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THE USE OF A ZONAL MODEL TO CALCULATE THE STRATIFICATION IN A LARGE BUILDING

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ABSTRACT

In the past, many churches were raised and in a church building no heating no heating system was installed, except a simple individual coal or peat stove, which could be rented by the churchgoers. The thick high stone walls of the church alleviated the fluctuations of the ambient air temperature and relative humidity. Accordingly, the indoor climate in the church building was quite stable. After the Second World War the living standard of the people increased and the increased prosperity also led to higher comfort demands in churches. As a consequence, many “local” heating systems were often replaced by rugged air inlets, designed to quickly heat the space to increase the thermal comfort. These were designed and operated without taking the effect of the fluctuating temperature and relative humidity on the artwork in the church into account. Consequently, the many artworks in the church building like the organ, the pulpit and the panel paintings are also exposed to this changing climate, leading to faster deterioration or even to damage. Evaluating the conservation conditions for the artworks in a large space requires knowledge of the stratification in temperature and humidity inside a large space that occurs during heating. The use of Computational Fluid Dynamics (CFD) is probably the most suitable method to predict the airflow pattern, but it is quite time consuming and requires a powerful computer. As an alternative a zonal model is a suitable method to predict the airflow in a large space in a simplified way. These models can be linked to a BES- software in which each zone is assumed to be perfectly mixed. By this coupling, the influence of the airflow on the temperature distribution and vice versa can be calculated, in order to judge the thermal comfort and preservation conditions in one zone. This paper presents the coupling of the existing thermal zonal model of Togari with the BES-software TRNSYS. In addition a moisture preservation equation was added to the thermal zonal model to predict the vertical relative humidity gradient in a large space. Further the model also was extended with a EMPD-model to include the moisture buffering of the walls in a simplified way.

KEYWORDS

Thermal stratification, zonal airflow model, dynamic simulation

INTRODUCTION

In the past , church buildings were usually unheated, except for a simple individual coal or peat stove, which could be rented by the churchgoers. The thick high stone walls of the church alleviated the fluctuations of the ambient air temperature and relative humidity. As a result, the indoor climate in the church building was quite stable (Schellen, 2002). After the Second World War the living standard of the people increased and the increased prosperity lead to higher comfort expectations (Camuffo et al., 2010), not only in residential dwellings, but also in churches. As a consequence many "local" heating systems were replaced by heating systems that heated the whole indoor air volume of the church. Often these systems were rugged air inlets, designed to quickly heat the space and achieve thermal comfort during services. These systems were designed and operated without taking the effect of the fluctuating temperature and relative humidity they caused on the artworks in the building into account. Consequently, the many wood based artworks usually present in the church, like the pulpit, the organ and the panel paintings are also exposed to this changing climate. According to the preservation needs determined by the ASHRAE conservation classes (Anon, 2011), changes in temperature and relative humidity or space gradients could lead to faster deterioration or even to damage at these artworks.

Table 1: ASHRAE conservation classes (Anon, 2011) related to the risk on damage at wooden artworks

Class		**Temperature**		**Relative Humidity**		**Risks**
		ΔT_{short} Space gradients	$\Delta T_{seasonal}$	ΔRH_{short} Space gradients	$\Delta RH_{seasonal}$	
AA		±2°C	±5°C	±5%	No changes	No risk of mech. damage
A	As	±2°C	+5°C -10°C	±5%	±10%	Small risk of mech. damage
	A	±2°C	+5°C -10°C	±10%	No changes	
B		±5°C	+10°C max. 30°C	±10%	±10%	Moderate risk to high-vulnerability artefacts but tiny risk to most paintings
C		<25°C (<30°C)			25-75%	High risk of mech. damage
D		-		<75%		Prevent dampness

To reconcile the heating demand due to comfort expectations of the visitors with the preservation needs for the artworks in designing a heating system for such churches including artwork, computer simulations can be helpful. During heating, thermal stratification occurs and in order to make a correct assessment of the stratification, in addition to the calculation of the energy exchange, the calculation of the airflow in the space is necessary. To predict the temperature and humidity distribution, different modelling approaches can be used: namely Computational Fluid Dynamics (CFD) models and the linking of a Building Energy Simulation (BES) software with an airflow model (the so-called zonal models).

The CFD-method is a widespread approach to model the airflow with a computational simulation. Models, based on this method, predict the temperature, velocity and other flow parameters by numerically solving the Navier-Stokes equations with a high degree of accuracy. However, for this research, the fluctuations in temperature and humidity over a longer time period and the effects of heating on the temperature and humidity distribution in large historical buildings, related to damage occurring at wooden artifacts are of interest. To simulate longer time periods, the mentioned CFD models are less suitable because of their need for powerful computers with a large amount of memory. In reality, these resources are often not available. To be able to predict the airflow in a building in a fast way and for a longer time period, the use of a macroscopic airflow model offers a solution. These so-called zonal models are an intermediate approach between the CFD and the multizone models used in BES, which consider the air as perfectly mixed. The difference with a CFD model is that the grid is much coarser and the differential equations between the cells are simplified. Based on the simplifications made, the zonal models can be further subdivided into pressure-zonal, temperature-zonal and momentum-zonal models.

As pointed out above, a BES-software considers a zone perfectly mixed. However, a BES software is commonly used for the evaluation or comparison of different heating techniques to meet sizing and energy operating cost requirements (Bouia & Dalicieux, 1991). Though, a BES software, because of this assumption of a well-mixed zone is unable to calculate the temperature stratification in the air. So to determine the thermal comfort and the preservation conditions the BES model has to be extended with an airflow model. Because the emphasis lies on predicting the airflow in a building in a fast way, we chose to implement a temperature-zonal model in the BES-software. For this purpose the thermal-zonal model of Togari(Togari, Arai, & Milura, 1993) was recoded in C++ and coupled to the commercial BES simulation environment TRNSYS (v17). In this paper the possibilities of the model and its shortcomings are discussed by applying it on the case of a typical church building.

THE COUPLED THERMAL ZONAL MODEL

The model proposed by Togari et al.(Togari et al., 1993) is a simplified model for calculating the vertical temperature distribution in a single large space. They assumed that the main components of the air movement in the large space are airflows along the vertical wall surfaces (so-called wall currents) and supply airstreams. Further, they also assumed that the horizontal temperature is uniform, except for the regions affected by supply air jet ventilation. Based on these assumptions a method was proposed to calculate the stratification.

First the space is divided into a finite number of horizontal layers or blocks. Each layer consist of a core cell and walls cells. The core cell represents a layer and when the layer is bounded by a wall, a wall cell is defined which accounts for the mass flows in the boundary layer. The method considers three major paths for the heat and mass transfer: (1) heat and mass transfer between the core cell and the wall cells, (2) heat transfer and air movement between the different core cells and (3) heat transfer through the wall. To calculate these transfers some decision were made:

- To predict the air and heat transport to the wall elements, the aid of the boundary layer theory is applied.
- Since the heat transfer through the walls, which is affected by both outdoor and indoor conditions, is rather complicated, a heat balance equation for coupled thermal conduction, convection and radiation is necessary to predict the wall surface temperature. For this the implementation of the model in a BES-software was used.

The Wall Currents

When the air in a space is warmed up or cooled down , the air is cooled at the colder walls or warmed up at the hotter walls. By consequence an airflow along the vertical walls is induced. The heat and air mass transfer along the wall is modelled using a so-called wall current model. This model assumes that the heat convection drives mass flow $m_{out(I,K)}$ with an average temperature $T_{D(i,K)}$ from layer i to its related boundary layer.

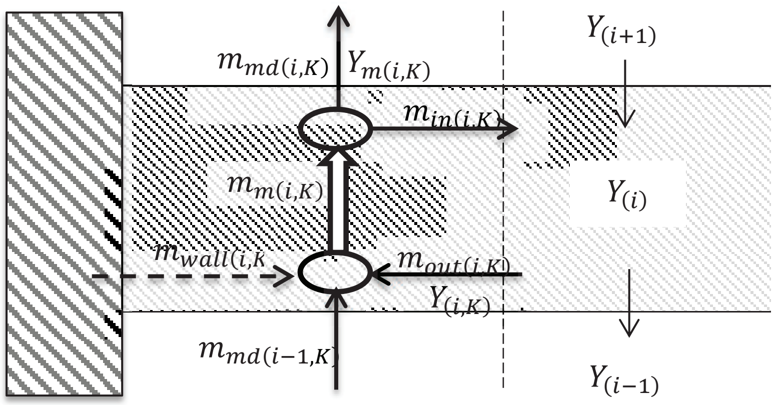

Figure 1: *Schematic representation of the composition of the wall currents and the mass flow in and between the wall cell and the core cells*

To calculate $m_{out(I,K)}$ with temperature $T_{D(i,K)}$ following equations were applied:

$$T_{D(i,K)} = 0.75T_{(i)} + 0.25T_{w(i,K)} \tag{17}$$

$$m_{out(I,K)} = 4\,\frac{\alpha_{C(i,K)}.A_{w(I,K)}}{C} \tag{18}$$

The current flow from layer i will combine with the current flow $m_{md(i+1,K)}$ from layer i+1 to form a total flow with mass $m_{m(i,K)}$ with average temperature $T_{m(i,K)}$, yielding:

$$m_{m(i,K)} = m_{out(i,K)} + m_{md(i+1,K)} \tag{19}$$

$$T_{m(i,K)} = \frac{m_{out(i,K)}T_{D(i,K)} + m_{md(i+1,K)}T_{m(i+1,K)}}{m_{m(i,K)}} \tag{20}$$

Some of the air of this current $m_{m(i,K)}$ returns to the air layer i ($m_{in(i,K)}$) and some continues to the cell down or up ($m_{md(i,K)}$), depending on the direction of the current flow. The splitting of the mass $m_{m(i,K)}$ into $m_{in(i,K)}$ and $m_{md(i,K)}$ is calculated by the ratio P(i,K):

$$m_{in(i,K)} = (1 - P_{i,K})m_{m(i,K)} \tag{21}$$

$$m_{md(i,K)} = P_{i,K}m_{m(i,K)} \tag{22}$$

The ratio P(i,K): is dependent on the relationship between the air temperature in zone i and in zone i+1 or zone i-1, associated with the current temperature $T_{m(i,K)}$. Table 1 presents the judgment criteria for the airflow pattern.

Table 2: judgment criteria for the airflow pattern

Flow direction	Temperature conditions	$P_{i,K}$
Descending	$T_{m(i,K)} \geq T_{(i)}$	0
	$T_{(i)} > T_{m(i,K)} > T_{(i-1)}$	$\frac{T_{(i)} - T_{m(i,K)}}{T_{(i)} - T_{(i-1)}}$
	$T_{m(i,K)} \leq T_{(i-1)}$	1
Ascending	$T_{m(i,K)} \leq T_{(i)}$	0
	$T_{(i)} < T_{m(i,K)} < T_{(i+1)}$	$\frac{T_{m(i,K)} - T_{(i)}}{T_{(i+1)} - T_{(i)}}$
	$T_{m(i,K)} \geq T_{(i+1)}$	1

2.2 Conservation equations

The calculation of the mass balance in every zone, was started from the lowermost zone. Flows are defined as positive in the upward direction and from the wall cells to the layers. The mass balance for a layer i is then calculated by:

$$0 = m_{source} + \sum_{k=1}^{m} (m_{in,i,K} - m_{out,i,K}) + m_{i-1} - m_i \tag{23}$$

Where m_i is the mass flow from layer i to i+1 [kg/s] and m_{source} is the mass flow coming from a source [kg/s].

In the air the heat transfer equation can be written as:

$$V_i \frac{(\rho C)^{t+\Delta t,m} T_i^{t+\Delta t,m} - (\rho C)^t T_i^{t}}{\Delta t} = Q_{source} + Q_{layer} + Q_{currents} + Q_b \tag{24}$$

Where Q_{source} is the heat coming from a source, Q_{layer} means the heat transfer between the layers and $Q_{currents}$represents the heat flow from or to the wall currents. $Q_b$expresses the heat transfer between the layers due to thermal stability yielding:

$$q_{b(i)} = C_{b,i} A_b (T_{i-1} - T_i) \tag{25}$$

Where $A_b$is the boundary area between zone i-1 and zone i. $C_{b(i)}$ is the heat transfer factor and is defined as "the value obtained when a stable (Cb = 2,3W/m²K) or unstable (Cb = 112W/m²K) temperature stratification is formed.

The model of Togari was extended by an equation for moisture conservation. Hereby the air is assumed as a mixture of dry air and water vapour, each component (vapour and dry air) obeying the ideal gas equation. Taking into account the time dependency of the moisture content of the air, the moisture balance equation for a layer i can be expressed as:

$$\rho_a V_i \frac{(Y_i^{t+\Delta t,m} - Y_i^{t})}{\Delta t} = G_{layer} + G_{currents} + G_{source} \tag{26}$$

G_{source} represents the vapour flow produced by people, systems activities such as washinh,…and G_{layer} is the vapour mass flow between the layers. The vapour mass flow $G_{currents}$ from the wall current is calculated by:

$$G_{currents} = \sum_{k=1}^{m} m_{in(i,K)} Y_{m(i,K)} - \sum_{k=1}^{m} m_{out(i,K)} Y_i^{t+\Delta t,m} \tag{27}$$

Where $Y_{m(i,K)}$ is the humidity ratio of wall current and is determined by the moisture flux $m_{wall(I,K)}$coming or going to the wall, the humidity ratio of the mass $m_{out,i,K}$coming from the adjacent layer and the humidity ratio from the wall current under or above (see Figure 197). The equation to determine the humidity ratio of the wall current $Y_{m(i,K)}$ is:

$$Y_{m(i,K)} = \frac{{Y_{i(i,K)}}^{p-1} m_{out(i,K)} + {Y_{m(i-1,K)}}^{p-1} m_{md(i-1,K)} + m_{wall(i,K)}}{m_{m(i,K)}} \tag{28}$$

The amount of water vapour exchange between room air and the wall (i,K) was modelled with a simplified algorithm. The chosen model was an effective moisture penetration depth (EMPD) model (Janssens, Rode, De Paepe, Woloszyn, & Sasic-Kalagasidis, 2008). This approach assumes that the moisture transfer takes place between the zone air and a thin fictitious layer of a uniform moisture content with a thickness $d_{buf,}$ which is related to the variation of water vapour pressure at the material surface. The effective penetration depth d_{buf} is calculated by equation 13, in which t_p is the period of cyclic variation (s).

$$d_{buf} = \sqrt{\frac{\delta(\varphi) P_{v,sat}(T) t_p}{\rho \xi(\varphi) \pi}} \tag{29}$$

Where $\delta(\varphi)$ is vapour permeability [s], $\rho\xi(\varphi)$ is the moisture capacity in terms of humidity derived from the material sorption isotherm [kg/m³] and $P_{v,sat}(T)$ represents the saturation water vapour pressure at temperature T [Pa]. The following equation is then solved together with the moisture balance equation for indoor air within a space under the non-steady-state.

$$m_{wall} = \frac{A(P_{v,i} - P_{v,buf})}{\frac{1}{\beta_i} + Z_{buf}} = A \rho \xi(\varphi) d_{buf} \frac{\left(\frac{P_{v,buf}}{P_{v,sat}(T_{buf})}\right)^{t+\Delta t} - \left(\frac{P_{v,buf}}{P_{v,sat}(T_{buf})}\right)^{t}}{\Delta t} \tag{30}$$

Where $P_{v,i}$ represents the water vapour pressure indoor and $P_{v,buf}$ is the average water vapour pressure in the layer[Pa]. β_i is the surface convection coefficient of the wall [m/s] and Z_{buf} is the vapour diffusion resistance between the surface and the moisture storage centre of the layer.

2.3 Adding infiltration

Infiltration is due to wind-driven or buoyancy-driven ventilation.With buoyancy-driven ventilation the pressure differences are due to air density differences, which in turn are due to temperature differences. Following equation was used to calculate the infiltration rate:

$$u_{inf} = \frac{u_{50}}{2} \left(\frac{\Delta \bar{p}}{50}\right)^{n} \tag{31}$$

Where u_{50} is the infiltration defined in air changes per hour; i.e. zone volume per hour, when the pressure difference between inside and outside is 50 Pa. 'n' is equal to 2/3 and $\Delta\bar{p}$ is the pressure difference[Pa] related to the height H [m] of the zone and the temperature difference [°C] which is calculated by:

$$\Delta \bar{p} = 0{,}01 H (T_{indoor} - T_{outdoor}) \tag{32}$$

2.4 Adding a jet flow model

To model a free jet stream, the following assumptions were made:

- **First the trajectory of the path was determined.**

For horizontally compact free jets, the trajectory of the centreline is described by Koestel (Awbi, 2003), yielding:

$$\frac{y}{\sqrt{A_0}} = 0{,}0522 \left(\frac{x}{\sqrt{A_0}}\right)^{3} Ar \tag{33}$$

The centreline velocity decay of compact jets can be described by the following equation.

$$\frac{U_{centerline}}{U_0} = \frac{K_1\sqrt{A_0}}{x} \tag{34}$$

Theoretical values of the characteristic K1 depend upon the type of velocity profile equation and supply conditions assumed (Goodfellow & Tahti, 2001). For a 3D-jet the value of K_1 equals 7,2 (Awbi, 2003). The temperature decay in the centreline of the jet was calculated with the following equation.

$$\frac{T_x - T_i}{T_0 - T_i} = \frac{K_2\sqrt{A_0}}{x} \tag{35}$$

In this equation the value for K2 was calculated by the formula of Shepelev (sec (Goodfellow & Tahti, 2001)):

$$K_2 = \frac{1}{K_1}\frac{(1 + Pr)}{2\pi 0{,}082^2} \tag{36}$$

- **To evaluate the total airflow rate transported by the jet to some distance from the diffuser, the entrainment rate was determined.**

For a given opening area A_0, the entrainment ratio is proportional to the distance x. For a compact jet, the following equation by Grimithlyn was used (sec (Goodfellow & Tahti, 2001)):

$$\frac{\dot{Q}_x}{\dot{Q}_0} = 0{,}29\frac{x}{\sqrt{A_0}} \tag{37}$$

3 VALIDATION OF THE MODEL

The validation case studied in this paper is the case that can be found in the report of Togari et al. (Togari et al., 1993) and that of Arai et al.(Arai, Togari, & Miura, 1994). The geometrically simple test room has a ground plane of 3m x 3m and measures 2,5m in height. The room consist of insulated boards (three vertical walls, ceiling and floor) and 1 glass wall. In the wall opposite to the glass wall, two openings were made in the symmetry plane: a supply inlet at 0,625m above the floor and a return outlet at 0,250m above the floor. The simulated cases are visualized in Table 3.

Table 3: Overview of the simulated case studies

Name Case	Supply air condition					Inlet location	Outside temperature
	outlet size [mm x mm]	air volume [m³/h]	velocity [m/s]	temperature [°C]	momentum [kg m/s²]		
N-11	500 x 74						42 (14)
N-10	500 x 74						12 (42)
H-100	500 x 74	150	1,13	40,7	0,053	zone 1	11

The room described above was fully modelled in TRNSYS. The space was vertically divided into 5 layers. In the model described by Togari the calculations of the stratification started based on measured wall surface temperatures, while in the model in TRNYS only outdoor conditions can be used. As a consequence the behaviour of the walls and the inside surface temperature is calculated by TRNSYS. For the case of natural convection (case N-11 and case N-10) good agreement was found (De Backer, Laverge, Janssens, & De Paepe, 2014). On Figure 198 results are shown for the case H-100 where the measured results by Togari were compared with the calculated results by the thermal-zonal model. The measured results were predicted well enough by the jet-model.

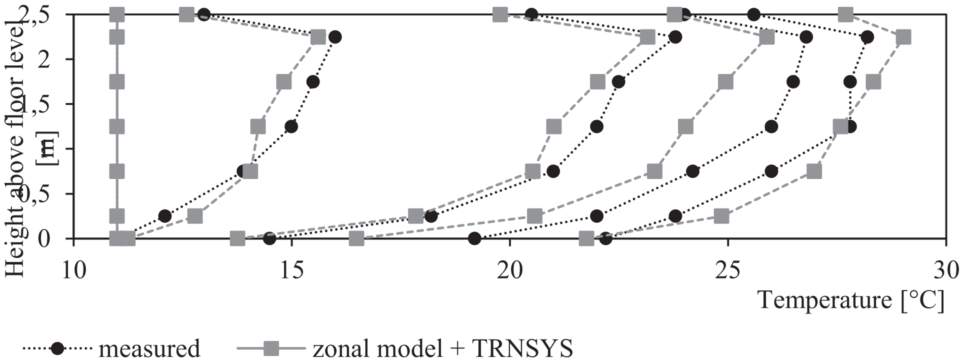

Figure 2: *The progress of the temperature in case of hot air supply condition (jet)*

4 A CASE STUDY

The goal of this study was to look how a church building could be modelled with the thermal zonal model. As already pointed out, the thermal zonal model had some limitations. One of the limitations is that it can only deal with simple geometries and the geometry can only be subdivided into horizontal layers. For that, the geometry of the church building had to be simplified.

For this study, a church was modelled with a floor plan and dimensions typical for a church in Belgium. The geometry of the church includes a rectangular nave of 32m in length, 17m in width and 15m in height (internal dimensions). The floor plan also containes side aisles. Each side aisle measures 5x5m. The church has 0,9m thick masonry walls . The windows are single-glazed. Based on literature (Fawcett, 2001), the following structure was assumed for the build-up of the church floor: a plain square paving stone, laid on a bed of lime mortar. The floor was modelled using the standard ISO 13370:2007. Figure 199 shows the model used in the simulation study, where the church building was divided into 5 layers. Initially, the attics of the church were not included in the model.

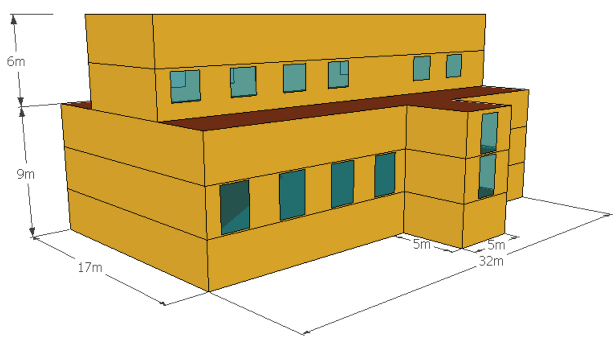

Figure 3: *Simplified geometrical model of a church building*

In the simulation study, 24 hours were simulated during which heating only occurred once, during service. The service started at 10h00 and ended at 11h00, so the heating device was operated from 9h30 till 11h30 with a set point temperature of 16°C. The temperature at the supply inlet was 45°C and the flow rate was 30000kg/h. The initial conditions in the church building were 8°C and 80%RH. Figure 200 and Figure 201 show results for the temperature and relative humidity profile in the church building. The temperature is the lowest zone rises to 16°C and achieves the set comfort temperature. The relative humidity decreased to 50%. In contrast, the temperature in the highest zone rose to 21°C, while relative humidity dropped down to 35%. For an artwork that encompasses several subsequent layers, like an organ, large relative humidity gradients occur during heating. Furthermore, there also can be concluded that the ASHREA class B , the highest class that can be reached in a church building (Ankersmit, december 2009), cannot be achieved if the church is only heated during a church service and one wants to achieve comfort temperature.

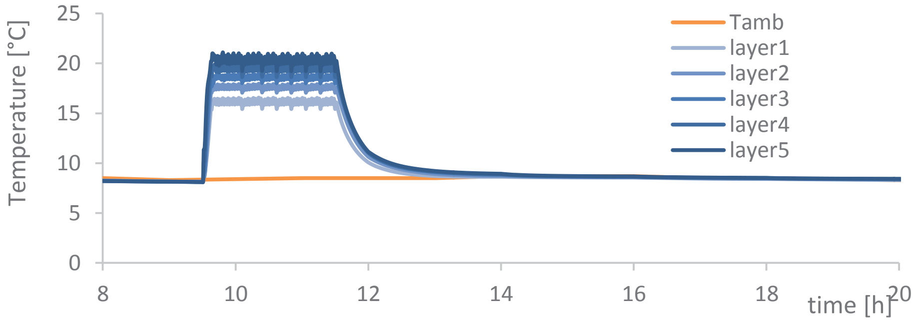

Figure 4: *Temperature distribution in the church building*

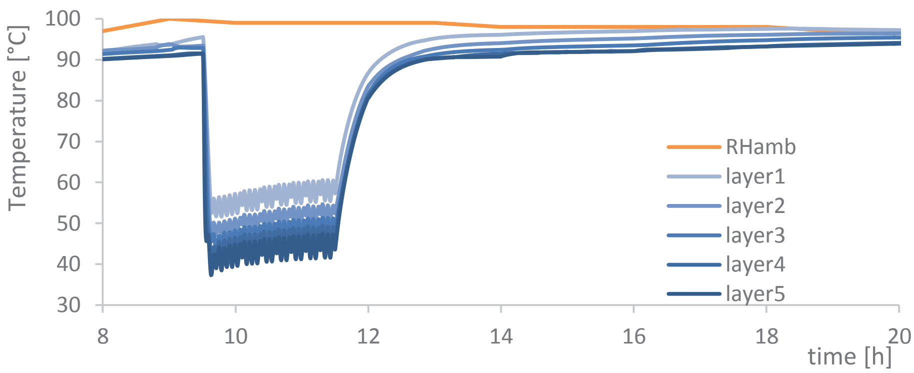

Figure 5: *Relative humidity distribution in the church building*

CONCLUSIONS

As an alternative to the complex CFD models, zonal models are a suitable method to predict the airflow in a large space in a simplified way. These zonal models can be linked to a BES-software, in which each zone is assumed to be perfectly mixed. By this coupling, the influence of the airflow on the temperature distribution and vice versa can be calculated, in order to assess the thermal comfort and preservation conditions in the zone. This paper presents the coupling of the existing thermal zonal model of Togari with the BES-software TRNSYS. In addition a moisture preservation equation was added to the thermal zonal model to predict the vertical relative humidity gradient in a large space. The model was additionally extended with a EMPD-model to include the moisture buffering of the walls in a simplified way. To validate the model, the cases described in the paper of Togari were simulated. Good agreement was found between the measurements and the thermal-zonal model. A case study of a typical church building was modelled. The results showed how stratification in the church building occurs during heating with a typical rugged air inlet, demonstrating that the ASHREA conservation class B, cannot be achieved when heating solely during church services.

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