PREDICTING TEMPERATURE AND MOISTURE DISTRIBUTIONS IN CONDITIONED SPACES USING THE ZONAL APPROACH

Kátia Cordeiro Mendonça[1], Etienne Wurtz[2] and Christian Inard[2]

[1]LST, Pontifical Catholic University of Paraná, Rua Imaculada Conceição, 1155, Prado Velho, Curitiba, PR, 80215-905, BRAZIL; Email: k.mendonca@pucpr.br
[2]LEPTAB, University of La Rochelle, Avenue Michel CREPEAU, 17042 La Rochelle CEDEX, France

ABSTRACT

Moisture is one of the main problems in buildings. In spite of the complexity to describe moisture physical phenomena, recent technological improvements have allowed them to be incorporated into building simulation programs. However, it is still unrealistic to predict whole building hygrothermal behavior using CFD models. As a result, the available whole hygrothermal building programs assume the room air to be perfectly mixed, which is a poor assumption for conditioned spaces.

To take into account the convective phenomena imposed by mechanical ventilation, we have used an intermediate approach (zonal method) to develop a model library to predict whole hygrothermal behavior in conditioned rooms.

In this paper, a zonal library, which includes two models in order to consider building envelope moisture buffering effects, is proposed. It is also discussed how to take into account the dynamic aspect of jet airflow in the zonal method. The zonal library is applied to a case study to show the impact of the external humidity on the whole hygrothermal performance of a room equipped with a vertical fan-coil unit.

INTRODUCTION

Moisture interacts in many ways with building elements, therefore affecting building performance. It causes deterioration of building materials, contributes to the poor indoor air quality and, in humid climates, represents one of the major loads in conditioned spaces. So, in order to reduce cooling energy while maintaining occupant comfort it is important to understand and model moisture transport correctly.

At present, some whole-building models that take into account moisture effects, including the buffering of moisture in construction materials, are available in the current literature: e.g. CLIM2000 (Woloszyn, 1999), BSIM2000 (Rode et al., 2001), DOMUS (Mendes et al., 2003) and WUFI (Holm at al., 2003). However, all of them assume room conditions to be uniform (*lumped-parameter model*), which is a non-realistic assumption in conditioned spaces. On the other hand, *Computational Fluid Dynamics* (CFD) models, that could describe in detail the non-uniform behavior of conditioned rooms, are time consuming. Consequently, they are difficult to apply to whole-building long-term simulations.

The intermediate approach, called *zonal method* (Inard, Bouia and Dalicieux, 1996; Musy, 1999), appears then as an alternative to these two models. This method is based on dividing a room into a relative small number of zones - typically on the order of tens to hundreds - compared to thousands and more for typical CFD simulations. In each zone (sub-volume) of the coarse-grid the state variables of air are considered uniform, except for the pressure that varies hydrostatically. While not as fine-grained as CFD simulation, zonal models do give useful information about temperature and moisture distributions, not available from lumped-parameter models, and which is important in comfort analysis.

In order to consider the convective phenomena imposed by mechanical ventilation in whole hygrothermal simulation of conditioned spaces, a zonal model library is proposed. This paper describes the proposed zonal library, which has been developed into the modular simulation platform SPARK (Sowell and Haves, 2001), and takes into account the main hygrothermal effects like moisture sources and moisture adsorption and desorption in construction materials.

MODEL DESCRIPTION

The proposed library was structured in three groups representing the three building domains: indoor air, envelope and HVAC system.

The first group, *indoor air sub-model*, is related to the indoor air space, where airflow speed can be considered low (where there are no driving flows). The second group, *envelope sub-model*, is related to the radiation exchanges between envelope and its neighborhood and to the heat and mass transfers through the envelope material. The latter can be represented by four sub-models of different complexity levels, with two of them taking into account moisture buffering effects in construction materials. Concerning the *HVAC system sub-model*, it refers to the whole system, which means the equipment, the control and the specific airflow from the equipment.

All these models were coupled into SPARK, where the resulting set of non-linear coupled equations is solved simultaneously.

Indoor air sub-model

Unlike CFD models, only the mass and energy conservation equations are solved for each zone into which the indoor air domain is divided. In the place of Navier-Stokes equations, a very simplified form of momentum equation based on the orifice flow equation is used to complete the model. The set of equations belonging to this sub-model is described by Mendonça et al. (2002).

Envelope sub-model

As mentioned before, this sub-model is subdivided into two other groups that represent radiation exchanges between envelope and its neighborhood, and heat and mass transfer across the building materials.

Internal radiation exchanges

Internal radiation exchanges can be modeled by two different simplified methods: *fictitious enclosure* (Walton, 1980) and the well-known *Radiant Mean Temperature*.

Basically, fictitious enclosure method calculates short and long wave radiation considering that each surface of the room envelope (in our case, each zone face adjacent to the building envelope) exchanges radiation only with a second surface that is equivalent to all other surfaces of the envelope.

Heat and mass transfers across building material

The four sub-models that describe heat and mass transfers through building materials correspond to:

- No heat nor moisture transfers;
- Only heat transfer;
- Coupled heat and moisture transfers (moisture transport in vapor phase);
- Coupled heat and moisture transfers (moisture transport in liquid and vapor phase).

Detailed description concerning these sub-models is available in Mora et al. (2003).

HVAC system sub-model

Because of the simplified hypothesis applied to momentum equation, the indoor air sub-model cannot represent airflow pattern generated by mechanical ventilation. As a result, besides describing the mechanical equipment and its control law, HVAC system sub-model also describes the specific airflow provided by the air supply diffuser.

In the current zonal library, all equations of this sub-model correspond to a vertical fan-coil unit. The equipment is represented by a steady state model of a cooling and dehumidifying coil from SPARK library, which is based on ASHRAE Secondary Toolkit (Brandemuehl, 1993). The room temperature control is modeled by a proportional law, acting on the cooling coil water flow rate and an on-off law, acting on the fan airflow rate.

Specific airflow generated by mechanical ventilation

In general, this sub-model consists of substituting the simplified momentum equation of indoor air sub-model for empirical or semi-empirical specific flow laws. In addition to maintaining the simplicity of the zonal method, preceding works (Wurtz, 1995; Musy, 1999; Riederer, 2002) have shown that this approach provides reasonable results.

As a first approach, the power-law function of momentum equation was replaced by the flow law concerning bi-dimensional isothermal jet (Rajaratman, 1976):

$$\dot{m}_{da} = \dot{m}_{da,0}\left(1+0.248\right)\sqrt{\frac{z}{e_0}} \qquad (1)$$

In order to preserve the modularity of the proposed library, the jet flow was modeled as illustrated in figure 1:

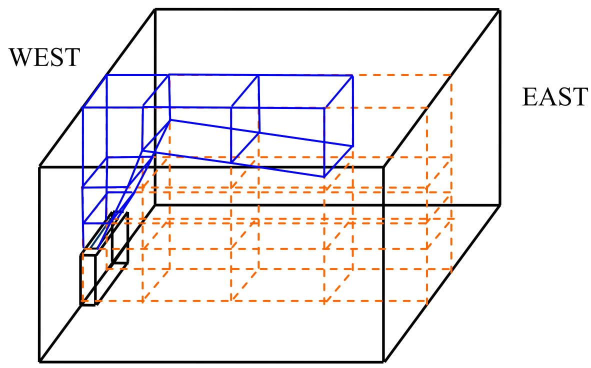

Figure 1 Zonal partitioning of a room containing a vertical jet

Figure 1 shows the middle plan of a room that was divided into sixteen volumes or zones. The volumes containing the jet are subdivided into two other volumes. One containing air belonging to the jet itself and another containing air from the surroundings. Note that this type of partitioning is also employed for the air-conditioning zone.

It was assumed that the jet does not develop laterally as it rises. Nevertheless, it entrains air through its side faces.

To take into account the jet dynamic behavior (the adaptation of the number of the zones occupied by the jet during the simulation), the jet was imagined to be always composed of two parts, a vertical and a horizontal one. Thus, the jet always rises up to the ceiling and contours it, as illustrated in figure 1. Then, the jet throw is calculated using equation 2 proposed by Goldmann at al. (1985) for an anisothermal wall jet:

$$\frac{Z_{max}}{e_0} = 4.5 Ar_0^{-0.389} \qquad (2)$$

where the initial Archimedes number is given by:

$$Ar_0 = \frac{g(T_s - T_a)}{T_r} \frac{e_0}{U_0^2} \qquad (3)$$

If the jet throw, Z_{max}, is greater than the distance Z between the air supply diffuser and the upper surface of a zone (or EAST surface for the horizontal part of the jet – see figure 1), the sub-volume containing the jet itself is represented by the jet equations (indoor air sub-model with empirical or semi-empirical specific flow laws). In case Z_{max} is less than the distance between the air supply diffuser and the middle of the zone, the sub-volume containing the jet is represented by the expressions associated to the indoor air sub-model for low airflow speed. In the remaining case, when Z_{max} is greater than the distance between the middle of the zone and the air supply diffuser, and less than that between the upper surface and the air supply diffuser, the sub-volume can be represented by either jet model or indoor air model, depending on which model had been used in the previous iteration. If in the previous iteration the sub-volume had been described by the jet model, the sub-volume will continue to be described by jet model equations. Otherwise, it will be described by indoor air model equations.

SIMULATION

To study the effect of external environment humidity on the whole hygrothermal performance (indoor air, envelope and air-conditioning system) of a conditioned room, the proposed zonal model was applied to a case study submitted to two different climatic conditions (hot and wet and hot and dry).

This case study represents a large office conditioned by a vertical fan-coil unit with a rectangular air supply diffuser (0.40 m x 1.05 m). Figure 2 shows a sketch of this office, its dimensions (m) and the location of the conditioning equipment.

The adopted zonal partitioning is illustrated in figure 3 for the plan containing the driving flow, as well as the zone dimensions (m), the position of the two openings for external ventilation, and the reference numbers of the zones.

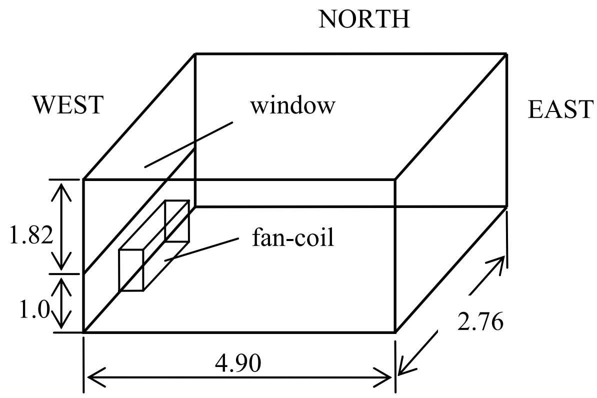

Figure 2 Sketch of the studied room

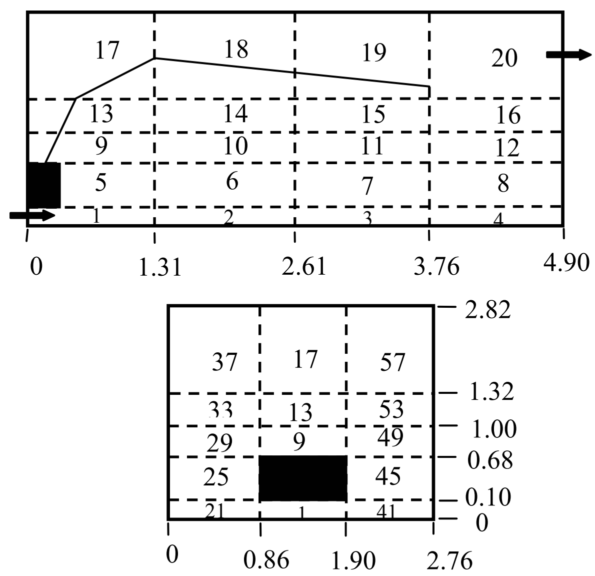

Figure 3 Zonal partitioning for the plan containing the jet

The cooling coil capacity of the fan-coil unit ranges from 2600 W to 4000 W, depending on the airflow and the water flow rates. In this case study, air was admitted to be supplied at a constant rate (620 m^3h^{-1}).

Except for the EAST and the WEST surfaces, all of the other walls of this office were thought to be in contact with other conditioned rooms. Therefore, the latter were modeled under constant boundary conditions (t=25°C e φ = 50%), while the EAST and the WEST walls were modeled under external boundary conditions. As the EAST surface was assumed to be adjacent to a corridor, it was not subjected to external radiation exchanges.

The simulations are made for five days. A weather file with hourly data for hot and wet climate conditions was chosen from SOLMET Miami weather data (1979). A variant of this weather data (30% lower relative humidity) was used to represent the hot and dry climate.

The room envelope is composed of a high hygroscopic material with the dry basis properties given in table 1.

Table 1
Dry-basis material for Mortar

PROPERTY	UNIT	VALUE
Cp_0	$Jkg^{-1}K^{-1}$	932
ρ_0	kgm^{-3}	2050
λ_0	$Wm^{-1}K^{-1}$	1.92
ε	%	18

The convective heat (h_T) and mass transfer (h_M) coefficients, for both external and internal surfaces, are given in table 2.

Table 2
Convective heat and mass transfer coefficients

WALL	EXTERNAL		INTERNAL	
	h_T	h_M	h_T	h_M
	$Wm^{-2}K^{-1}$	ms^{-1}	$Wm^{-2}K^{-1}$	ms^{-1}
WEST	11	0.030	3	0.0003
EAST	3	0.0003	3	0.0003
NORTH	3	0.0003	3	0.0003
SOUTH	3	0.0003	3	0.0003
FLOOR	3	0.0003	3	0.0003
CEILING	3	0.0003	3	0.0003
WINDOW	20	0	20	0

The NORTH and the SOUTH surfaces are 0.1 m thick while the other surfaces are 0.2 m.

Heat and mass transfers across the building envelope were simulated by the sub-model that takes into account moisture transport in liquid and vapor phases.

Concerning the internal radiation exchanges, they were modeled by Radiation Mean Temperature sub-model, using a radiant transfer coefficient, $h_R = 6\ Wm^{-2}K^{-1}$.

Fifty percent of the radiant load coming from the window was assumed to reach the floor surface. The remaining fifty percent was assumed to be distributed to the other surfaces proportionally to their areas.

To simulate heat and moisture gains from an occupant, internal sensible-heat and moisture sources corresponding respectively to 65 W and 0.022 gs^{-1} were admitted, from 8 am to 6 pm every day.

A constant ventilation of 0.5 ach was also admitted.

Initial indoor conditions were considered equal to the initial external conditions at midnight on 3rd July: 23.1°C and 97% for the hot and wet case, and 23.1°C and 67% for the dry and hot case. In both cases, initial conditions for building materials were 25°C and 50%.

DISCUSSION AND RESULT ANALYSIS

Figure 4 shows the temperature distribution vs. time in the external environment and in the occupant space (represented by the zones numbered 10, 11 and 12 in figure 3) for hot and wet climates.

Temperature distributions in the three zones are quite similar and remain around 24°C, 1°C below the set-point. This occurs because the sensor was assumed to be integrated to the equipment. Therefore, the temperature control is based on the mixing of the return airflow and the outside airflow.

Similar results were found for the hot and dry case, since its sensible loads are identical to those of the humid case.

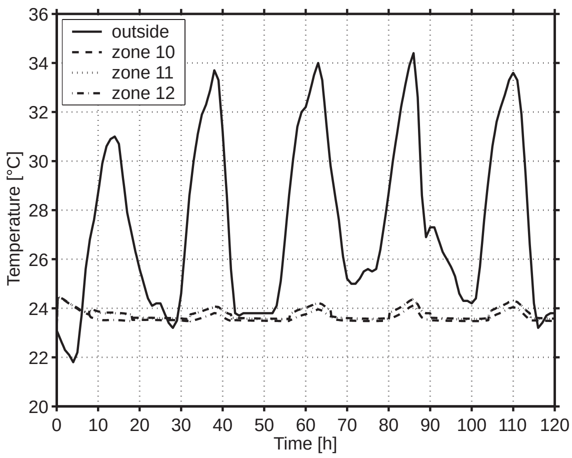

Figure 4 Temperature distributions vs. time for the hot and wet case

The corresponding relative humidity distribution vs. time for the three zones and the external environment are shown in figures 5 and 6, respectively, both for humid and dry cases.

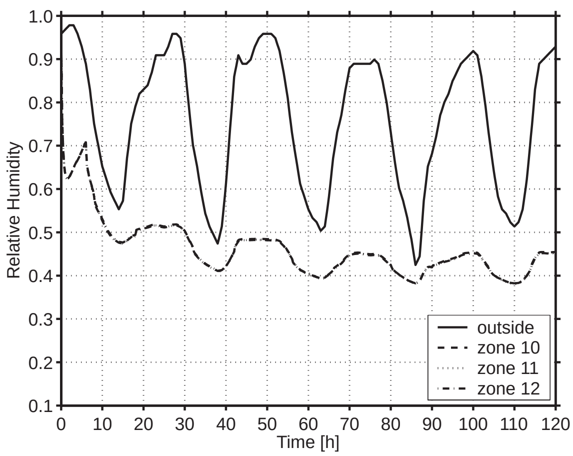

Figure 5 Relative humidity distributions vs. time for the hot and wet case

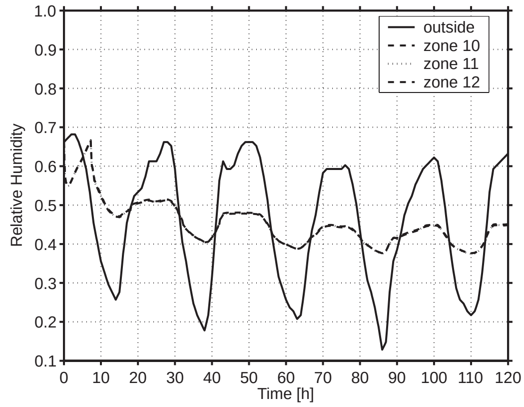

Figure 6 Relative humidity distributions vs. time for the hot and dry case

Although there is an important difference in external relative humidity distribution between the two cases, in both of them the relative humidity distribution in the occupant space has remained at the same level. The cooling coil capacity of the fan-coil unit was able to absorb the higher latent thermal load in the humid case.

Similarly to the temperature distribution, there is no difference between the relative humidity distributions in the three chosen zones. We attribute this fact to the quite homogeneous boundary conditions imposed to the problem.

Figure 7 illustrates, for both cases, the distribution vs. time of water vapor extracted from or added to air due to moisture sorption effects, external ventilation and dehumidification by the fan-coil unit.

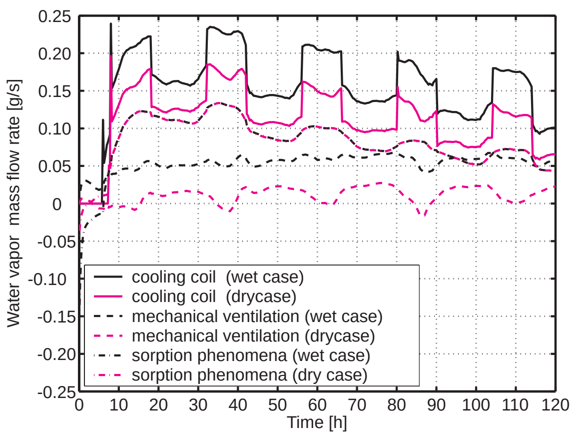

Figure 7 Net water vapor mass flow rate distributions vs. time due to cooling coil dehumidification, sorption phenomena and mechanical ventilation for humid and dry cases

Figure 7 shows the curves related to sorption phenomena for the two cases are coincident. Besides that, they are positive, which means the desorption was the dominant phenomenon. The amount of water vapor added to air by this phenomenon is equivalent or greater than to that added by external ventilation. Since the net amount of water vapor added to air by sorption phenomena and the internal moisture sources are equal for both cases, the higher amount of water vapor extracted from air by the cooling coil in the humid case refers to the higher amount of water vapor introduced by external ventilation.

Validation of the complete zonal library against other numerical models is under way by means of Annex 41: Whole-Building Heat, Air And Moisture Response (Moist-Eng), from the International Energy Agency (IEA). Until now, only the envelope sub-models have been evaluated. Temperature and moisture distributions for a single wall predicted by the two sub-models that consider sorption effects were compared to those predicted by other envelope moisture calculation tools. A good agreement between their results was obtained.

CONCLUSION

A zonal model library to predict whole hygrothermal behavior in conditioned rooms has been proposed. Due to the zonal aspect of the model, the non-uniform behavior of air imposed by mechanical ventilation can be taken into account. The proposed zonal library has been developed into a modular simulation environment, making it possible to incorporate different models to consider heat and moisture transfers across the building envelope.

The proposed library has been used to evaluate the effect of the external environment humidity on the hygrothermal behavior of a conditioned room. In the studied case results showed that building material moisture buffering effects can be as important as external ventilation.

ACKNOWLEDGMENT

The authors thank CNPq – Brazilian Research Council - for supporting this work.

REFERENCES

Brandemuehl, M. J. 1993. HVAC 2 Toolkit. A Toolkit for secondary HVAC System Energy Calculations. Joint Center for Mangement, University of Colorado, USA.

Goldman, D., Jaluria, Y. 1985. Effect of Opposing Buoyancy on the Flow in Free and Wall Jet, *Journal of fluid mechanics*, **166**, pp. 41-56.

Holm, A., Kuenzel, H.M., Sedbauer, K. 2003. The Hygrothermal Behaviour of Rooms: Combining Thermal Building Simulation and Hygrothermal Envelope Calculation, *Proceedings of the 8th International IBPSA Conference*, pp. 499-505, Eindhoven, The Netherlands.

Inard C, Bouia H, Dalicieux P. 1996. Prediction of air temperature distribution in building with a zonal model. *Energy and Buildings*, **24** (2), pp 125-132.

Mendes, N., Oliveira, R. C. L. F., Santos, G. H. 2003. DOMUS 2.0: A Whole-Building Hygrothermal Simulation Program, *Proceedings of the 8th International IBPSA Conference*, pp. 863-870, Eindhoven, The Netherlands.

Mendonça, K.C., Inard, C., Wurtz, E., Winkelmann, F.C., Allard, F. 2002. A Zonal Model for Predicting Simultaneous Heat and Mass Transfers in Buildings, *Proceedings of the 9th International Conference on Indoor Air Quality and Climate*, pp. 518-523, Monterey, CA, USA.

Mora, L., Mendonça, K.C., Wurtz, E., Inard, C. 2003. SIMSPARK: An Object-Oriented Environment to Predict Coupled Heat and Mass Transfers in Buildings, *Proceedings of the 8th International IBPSA Conference*, pp. 903-910, Eindhoven, The Netherlands.

Musy, M. 1999. Génération Automatique de Modèles Zonaux pour l'Etude du Comportement Thermo-Aéraulique des Bâtiments, *Ph.D. Thesis*, Université de La Rochelle, France.

NATIONAL CLIMATIC DATA CENTER. 1979. SOLMET Hourly solar radiation – surface meteorological observations. Final Report TD - 9724, Asheville, NC, 1979.

Rajaratnam, N. 1976. Turbulent Jets, *Elsevier Scientific Publishing Company*, Amsterdam, The Netherlands.

Riederer, P. 2002. Thermal Room Modelling Adapted to the Test of HVAC Control Systems, *Ph.D. Thesis*, Ecole des Mines de Paris, France.

Rode, C., Grau, K. 2001. Synchronous Calculation of Transient Hygrothermal Conditions of Indoor Spaces and Building Envelopes, *Proceedings of the 7th International IBPSA Conference*, pp. 491-498, Rio de Janeiro, Brazil.

Sowell, E. F., Haves, P. 2001. Efficient Solution Strategies for Building Energy System Simulation, *Energy and Buildings*, **33**, pp. 309-319.

Walton, G. 1980. A New Algorithm for Radiant Exchange in Room Loads Calculations, *ASHRAE Transactions*, **86** (2), pp. 190-208.

Woloszyn, M. 1999. Modélisation hygro-thermo-aérauliques des bâtiments multizones : proposition d'une stratégie du système couplé, *Ph.D. Thesis*, INSA de Lyon, France.

Wurtz, E. 1985. Modélisation Tridimensionnelle des Transfers Thermiques et Aérauliques dans le Bâtiment en Environnement Orienté-Objet, *Ph.D. Thesis*, Ecole Nationale des Ponts et Chaussées, France.

NOMENCLATURE

Ar_0:	Initial Archmedes number
Cp_0:	Dry porous materials specific heat [$Jkg^{-1}K^{-1}$]
e_0:	Thickness of the jet when $z=h_0$, [m]
g:	Gravitational acceleration, [ms^{-2}]
h_0:	Distance from the air supply diffuser to the fictitious origin of the jet, [m]
$\dot{m}_{da}$:	Dry air mass flow rate, [kgs^{-1}]
$\dot{m}_{da,0}$:	Initial dry air mass flow rate [kgs^{-1}]
t:	Temperature [°C]
T_a:	Temperature of indoor air, [K]
T_s:	Temperature of wall surface, [K]
T_a:	Temperature of reference, [K]
U_0	Initial velocity, [ms^{-1}]
Z:	Distance measured from the air supply diffuser [m]
Z_{max}:	Jet throw, [m]
ε:	Porosity [-]
λ_0:	Dry porous material thermal conductivity [$Wm^{-1}K^{-1}$]
ρ_0:	Dry porous material density [kgm^{-3}]
φ:	Relative humidity [-]