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Building hygrothermal modeling by nodal method

Farah Gabsi
Université de Lorraine
CNRS, CRAN, F-54000 Nancy, France
LGIPM, F-57000 Metz, France
Farah.Gabsi@univ-lorraine.fr

Frédéric Hamelin
Université de Lorraine
CNRS, CRAN, F-54000 Nancy, France
Frederic.Hamelin@univ-lorraine.fr

Nathalie Sauer
Université de Lorraine
LGIPM, F-57000 Metz, France
Nathalie.Sauer@univ-lorraine.fr

Abstract—This article deals with the problem of modeling a multi-zone office building to ensure its hygrothermal comfort. An analysis of the hygrothermal behavior is carried out using a nodal method based on the SimscapeTM library in MATLAB/Simulink® environment. The application target is the “Eco-Safe” platform (CRAN Nancy, France) that fits in a global context of smart home [1].

Index Terms—buildings, hygrothermal modeling, thermal comfort, nodal analysis.

I. INTRODUCTION

The park of existing buildings in France has 2/3 of dwellings built before 1975. The renewal rate of this park is less than 1% per year. It therefore represents a significant potential for achieving the objectives of “Factor 4”, validated by the Grenelle Environnement in 2007, which aims to divide by 4 greenhouse gas emissions by 2050. The impact of mass transfers on the real energy behavior of buildings has been highlighted for several years [2]. The transfer equations are known, but very few works [3], [4] integrate these models in order to implement energy efficient control laws that ensure hygrothermal comfort [5] within buildings. Yet, moisture directly impacts comfort in buildings since its rate reflects both the health of buildings and the quality of indoor air. The main causes of development of moisture [6] are related to defects in HVAC systems (direct infiltration of moist air, negative pressure gradient across the envelope, inadequate ventilation, etc.) or inappropriate use of the building (high occupancy density, bad air conditioning settings, etc.).

In this paper, a complete methodology of modeling the hygrothermal behavior of a multi-zone office building is proposed. It uses the Simscape™ library in MATLAB/Simulink® environment. The application target is the “Eco-Safe” platform (CRAN Nancy, France) that fits in a global context of smart home [1].

II. THERMAL COMFORT

The notion of comfort is closely related to one-off adjustments in body regulation, both global and local. The perception of cold and heat depends on many parameters. However, it is possible to statistically characterize average

comfort zones. If we consider a volume of air ① (called afterwards zone ①) bounded by n walls, the most determining parameters of atmosphere are the ambient temperature of the air $T_{o\oplus}$, the temperature of the n walls $T_{k\oplus}$ ($1 \leq k \leq n$), the relative humidity of the air $\phi_{\oplus}$ and the flow rate of air [7]. The objective of this paragraph is to specify the requirements of hygrothermal comfort in current situations (for resting persons).

A. Operative temperature

The operational temperature $T_{o\oplus}$ in zone ① is a simplified measure of human thermal comfort derived from air temperature, mean radiant temperature and air speed. When the air speed is less than 0.2 m/s, (as is typical in buildings), $T_{o\oplus}$ can be approached by the arithmetic mean of air temperature in zone ① ($T_{o\oplus}$) and mean radiant temperature $T_{mr\oplus}$ [7]:

$$T_{o\oplus} \approx \frac{T_{mr\oplus} + T_{o\oplus}}{2} \quad (1)$$

where $T_{mr\oplus}$ is defined by:

$$T_{mr\oplus} = \frac{\sum_k S_{k\oplus} \times T_{k\oplus}}{\sum_k S_{k\oplus}} \quad (2)$$

A model of adaptive thermal comfort was proposed by [8] based on objective investigations carried out in buildings in situ. The model consists of a simple linear regression of the comfort temperature $T_{comf}(k)$ depending on the filtered outside air temperature:

$$T_{comf}(k) = \begin{cases} 0.049 T_{rm}(k) + 9.2 & \text{if } T_{rm}(k) \leq 283.15 \\ 0.206 T_{rm}(k) - 34.85 & \text{if } T_{rm}(k) > 283.15 \end{cases} \quad (3)$$

with $T_{rm}(k)$ such that:

$$T_{rm}(k) = 0.8 T_{rm}(k-1) + 0.2 T_{dm}(k-1) \quad (4)$$

and $T_{dm}(k-1)$ the daily mean outdoor temperature $T_{\odot}$ on day $(k-1)$.

A first criterion of thermal comfort is to ensure that operational temperature $T_{o\oplus}$ (1) belongs to the comfort temperature interval defined by $T_{comf} \pm 2$ K. Nevertheless, a source of thermal discomfort remains if the difference between the ambient temperature $T_{\odot}$ and the wall surface temperature $T_{k\oplus}$ is too big. So, the comfort objectives have to include $|T_{\odot} - T_{k\oplus}| < 8$ K for glass areas and 5 K for opaque walls.

B. Humidity

Moisture may condense on surface (Σ_k) bounding zone ① when the relative humidity of the air $\phi_{\textcircled{1}}$ is important and when the difference of temperature between $T_{\textcircled{1}}$ and that of the walls $T_{k\textcircled{1}}$ is too big. $\phi_{\textcircled{1}}$ depends on the vapor pressure $P_{v\textcircled{1}}$ and the saturation vapor pressure $P_{vs\textcircled{1}}$ in zone ① according to the relation:

$$\phi_{\textcircled{1}}(\%) = 100 \frac{P_{v\textcircled{1}}}{P_{vs\textcircled{1}}} = 100 \frac{\rho_{v\textcircled{1}}}{\rho_{vs\textcircled{1}}} \quad (5)$$

where $\rho_{v\textcircled{1}}$ and $\rho_{vs\textcircled{1}}$ are respectively the densities of water vapor in zone ① associated with vapor pressure $P_{v\textcircled{1}}$ and $P_{vs\textcircled{1}}$. Saturation vapor pressure $P_{vs\textcircled{1}}$, which does not decrease with the barometric pressure, corresponds to the maximum partial pressure that can reach $P_{v\textcircled{1}}$ without generating condensation. It can be estimated from Tetens' formula [7]:

$$P_{vs\textcircled{1}} = 610,78 \times \exp\left(\frac{17.269(T_{\textcircled{1}} - 273.15)}{T_{\textcircled{1}} - 35.85}\right) \quad (6)$$

To avoid the surface condensation on wall (Σ_k) , it is therefore necessary that the temperature $T_{k\textcircled{1}}$ be higher than the dew-point temperature $T_{d\textcircled{1}}$ of this zone ①, which is the temperature from which the air becomes saturated with water vapor and that water condenses. For a relative humidity equal to $\phi_{\textcircled{1}}$ and an internal temperature $T_{\textcircled{1}}$, $T_{d\textcircled{1}}$ is such that:

$$610,78 \times \exp\left(\frac{17.269(T_{d\textcircled{1}} - 273.15)}{T_{d\textcircled{1}} - 35.85}\right) = \frac{\phi_{\textcircled{1}}}{100} P_{vs\textcircled{1}} \quad (7)$$

In addition, just as the difference in temperature between two faces of a (Σ_k) creates a heat flow, the difference in humidity revealed by the vapor pressure creates a gradient in the material and therefore a "flow" of moisture. To avoid any condensation phenomenon in the heart of the wall, it is imperative that at all points of this wall the vapor pressure is lower than the saturation vapor pressure $P_{vs\textcircled{1}}$.

Furthermore, air quality standards recommend for France a humidity rate $\phi_{\textcircled{1}}(\%)$ between 40 and 60 %. So, overall, we will have to ensure in terms of humidity:

$$\begin{cases} 40\% < \phi_{\textcircled{1}}(\%) < 60\% \\ P_{vk\textcircled{1}} < P_{vs\textcircled{1}} \Rightarrow T_{k\textcircled{1}} > T_{d\textcircled{1}} \\ P_{vk} < P_{vs\textcircled{1}} \end{cases} \quad (8)$$

III. HYGROTHERMAL MODELING FOR BUILDINGS

The different models of building components are developed in this section. The approximations made are justified by taking as an example the city of Nancy (France, ≈ 230 m altitude).

A. Moisture and heat balances into one zone

1) *Sensible heat balance*: Taking into account the thermal capacity of moist air, i.e that of dry air and water vapor, the energy balance in a ① area delimited by n walls consists in equaling the sum of all incoming and outgoing heat fluxes

TABLE I
NOMENCLATURE

Par.	Description	SI units
C_{mk}	Specific moisture capacity of material (Σ_k)	kg/(kg·Pa)
C_{pda}	Specific heat capacity of dry air (= 1 006)	J/(kg·K)
C_{pk}	Specific heat capacity of material (Σ_k)	J/(kg·K)
C_{pv}	Specific heat capacity of water vapor (= 1 860)	J/(kg·K)
h_c	Heat transfer coefficient (≈ 8.3 (interior) and ≈ 34 (exterior; normal conditions) [9])	W/(m ² ·K)
h_m	Water vapor transfer coefficient ($\approx 7e^{-9} \times h_c$)	kg/(m ² s·Pa)
h_v	Latent heat of vaporization of water (= $2.5e^6$)	J/kg
n_a	Number of moles of air per m ³ (≈ 41)	mol/m ³
$P_{da\textcircled{1}}$	Dry air pressure of zone ①	Pa
R_{da}	Gas constant for dry air = 287.1	J/(kg·K)
R_v	Gas constant for water vapor = 461.5	J/(kg·K)
$S_{k\textcircled{1}}$	Area of surface (Σ_k) in contact with zone ①	m ²
$V_{\textcircled{1}}$	Volume of zone ①	m ³
V_k	Volume of material (Σ_k)	m ³
δ_{pk}	Vapor permeability of material (Σ_k) ($\frac{\epsilon_k}{\delta_{pk}} = R_k$ is the moisture resistance)	kg/(m·s·Pa)
λ_k	Thermal conductivity of material (Σ_k) ($\frac{\epsilon_k}{\lambda_k} = R_k$ is the thermal resistance)	W/(m·K)
$\rho_{da\textcircled{1}}$	Density of dry air in zone ①	kg/m ³

with the temporal variation of enthalpy $H_{\textcircled{1}}(t)$ of moist air (See table I):

$$\begin{aligned} \dot{H}_{\textcircled{1}}(t) &= (\rho_{da\textcircled{1}}(t)C_{pda} + \rho_{v\textcircled{1}}(t)C_{pv}) V_{\textcircled{1}} \dot{T}_{\textcircled{1}}(t) \\ &= C_{pda} \frac{P_{da\textcircled{1}}(t)}{R_{da}T_{\textcircled{1}}(t)} \left(1 + 1.15 \frac{P_{v\textcircled{1}}(t)}{P_{da\textcircled{1}}(t)}\right) V_{\textcircled{1}} \dot{T}_{\textcircled{1}}(t) \quad (9) \\ &= \sum_j q_{j\textcircled{1}}(t) \end{aligned}$$

$$\text{knowing that } \rho_{da\textcircled{1}}(t) = \frac{P_{da\textcircled{1}}(t)}{R_{da}T_{\textcircled{1}}(t)}, \rho_{v\textcircled{1}}(t) = \frac{P_{v\textcircled{1}}(t)}{R_vT_{\textcircled{1}}(t)}.$$

This sensible heat balance is a nonlinear differential equation. However, as in general the pressure at sea level is between 980 and 1040 hPa (usual meteorological conditions in France) and that the temperature in a ① zone is such that $288 < T_{\textcircled{1}}(t) < 308$ K, (9) can be approached for the city of Nancy by the following linear differential equation with a maximum error of $\pm 9\%$ on the estimation of the term weighting $\dot{T}_{\textcircled{1}}(t)$:

$$1.15 \times C_{pda} V_{\textcircled{1}} \dot{T}_{\textcircled{1}}(t) = \sum_j q_{j\textcircled{1}}(t) \quad (10)$$

2) *Moisture balance*: Into zone ①, the density of water vapor $\rho_{v\textcircled{1}}(t)$ does not vary as the temperature or pressure changes, as long as moisture is not added or removed. It follows:

$$V_{\textcircled{1}} \dot{\rho}_{v\textcircled{1}}(t) = \sum_j g_{j\textcircled{1}}(t) \quad (11)$$

3) *Convective transfer*: Since the area ① is assumed to be in contact with n walls $(\Sigma_{k\textcircled{1}})$, each surface side $S_{k\textcircled{1}}$, of interior temperature $T_{k\textcircled{1}}(t)$ and of water vapor density equal

to $\rho_{v\oplus}(t)$, is the place of a convective heat / moisture flow with the air of the $\oplus$ zone according to the relations [10]:

$$\begin{cases} g_{1\oplus}(t) = -\sum_{k=1}^n \frac{P_{\oplus}(t)}{n_a} h_m S_{k\oplus} (\rho_{v\oplus}(t) - \rho_{vk\oplus}(t)) \\ q_{1\oplus}(t) = -\sum_{k=1}^n h_c S_{k\oplus} (T_{\oplus}(t) - T_{k\oplus}(t)) + h_v g_{1\oplus}(t) \end{cases} \quad (12)$$

where h_m , h_c , h_v and n_a are defined in table I and $\frac{P_{\oplus}(t)}{n_a} \approx 2500$.

Temperatures $T_{k\oplus}$ and densities of water vapor $\rho_{vk\oplus}(t)$ in relation (12) can be determined by means of moisture and heat balances in walls surrounding zone $\oplus$.

B. Moisture and heat balances in walls / windows

Let us consider a homogeneous and isotropic material (Σ_k) of thickness e_k delimited by two parallel surfaces ($\Sigma_{k\oplus}$) and ($\Sigma_{k\ominus}$), this environment being able to represent for example a wall, a window, a door, etc. It is assumed that the temperature and the partial pressure of water vapor are identical at every point of each surface (one-dimensional conduction through the material). Let $(T_{k\oplus}(t), P_{vk\oplus}(t), \rho_{vk\oplus}(t))$ and $(T_{k\ominus}(t), P_{vk\ominus}(t), \rho_{vk\ominus}(t))$ be the temperature, the partial vapor pressure and the vapor density characterizing each of the two surfaces ($\Sigma_{k\oplus}$) and ($\Sigma_{k\ominus}$) respectively. It follows from this hypothesis that the mass and thermal flows are carried out perpendicularly to these surfaces. The remainder of this section presents the mass and sensible heat balances performed on the equidistant central surface of $e_k/2$ of ($\Sigma_{k\oplus}$) and ($\Sigma_{k\ominus}$). Let $T_k(t)$, $P_{vk}(t)$ and $\rho_{vk}(t)$ be the temperature, the vapor pressure and the density of vapor in (Σ_k). The model of the coupled heat and moisture transport is based on the well-known laws of Fourier, Fick and Darcy.

1) *Moisture balance*: In porous unsaturated materials, moisture transfer occurs in two forms: liquid and water vapor. The transfer of liquid water is induced by a capillary pressure gradient but is not considered in this work (building supposed healthy). The transport of water vapor is a diffusional process under a vapor pressure gradient, governed by Fick's law ($\frac{g_v}{S_k} = -\delta_{pk} \text{grad}(P_v)$). The mass balance at (Σ_k) is given by (See table I):

$$V_k \dot{\rho}_{vk}(t) = \frac{-2S_k}{\rho_k C_{mk}} \times \sum_{\oplus=\ominus, \oplus} \frac{\delta_{pk}}{e_k} (\rho_{vk}(t) - \rho_{vk\oplus}(t)) \quad (13)$$

with:

$$\begin{cases} \rho_{vk}(t) = \rho_k C_{mk} P_{vk}(t) \\ \rho_{vk\oplus}(t) = \rho_k C_{mk} P_{vk\oplus}(t) \end{cases} \quad (14)$$

2) *Heat balance*: The transfer of heat in the porous environment occurs under three forms : a flow of conduction under a gradient of temperature expressed by the law of Fourier ($\frac{q_{cond}}{S_k} = -\lambda \text{grad}(T)$), an advection of sensitive heat through the flows of liquid and steam (classically neglected [11]) and a transfer of latent heat of change of phase driven by the steam ($\frac{q_v}{S_k} = -h_v \delta_{pk} \text{grad}(P_v)$). The balance equation of the energy conservation is (See table I):

$$\begin{aligned} \rho_k C_{pk} V_k \dot{T}_k(t) = & -2S_k \times \sum_{\oplus=\ominus, \oplus} \frac{\lambda_k}{e_k} (T_k(t) - T_{k\oplus}(t)) \\ & - \frac{2S_k \sigma h_v}{\rho_k C_{mk}} \times \sum_{\oplus=\ominus, \oplus} \frac{\delta_{pk}}{e_k} (\rho_{vk}(t) - \rho_{vk\oplus}(t)) \end{aligned} \quad (15)$$

Remark 1: For a wall (Σ_k), if the Biot numbers defined for the transfers of heat and transfer of mass respectively by:

$$Bi = \frac{h_c e_k}{\lambda_k}; \quad Bi_m = \frac{h_m e_k}{\delta_{pk}} \quad (16)$$

are small in front of 1 ($Bi < 0.1$ and $Bi_m < 0.1$ by considering the coefficients h_c and h_m for both surfaces ($\Sigma_{k\oplus}$) and ($\Sigma_{k\ominus}$)) then the resistance to conduction within (Σ_k) is much lower than the resistance to convection across the air boundary layer. In this case, the assumption of uniform temperature/partial vapor distribution in (Σ_k) is reasonable and Equations (15) and (13) become

$$\begin{cases} \rho_{vk}(t) = \rho_{vk\oplus}(t) = \rho_{vk\ominus}(t) \\ T_k(t) = T_{k\oplus}(t) = T_{k\ominus}(t) \end{cases}$$

C. Mass and heat exchanges within a zone

1) *linked to the air movements*: We consider air exchange and denote in m^3/s the effective flow rate of outdoor air, including infiltration and mechanical ventilation. Assuming that the air is renewed from outside at a temperature $T_{\oplus}(t)$ due to a constant flow Q_a , a mass algebraic flow $g_{2\oplus}(t)$ / heat $q_{2\oplus}(t)$ exists and is defined by:

$$\begin{cases} g_{2\oplus}(t) = -Q_a (\rho_{v\oplus}(t) - \rho_{v\oplus}(t)) \\ q_{2\oplus}(t) = -1.15 \times Q_a C_{pda} (T_{\oplus}(t) - T_{\oplus}(t)) \end{cases} \quad (17)$$

Let us assume that a CMV is used to circulate the air between rooms $\oplus$ and $\ominus$. It is assumed that the air taken in the 'source' room is replaced by the outside air through the openings at the level of the windows. On the other hand, air received by the 'destination' room replaces a part of its air that is equally displaced through the openings at the level of the windows. Two air masses of different psychrometric characteristics are thus mixed. Let $(\rho_{v\oplus}, T_{\oplus})(t_{on})$ and $(\rho_{v\ominus}, T_{\ominus})(t_{on})$ be these characteristics within each zone $\oplus$ and $\ominus$ at the moment t_{on} the CMV is switched on. If we take the example that the air goes from zone $\oplus$ to zone $\ominus$, the humidity / heat balance is:

$$\begin{cases} g_{3\oplus}(t) = -Q_{cmv} (\rho_{v\oplus}(t) - \rho_{v\oplus}(t)) \\ q_{3\oplus}(t) = -1.15 \times Q_{cmv} C_{pda} (T_{\oplus}(t) - T_{\oplus}(t)) \end{cases} \quad (18)$$

$$\begin{cases} g_{3\oplus}(t) = -Q_{cmv} (\rho_{v\oplus}(t) - \rho_{v\ominus}(t)) \\ q_{3\oplus}(t) = -1.15 \times Q_{cmv} C_{pda} (T_{\oplus}(t) - T_{\ominus}(t)) \end{cases} \quad (19)$$

with Q_{cmv} the air flow of the CMV in m^3/s .

2) *linked to heating/cooling systems and air conditioning:* Air treatment systems in buildings do not just cool/heat but also treat, filter and humidify indoor air. In this paper, we assume that the temperature T_h, T_c (in K) and the vapor density $\rho_{v,h}, \rho_{v,c}$ of the forced-air (in kg/m^3) in heating / cooling mode are known. The focus is on reversible heat pumps (HP). A heat pump is a thermodynamic device that allows capturing heat from outdoor air and transmitting it inside the building. If it is reversible, the HP is capable of providing air conditioning in the summer. Essentially, and without taking into account the transitional aspects, the equation translating this energy flow in zone $\oplus$ is in heating / cooling mode:

$$\begin{cases} g_{4\oplus}(t) = -Q_{HP} (\rho_{v\oplus}(t) - \rho_{v,h/c}(t)) \\ q_{4\oplus}(t) = -1.15 \times Q_{HP} C_{pda} (T_{\oplus}(t) - T_{h/c}(t)) \end{cases} \quad (20)$$

with Q_{HP} the air flow of the HP in m^3/s .

3) *linked to occupancy:* Table II specifies the energy and the water vapor released by one person depending on his activity and the temperature of zone $\oplus$.

TABLE II
HEAT/MOISTURE FLUX

	Rest to light activity		Light to sustained activity		Heavy physical activity	
$T_{\oplus}$ (C)	$g_{5\oplus}$ (g/h)	$q_{5\oplus}$ (W)	$g_{5\oplus}$ (g/h)	$q_{5\oplus}$ (W)	$g_{5\oplus}$ (g/h)	$q_{5\oplus}$ (W)
18	35	125	95	190	165	270
21	35	120	110	190	185	270
22	40	120	125	190	215	270
23	50	120	135	190	225	270
24	60	115	140	190	230	270
25	60	115	145	190	240	270
26	65	115	150	190	250	270

4) *linked to the absorbed shortwave solar radiations:* The sun provides heat that greatly influences the temperature within the platform rooms. To estimate the solar energy transmitted, it is necessary to follow the positioning of the sun and to take into account direct solar and diffuse radiation. In order to understand the calculations to quantify the global radiation, it is important to define not only the various solar components, but also the different angles used for calculations [12]. These angles are the local latitude (Φ), the hour angle (ω) in the local solar time, the solar declination (δ_s) and the solar azimuth (ϕ_s). The energy related to direct solar radiation is such that:

$$q_{6\oplus}(t) = 1260 S_{wk} \exp\left(\frac{-1}{2.3 (\alpha_s(t) + 3) \sin(\alpha_s(t))}\right) \times \cos(\beta(t)) \quad (21)$$

where S_{wk} is the surface of the windows of (Σ_k), $\alpha_s(t)$ is the solar elevation angle and $\beta(t)$ is the angle of incidence

Fig. 1. The “Eco-Safe” platform.

between the sun's rays and the face of windows. As for the diffused radiation energy, it is calculated by the relationship:

$$q_{7\oplus}(t) = S_{wk} (62.5 \times (1 + \cos(i)) \sin(\alpha_s(t)^{0.4}) + 108 \times (1 - \cos(i)) \sin(\alpha_s(t)^{1.22})) \quad (22)$$

IV. HYGTROHERMAL MODELING BY NODAL METHOD – APPLICATION TO THE “ECO-SAFE” PLATFORM

By this type of approach, each node represents volumes of material or components, while inter-node connections represent the flow paths of thermal and mass fluxes (walls, doors, windows, etc.) [1], [13]. The graphic representation of the model has the same appearance as an electrical diagram, hence the analogue method name where:

- the temperature and density of water vapour are represented by electrical potentials;
- energy flows and mass transfers are represented by current intensities;
- thermal and moisture resistances are represented by electrical resistances;
- the thermal and moisture retention capacities are represented by capacitors.

For example illustration, the “Eco-Safe” platform shown in Figure 1 is considered. The electrical circuits translating the energy and mass models within this platform are represented at the top of Figure 2. For each zone, a global model is defined by assembling the different hygrothermal blocks. It consists of four walls, a double glazed window, a floor and a ceiling. The model of this part is illustrated for room $\oplus$ at the bottom right corner of Figure 2. The consideration of heterogeneous walls (e.g. window and wall) result in as many circuits RC paralleled. An example of a wall model considering a window with double glazing is represented at the bottom left corner of Figure 2. For the sake of simplicity and due to the low impact of term $h_v g_{1\oplus}(t)$ in (12) during winter conditions, the latter has been neglected.

Fig. 2. Hygrothermal modeling of "Eco-Safe" platform.

Parameters R and C of the simulator were determined from the nature and characteristics of the different materials used in the wall dividers, walls and windows of the platform. They have been initialized taking into account theoretical values of certain parameters specified in table III and according to the following analogies:

- for zone $\textcircled{i}$:

$$\begin{cases} R_{T,k\textcircled{i}} = \frac{1}{h_c S_{k\textcircled{i}}} \\ R_{H,k\textcircled{i}} = \frac{1}{2500 h_m S_{k\textcircled{i}}} \end{cases}; \begin{cases} C_{T,k\textcircled{i}} = 1.15 C_{pda} V_{\textcircled{i}} \\ C_{H,k\textcircled{i}} = V_{\textcircled{i}} \end{cases} \quad (23)$$

- for wall / windows / ceil / floor k :

$$\begin{cases} R_{T,k} = \frac{e_k}{2\lambda_k} \\ R_{H,k} = \frac{e_k}{2\delta_k} \end{cases}; \begin{cases} C_{T,k} = \rho_k C_{pk} e_k \\ C_{H,k} = \rho_k C_{mk} e_k \end{cases} \quad (24)$$

They have been adjusted from the temperature curves actually observed on the system. According to figures 3 and 4, it appears that the profiles of actual and estimated temperature values are found to be similar. The simulated curves were generated from the entries $T_{\textcircled{i}}(t)$ and $\rho_{v\textcircled{i}}(t)$ (Figure 5), which are the data issued from the weather station over the period 3/5/2018 - 3/9/2018.

TABLE III
THEORETICAL VALUES OF HYGTHERMAL PARAMETERS [14]

	ρ_k	$\delta_{pk}(e^{-12})$	C_{pk}	λ_k
Concrete	2400	1.4	1000	2
Air	1.23	185	1006	0.025
Polystyrene	40	3.1	1450	0.04
Glass	2580	0	720	1.2

Fig. 3. Measured (solid line) and estimated (dashed line) temperatures for zoom ①. An estimation of the operative temperature is represented in dotted line.

Fig. 4. Measured (solid line) and estimated (dashed line) humidity rates in zone ①.

V. CONCLUSION

In this paper, a complete methodology has been established for modeling the hygrothermal behavior of a multi-zone office building. The application target is the “Eco-Safe” platform (CRAN Nancy, France). This study takes into account not only all the walls/windows and equipments (CMV, heat pump) fitted to each of the zones but also the effects of solar radiations and occupants. In a practical context, in order to facilitate the determination of a hygrothermal model of a multi-zone

Fig. 5. Outside temperature (solid line) and humidity rate (dashed line) of outdoor air (3/5/2018 - 3/9/2018).

building, the proposed nodal method is based on the use of the thermal electric analogy. In perspectives, it would be interesting to implement an efficient control law based on this model.

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