Proceedings of Building Simulation 2011:
12th Conference of International Building Performance Simulation Association, Sydney, 14-16 November.

A BUILDING SIMULATION ENVIRONMENT FOR ANALYSIS OF WATER SPRAY EVAPORATIVE COOLING

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ABSTRACT

Evaporative cooling is an attractive energy-efficient technique for producing a comfortable indoor environment. The efficiency and low cost of water spray evaporative cooling systems makes them a good alternative to reduce energy use. Even so, building simulation software does not currently incorporate direct evaporative cooling spray models because, among other reasons, an accurate prediction of spray evaporation performance is difficult to achieve due to the complex physical phenomena.

In this work, simulations are carried out to illustrate the capability of the integration between a vaporization spray model and a whole-building hygrothermal model to predict hygrothermal performance of a building with water spray evaporative cooling system. The evaporation and building thermodynamics behavior are examined in a series of parametric tests, which demonstrate that the combined model is consistent and responds qualitatively well to variations in input parameters

INTRODUCTION

Energy consumed by cooling systems has a direct impact on the operational cost of a building and an indirect impact on the environment so that since the early 70`s, with the energy world crisis, many research works have been focused on building energy efficiency, aiming to simulate whole buildings and their HVAC systems.

However, in building energy analysis, calculated heat conduction through walls usually neglects the storage and transport of moisture in the porous structure of the walls, which are normally submitted to both temperature and moisture content gradients so that an accurate heat transfer determination requires a simultaneous calculation of both sensible and latent effects, especially when vapor is generated for cooling down building zones.

Therefore, there is a need to develop 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., 2003b; Holm et al., 2002; Simonson et al., 2002).

Despite the importance of the subject concerning moisture and its impact on the cooling system performance, just a few publications can be found in the literature. Knabe and Le (2001) combined building simulation with a split air conditioning system, considering the envelope hygrothermal effect by using the program TRNSYS-TUD. However, no work has mentioned the combined simulation of a spray based evaporative cooling with a detailed hygrothermal model for the building envelope.

Due to the lack of a building simulation program that can simulate in details the combined heat, vapor and liquid transfer in porous elements and a water-spray based direct cooling system, a generic and flexible educational computational algorithm has been elaborated in order to integrate both water spray systems and multizone hygrothermal building model.

Consequently, in order to verify the effects of the moisture buffer capacity on the hygrothermal performance of buildings equipped with water spray systems, a model to simulate spray systems is combined to a full hygrothermal model. The mathematical components models are described in the next section.

In the algorithm, models bring together the scales of the droplet, spray and the building, considering heat and moisture transfer through porous elements. The spray model is based on *discrete particle model in separate flows*, while the combined heat and moisture transfer through the walls is based on the *Philip and De Vries model* for both funicular and pendular states.

The hygrothermal envelope model is based on the work presented by Mendes et al.(2002), which has a robust method to calculate simultaneously the temperature and moisture content distributions within the porous envelope and was conceived to preserve numerical stability.

Results are presented in terms of indoor air temperature and relative humidity, comparing the effect of the vaporization system.

SPRAY VAPORIZATION MODEL

The mathematical modeling of sprays was detailed and discussed in Silva et al. (2004) and Silva et al. (2006). The mathematical spray model includes the solution of equations of momentum, energy and mass conservation, for each phase. The *discrete particle model in separate flows* was adopted, in which the spray is divided into samples of discrete droplets, whose motion and transport are tracked through the flow field, using a Lagrangian formulation, while air is treated through a Eulerian formulation (Faeth, 1983). The effect of droplets on the gas phase is considered by introducing appropriate source terms into the gas phase equations of motion.

Dynamic Behavior

In order to determine the coefficients for the heat and mass transfer between the droplets and the air, the correct balance of velocities and distances reached by the spray is so important. In this process, when a liquid is sprayed into a non-condensing gas environment, it leads to an exchange in momentum between the droplets of the spray and the gas, in this case the air. The droplets decelerate due to aerodynamic drag and the air acquires the momentum lost by the droplets. This creates a flow field in which air is continually entrained into the spray, as shown in Figure 1a.

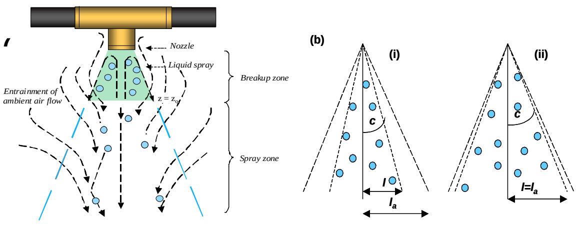


Figure 1- (a) Schematic diagram of the zones and breakup of the typical liquid film of a spray (b) Definition sketches for narrow (i) and wide (ii) sprays.

From Ghosh and Hunt (1994), the governing equations that define the numerical solution of the dynamic behavior follow:

- Entrainment law, assuming that $c << 1$ (one)

$$\frac{d}{dz}\pi l_a^2 V_a = 2\pi\beta V_a l_a \quad \textit{(narrow spray)} \qquad (1)$$

$$\frac{d}{dz}\pi l_a^2 V_a = \frac{d}{dz}\pi l^2 V_a \text{ for } c>0.194 \textit{ (wide spray)} \qquad (2)$$

where the entrainment coefficient β is 0.11 and c is the tangent of the half-angle of the spray, as shown in Figure 1b.

- Volume fraction

$$\alpha = \frac{Q_l}{V_l \pi l^2} \qquad (3)$$

where $\qquad l = cz \qquad (4)$

- Force on the droplets

$$\frac{F_z}{\rho_a} = -\frac{3}{8a} C_D (V_l - V_a)^2 \alpha \qquad (5)$$

where a is the radius of the droplet, ρ_a is the specific mass of air and C_D is the drag coefficient

- Rate of change of the average momentum of the droplets

$$V_l \frac{dV_l}{dz} = \left(\frac{F_z}{\rho_a}\right)\left(\frac{\rho_a}{\rho_l}\right)\alpha^{-1} \qquad (6)$$

- Rate of change of momentum flux of the air jet

$$\frac{d}{dz}(\pi l_a^2 V_a^2) = -\left(\frac{F_z}{\rho_a}\right)\pi l^2 \qquad (7)$$

At the point where the disruption of the liquid film occurs, the initial droplet velocity equals that of the liquid sheet, and the air jet width (l_a) is approximately equal to the spray width (l_0). At $z=z_0$, the initial conditions are

$$V_a = V_{a_0}; \qquad V_l = V_{l_0} \quad \text{and} \quad l_a = l_0 \qquad (8)$$

Discrete particle model

Using the discrete particle model it is possible to analyze the transference of heat and mass between a droplet and the air that surrounds it. It is considered that the conditions of the environment are known and constant during each time interval of the quasisteady process and that the equations are corrected to include the effect of relative movement between the droplet and the air, as suggested by Sirignano (1999) and Faeth (1983).

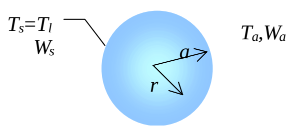
The boundary conditions for the equations of mass, energy and species conservation are:

$$r = a: \qquad T = T_s = T_l \qquad W = W_s$$

$$r = \infty: \qquad T = T_a \qquad W = W_a \qquad (9)$$

where T_a and W_a are the conditions outside the boundary layer of the droplet, as shown in Figure 2.

Figure 2- Liquid particles interacting with gaseous atmosphere



- Rate of change of the droplet radius

$$\frac{da}{dz} = -\frac{\dot{m}}{4\pi a^2 \rho_l V_l} \qquad (10)$$

where $\dot{m}$ is the mass transfer rate from the droplet to the air, corrected for Re $\neq 0$ (Faeth, 1983),

$$\dot{m} = \left\{1 + \frac{0.278\,\mathrm{Re}^{1/2} Sc^{1/3}}{\left[1 + \frac{1.232}{\mathrm{Re} Sc^{4/3}}\right]^{1/2}}\right\}\dot{m}_{\mathrm{Re}=0} \qquad (11)$$

for air under normal conditions of temperature and pressure, Schmidt number (Sc) $=\nu_a/D_{AB}$. The mass transfer rate from droplet to air for Re = 0 ($\dot{m}_{\mathrm{Re}=0}$) is defined, by Faeth (1983), as

$$\frac{\dot{m}_{\mathrm{Re}=0}}{4\pi a \rho_g D_{AB}} = \ln\left[\frac{1+W_s}{1+W_a}\right] \tag{12}$$

where D_{AB} is mass diffusivity, ρ_g is the density of the gas mixture and W_s is the saturation humidity ratio (at the droplet surface)

- Rate of change of the droplet temperature

$$\frac{dT_l}{dz} = \frac{3}{4}\frac{\dot{m}H_l}{\pi a^3 \rho_l c_{p_l} V_l}\left[\frac{B_T}{B_Y}Le^{2/3} - 1\right] \tag{13}$$

where c_{p_l} is the specific heat of the water, H_l is the latent heat of vaporization, Le is Lewis number and B_T and B_Y are

$$B_T = \frac{c_{p_a}(T_a - T_l)}{H_l} \tag{14}$$

$$B_Y = \frac{W_s - W_a}{1+W_a} \tag{15}$$

Vaporization in spray

Following the discrete particle model in separate flows, it was possible to analyze the conservation of mass and energy. At the scale of the spray the volume fraction (α) is used to consider the variables calculated in the discrete particle model, such as mass flows and outflow that correspond to each control volume.

The control volume ($d\forall$) is defined by equation:

$$d\forall = \pi l^2 dz \tag{16}$$

Considering the evaporation of the droplets, the outflow of liquid (Q_l) will not be constant as a function of z, although the number of droplets remains constant until their complete evaporation. Using conservation laws, the variation of the liquid outflow (Q_l) can be described by the expression:

$$\frac{1}{Q_l}\frac{dQ_l}{dz} = -\frac{3}{4}\frac{\dot{m}}{\pi a^3 \rho_l V_l} \tag{17}$$

Thus, in the one-dimensional model, all the variables are functions only of z and the balance equations can be written as follows:

- Mass vapor conservation

$$\frac{d}{dz}\left[\rho_a V_a \pi l^2 W_a (1-\alpha)\right] - \pi \rho_i W_i \frac{d}{dz}\left[l^2 V_a\right] - \frac{3}{4}\frac{\dot{m}Q_l}{\pi a^3 V_l} = 0 \tag{18}$$

where W_a is the humidity ratio of the air inside of the control volume and W_i is the humidity ratio of induced air.

- Energy conservation

$$\frac{d}{dz}\left[\rho_a V_a \pi l^2 h_a (1-\alpha)\right] - \pi \rho_i h_i \frac{d}{dz}\left[l^2 V_a\right] + \rho_l c_{p_l}\frac{d}{dz}(T_l Q_l) = 0 \tag{19}$$

where h_i and h_a are, respectively, the enthalpy of the induced air and the enthalpy of the air inside of control volume.

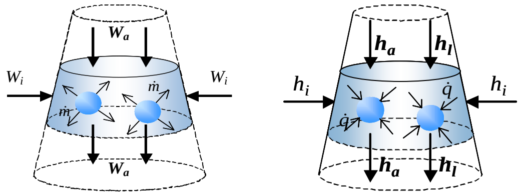


Figure 3- Schematic diagram of (a) the mass vapor conservation and (b) the energy conservation

Solving this equation system [(1) – (19)], and other complementary equations, by the BDF method of the Fortran Library, it was possible to verify the outflow (Q_a), temperature (T_a) and humidity ratio (W_a) of the treated air, at the end of the plume. These output data will be used to simulate the thermal performance of the building.

As has been emphasized, an extensive number of parametric tests were used to check the effect of each variable of the spray model. In addition to these parametric tests, the spray model was also simulated as an adiabatic saturator, in which

$$\frac{dT_a}{dz} = 0 \quad \text{and} \quad \frac{dW_a}{dz} = 0 \tag{20}$$

whose liquid temperature (T_l) has reached the wet bulb temperature (WBT) with great precision. These tests, including that for adiabatic saturation, were carried out for different liquid input temperatures and for different angles of spray. As these parametric tests are so extensive, only the results for the base case of the spray model will be shown in this paper.

INTEGRATION TO A BUILDING HYGROTHERMAL MODEL

The present spray model can be easily integrated to building simulation programs, even using a dynamic model for the analysis of a whole-building hygrothermal behavior. The thermodynamics properties of moist air at the spray inlet and outlet constitute the main connection between the building and the vaporization system. Figure 4 represents the iteration between the spray and the environment, where the air outflow of the spray model provides data for input to the building model and surrounding air is a variable, at each time step, for the spray model.

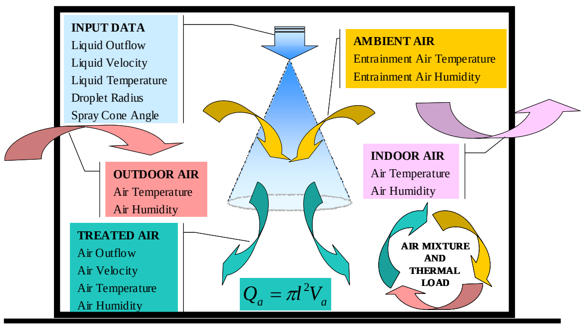


Figure 4- Schematic diagram of coupling model

Therefore, between the solution of the spray model and the evaluation of the building behavior there is a process of mixing of air masses under different conditions of temperature and humidity. The indoor and outdoor air and the treated air outflow for the nozzles interact at each time step with the sources of sensible and latent loads of the building.

In the present work, a lumped formulation for calculating both room air temperature and humidity ratio is considered for each building zone.

Equation (21) describes the energy balance, for a building room submitted to loads of conduction, convection, short-wave solar radiation, inter-surface long-wave radiation, infiltration, ventilation and the sensible and cooling loads for the HVAC system.

$$\dot{E}_t + \dot{E}_g = \rho_{air} c_{air} V_{air} \frac{dT_{db}}{dt} \qquad (21)$$

where,

- $\dot{E}_t$, energy flow that crosses the room control surface (W);
- $\dot{E}_g$, internal energy generation rate (W);
- ρ_{air}, air density (kg/m^3);
- c_{air}, specific heat of air (J/kg-K);
- V_{air}, room volume (m^3);
- T_{db}, room air dry-bulb temperature (ºC).

The term $\dot{E}_t$, on the energy conservation equation, includes loads associated to the building envelope (sensible and latent conduction heat transfer), furniture (sensible and latent), fenestration (conduction and solar radiation), openings (ventilation and infiltration) and HVAC systems. For the spray, this term is divided into sensible and latent terms as:

$$\dot{E}_{t,spray,sensible} = m_{spray\_air} c_{p,air} \left(T_{spray\_outlet} - T_{int} \right) \qquad (22)$$

and

$$\dot{E}_{t,spray,latent} = m_{spray\_air} h_{LV} \left(W_{spray\_outlet} - W_{int} \right) \qquad (23)$$

Where the mass flow represents the dry-basis air mass flow through the vaporization system. The term h_{LV} represents the latent heat of vaporization calculated at the inlet dry-bulb temperature.

On the other hand, the term $\dot{E}_g$ accounts for the internal heat gain from people, equipment and lighting.

The sensible and latent loads associated to the combined heat and moisture transfer problem through the building zone porous walls are calculated as:

$$Q_{wall,S}(t) = \sum_{i=1}^{m} h_{c,i} A_i \left[T_{i,x=L}(t) - T_{db}(t) \right] \qquad (24)$$

$$Q_{Wall,L}(t) = \sum_{i=1}^{m} L(T_{i,x=L}(t)) h_{m,i} A_i \left[\rho_{v,n,i}(t) - \rho_{v,int}(t) \right] \qquad (25)$$

In Eqs. (24) and (25), A_i represents the area of the *i-th* surface, h the convective transfer coefficients for heat (h_c) and for mass (h_m), $T_i(t)$ the temperature at the *i-th* internal surface of the considered zone, L the vaporization latent heat e ρ_v the water vapor density.

The temperature and the vapor density distributions - needed to determine the sensible and latent conduction loads given by Eqs. 26 and 27 - are calculated by using the combined heat and moisture transfer model based on the Philip and DeVries theory and the method presented in (Mendes *et al.* , 2002), which solves the following set of partial different governing equations for each control volume within the porous building element:

$$\rho_0 c_m(T,\theta) \frac{\partial T}{\partial t} = \frac{\partial}{\partial x}\left(\lambda(T,\theta) \frac{\partial T}{\partial x}\right) + L(T)\rho_l \frac{\partial}{\partial x}\left(D_{Tv}(T,\theta) \frac{\partial T}{\partial x} + D_{\theta v}(T,\theta) \frac{\partial \theta}{\partial x}\right) \qquad (26)$$

$$\frac{\partial \theta}{\partial t} = \frac{\partial}{\partial x}\left(D_T(T,\theta) \frac{\partial T}{\partial x} + D_\theta(T,\theta) \frac{\partial \theta}{\partial x} \right) \qquad (27)$$

where ρ_0 is the solid matrix density, c_m, mean specific heat, T, temperature, t, time, λ, thermal conductivity, L, latent heat of vaporization (= h_{LV}), θ, volume basis moisture content, j_v, vapor flow, ρ_l, the water density, D_{Tv}, vapor phase transport coefficient associated to a temperature gradient, $D_{\theta v}$, vapor phase transport coefficient associated to a moisture content gradient, D_T, mass transport coefficient associated to a temperature gradient [m^2/s-°C] and D_θ mass transport coefficient associated to a moisture content gradient [m^2/s].

Internally, the wall is exposed to convection and phase change and externally (x=0) it is exposed to solar radiation ($\alpha\, q_r$), convection $h_{ext}(T_{ext} - T(0))$ and phase change ($h_{m,ext}(\rho_{v,ext} - \rho_{v,x=0})$), so that the energy equation becomes

$$-\left(\lambda(T,\theta)\frac{\partial T}{\partial x}\right)_{x=0} - \left(L(T) j_v\right)_{x=0} = h_{ext}\left(T_{ext} - T_{x=0}\right) + \alpha q_r + L(T) h_{m,ext}\left(\rho_{v,ext} - \rho_{v,x=0}\right) \qquad (28)$$

The conservation governing equations are then discretized by using the control-volume formulation method with a central difference scheme and linearized vapor concentration difference at the boundaries in terms of temperature and moisture content. The resulting algebraic equations are solved using the MultiTriDiagonal-Matrix Algorithm (MTDMA) as described in Mendes and Philippi (2005).

In terms of water vapor balance, different contributions have been considered: ventilation, infiltration, internal generation, porous walls, furniture, Spray system and people breath. In this way, the lumped formulation becomes:

$$(\dot{m}_{\text{inf}}+\dot{m}_{vent})(w_{ext}-w_{\text{int}})+J_b+J_{ger}+J_{poroussurface}+J_{Spray}=\rho_{air}V_{air}\frac{dW_{\text{int}}}{dt} \tag{29}$$

In the water vapor mass balance Eq. 29, the term J_{Spray} is calculated as:

$$J_{Spray}=\dot{m}_{spray\_air}\left(W_{Spray\_outlet}-W_{\text{int}}\right) \tag{30}$$

and the term $J_{porous\ surface}$ is calculated in the same way as the latent conduction load.

The room air temperature (T_{int}) is calculated by the energy conservation equation, while the supply air temperature from the vaporization system by using the spray model.

In the integrated system, the user can pre-define environmental settings including temperature and humidity set points and dead bands. The interval of time, Δt, represents the ON and/or OFF time of the spray calculated from user configuration controls and thresholds.

The treated air outflow, while the spray (or set of sprays) is ON, is then mixed with indoor air and the ventilation air. The changes in sensible and latent load are computed and updated as a function of the spray ON and/or OFF time.

SIMULATION PROCEDURE

This section presents the weather data and the building and system simulation parameters used to perform the simulations.

Weather Data

The Test Reference Year (TRY) of the city of Curitiba, Brazil (latitude: -25.40º; longitude: 47.19º; GMT: -3h; altitude: 949.0m) 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.

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

Table 1: Monolithic cellular concrete physical characteristics.

λ (W/mK)	0.18
Thickness (m)	0.15
U (W/m^{2}K)	1.33
R (m^{2}K/W)	0.750
Density	650
C_p (J/kgK)	840

Figure 5 shows the building geometric characteristics of zone building. All zone walls, floor and roof have the same physical characteristics shown in Table 1.

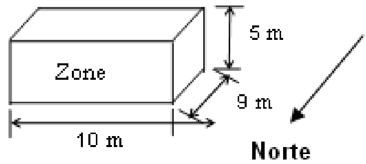


Figure 5: Zone building geometric characteristics.

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

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

	Facade	Windows quantity	Height x Width
Zone	East	1	1.5 m x 4.0 m
	West	1	1.5 m x 4.0 m

Internal thermal gains from people, lighting and equipment have been considered during working days (from Monday to Friday) from 8 am to 6 pm. Ten occupants have been considered as individual sources dissipating 60 W of sensible heat and 80 W of latent heat, while lighting power density of 45W/m^2 and ten equipments with power of 100 W have been selected for the present study.

The ASHRAE cooling load weighting factors have been used in order to calculate the radiative gain on the internal envelope surfaces.

Spray Data Description

Table 3 presents the evaporative cooling system input parameters for the simulations. The number of nozzles is evaluated in the results section.

Table 3: Simulation parameters for the evaporative cooling system.

Parameter	**Value**
Spray half-angle	30 graus
Water flow	1.567E-6 m^3/s
Initial temperature of droplets	28.0 ºC
Initial speed of droplets	83.0 m/s
Radius of droplets	5.5e-6 m
Air speed at the inlet	45.0 m/s

PROGRAM INTERFACE - POWERDOMUS

The building hygrothermal model and the spray model were integrated using the same object-oriented programming language C++ Builder. The Power Domus interface is modularly divided in parts.

The Building Description Module (Fig. 6) 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.

The user can interact with building envelope and openings and after defining the geometry, a text file will be automatically created so that the user can verify all the inputs. The software edition interface is 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 and colors. Domus makes use of the OpenGL API to render both 2-D and 3-D panels. Nowadays OpenGL is the premier environment for developing portable, interactive 2-D and 3-D graphics applications. If hardware acceleration is detected, then high quality graphics can be affordable. The wall layers are edited in a particular window, where all the available materials, geographical orientation (wall azimuth and inclination), soil properties, painting and other parameters are shown.

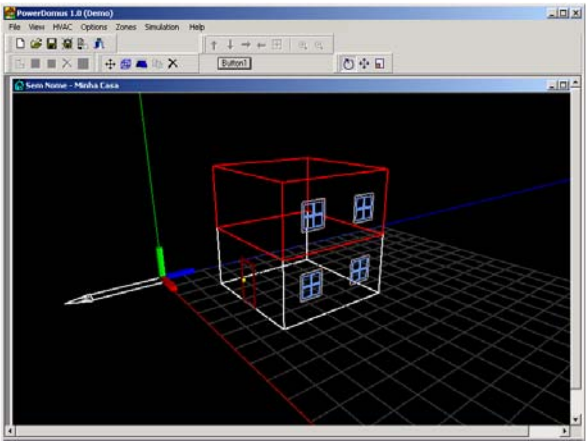


Figura 6: Example of a two-zone building edition

Figure 7 presents the evaporative cooling system interface where the user provide information on the parameters shown in Table 3 to simulate the spray.

The software hourly weather files provide dry bulb temperature, relative humidity, direct and diffuse solar radiation and wind speed and direction. The program also provides a weather summary window and a psychrometrics charts where temperature and relative humidity data presented in the weather file can also be plotted in order to analyse quickly the potential of using an evaporative cooling.

Figure 8 presents the control interface for the evaporative cooling system where the user can choose either an on/off or a PID control strategy. On that window, the user also provide the *set-point* and dead band values for both temperature and relative humidity. For the PID control technique, the user also provides the proportional, integral and derivative constants (K_p, K_i,*and* K_d).

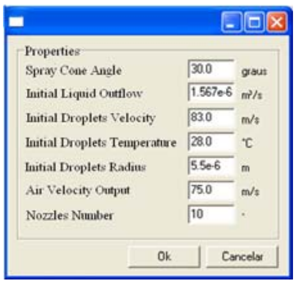


Figura 7: Interface for entering evaporative cooling parameters.

Figure 8: Edition interface for the evaporative cooling system control.

RESULTS

In this section, results obtained through simulations are presented according to the building and spray combined model and the input parameters as described in the previous sections. The simulations were carried out using the most complete wall model of DOMUS building program which considers the phenomenon of adsorption and desorption of moisture within the walls (level 0). It has been adopted for these simulations a time step of 1 minute, making possible to obtain a greater sensitivity on the variation of the internal temperature of the zones.

In order to evaluate the performance of the models, the number of the nozzes (NA) and the ventilation rate (it terms of air changes per hour, n) so that the system meets the standard of thermal comfort provided by the control system configuration. The results presented below were obtained with a ON/ OFF system control acting both on the indoor air temperature and relative humidity. The system was configured according to a 24ºC temperature set-point and a 1°C hysteresis and a 90% relative humidity set-point with a 10% hysteresis, scheduled to work from 8:00 am 6:00 pm.

Figure 9 and 10 shows the variation of temperature and relative humidity within a 10-ach (air changes per hour) zone , on January 5, comparing the performance under the presence of five-nozzles (Fig. 9) and 20-nozzle spray systems.

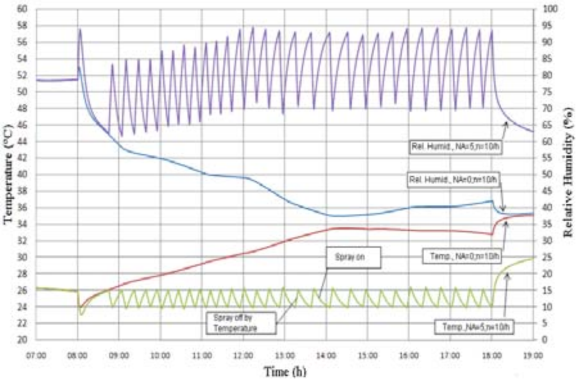


Figure 9: Temperature and relative humidity variation in zone with (NA= 5) and without spray (NA = 0)

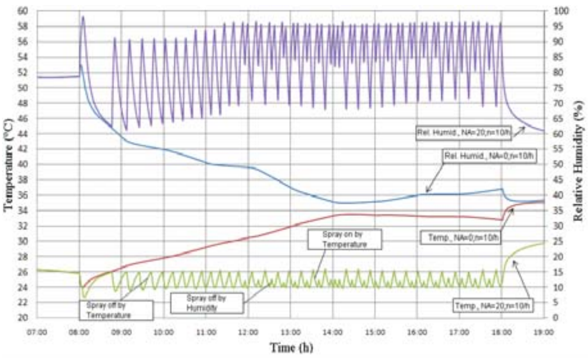


Figure 10: Temperature and relative humidity variation in zone with (NA= 20) and without spray (NA = 0)

The results indicate in both cases that temperature and relative humidity can be maintained by the spray control system. The higher reduction on the zone temperature by the evaporative cooling system was 9.5°C around 2:30 p.m..

For the case of five and twenty nozzles, the ON/ OFF control system has respectively switched 29 and 55 times. This increase can be explained by the fact that in this case the spray was turned off as for the temperature as the humidity control. This occurs because the relative humidity of the zone exceeds the maximum limit set in the control system, due to a great number of nozzles.

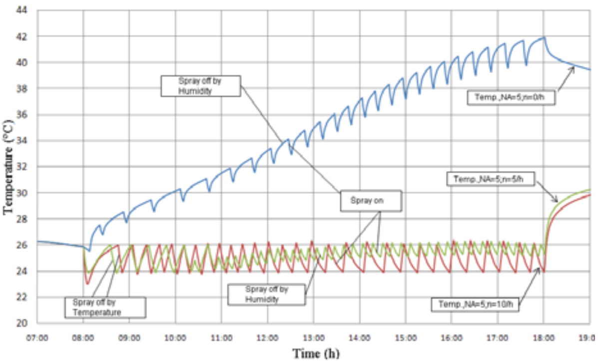


Figure 11: Effect of air change rate on air temperature in zone with spray system.

Figures 11 and 12 show respectively the temperature and relative humidity variation within the zone for three different cases varying the number of air changes per hour (n = 0/h, n = 5/h and n = 10/h), using the five-nozzle evaporative system (NA = 5). One can notice, in Figures 11 and 12, that the lower the ventilation rate the higher the ON/OFF cycles frequency, without reaching a good temperature control (25°C). This occurred due to the fact the system was conceived to control both temperature and relative humidity. Indeed, it has been observed that the temperature control using a 10-ach ventilation system was the most effective.

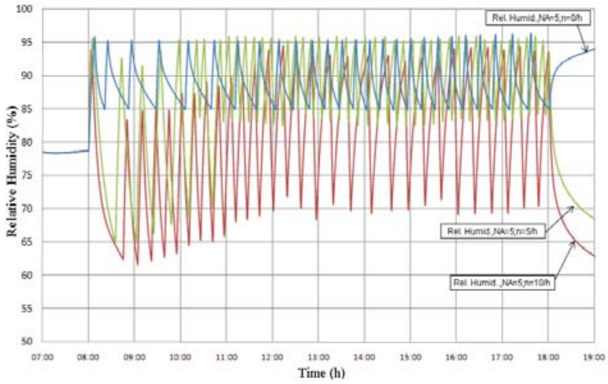


Figura 12: Effect of air change rate on air relative humidity in zone with spray system

CONCLUSIONS

This work has been focused on the integration of a direct evaporative cooling spray model into a hygrothermal building simulation tool.

The integration of the evaporative cooling spray model, developed by Silva et al. (2004), into the building simulation program PowerDomus (Mendes et al., 2005) makes possible to bring together the scales of the droplet, spray and building, through an integral building simulation model.

Aiming to analyze the performance of the models integration, some simulations have been carried out varying the number of nozzles and the ventilation rate to verify their effect on the indoor air temperature and relative humidity.

The results presented show consistency of models integration and how effective an evaporative system

can be to control dry-bulb temperature.

It has also been noticed the effects of the number of the nozzles and the ventilation rate. As the control system acts on both temperature and relative humidity, a great number of nozzles may reduce the thermal control efficiency. In addition, is has been shown the importance of an appropriate air change rate on the spray efficiency to control the indoor air psychrometrics conditions.

In conclusion, the integration of those models may assist the design of micro aspersion systems for cooling and/or humidification of buildings and the design of ventilation systems in order to reduce both energy and water consumption. It also makes possible to analyze and compare the cost/benefits relation between the microaspersion system and traditional cooling systems. Additionally, the integrated spray and building hygrothermal model is also useful for the development of new control systems for combined microaspersion and air conditioning systems.

ACKNOWLEDGEMENTS

The authors thank CNPq - *Conselho Nacional de Desenvolvimento Científico*, CAPES- *Coordenação de Aperfeiçoamento de Pessoal de Nível Superior* and Eletrobrás for support of this work.

NOMENCLATURE

a	radius of the droplet [m];
B_T	heat transfer number
B_Y	Spalding's mass transfer number
c	tangent of the half-angle of the spray
C_D	drag coefficient
c_p	specific heat [kJ/kg.K];
D_{AB}	mass diffusivity [m^2/s];
F_z	average force per unit volume [N/m^3];
h	enthalpy [kJ/kg];
h_{lv}	latent heat of vaporization [kJ/kg];
l	radius of spray jet [m];
l_a	radius of air jet [m];
$\dot{m}$	mass flow [kg/s];
n	air change rate [h^{-1}];
NN	number of nozzles
$\dot{q}$	heat flow at droplet surface [kW/m^2];
Q	volumetric outflow [m^3/s];
q_s	sensible heat load [kW];
q_l	latent heat load [kW];
r	radial distance [m];
Re	Reynolds number
RH	relative humidity [%];
Sc	Schmidt number
T	temperature [°C];
V	velocity [m/s];
W	humidity ratio [kg/kg];
z	vertical distance from nozzle [m];
α	volume fraction
β	entrainment coefficient
ρ	specific mass [kg/m^3];
ν	kinematic viscosity [m^2/s];

Subscripts

a	air inside of the control volume
e	outdoor conditions
g	gas mixture (air + vapor)
l	liquid
i	induced air, indoor conditions
s	saturation, droplet surface
o	initial condition

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