$$Pot_{comp(T_{cd,in},T_{ev,out})} = Pot_{ref} \, x \, CAPFT \, x \, EIRFT \, x \, EIRPLR \quad (4)$$

The values of $CAPFT$, $EIRFT$ and $EIRPLR$ are dimensionless values, which are obtained in the following way:

$$CAPFT = \frac{Q_{ev,available}}{Q_{ev,rat}}; \quad (5)$$

$$EIRFT = \frac{Pot_{comp,max}}{Q_{ev,available}} \frac{Q_{ev,rat}}{Pot_{comp,rat}}; \quad (6)$$

$$EIRPLR = \frac{Pot_{comp}(PLR)}{Pot_{comp,max}}, \quad (7)$$

where,

- $Q_{ev,rat}$, is the chiller cooling capacity at nominal conditions (W);

- P_{otcomp} (PLR), the chiller compressor electric power at part-load conditions (W);

- $P_{otcomp,rat}$, the chiller compressor electric power at rating conditions (W);

- $Pot_{comp,max}$, the maximum chiller compressor electric power at full load conditions (W).

Cooling Tower Model

The cooling tower model used is based on the model included in ASHRAE HVAC 1 Toolkit (Bourdouxhe and Grodent 1999). Cooling tower performance is modeled using effectiveness-NUT relationships for counter flow heat exchanger. The model input data is taken from catalogs. The fan power consumption is given as a function of the fraction of time (ω), in which the tower fan should operate, where this fraction is obtained through the following equation (EnergyPlus 2004):

$$\omega = \frac{T_{set} - T_{wout,off}}{T_{wout,on} - T_{wout,off}}, \quad (8)$$

Where T_{set} is the outlet water set point temperature (ºC), $T_{wout,off}$ the outlet water temperature with tower fan off (ºC) and $T_{wout,on}$ the outlet water temperature with tower fan on (ºC).

Pump Model

The pump model is based on the model presented in (Brandemuehl et al. 1993). The model calculates pump power consumption and exiting fluid temperature for a given flow rate and entering fluid conditions. The model considers both variable and constant flow pumps. The pump power for variable flow is calculated from a regression of part-load power consumption as a function of part-load flow. The pump power consumption is determined by the following expression:

$$\dot{W}_t = FFLP \, x \, \dot{W}_{t,rat}, \quad (9)$$

where W_t and $W_{t,rat}$ are respectively the actual pump/motor power at part-load conditions and full load power. The factor $FFLP$ is a polynomial function in function of the partial-load-ratio (PLR), in the following way

$$FFLP = C_0 + C_1 PLR + C_2 PLR^2 + C_3 PLR^3. \quad (10)$$

The factor PLR is defined as the ratio between the actual volumetric flow and the rated flow $PLR = \dot{V}_{actual} / \dot{V}_{rat}$. The coefficients C_0, ..., C_3 are obtained from data catalogs.

Mixing Box Model

The model calculates the output air conditions for a mixing process based on the energy and mass conservation equations for the moist air.

Applying the mass and energy conservation Equation (3) to the mixed humidity ratio and enthalpy are given as:

$$\dot{m}_{a,out} = \dot{m}_{a1,in} + \dot{m}_{a2,in}; \quad (11)$$

$$W_{a,out} = \frac{\dot{m}_{a1,in} W_{a1,in} + \dot{m}_{a2,in} W_{a2,in}}{\dot{m}_{a,out}} \quad (12)$$

$$h_{a,out} = \frac{\dot{m}_{a1,in} h_{a1,in} + \dot{m}_{a2,in} h_{a2,in}}{\dot{m}_{a,out}} \qquad (13)$$

where, $\dot{m}$, W, h are respectively air mass flow rate, humidity ratio and enthalpy in the mixing box inlet and outlet .

Cooling and Dehumidification Coil Model

For the cooling and dehumidification coil model, the CCDET model (Brandemuehl *et al.* 1993) has been used as presented in ASHRAE HVAC 2 Toolkit. The coil is modeled as a counterflow heat exchanger adapted for air conditioning systems with at least four rows. This model considers the external condensation at the coil surface and requires geometric data such as the internal and external tube diameter, fin area, fin thickness etc. The model calculates the water temperature, air humidity ratio and temperature at the coil outlet, sensible heat transfer rate and the fraction of wet surface area.

Humidifier Model

A steam injection humidifier model based on the ASHRAE HVAC 2 Toolkit (Brandemuehl *et al.* 1993) has been implemented, which uses a simple thermal balance in the humidifier taking into account the air saturation. This model determines the air temperature and air humidity rate, where steam flow is injected in the air stream increasing its humidity ratio.

Fan Model

This model calculates the fan power and leaving air conditions, as well the air temperature and humidity for a given flow rate and entering air conditions. The fan model used is based on the model presented in Brandemuehl *et al.* (1993). The fan power consumption is then determined in the following way:

$$W_t = FFLP \ x W_{t,rat}, \qquad (14)$$

where W_t and $W_{t,rat}$ are respectively the actual shaft power at part-load conditions and full load power. The factor *FFLP* is a polynomial curve fit as a function of part-load ratio (*PLR*), in the following way:

$$FFLP = C_0 + C_1 PLR + C_2 PLR^2 + C_3 PLR^3. \qquad (15)$$

The factor *PLR* is defined as the ratio of the actual volumetric flow rate to the rated flow rate $PLR = \dot{V}_{actual} / \dot{V}_{rat}$. The coefficients $C_0,..., C_3$ are obtained from catalogs data.

INTEGRATION TO A BUILDING HYGROTHERMAL MODEL

The present model for simulating HVAC systems has been integrated into the energy simulation program PowerDomus, which has a dynamic model for the analysis of a whole-building hygrothermal behavior. In the present work, a lumped formulation for calculating both room air temperature and humidity ratio is considered for each building zone. The formulation for hygrothermal envelope model is based on the work presented by Mendes et al. (2002) and Mendes and Philippi (2005), using temperature and moisture content as driving potentials and considering all transport coefficients moisture-content dependent.

The conservation partial differential governing equations are discretized by using the control-volume formulation method with a central difference scheme. The resulting algebraic equations are solved using the MultiTriDiagonal-Matrix Algorithm (MTDMA) described in Mendes and Philippi (2004).

The building zone model has to be fed with the sensible and latent HVAC system cooling loads, while the HVAC model needs the zone leaving air psychrometric state and the airflow return rate. The moisture adsorption/dessorption related loads are taken into account in both energy and mass balances and the vapor concentration difference is linearized in terms of temperature and moisture content differences, making the numerical solution more stable.

SOFTWARE INTERFACE

The software has been developed in C++ language under C++ Builder environment. Also the OpenGL API was used to render both 2-D and 3-D panels, in this way it was possible to create an application with great visual quality. The program is available on the PC platform under Windows 9x and NT operating systems. The user interface consists of a series of windows in which the user can enter the relevant input data for the buildings and the HVAC system and review the results.

The software interface is modularly divided into different parts, which are briefly described in the paragraphs below.

Building Description Module

This module is responsible by the definition of the building geometry and dimensions. A building can have several zones and each zone can have several walls and each wall can have several layers and each layer shall be defined by its physical characteristics. Figure 1 shows the building description module.

The Domus edition interface is very simple, allowing the user to build a construction without much of specific knowledge. For the 3-D visualization panel, it is provided tools to rotate, translate and change building scale.

HVAC System

In the HVAC window, the user will be able to specify the HVAC primary system and secondary system (fan coil) for each zone. The user will also be able to define controllers for each fan coil equipment in each zone.

Figure 2 is showing the HVAC primary system module. The user edit the primary system inserting components such as pumps, water cooled chillers, air cooled chillers, boilers and towers.

Figure 3 is depicting the HVAC secondary system. In this screen is only possible to insert fan coil components. After inserting a fan coil, the user can edit it choosing its components (fans, humidifier, mixing box, etc.) and controllers.

Figure 4 shows the fan coil editing screen. In this window the user chooses the fan coil components and in which sequence they will be simulated.

Output Reports

The program provides a great variety of report options such as hourly reports for temperature, relative humidity, thermal comfort level (PMV and PPD), temperature and moisture content distributions within any building wall and any HVAC components, energy consumption, thermal loads, energy costs, and monthly general statistics.
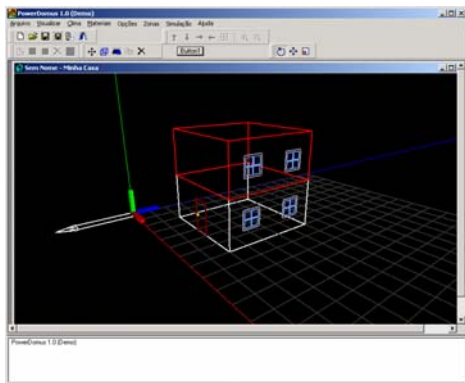

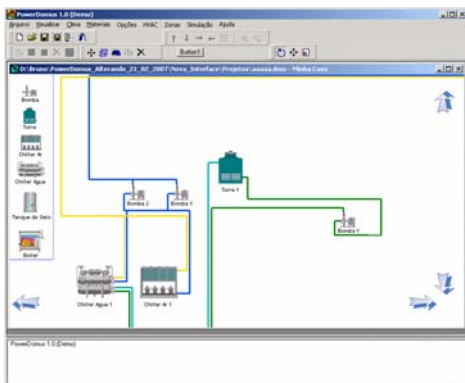*Figure 1: Example of a two zone building edition.*

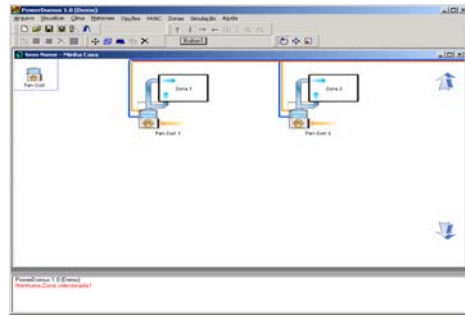*Figure 2: A HVAC primary system configuration.*
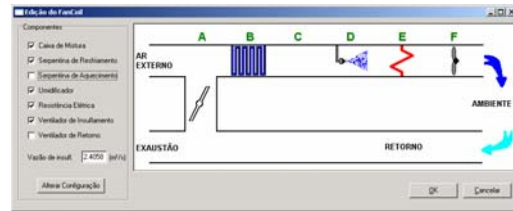

Figure 3: A HVAC secondary system configuration.

Figure 4: PowerDomus window used to edit the Air Handling Unit components.

SIMULATION PROCEDURE

Weather Data

The Test Reference Year (TRY) of the city of Belém, north of Brazil (latitude: -1.38º; longitude: 48.48º; GMT: -3h; altitude: 16m) has been used.

Building Description

The building envelope has been considered as a 15-cm monolithic cellular concrete layer, with external and internal permeances of 9e-10 kg/(Pa-m^2-s) and 2e-10 kg/(Pa-m^2-s), while the external and internal convective heat transfer coefficients have been set as 29.30 and 8.39 W/m^2-K. For determining the convective mass transfer coefficients, a unitary Lewis number has been used. Table 1 shows the monolithic concrete physical characteristics used for the building envelope. No ground contact has been considered, *i.e.*, the building has been placed in the air to avoid simplifications on the heat and moisture transfer via the ground.

Table 1: Monolithic cellular concrete physical characteristics.

λ	T	U	R	RO	C$_P$
0.18	0.15	1.33	0.750	650	840

λ is the thermal conductivity (W/m²K); T the thickness (m) ; U the overall heat transfer coefficient (W/m^{2}K); R the thermal resistance (m^{2}K/W); ρ the density (kg/m^3) and C$_p$ the specific heat (J/kg-K).

A vapor permeability of 3e-11 kg/(m•s•Pa) and an average sorption isotherm linear function - $\theta = 0.042965\ \phi$ - have been considered, where ϕ represents the relative humidity. Figure 5 shows the

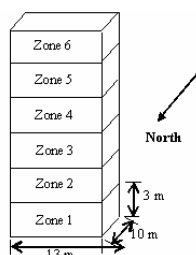building geometric characteristics (dimensions) of a six-storey building.

Figure 5: Multizone building geometric characteristics.

Double-glazed windows with unitary solar heat gain coefficient (at normal incidence), no internal shading device and a global heat transfer coefficient of 3 W/m²-K have been distributed on the building facades as presented in Table 2.

Table 2: Size and quantity of windows on the building facades.

	FACADE	WINDOWS QUANTITY	HEIGHT X WIDTH
Zone 1	N	1	1.5 m x 4.0 m
	S/E/W	2	1.5 m x 4.0 m
Zones 2-6	N	2	1.5 m x 5.0 m
	S/E/W	4	1.5 m x 2.0 m

Internal thermal gains from people, lighting and equipment have been considered during working days (Monday to Friday) from 8 am to 6 pm. Twenty occupants have been considered as individual sources dissipating 60 W of purely convective sensible heat and 80 W of latent heat, while lighting power density of 45 W/m² and ten pieces of equipment at 100 W have been selected for the present study. The ASHRAE cooling load weighting factors have been used in order to calculate the long-wave radiative heat gain on the internal envelope surfaces.

HVAC System and Equipment Description

Primary Systems (Plant) with air-cooled chiller (SP-AIR)

The primary system is composed of one air cooled chiller and two cold water circulation pumps. Table 3 presents the nominal data chiller obtained from catalog, while Table 4 shows the capacity (CAPFT) and efficiency (EIRFT, EIRFPLR) regression coefficients.

Table 3: Air-cooled chiller simulation parameters.

NOMINAL PARAMETERS	EVAPORATOR
Capacity (*kW*)	245.90
Temperature rise (*ºC*)	5.5
Water flow (*L/s*)	10.68
COP	3.05
Water leaving temperature (*ºC*)	7.0

Table 4: Air-cooled chiller capacity and efficiency coefficients.

	A	B	C	D	E	F
CAPFT	1.08	3.5e-2	2.1e-4	-6.9e-3	-3.6e-5	-2.3e-4
EIRFT	0.64	-6.5e-3	4.6e-4	-1.7e-5	4.4e-4	-6.2e-4
EIRPLR	0.60	7.7e-2	3.4e-1	–	–	–

Table 5 shows the nominal parameters pumps obtained from catalog data.

Table 5: Model parameters for the pumps.

NOMINAL PARAMETERS	PUMP 1	PUMP 2
Required Power (kW)	0.6847	0.6847
Motor Efficiency (%)	70	70
Rotation (rpm)	1750	1750
Water flow (L/s)	5.34	5.34
Manometric Pressure (m)	9.15	9.15

Primary Systems (Plant) with water-cooled chiller (SP-AIR)

This primary system is composed of one water-cooled chiller, two cold- (1 and 2) and two condensation- water (3 and 4) circulation pumps and three cooling towers. Table 6 presents the nominal data water cooled chiller obtained from catalog, while Table 7 shows the capacity (CAPFT) and efficiency (EIRFT, EIRFPLR) regression coefficients. Table 8 shows the nominal parameters pumps obtained from catalog data and the Table 9 presents the cooling towers simulation parameters.

Table 6: Water-cooled chiller simulation parameters.

NOMINAL PARAMETERS	EVAP.	COND.
Capacity (*kW*)	261.6	322.3
Temperature rise (*ºC*)	5.5	5.5
Water flow (*L/s*)	11.36	14
COP	4.138	–
Water leaving temperature (*ºC*)	7.0	–
Water entering temperature (*ºC*)	–	29.44

Table 7: Waterr-cooled chiller capacity and efficiency coefficients.

	A	B	C	D	E	F
CAPFT	0.78	3.8e-2	1.5e-4	-8.4e-3	-2.7e-4	-3.3e-4
EIRFT	0.66	-9.6e-3	7.3e-4	-6.7e-4	6.1e-4	-8.8e-4
EIRPLR	0.78	-3.7e-1	5.9e-1	–	–	–

Table 8: Model parameters for the pumps.

NOMINAL PARAMETERS	PUMPS 1 AND 2	PUMPS 3 AND 4
Required Power (*kW*)	0.7283	0.6338
Motor Efficiency (*%*)	70	66
Rotação (*rpm*)	1750	1750
Water flow (*L/s*)	5.68	7.0
Manometric Pressure (*m*)	9.15	4.50

(m³/h)					

Table 9: Cooling towers simulation parameters.

NOMINAL PARAMETER	TOWER 1	TOWER 2	TOWER 3
Capacity (*kW*)	107.40	107.40	107.40
Water flow (*L/s*)	4.67	4.67	4.67
Water entering temperature (*ºC*)	34.94	34.94	34.94
Water leaving temperature (*ºC*)	29.44	29.44	29.44
Air Flow (*m³/h*)	15500	15500	15500
Fan Power (*kW*)	2.237	2.237	2.237

Secondary System

The secondary system is composed of three components: mixing box, cooling coil and suply fan. For the mixing box, a constant air change rate of 30% in volume of the total suply airflow has been assumed. The largely used on-off based control system has been employed for all zones. The set-point temperature is 24 ºC with a hysteresis of 2 ºC, *i.e*., the temperature is kept between 23 ºC and 25 ºC.

Table 10 shows the cooling coil simulation geometric parameters for all building zones. Table 11 describes the fan simulation parameters for each zone.

Table 10: Cooling coil simulation geometric parameters.

COIL PARAMETERS	ZONE 1	ZONES 2 TO 4	ZONE 6
Height (mm)	457	457	457
Length (mm)	1050	850	1050
Width (mm)	198	264	198
Distance between face tubes (mm)	38.08	38.08	38.08
Row spacing (mm)	33.0	33.0	33.0
External tube diameter (mm)	15.87	15.87	15.87
Internal tube diameter (mm)	13.87	13.87	13.87
Fin thickness (mm)	0.190	0.190	0.190
Fins number per meter	314.9	314.9	314.9
Rows number	6	8	6
Tubes number per row	12	12	12
Circuits number	6	6	6
Fin thermal conductivity (W/m K)	230	230	230
Tube thermal conductivity (W/m K)	380	380	380
Water flow (L/s)	1.20	1.10	1.30

Table 11: Fans simulation parameters.

Parameter	Zone 1	Zone 2	Zone 3	Zone 4	Zone 5	Zone 6
Power (kW)	0.74	0.74	0.74	0.74	0.74	0.74
Efficiency (%)	68	67	68	67.5	67.9	69
Air flow	3846	3342	3285	3228	3349	4327

RESULTS

This section presents the simulation results using the mathematical models for the primary and secondary systems components integrated to the PowerDomus hygrothermal building model. The simulation parameters used were described in the simulation procedure section. A 1-min simulation time step has been used and results are presented for the city of Belém – Brazil on January 14.

Figures 6a and 6b show humidity ratio and temperature evolution on the secondary system components outlets, between 4 and 4:30 pm. The humidity ratio and temperature values in the mixing box are closer to the room air values due to the low air change rate of 30%. It can also be noticed a decrease on temperature and on humidity ratio due to the combined transfer of sensible and latent heat. In addition, the fan motor releases heat so that a temperature increase of 0.84ºC has been noticed. When the valve is closed, the thermal equilibrium is reached between the HVAC secondary system components.

Figure 7 presents a comparison in terms of yearly HVAC energy consumption between primary system with air-cooled chiller (SP-AIR) and water-cooled chiller (SP-WATER). For the SP-AIR are presented the air-cooled chiller, cold-water pumps and secondary system fans energy consumption. While for SP-WATER the condensation-water pumps and cooling towers fans energy consumption have been additionally added. The water cooled chiller show a relative difference of 13% less than air cooled chiller consumption energy. While the water cold pumps showing energy consumption relative difference of 6% greater compared to SP-AIR. The equipment total yearly consumption of the system with water-cooled chiller was approximately 1.6% lower than the air-cooled chiller primary system.
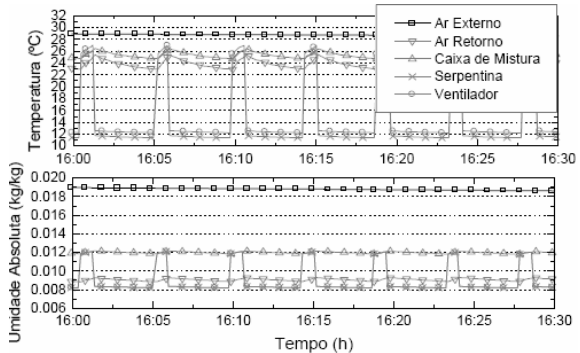

Figure 6: Humidity ratio (a) and temperature (b) evolution at the zone-1 outlets of the secondary system, between 4 and 4:30 pm.

The next simulations results show the building envelope moisture effects on the temperature and humidity ratio zone and the HVAC energy consumption. The interaction between the porous building envelope, room air and HVAC system is presented by contrasting a pure thermal model (no moisture) and a hygrothermal model (with Moisture) to calculate the conduction loads through the envelope.
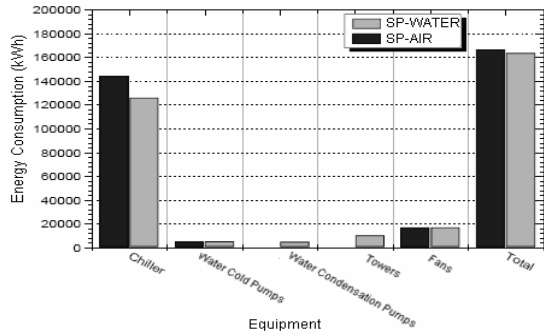

Figure 7: Yearly HVAC energy consumption.

Figure 8 illustrates the zone-6 air temperature and relative humidity evolution, where the HVAC equipment operates between 8 am and 6 pm. It can be observed that the predicted temperatures are lower when the HVAC system is off using the model that considers the moisture interaction with the building envelope. Figure 8 shows also that even when the HVAC system is on the zone relative humidity is greater when moisture is considered as the vapor flow from walls is important on the zone water vapor mass balance. In addition, Figure 8 depicts a better performance for temperature control considering moisture as the system has been sized using the hygrothermal model.
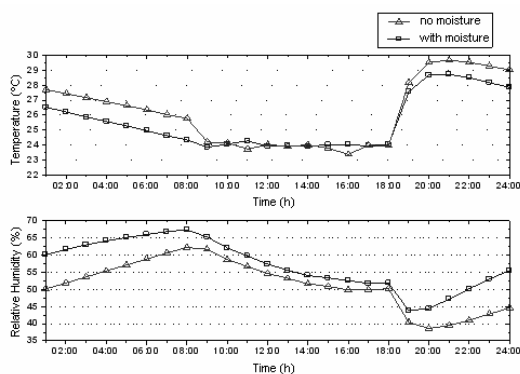

Figure 8: Zone-6 temperature and relative humidity evolution, with and without moisture.

Figure 9 presents a comparison in terms of yearly HVAC energy, predicted with both hygrothermal and thermal building models, showing that ignoring moisture underestimates the yearly energy consumption in 4%, especially because the latent conduction load is not added to the sensible one,

besides the fact that thermal conductivity is higher. The effect of moisture on the wall thermal capacity is of minor importance.

In this way, when moisture is disregarded, there is a risk of oversizing HVAC system and underestimating energy consumption, as already argued in Mendes (1997) and Mendes *et al.* (2003).
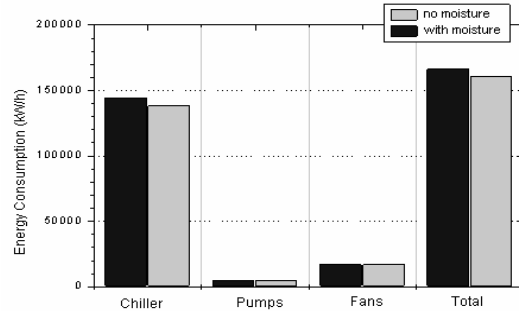

Figure 9: Yearly HVAC energy consumption

CONCLUSION

A generic model for HVAC systems integrated to a hygrothermal whole-building simulation model has been presented. Then, to show the model capability, a 6-storey building - composed of an air-cooled chiller, water-cooled chiller and a secondary system with mixing box, cooling coil and supply fan - has been simulated, showing results for the six building zones.

Simulation results have shown a great economy higher than 10% when water-cooled chillers are used instead of air-cooled ones. However, when all HVAC systems components are taking into account, that difference decreases to 1.6% It has also been noticed that when moisture is taken into account the energy consumption may be 4% higher than that obtained disregarding the presence of moisture for the present case study.

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