Implementation of Models for Building Envelope Air Flow Fields in a Whole Building Hygrothermal Simulation Tool

C. Rode
Technical University of Denmark, Department of Civil Engineering, Denmark

K. Grau
Danish Building Research Institute, Aalborg University, Denmark

ABSTRACT: Simulation tools are becoming available which predict the heat and moisture conditions in the indoor environment as well as in the envelope of buildings, and thus it has become possible to consider the important interaction between the different components of buildings and the different physical phenomena that occur. However, there is still room for further development of such tools. This paper will present an attempt to integrate modelling of air flows in building envelopes into a whole building hygrothermal simulation tool. Two kinds of air flows have been considered: (1) Air flow in a ventilated cavity such as behind the exterior cladding of a building envelope, i.e. a flow which is parallel to the construction plane. (2) Infiltration/exfiltration of air through the building envelope, i.e. a flow which is perpendicular to the construction plane. The paper presents the models and how they have been integrated into a whole building simulation tool. Furthermore, the paper shows some calculation results for validation and for illustration of the importance of the phenomena.

1 INTRODUCTION

The ventilated exterior cladding is a rather common building component at the exterior side of building envelopes. It is typically used in wall systems, at least in Northern Europe, but also elsewhere in the world. The ventilated cavity behind the cladding provides good safety against ingress of precipitation that meets the exterior face of the wall. Because the cavity has openings that make it possible to have an immediate exchange of air between the cavity and the air outside, then there will be an almost immediate pressure equalization of wind gusts that meet the façade. Thus there is only little potential to push water through cracks and openings in the exterior cladding. Besides, the cavity itself constitutes a drainage capability for precipitation water that *may* have entered into the cavity, as provision can be made so it can run down and out.

The outdoor ventilated attic, or ventilation under a roofing material, may also be modelled in a similar way as the ventilated exterior cladding of a wall. But naturally for these constructions, the cavity will have a much lower slope, and the cavity may have considerable extension ("thickness") perpendicular to the flow direction.

The flow in ventilated cavities is predominantly governed by wind forces and stack effect.

In all situations, the ventilated claddings have an influence on the thermal, moisture and air flow conditions in the building envelope. Despite of this, it is only rare, if ever, that ventilated cavities are considered in whole building heat, air and moisture simulation models. Our intent with the work presented in this paper is to demonstrate a possible means how to integrate a simplified cavity air flow model in the whole building hygrothermal simulation model, *BSim* (Rode & Grau, 2003 and 2004). The heat and moisture flows in this model, like in most energy simulation tools, are modelled one-dimensionally with flows that pass through the building elements perpendicular to their surface. However, when ventilated cavities are introduced, the flow in these parts follows a path which is parallel to the construction, and thus, the whole situation should be modelled two- (or even three-) dimensionally. We will introduce an approximate model which allows to build upon the overall one-dimensional nature of the heat and moisture flow calculations in the tool.

Another development regards a model of air flow passing through the building envelope perpendicular to its plane, i.e. by exfiltration or infiltration. It may have a quite notable influence on the heat flow through the wall. In addition, the impact on the moisture conditions can, even for small air flows, completely change the situation, e.g. from a healthy construction to a disaster. Air flows from warm/humid environment towards colder climates should generally be avoided, whereas air flows in the opposite direction could have a drying effect.

When infiltrating air flows can be seen to provide some of the required fresh air supply to the indoor

environment, then the so-called dynamic wall principles apply, and the wall itself works as a heat exchanger where some of the heat lost by conduction is recovered by the ingoing air.

For these reasons, it has been decided to implement a model for advective heat and moisture flow in a whole building simulation tool. Again, *BSim* from the Danish Building Research Institute has been chosen as one such tool. A multi-zone model for air flow between zones (rooms) within buildings has been implemented into *BSim* as described previously by Grau & Rode, 2005. For more information about BSim, see www.bsim.dk.

2 VENTILATED AIR CAVITY MODEL

The ventilated cavity is set up as shown in Figure 1 as one of the layers in the sequence of a one-dimensional multilayered structure.

The ventilated cavity is modelled as another hygroscopic material in the sequence of layers that form the multilayered structure. However, the material is chosen such that is has some, but only rather limited hygroscopic capacity and diffusion resistance. However, this material is just used as a dummy layer to hold the properties of the air that is passing through the cavity.

The air flow is driven by an air pressure difference between inlet and outlet, which is caused by (1) wind induced air pressure differences between the loci for inlet and outlet to the cavity, and (2) stack effect:

$$\Delta P_{wind} = \Delta c \cdot \tfrac{1}{2} \cdot \rho \cdot v^2 \tag{1}$$

where Δc = difference in pressure coefficients between inlet and outlet - the value is calculated as c_{top} - c_{bottom}; ρ = density of air, kg/m^3; v = wind speed in 10 m's height, m/s.

$$\begin{aligned} \Delta P_{stack} &= g \cdot \frac{\partial \rho}{\partial T} \cdot \Delta T \cdot \Delta h \\ &= g \cdot \frac{-P_{atm} \cdot M_{air}}{R \cdot T^2} \cdot \Delta T \cdot \Delta h \\ &\approx -0.043 \cdot \Delta T \cdot \Delta h \end{aligned} \tag{2}$$

where g = acceleration of gravity = 9.81 m/s^2; T = absolute temperature, K; M_{air} = molar mass of air = 28.96 g/mole; R = universal gas constant = 8.31 J/(mole·K); ΔT = temperature difference between air in the cavity and the outdoor ambient, K (calculated as $T_{ambient} - T_{cavity}$); Δh = vertical distance between inlet and outlet ("Height"), m.

Ideal gas assumptions are applied in the second line of Equation (2).

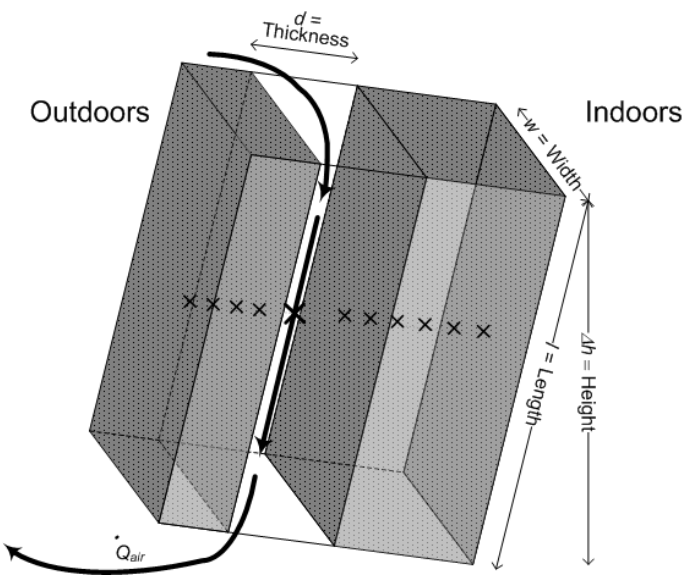

Figure 1. Configuration of the ventilated cavity. The 'x'-es denote the node points for the calculation. They may represented separate materials or subdivisions or one material.

The total pressure difference to drive air flow through the cavity is:

$$\Delta P_{cavity} = \Delta P_{wind} + \Delta P_{stack} \tag{3}$$

The two pressure contributions are added with the signs from the two previous equations as they may work in opposite directions. We apply a sign convention such that positive pressure differences are those which lead to downward air flow.

The air velocity in the duct is calculated as follows:

$$v_{duct} = \sqrt{\frac{\left|\Delta P_{cavity}\right|}{\tfrac{1}{2} \cdot \rho \cdot \xi_{total}}} \tag{4}$$

where v_{duct} = air velocity in the ventilated cavity, m/s; ξ = loss coefficient, -

The total loss coefficient is the sum of the coefficients for inlet, exit and duct losses. ξ_{inlet} and ξ_{exit} are set to 0.6 and 1.0, respectively, while ξ_{duct} is calculated as follows:

$$\xi_{duct} = f \frac{l}{d_h} \tag{5}$$

where f = friction factor; l = duct length in flow direction ("Length"), m; d_h = hydraulic diameter.

For laminar flow in the duct, the friction factor is:

$$f = \frac{96}{\mathrm{Re}} = \frac{96 \cdot \nu}{v_{duct} \cdot d_H} \tag{6}$$

where ν = kinematic viscosity, m²/s.

For a rectangular cavity:

$$\begin{aligned} d_H &= \frac{2 \cdot w \cdot d}{w + d} \\ &\approx 2 \cdot d \qquad \text{(for a narrow cavity)} \end{aligned} \tag{7}$$

where w = cavity width (extent in horizontal direction in the plane of the cavity), m; d = cavity thickness (distance across the cavity perpendicular to its plane), m.

The air velocity is itself a parameter in determining ξ_{total}, and therefore an iterative procedure is followed to determine v_{duct}. Once the air flow in the cavity is known, the heat and air balances for the cavity air can be determined. However, a fundamental issue concerns the fact that the hygrothermal conditions in the cavity develop over the path length of the cavity, while the flows through the building envelope are calculated in node points along a line which is perpendicular to the cavity. Thus, since it is not intended with the work presented in this paper to subdivide the construction into several sub-areas along the cavity, i.e. to make a multi-dimensional model, the correct flow field of heat, air and moisture transfer cannot be captured with full correctness of this model. We introduce an approximation by distributing the effect of the cavity equally over its path length[1].

In this model, the cavity is regarded as a fully mixed volume. The air enters at the inlet at the condition of the outdoor air, and it exchanges heat and moisture with the cavity walls facing the interior and the exterior parts of the construction.

Indeed, in the model the ventilated cavity is calculated as just another non-fluid layer filled with a material which has thermal conductivity and water vapour permeability in the same order of magnitude as still air. The material chosen is inorganic felt, since this is a rather open material with quite limited heat and moisture capacity, and therefore it does not significantly "disturb" the conditions in the cavity. By adjusting the thickness of this layer, one can achieve a desired thermal resistance of the cavity.

The reason for not using "still air" as the hygroscopic material filling the cavity is that the "sorption curve" for air will be very strongly temperature dependent, and the model would need quite some adjustment to take this into account.

The effect of the cavity ventilation is calculated by adding heat and moisture sources (or sinks) to this layer which evenly distributes the source power over the transmission area of the cavity as follows.

Heat source per wall transmission area brought by ventilation with outdoor air (W/m^2):

$$q_{vent} = \frac{\dot{Q}_{air}\left(\rho \cdot c_p\right)_{air}\left(T_{out} - T_{cav}\right)}{w \cdot l} = \frac{d}{l} v_{duct} \cdot \left(\rho \cdot c_p\right)_{air}\left(T_{out} - T_{cav}\right) \tag{8}$$

since $\dot{Q}_{air}$ = air flow in duct = $d \cdot w \cdot v_{duct}$; and ρ = density, kg/m^3; and c_p = specific heat, J/kg·K.

Moisture source per wall transmission area brought by ventilation with outdoor air (kg/m^2·s):

$$g_{vent} = \frac{\dot{Q}_{air}}{w \cdot l}\frac{\left(p_{out} - p_{cav}\right)}{R_v \cdot T} = \frac{\dot{Q}_{air}}{w \cdot l}\frac{\left(p_{out} - p_{cav}\right)}{130{,}600 \frac{\mathrm{N \cdot m}}{\mathrm{kg}}} = \frac{d}{l} v_{duct} \frac{\left(p_{out} - p_{cav}\right)}{130{,}600 \frac{\mathrm{N \cdot m}}{\mathrm{kg}}} \tag{9}$$

where p = water vapour pressure, Pa; R_v = Gas constant for water vapour, J/kg·K.

As *BSim* uses an implicit finite control volume solver, these convective source terms can be introduced in the control volume equation for the cavity in such a way that the source terms do not cause instability of the numerical procedure. Also for this reason, physical unrealistic conditions, such as negative vapour pressure, will not occur.

3 PROCEDURES TO CALCULATE HEAT AND MOISTURE TRANSPORT THROUGH POROUS WALLS WITH A FILTRATING AIR FLOW

The filtration air flow $q_{inf,m}$ is calculated according to the procedures described in the previous section.

Both temperature and vapour pressures are calculated in *BSim* using an implicit finite control volume scheme. For heat conduction, the finite difference form of the heat balance equation form for one control volume looks:

$$\left(\rho c_p\right)_i \Delta x_i \frac{T_i^{j+1} - T_i^j}{\Delta t} = \frac{T_{i+1}^{j+1} - T_i^{j+1}}{\frac{\Delta x_i}{2\lambda_i} + \frac{\Delta x_{i+1}}{2\lambda_{i+1}} + R_{i+½}} + \frac{T_i^{j+1} - T_{i-1}^{j+1}}{\frac{\Delta x_{i-1}}{2\lambda_{i-1}} + \frac{\Delta x_i}{2\lambda_i} + R_{i-½}} \tag{10}$$

where ρ = density of the material, kg/m^3; c_p = specific heat, J/(kg·K); Δx = width of the control volume, m; T = temperature, K; Δt = time step, s; λ = thermal conductivity, W/(m·K); R = possible interface resistance between control volumes, m^2·K/W; i = index for control volume number; j = index for time step.

With air flow, Equation (10) is modified as follows:

[1] It should be acknowledged that the heating/cooling and drying/humidification that take place of the air in the cavity along its length can be calculated as functions that develop exponentially over the length. For small air flows, the conditions in the cavity will within a short distance reach the same equilibrium conditions as in an unventilated cavity while, for large air flows, the inlet conditions will prevail most or all of the way through the cavity. This is described in a section on air flows in channels with transverse heat flows by Hagentoft, 2001.

$$\left(\rho c_p\right)_i \Delta x_i \frac{T_i^{j+1} - T_i^j}{\Delta t} = \frac{T_{i+1}^{j+1} - T_i^{j+1}}{\frac{\Delta x_i}{2\lambda_i} + \frac{\Delta x_{i+1}}{2\lambda_{i+1}} + R_{i+½}} + \frac{T_i^{j+1} - T_{i-1}^{j+1}}{\frac{\Delta x_{i-1}}{2\lambda_{i-1}} + \frac{\Delta x_i}{2\lambda_i} + R_{i-½}} + q_{air}\rho c_p\left(T_{upstream}^{j+1} - T_i^{j+1}\right) \quad (11)$$

where T_{upstream} = temperature in the adjacent control volume from which the air comes, K.

For vapour diffusion, the finite difference form of the moisture balance equation looks:

$$(\rho\xi)_i \Delta x_i \frac{\frac{p_i^{j+1}}{p_{s,i}^{j+1}} - \frac{p_i^j}{p_{s,i}^j}}{\Delta t} = \frac{p_{i-1}^{j+1} - p_i^{j+1}}{\frac{\Delta x_{i-1}}{2\delta_{i-1}} + \frac{\Delta x_i}{2\delta_i} + Z_{i-½}} + \frac{p_{i+1}^{j+1} - p_i^{j+1}}{\frac{\Delta x_i}{2\delta_i} + \frac{\Delta x_{i+1}}{2\delta_{i+1}} + Z_{i+½}} \quad (12)$$

where ξ = moisture capacity (slope of sorption isotherm), kg/kg; p = vapour pressure, Pa; p_s = saturation vapour pressure, Pa; δ = water vapour permeability, kg/(m·s·Pa); Z = possible interface water vapour resistance between control volumes, Pa·m^2·s/kg.

With air flow, Equation (12) is modified as follows:

$$(\rho\xi)_i \Delta x_i \frac{\frac{p_i^{j+1}}{p_{s,i}^{j+1}} - \frac{p_i^j}{p_{s,i}^j}}{\Delta t} = \frac{p_{i-1}^{j+1} - p_i^{j+1}}{\frac{\Delta x_{i-1}}{2\delta_{i-1}} + \frac{\Delta x_i}{2\delta_i} + Z_{i-½}} + \frac{p_{i+1}^{j+1} - p_i^{j+1}}{\frac{\Delta x_i}{2\delta_i} + \frac{\Delta x_{i+1}}{2\delta_{i+1}} + Z_{i+½}} + q_{air}\frac{\left(p_{upstream}^{j+1} - p_i^{j+1}\right)}{R_v T_i} \quad (13)$$

where p_{upstream} = vapour pressure in the adjacent control volume from which the air comes, Pa; R_v = gas constant for water vapour = 461.5 J/(kg·K); T = absolute temperature, K.

A "new" vapour pressure distribution has been determined. However, since the whole problem is non-linear (because the moisture capacity is not a constant value), a special procedure must be followed to ensure that the mass balance will be correct. The vapour pressures just found must be regarded as preliminary indications of the new values. It *could* even be that some of the preliminary vapour pressures have values that exceed the values of p_s at the same location, or they may be negative. This is either neglected at first, or the time step is repeated with a smaller Δt. The found vapour pressures are used in the moisture balance of each control volume as follows.

The vapour flux across the interface between control volumes, *i* and *i+1*, for the time step from *j* to *j+1* is:

$$g_{i-½}^{j+1} = -\frac{p_i^{j+1} - p_{i-1}^{j+1}}{\frac{\Delta x_{i-1}}{2\delta_{i-1}} + \frac{\Delta x_i}{2\delta_i} + Z_{i-½}} \quad (14)$$

where g = water vapour flux, kg/m^{2}s.

The vapour flux is calculated from the vapour pressures at the end of the time step, as predicted by Equation (13). A similar equation to Equation (14) is set up for $g_{i+½}$ and the increase in moisture content control volume *i* is found as follows:

$$\rho_i \Delta x_i \frac{u_i^{j+1} - u_i^j}{\Delta t} = -\left(g_{i+1}^{j+1} - g_i^{j+1}\right) \quad (15)$$

where u = moisture content, kg/kg

The equilibrium relative humidity is determined for each control volume by taking the calculated moisture contents and using the sorption curve for the relevant material (that is, an inverse expression for the sorption curve will be used).

4 EXAMPLES

4.1 *BESTEST building*

An existing *BSim* model of the BESTEST case building (Judkoff and Neymark. 1995) has been taken as a starting point for illustrating the capabilities of the ventilated cavity model. The original model consists of just one thermal zone. Two thermal zones have been added representing a bathroom and a kitchen, and the original thermal zone representing a living room (see Figure 2). Inspired by Woloszyn et al. (2005) has been added a bathroom and a kitchen within the BESTEST building, while keeping the same materials for the exterior building envelope – monolithic constructions of aerated concrete.

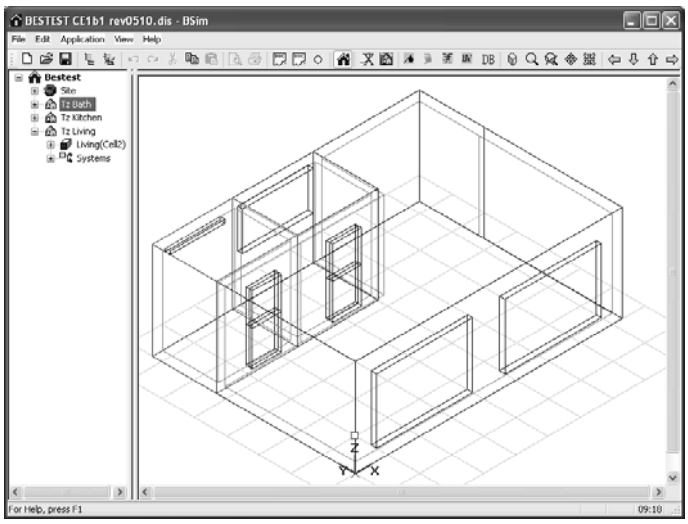

Figure 2. BESTEST model

The façade facing east has been replaced by a construction with a ventilated cavity (see Figure 3).

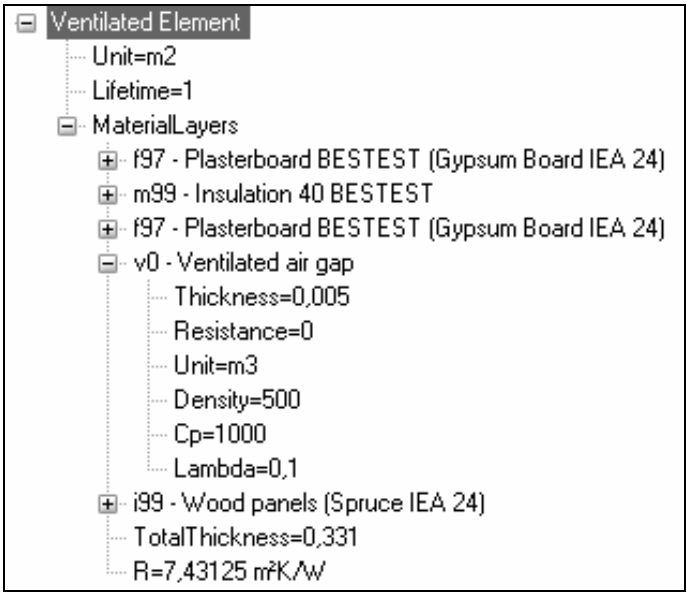

Figure 3. Construction with ventilated cavity. The material layers are listed from the inside of the construction towards the outside. The thickness of the insulation is 285 mm.

Seen from inside, the construction consists of 1.3 cm plasterboard, 28.5 cm insulation, 0.9 cm plasterboard, 3.8 cm ventilated cavity, and 1.9 cm wood panel facing outdoors. The ventilated air gap has material properties as a material with only small moisture resistances, and only small heat and moisture capacities and equivalent thickness 0.005 m.

In order to activate the air flow in the cavity driven by the air pressure difference between inlet and outlet, a profile for the wind pressure coefficient must be assigned to the constructions.

The wind pressure coefficient is depending on the building geometry, and is in general difficult to obtain. The most precise is to measure the coefficient, or to make a calculation by a CFD program. A less precise method is to obtain the coefficients from standard tables defined for common building geometries.

The pressure coefficient is dependent on the wind direction. By providing the coefficients for the wind directions north, east, south and west, the profile can be calculated for any wind direction (Axley, 2001).

The model has been simulated taking the ventilated cavity into account, and compared to a simulation where the ventilated cavity is not taken into account.

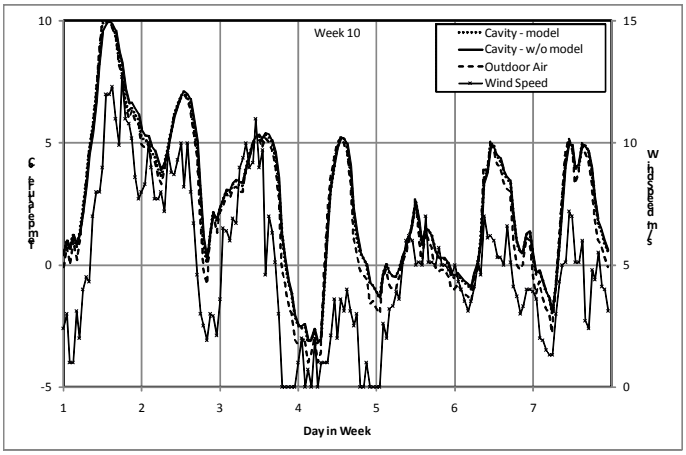

Figure 4. Temperature in the ventilated cavity. One calculation ("model") is based on the model for ventilated air cavity, while the other ("w/o model") is for a similar calculation which does not consider the air flow in the cavity.

The temperature within the ventilated cavity has been compared for a period in March (of the Danish Design Reference Year). As can be seen in Figure 4, the temperature is somewhat dependent on the wind speed. With high wind speed the temperature is the same as the outside temperature, and with low wind speed the temperature in the cavity is higher. When not taking the ventilation of the cavity into account, the temperature is very similar, or if any different, it is slightly higher and less fluctuating.

The calculation of the ventilated cavity has no significant effect on the relative humidity (nor the temperature) of the air in the room facing the ventilated construction, see Figure 5.

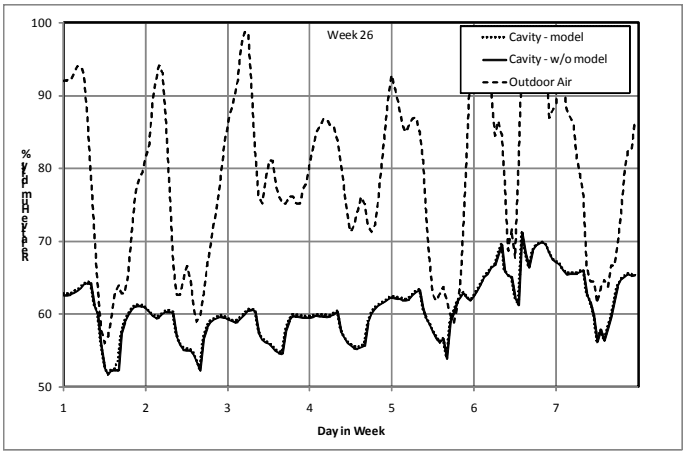

Figure 5. Relative humidity of the living room.

The relative humidity within the construction with the ventilated cavity has been compared, taking the ventilation into account or not. See Figure 6, which shows the equilibrium relative humidity for the exterior wooden cladding and for the interior plasterboard during a week in the end of June.

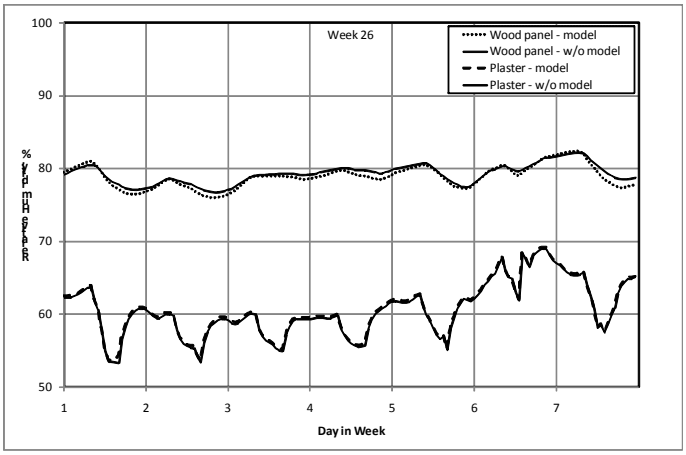

Figure 6. Relative humidity of the ventilated construction. Results entitled "w/o model" are from the model without ventilated cavity.

In the material layer facing the room (*Plaster*) the relative humidity is as expected nearly the same with and without taking the ventilation into account. In the wood layer separating the ventilated cavity and the outside the relative humidity has been slightly reduced by taking the ventilation of the cavity into account. Annual duration curves for the moisture content in the wood siding as predicted with and without the model for the ventilated cavity, are shown in Figure 7.

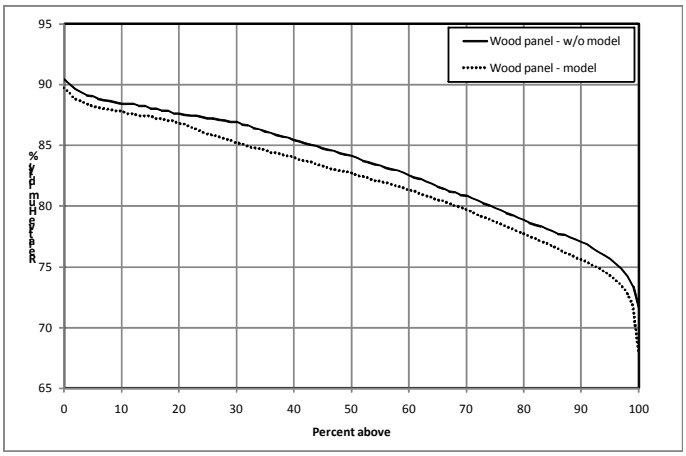

Figure 7. Annual distribution of relative humidity in the exterior wood based siding as calculated with ("model") and without ("w/o model") the ventilated cavity model.

4.2 *Example – Validation of filtration model*

An existing *BSim* model (the BESTEST building) has been taken as a starting point for illustrating the new model with filtration air flow in the building envelope. The building consists of 150 mm solid aerated concrete walls, and the walls have no surface coating.

One of the exterior walls of the building is split up such that half of the wall has a filtration air flow going through it, while the other is calculated as being air tight. See Figure 8. The building is exposed to an outdoor climate which artificially was constant at 20ºC and 30% RH, while the indoor climate was assumed operated so it was constantly 30ºC and 50% RH. Outside, the wind was assumed blowing from the west at 5 m/s, so by turning the building so the façade with filtration air flow either faces the wind, or is on the leeward side, then the heat and moisture flows in the building envelope could be investigated with the advection phenomena. With constant boundary conditions and constant air flows, the results can be compared with results of analytical calculations. The air flows were adjusted to always give a mass flow rate of 0.0005 kg/(m^{2}s), corresponding to 1.57 m^3/(m^{2}h) – either infiltrating or exfiltrating.

The mesh had control volumes of thickness 1.7 mm closest to the indoor climate, increasing to 10 mm through most of the wall, so there were a total of 17 control volumes to represent the wall.

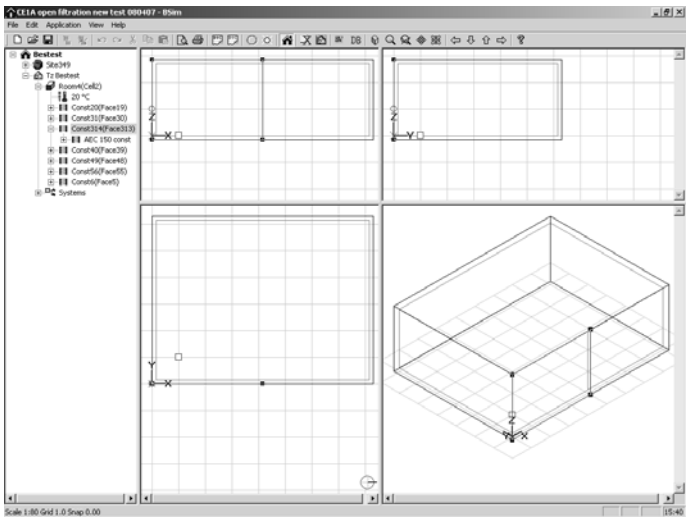

Figure 8. BESTEST building modelled in BSim. The wall that that is split in two halves has one half calculated with the filtration model, whereas the other half is calculated as being air tight.

The analytical solution for the temperature is given by (Hagentoft, 2001):

$$T(x) = T_{out} + (T_{in} - T_{out}) \frac{e^{\frac{x}{\ell}} - 1}{e^{\frac{L}{\ell}} - 1} \tag{16}$$

where L = thickness of the wall, m; ℓ = a quantity of dimension length that characterizes the interaction between conductive and convective heat flows, such that:

$$\ell = \frac{\lambda}{c_{air} \dot{M}_{air}} \tag{17}$$

where c_{air} = heat capacity of air = 1005, J/(kg·K); $\dot{M}_{air}$ = mass flow rate of air, kg/(m^{2}s).

Figure 9 shows the comparison between the temperatures calculated numerically using *BSim* and the analytical solutions for exfiltration, no air flow, and infiltration, respectively.

Table 1 gives a comparison between the analytically and numerically calculated temperatures in the middle of the wall.

Table 1. Analytically and numerically calculated temperatures in the middle of the wall.

	Analytical	Numerical
Exfiltration	25.52	25.55
No air flow	25.00	25.04
Infiltration	24.48	24.50

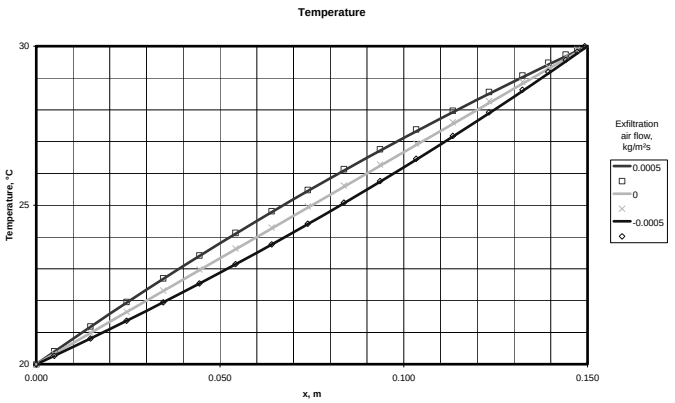

Figure 9. Analytical (solid lines) and numerical calculation (symbols) of temperatures through the wall with exfiltration (□), no air flow (x) and infiltration (◊) air flow through the outer wall.

The temperature determines the saturation vapour pressure. An approximation of the analytical vapour pressure is found as:

$$p(x) = p_{out} + (p_{in} - p_{out}) \frac{e^{\frac{x}{\ell_v}} - 1}{e^{\frac{L}{\ell_v}} - 1} \tag{18}$$

where ℓ_v = a quantity of dimension length that characterises the interaction between conductive and convective water vapour flows, such that:

$$\ell_v = \frac{\delta}{c_v \dot{M}_{air}} \tag{19}$$

where c_v = ratio between humidity of the air (x) and the vapour pressure = 0.622/(101.325 Pa - p), Pa^{-1}

c_v is not quite constant since p varies with location in the construction, and therefore, the analytical solution is only approximate (with a few percents' deviation).

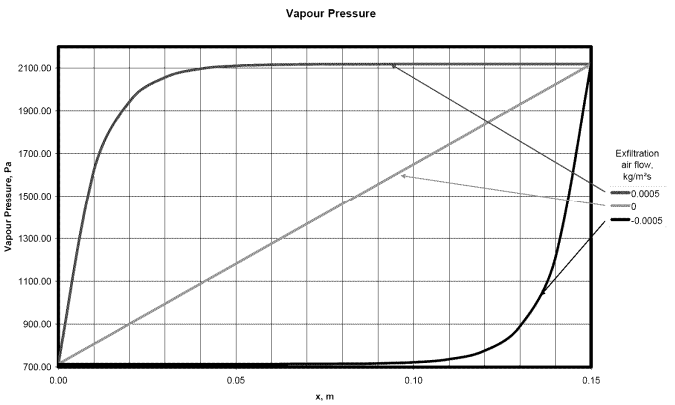

Figure 10. Vapour pressure calculated according to Equation (18).

Finally, RH can be determined as p/p_s. Figure 11 shows the comparison between the temperatures calculated numerically using *BSim* and the analytical solutions for exfiltration, no air flow, and infiltration, respectively.

Table 2 gives a comparison between the analytically and numerically calculated relative humidity in the middle of the wall.

Table 2. Analytically and numerically calculated relative humidity in the middle of the wall.

	Analytical	Numerical
Exfiltration	66.8	64.9
No air flow	44.7	44.7
Infiltration	22.5	23.4

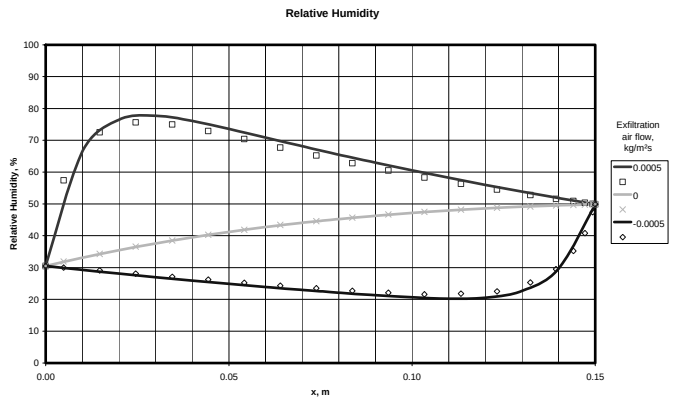

Figure 11 Analytical (solid lines) and numerical calculation (symbols) of relative humidity through the wall with exfiltration (□), no air flow (x) and infiltration (◊) air flow through the outer wall.

It can be concluded that the advection model in *BSim* predicts the analytical temperature distribution with a good satisfaction, while also the relative humidity distribution is predicted with a fair amount of accuracy.

5 CONCLUSION

A model for calculating air flows in ventilated cavities has been implemented in the whole-building hygrothermal simulation tool *BSim*. An existing model of the BESTEST case has been taken as a starting point for illustrating the capabilities of the ventilated cavity model. Finally a model of an existing research house with façade elements with ventilated cavities has been simulated.

A preliminary model for filtration air flow through a wall has been implemented to investigate its effect on heat and moisture conditions in the wall.

6 REFERENCES

Axley, J.W. 2001. Residential Passive Ventilation Systems: Evaluation and Design. AIVC Technical Note 54, AIVC, University of Warwick Science Park.

Grau, K. and Rode, C. 2005. Whole building HAM simulation with a multizone air flow model. IEA ECBCS Annex 41, Working meeting, Trondheim, October 2005. Paper A41-T1-DK-05-6.

Grau, K. and Rode, C. 2005. A model for air flow in ventilated cavities implemented in a tool for whole-building hygrothermal analysis. IEA ECBCS Annex 41, Working meeting, Lyon, October 2006. Paper A41-T1-DK-06-

Hagentoft, C.-E. 2001. Introduction to Building Physics. Studentlitteratur.

Jensen, Rasmus Lund, 2005, Modelling of Natural Ventilation and Night Cooling - by the Loop Equation Method. Ph.D. Thesis. Department of Building Technology and Structural Engineering, Aalborg University, ISSN 1397-7953 R0402. In Danish.

Judkoff, R. and J. Neymark. 1995. Building energy simulation test (BESTEST) and diagnostic method. NREL/TP-472-6231. Golden, Colo.: National Renewable Energy Laboratory.

Rode, C. and Grau, K. 2003. Whole Building Hygrothermal Simulation Model, American Society of Heating, Refrigriation and Air-conditioning Engineers. Recent Advances in Energy Simulation: Building Loads, Symposium CH-03-09, Chicago, USA.

Rode, C. and Grau, K. 2004. Calculation Tool for Whole Building Hygrothermal Analysis. IEA ECBCS Annex 41, Working meeting, Zürich, May 2004. Paper A41-T1-DK-04-1

Woloszyn, M., Shen, J., Mordelet, A. and Brau, J. 2005. Numerical Simulations of Energy Performance of a Ventilation System Controlled by Relative Humidity. AIVC Conference 'Ventilation in relation to the energy performance of buildings', Brussels, September 21-23, 2005.