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SPRAY VAPORIZATION FOR EVAPORATIVE COOLING OF BUILDINGS

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ABSTRACT

This article presents a one-dimensional model of spray vaporization that can be easily used in thermal performance simulations of buildings. A mathematical model of momentum, heat and mass transfer in the atomization zone was developed.

Using a discrete particles model with separate flows and solving a non-homogeneous ordinary differential equations system, it is possible to verify the outflow, temperature and humidity of the treated air, at the end of the plume.

These algorithms could be coupled to a building simulation code.

INTRODUCTION

Sprays have many engineering applications, such as, in combustion systems, agricultural and industrial processes, dust control, fire fighting, spray drying, transport systems, nuclear reactor core cooling and evaporative cooling, among others. In many parts of the world, evaporative cooling of air is an attractive energy efficient technique for producing a comfortable indoor environment.

This kind of device is being used to reduce the use of air conditioning in large industrial and commercial buildings. In buildings with mechanical air-conditioning systems, on the other hand, water spray evaporative cooling helps to reduce the energy use when used as a precooling system. Despite this, direct and indirect evaporative coolers are still considered as *emerging technologies* with regard to promote energy efficiency in the building sector (Vine, 2002).

Accurate prediction of spray evaporation is extremely difficult due to the complex physical phenomenon. Therefore, building simulation softwares do not incorporate direct evaporative cooling spray models.

Over the years, numerous papers have been published on spray models. However, the great majority of studies on sprays (Faeth, 1983; Sirignano, 1999; Baskaya, 1998; Catoire et al., 1998; Mostafa and Elghobashi, 1985; Chen and Pereira, 1996; Sommerfeld et al., 1993; Masodi and Sirignano, 2000; among others) relate to fuel injection in combustion chambers, in which the emphasis is on the heat and mass transfer that occur at higher temperatures. Other studies approach the dynamics of spray without considering the heat and mass exchanges (Lee and Tankin, 1984; Ghosh and Hunt, 1994; Ghosh and Hunt, 1998).

Kachhawaha et al. (1998) present a study whose objectives are close to those of this paper. However, their work was developed in a wind tunnel with known amounts of air the momentum exchange between the droplets and the surrounding air being neglected. Consequently, air entrained by the droplets was negligible, and this is an essential factor to be considered for sprays in free jet streams.

To consider all variables, extensive studies resulted in developing more sophisticated versions of models, including also commercial packages such as Flow3D and Phoenix. It should be stressed, however, that a practical application of the models requires profound knowledge of spray and some experience in programming and calculation of systems with distributed parameters. Therefore, there is still a considerable demand for simple models, which can be used to optimize the decisions of the designer (Zbicinski, 1995).

By being able to estimate the leaving temperature and humidity of the air the designer may explore possibilities for using evaporative cooling on specific applications for cooling, humidifying and ventilating systems in all types of commercial and industrial applications in all climates.

Therefore, a study is being developed that considers the evaporative cooling by micron aspersion of water, bringing together the scales of the droplet, spray and building.

SPRAY CHARACTERISTICS

Sprays may be produced in various ways. Essentially, the atomization principle needs a high relative speed between the liquid to be atomized and the surrounding air. In the pressure atomizers a jet or sheet of liquid disintegrates due to the exchanges of momentum between the liquid, at high speed, and surrounding air.

Thus, sprays are generated simply by forcing liquid under pressure through small orifices. The main purpose of sprays is to increase the surface area of the injected liquid, thereby increasing the heat and mass transfer rates. For this study we consider a pressure atomizer with a plain orifice of 0.2mm diameter.

As in practice nozzles do not produce sprays of uniform droplet size, the representative mean diameter of the droplets was calculated through SMD (Sauter Mean Diameter) from frequency distribution curves, supplied by the manufacturer. Based on the preceding consideratoins, the SMD calculated is 11μm.

The initial droplets velocity (V_{lo}) was derived from flow number terms (FN), and is expressed as the ratio between mass flow and the effective flow area of a pressure atomizer (Lefebvre, 1989):

$$FN = \frac{\dot{m}_l}{(\Delta P_L \rho_l)^{0.5}} \tag{1}$$

thus

$$V_{l0} = \frac{\dot{m}_l}{FN.\rho_l} \tag{2}$$

where $\dot{m}_l$ = liquid mass flow (kg/s), ΔP_L = pressure differential (Pa) and ρ_l = liquid density (kg/m^3).

PHYSICAL MODEL

The study of sprays involves many engineering areas, from fluid dynamics to heat and mass transfer, atomization and multiphase dynamics. The solution of this kind of systems includes the solution of equations of momentum, energy and mass conservation, for each phase. In some applications the airflow is not affected by the presence of the droplets, but in others it is dependent on them, as in the present case.

A major objective of modeling sprays is to reduce the time and cost of the development process. From the existing lines of spray modeling we will adopt the *discrete particle model in separate flows*, in which 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). In this procedure, the behavior of spray is derived from the description of a finite number of particles. The effect of droplets on the gas phase is considered by introducing appropriate source terms into the gas phase equations of motion. One of the great advantages of this model is the computational economy, since it is not necessary to formulate the whole spray field (Faeth, 1983). According to Sirignano (1999), the Lagrangian formulation is preferred because it reduces the numerical error due to artificial diffusion.

Dynamic behavior

The correct balance of velocities and distances reached by the spray is also important, in order to determine the heat and mass transfer coefficients between the droplets and the air.

When a liquid is sprayed into a non-condensing gas environment, it leads to an exchange in momentum between the droplets of spray and the gas, in this case the air. The droplets decelerate by 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 the entrained gas enters the spray, it drags small liquid droplets at the outer regions of the spray inward, and may even lead to the eventual contraction of the spray (Ghosh and Hunt, 1994), as shown in Figure 1.

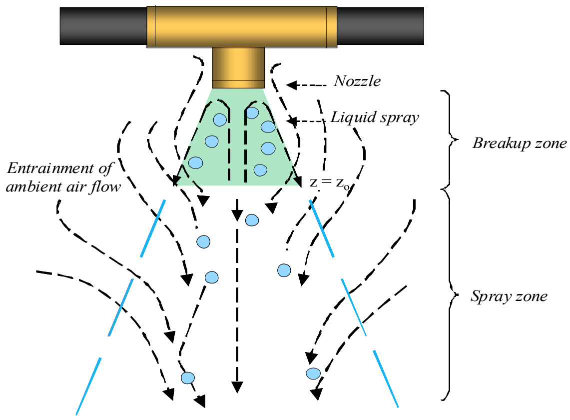

Figure 1 Schematic diagram of the zones and breakup of the typical liquid film of a spray

As in momentum jets the turbulence drags the external flow with it and therefore the volume flux of the gas in the spray increases, which implies that external air is entrained into the spray at its boundary. Initially, the velocities of the droplets are much greater than that of the air stream and are not much affected by it, but subsequently they slow down and become comparable with the air speed (Ghosh and Hunt, 1994).

It is important to point out that spray is effectively formed when the sheet of liquid disintegrates into steady droplets. References on the behavior of the liquid film can be found in Lee and Tankin (1984) and Lefebvre (1989).

The characterization of the vertical zones of spray, illustrated in Figure 2, can be summarized as follows (Ghosh and Hunt, 1994):

- Zone I (near): the speed of the water is so great ($V_l \gg V_a$) and the amount of air is so small ($l_a = l_0$) that this does not reduce the speed of the liquid and, consequently, the speeds of the droplets ($V_l = V_{l0}$) and air ($V_{a0} = b.V_{l0}$) are very close to the initial speed of the liquid (V_{l0}), where $b<1$.
- Zone I (forced): the amount of air in the spray increases reducing the speeds of the droplets and the air that had been sped up in the previous sub zone. The reductions of V_a

and V_l are accented and this reduction in speed is compensated with the increase in l_a and l.

- Zone II: even if $V_l \geq V_a$, the air volume in the spray is so much greater that the droplets do not determine the flow of air. At this moment there is an uncoupling between the behavior of the air and the droplets.
- Zone III: in this zone the droplets reach their terminal speed ($V_l = V_t$) and the air tends to have the same speed as the droplets or, eventually, even less ($V_l \geq V_a$).

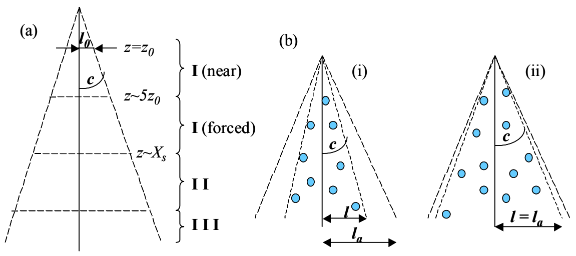

Figure 2 (a) Three main spray zones in a vertical spray. (b) Definition sketches for narrow (i) and wide (ii) spray.

Through a one-dimensional analysis it is possible to calculate the variation of the air jet width l_a and the mean axial velocity of the gas $< V_a >$, as a function of distance z in the spray zone.

In the initial zone of droplets disruption there is a large component of the airflow normal to the jet axis. However, below the height $z=z_0$, which defines the beginning of the spray zone, the air that flows through the spray is largely axial, and the normal entrainment velocity is small compared to the gas velocity within the spray jet.

The outflow of liquid Q_l that is discharged from the nozzle distributes spherical droplets in the level $z=z_0$, where the average ray of spray is l_0 and $z_0=l_0/c$. These droplets have an initial velocity V_{l0} and occupy a volume fraction $<\alpha_0>$ of the volume of spray.

The average force per unit volume F_z at a point (r, z) is the product of the force on each particle $f_z\ (r, z)$ and of the volume fraction $\alpha\ (r, z)$.

For one-dimensional analysis it is necessary to average the relevant variable across the spray, as a function only of z. These variables are: l_a, V_a, V_l, α and F_z/ρ_a, where the latter four are averaged across the spray cross-section.

Ghosh and Hunt (1994) assume that the entrainment is similar to that in a turbulent jet and therefore proportional to the average air velocity in the jet. For a typical jet the entrainment coefficient β is 0.11. This average value is also derived through the use of the equations of momentum conservation, presented by Stoecker (1967). Following Stoecker (1967), smoke tests of a circular jet show a pattern roughly similar to a cone, whose angle is approximately 22°. But for sprays with a wide angle ($c \gg \beta$) the width of the air flow is not determined by the turbulent spreading of the air jet but by the spray itself, so that $l_a=l$ (Ghosh and Hunt, 1994).

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

- Entrainment law, assuming formally that $c \ll 1$

$$\frac{d}{dz}\pi l_a^2 V_a = 2\pi\beta V_a l_a \tag{3}$$

$$\frac{d}{dz}\pi l_a^2 V_a = \frac{d}{dz}\pi l^2 V_a \qquad \text{for} \quad c \gg \beta \tag{4}$$

where the entrainment coefficient β is 0.11 and the tangent of the half-angle of spray c is defined as l_0/z_0.

Substituting Equation (4) into (3) and solving for β, we obtain

$$\beta = c + \frac{l_a}{2V_a}\frac{dV_a}{dz} \tag{5}$$

for wide spray ($c \gg \beta$), when $l_a=l$ and $c \ll 1$.

- Volume fraction

$$\alpha = \frac{Q_l}{V_l \pi l^2} \tag{6}$$

where $l = cz$ (7)

- Force on the droplets

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

where a is the radius of the droplet, ρ_a is the specific mass of air and C_D is the drag coefficient [Wallis, 1969, cited by Ghosh and Hunt, 1994]

$$C_D = \frac{24}{\mathrm{Re}}\left(1 + 0.15|\mathrm{Re}|^{0.687}\right) \tag{9}$$

for $|Re| \leq 1000$. Reynolds Number (Re) is defined as

$$\mathrm{Re} = \frac{2a}{\nu_a}|V_l - V_a| \tag{10}$$

where ν_a is the kinematic viscosity of air.

- Rate of change of the average momentum of droplets

$$V_l \frac{dV_l}{dz} = \left(\frac{F_z}{\rho_a}\right) \cdot \left(\frac{\rho_a}{\rho_l}\right) \cdot \alpha^{-1} \tag{11}$$

where ρ_l is the specific mass of water.

- Rate of change of momentum flux of the air jet

$$\frac{d}{dz}(\pi l_a^2 V_a^2) = -\left(\frac{F_z}{\rho_a}\right)\pi l^2 \tag{12}$$

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

$$V_a=V_{a0}; \qquad V_l=V_{l0} \qquad \text{and} \qquad l_a=l_0 \tag{13}$$

Without an external driving force for the airflow, the initial average air velocity V_{a0} must always be less than the average liquid velocity, which means that $b<1$, in $V_{a0}=b.V_{l0}$, where b is a arbitrary coefficient. Ghosh and Hunt (1994) showed that the value of V_{a0} does not have a significant effect on the values of V_a and l_a downstream.

The dynamic behavior is obtained solving equations (3) – (13).

Description of the droplet model

Liquid particle evaporation is usually associated with a relative motion between the droplet and surrounding air. These velocity profiles have a large impact on the mass, momentum and energy exchanges between the air and the droplets, which can be dealt with different levels of complexity.

In this study a model for the vaporization of droplets with infinite thermal diffusivity was used. This model produces results with excellent correlation with the experimental data (Faeth, 1983). The following assumptions, typical of most droplet models, are adopted in this work (Faeth, 1983):

i. The droplet is assumed to be spherical.

ii. The spray is assumed to be dilute. Under this assumption droplet collisions are ignored and the effect of adjacent droplets on droplet transport rates are neglected.

iii. The flow around the droplet is assumed to be quasisteady, i.e., the flow immediately adjusts to the local boundary conditions and droplet size at each instant of time.

iv. The radial velocity of the liquid surface due to the evaporation of liquid is neglected.

v. Effects of drag and forced convection are represented by empirical correlations.

vi. Gas phase transport is based on mean ambient properties and the effect of turbulent fluctuation is ignored.

vii. The liquid surface is assumed to be in thermodynamic equilibrium with negligible temperature jumps due to finite rates of evaporation. Furthermore, the effect of surface tension is neglected when determining phase equilibrium at the liquid surface.

viii. The pressure is assumed to be constant and equal to the local mean ambient pressure.

ix. Only concentration diffusion is considered, neglecting thermal diffusion.

x. Radiation between the droplets and their surroundings is neglected.

xi. The gas phase Lewis number is assumed to be unity in the droplet model (Faeth, 1977).

xii. The properties of the gas flow field are assumed to be constant at each instant of time.

xiii. The transport process within the droplet is neglected and its properties are assumed to be constant at each instant of time.

xiv. The spray is assumed to be mono-dispersed.

Discrete particle model

Through 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:

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

$$r = \infty: \qquad T = T_a \qquad W = W_a \tag{14}$$

where T_a and W_a are the conditions outside the boundary layer of the droplet.

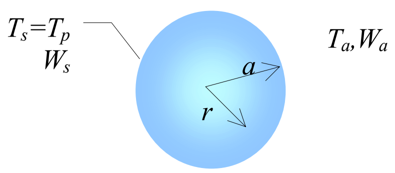

Figure 3 Liquid particles interacting with gaseous atmosphere

- Rate of change of the droplet radius

$$\frac{da}{dz} = -\frac{\dot{m}}{4\pi a^2 \rho_l V_l} \tag{15}$$

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

$$\dot{m} = \left\{ 1 + \frac{0.278 \mathrm{Re}^{1/2} Sc^{1/3}}{\left[1 + \frac{1.232}{\mathrm{Re}\, Sc^{4/3}} \right]^{1/2}} \right\} \dot{m}_{\mathrm{Re}=0} \qquad (16)$$

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

$$\frac{\dot{m}_{\mathrm{Re}=0}}{4\pi a \rho_a D_{AB}} = \ln\left[\frac{1+W_s}{1+W_a} \right] \qquad (17)$$

where D_{AB} is mass diffusivity and W_s is gotten by the expression

$$W_s = 0.62198 \frac{p_{ws}}{p - p_{ws}} \qquad (18)$$

where p is the total atmospheric pressure and p_{ws} the saturation pressure of the pure water

$$p_{ws} = 1000 \exp(A T_l^2 + B T_l + C + D T_l^{-1}) \qquad (19)$$

For temperatures (T_l) between 273.15 and 322.15K (ASHRAE, 1996):

A= 0.1255001965x10^{-4}

B= -0.1923595289x10^{-1}

C= 0.2705101899x10^{2}

D= -0.6344011577x10^{4} (20)

- Rate of change of the droplet temperature

$$\frac{dT_l}{dz} = \frac{3}{\rho_l c_{p_l} V_l} \left(\frac{\dot{q}}{a} - \frac{\dot{m} H_l}{4\pi a^3} \right) \qquad (21)$$

where c_{p_l} is the specific heat of the water, H_l is the latent heat of vaporization and $\dot{q}$ is the total exchange of heat per unit area between the surface of the droplet and the surrounding air, corrected for Re ≠ 0 (Faeth, 1983).

However, following Faeth (1977), Equation (21) can be rearranged so that the wet bulb state can be interpreted as follows

$$\frac{dT_l}{dz} = \frac{3}{4} \frac{\dot{m} H_l}{\pi a^3 \rho_l c_{p_l} V_l} \left(\frac{B_T}{B_Y} - 1 \right) \qquad (22)$$

where

$$B_T = \frac{c_{p_a}(T_a - T_l)}{H_l} \qquad (23)$$

and

$$B_Y = \frac{W_s - W_a}{1 + W_a} \qquad (24)$$

Vaporization in spray

Following the discrete particle model in separate flows, we analyze the conservation of mass and energy. At the scale of the spray the volume fraction, α, is used to ponder the variables calculated in the discrete particle model, such as mass flows and outflow that arrive at each control volume.

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:

$$\frac{1}{Q_l} \frac{dQ_l}{dz} = -\frac{3}{4} \frac{\dot{m}}{\pi a^3 \rho_l V_l} \qquad (25)$$

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

$$\frac{d}{dz}\left[\rho_a V_a \pi l^2 W_a (1-\alpha) \right] - 2\pi l \beta V_a \rho_i W_i - \frac{3}{4} \frac{\dot{m} Q_l}{\pi a^3 V_l} = 0 \qquad (26)$$

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

$$\frac{d}{dz}\left[\rho_a V_a \pi l^2 h_a (1-\alpha) \right] - 2\pi l \beta V_a \rho_i h_i + \rho_l c_{p_l} \frac{d}{dz}(T_l Q_l) = 0 \qquad (27)$$

where h_i and h_a are

$$h_i = c_{p_a} T_i + W_i (2501 + 1.805 T_i) \qquad (28)$$

$$h_a = c_{p_a} T_a + W_a (2501 + 1.805 T_a) \qquad (29)$$

respectively, the enthalpy of the induced air and the enthalpy of the air inside of control volume.

SIMULATION AND RESULTS

Silva et al. (2002) using Equation (21) solved this system of equations for narrow spray (β=0.11) in the SOPHT program (Kaviany, 2002) and verified the spray performance under different conditions of temperature and humidity.

In this simulation an attempt was made to evaluate one spray that is usually used in evaporative cooling of buildings. Thus, we will simulate a high-pressure atomizer (5,516 to 6,895 kPa) with a plain orifice of 0.2mm of diameter which produces droplets with 11µm in a spray cone angle of 60°.

In this kind of spray, with very small droplets, the air and droplet velocities are very close, as shown in Figure 4. The initial droplets velocity (V_{lo}) calculated in Equation (2) is 83 m/s for liquid mass flow ($\dot{m}_l$) of 0.001567 kg/s.

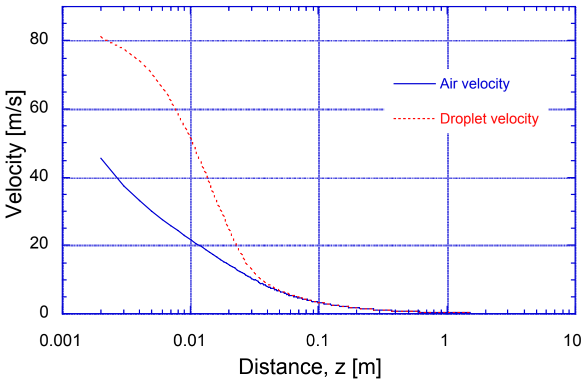

Figure 4 Air and droplet velocities

The reach of the spray and the properties of the air in the spray depend on the temperature and humidity conditions of the surrounding air. A sample of the results is shown in Figure (5), whose initial values are in Table 1.

Table 1
Initial values for the numerical solution of the Figures (5), (6) and (7)

	Condition 1	Condition 2	Condition 3
T_i	**45**	**35**	**35**
W_i	**0.005945**	**0.01054**	**0.021443**
RH_i	**10**	**30**	**60**
$z=z_0$	0.001	0.001	0.001
V_{l0}	83	83	83
V_{a0}	75	75	75
l_{a0}	5.77e-4	5.77e-4	5.77e-4
a_0	5.5e-6	5.5e-6	5.5e-6
Q_{l0}	1.567e-6	1.567e-6	1.567e-6
T_{a0}	45	35	35
W_{a0}	0.005945	0.01054	0.021443
RH_{a0}	10	30	60
T_{l0}	30	30	30

See nomenclature for units.

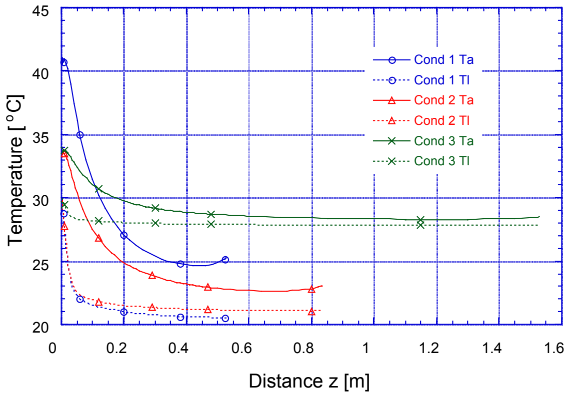

Figure 5 Temperature and humidity effects on spray

When a droplet is subjected to a fixed environment at moderate pressures, it approaches a wet-bulb state after a short transient heating or cooling period. In the wet–bulb state, all the heat reaching the droplet is utilized in the vaporization heat of the evaporating liquid and the droplet temperature remains essentially constant (Faeth, 1983).

In all the situations in Figure (5) the temperature of the droplet tends to WBT with a small difference ($WBT – T_l$) of precision in the model that varies from -0.4°C (-1.4%) to -0.9°C (-4,1%), respectively, in condition 3 and 1. This uncertainty is probably a result of empirical correlations of drag and forced convection.

At the end of the spray when the droplets had evaporated, a trend of increase in the air temperature can be observed. In the same way, it can be observed in Figure 6 that the relative humidity tends to decrease after the total evaporation of the droplets.

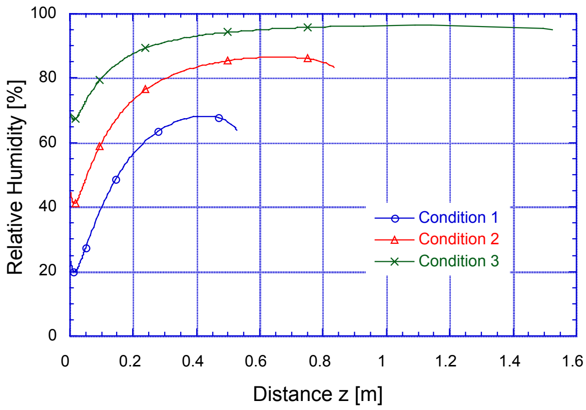

Figure 6 Effect of the surrounding conditions on relative humidity of the air treated by spray

Figure 7 shows the distance reached by the spray before total evaporation of the droplets as a function of ambient conditions.

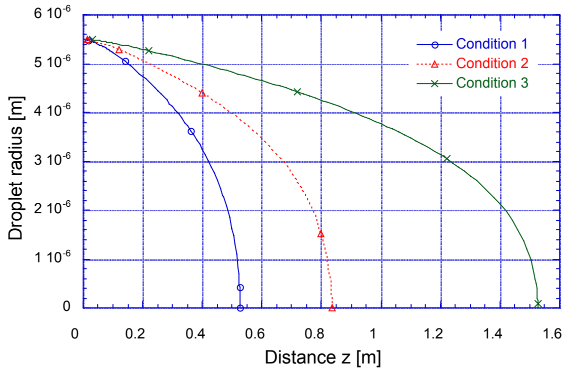

Figure 7 Effect of the temperature and humidity surrounding on distance reached by spray

Table 2
Final values from the numerical solution of the Figures (5), (6) and (7)

	Condition 1	Condition 2	Condition 3
T_i	45	35	35
W_i	0.005945	0.01054	0.021443
RH_i	10	30	60
z	0.5266	0.8352	1.5271
V_l	0.64	0.40	0.22
V_a	**0.64**	**0.40**	**0.22**
l_a	**0.3039**	**0.4819**	**0.8811**
a	1.25e-13	1.57e-13	2.15e-13
Q_l	1.88e-29	3.66e-29	9.42e-29
T_a	**25.3**	**23.0**	**28.5**
W_a	**0.012907**	**0.014742**	**0.023535**
RH_a	**63.9**	**83.2**	**95.0**
T_l	20.6	21.1	27.8
$(T_i - T_a)$	19.7	12.0	6.5

See nomenclature for units.

From bold values in Table 2 it is possible to get the treated air outflow and its properties, as shown in a schematic diagram in Figure 8.

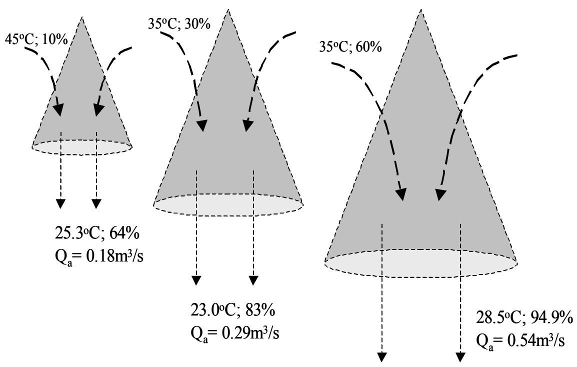

Figure 8 Schematic diagram comparing the treated air outflow and its properties at different conditions

An attempt was made to verify the effect of liquid temperature on air conditions. No significant influence was observed, as shown in Table 3.

Table 3
Effect of liquid temperature on air conditions

T_{l0}	**30°C**	**21.5°C = WBT**	**15°C**
z	0.8352	0.8426	0.8486
T_a	23.1	23.0	22.9
RH_a	83.3	83.3	83.4

See nomenclature for units.

CONCLUSIONS

The parametric analysis presented in this work show that the model is sufficiently consistent, considering the number of interdependent variables. The objective to simplify spray behavior for evaporative cooling of buildings has been obtained with a small difference between T_l and *WBT*, that theoretically should be reached.

Experimental data for simulated spray were not found, hindering direct comparison of results.

The outflow data for treated air and its properties will be used as input data for iterative processes with building simulation software. Thus, the values of T_i and W_i won´t be constant at the time of simulation.

This paper shows how to simulate spray behavoir and take information to use in the buildings simulation. The next step in this study will be to elaborate a bidimensional model, considering the eventual contraction of the spray .

ACKNOWLEDGMENTS

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NOMENCLATURE

a	radius of the droplet (m),
b	arbitrary coefficient
c	tangent of the half-angle of spray
C_D	drag coefficient
c_p	specific heat (kJ/kg.K)
D_{AB}	mass diffusivity (m^2/s)
FN	flow number
F_z	average force per unit volume (N/m^3)
h	enthalpy (kJ/kg)
H_l	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),
p	total atmospheric pressure (Pa)
p_{ws}	pressure of saturation (Pa)
$\dot{q}$	heat flow at droplet surface (kW/m^2)
Q	volumetric outflow (m^3/s)
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
ΔP_L	pressure differential (Pa)
ρ	specific mass (kg/m^3)
ν	kinematic viscosity (m^2/s)

Subscripts

a	air inside of the control volume
l	liquid
i	induced air
s	saturation, droplet surface
0	initial condition

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