MODELLING THE SOLVENT VENTILATED WINDOW FOR WHOLE BUILDING SIMULATION

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Abstract

Whole building simulation may play a key role in the optimization and assessment of the market potential of new building components. ESP-r was used for these purposes in the development of a novel ventilated reversible glazing system. The innovative character of the ‘SOLVENT’ window required the development of a specific simulation approach within this whole building simulation software, in order to account for buoyancy in the air channel. A multi-zone approach with an air flow network was developed, and several variations studied. Parametric studies assessed the effect of the number of zones into which the window is divided, heat transfer correlations for the air gap and for the external surface of the window, local loss coefficients for the air flow network and the use of the \textit{ish} module for detailed solar radiation treatment. An experimental measurement campaign
performed in the PASSYS test cell in Porto allowed calibration and verification of the simulation model, as well as an analysis of the accuracy achieved.

1. INTRODUCTION

The use of clear windows in buildings is a common way to provide daylight and solar energy to rooms in buildings. In many buildings, especially those located in places with frequent sunny weather, visual comfort problems arise, e.g., glare. Users are often prompted to use internal or external solar protection devices, which in turn reduce energy gains in the heating season and may prompt them to use electrical lighting.

In an attempt to solve the visual comfort problems without compromising energy performance, Etzion and Erell proposed an innovative glazing system, here called the “SOLVENT window” [1].

The SOLVENT window is shown in figure 1. It is essentially a ventilated window with a conventional double clear glazing on one side and an absorptive glazing on the other side of the channel. The window frame allows a rotation in such a way that the absorptive glazing faces indoors in the winter mode and outdoors in the summer mode [2].

[Insert Figure 1 about here]
The physical principle of the system is the interception of solar radiant energy by the absorptive glazing and its conversion into convective heat and long wave radiation. As the absorptive glazing heats up, it will create natural convective air flows in the air channel and along its free face. An accurate quantification of the energy flows is thus essential for evaluation of the SOLVENT window energy performance when integrated in realistic buildings.

The development of this glazing system was the aim of a research project called SOLVENT, supported by an EU grant. One of the project goals was to study the energy impact of the window on several types of buildings in various climates. The thermal simulation software ESP-r was selected for this purpose [3, 4].

Modelling the SOLVENT window within ESP-r in a conventional way, as in other whole building simulation (WBS) software, could not account for the flow in the air channel and the heat convected by this means. A special procedure was thus needed, capable of accounting for the buoyancy in the air channel and of describing the induced air flow.

2. MODELLING APPROACH

The combined effect of heat and mass flow in buildings may be simulated by two general methods:

   i) CFD simulation
ii) Air flow network coupled with energy balance.

The first alternative probably has the potential to achieve more accurate results. However, it may also be more difficult to implement in practice. As one of the goals is to obtain results for long periods (frequently a complete year), the dynamic simulation of the air flow in the channel, if at all possible, would require computational resources much beyond the scope of the study. Along with this difficulty, the experimental setup necessary for calibrating and validating a CFD model is usually complex.

The second alternative provides a method compatible with simulation of long periods, and it is easier to calibrate and integrate with other heat fluxes such as absorption of solar radiation in the glazings, long wavelength exchanges between the glazings, etc. This kind of approach has been used in the simulation of PV façade elements [5] and double-skin façades with one opaque wall [6].

The use of this technique requires that the air channel be divided into two or more thermal zones, linked to each other and to external nodes by network components such as air ducts and openings. Figure 2 represents an ESP-r model of the SOLVENT window constructed according to this approach.

Insert Figure 2 about here
While the physical description of the system and the inclusion of the angular-dependent optical properties of the absorptive glazing and of the double-clear glazing are straightforward, there are a number of questions that may require some research:

i) The number of zones into which the window must be divided.

ii) The heat transfer coefficient values or correlations needed to characterise the surfaces of the air channel.

iii) The localized pressure loss coefficients for the inlet and outlet of the channel.

The assessment of the adequacy of the possible parameters and calibration of the simulation model were obtained by monitoring a SOLVENT window mounted in a PASSYS test cell in Porto [7]. An ESP-r model of the test cell integrating the SOLVENT window was developed for comparison of simulation results with measurements. Figure 3 shows a representation of the model geometry.

The base case was proposed before specific measured data became available, so it may represent the approach of a modeller who has no prior measured data. In the context of the parametric studies described later in this paper, the following model comprises the default values used in the simulation. The window was divided into four vertically interconnected thermal zones. The division into several thermal zones was required to account for the pressure variation and buoyancy effects, but the number of zones itself (4) was arbitrary. The default heat transfer coefficients
computed by ESP-r were used for all surfaces. Localized pressure drop coefficients were computed from standard fluid mechanics relations [8], using $K_{in}=0.5$ for the inlet and $K_{out}=1$ for the outlet. Curve effects were neglected because the elbows were coincident with the inlet/outlet and because they were very smooth.

3. EXPERIMENTAL SETUP

The experimental setup comprised a SOLVENT window installed in the PASSYS test cell located in Porto and instrumentation to monitor its thermal and daylighting behaviour. Figure 4 shows a view of the SOLVENT window (in summer mode) installed in the test cell with some of the associated instrumentation [9]. The main monitored quantities were, in addition to outdoor climatic variables, the temperature of the absorptive glazing, the temperature of both clear-glazed panes, the air temperature at different heights in the air channel and the air velocity at the centre of the channel. Figure 5 shows a scheme of the window and measuring instrumentation. Difficulties associated with temperature measurement under incidence of strong solar radiation were identified and are discussed elsewhere [10].
4. NUMBER OF ZONES

It has been reported that the division of a window into several zones linked by an air flow network has the potential to account for buoyancy and natural convection effects [5, 6, 11]. However, no rule has been given which allows the determination of the number of divisions needed.

This section presents a parametric study of the number of zones into which the window air channel should be divided and the comparison of results with measurements in the test cell. Alternatives considered comprised modelling the air gap as only one thermal zone, or with several vertically overlapping zones (2, 4, 8 and 16 zone models). An alternative $4\times2$ division was also implemented, based on the 4-zone model, but sub-dividing each level into two horizontal zones. Convection calculations were left in ESP-r default mode and the local head loss coefficients were also kept at $K=0.5$ for the inlet and $K=1.0$ for the outlet of the air channel. Simulations were performed for the winter mode (i.e., with the channel open to the building interior) in order to minimise the effect of wind. A time step of 1-minute was used in the simulation, together with hourly integration for results output.

Figure 6 shows the evolution of outdoor temperature and solar radiation during the period 29th October-11th November 2001, which was used as reference for
comparing measurements and simulation results. The three-day period 8th-10th November, for which full measurements of velocity in the air gap were available, was the subject of more detailed analysis.

Results presented in figures 7 to 10 show hourly averages of system temperatures and air velocity for a period of three days starting at 0:00 h of November 8th 2001, and daily cooling needs for the period 29 October-13 November. An analysis of simulation vs. measurement deviation is also presented in table 1, showing the absolute mean deviation between simulation and measurements as computed by:

\[
AMD = \frac{\sum |X_{\text{sim},i} - X_{\text{meas},i}|}{n} \quad (\text{eq. 1})
\]

Insert Table 1 about here
The number of zones used affects simulation results substantially. However, each of the parameters investigated is affected in a different manner:

- The maximum daytime temperature of the absorptive glass is predicted best by the single-zone models. However, the multi-zone models are better at predicting glass temperature at night, and have lower overall mean deviations between predicted and measured values. All models appear to overestimate the effects of thermal inertia (figure 7).

- Simulated values for the daytime temperature of air at mid-height in the ventilated channel were far too high when calculated using the 1-zone model, probably as a result of air stagnation since no air flow was predicted. Increasing the number of zones gave values that were too low, but were better at predicting night-time minima, and had lower overall deviations (figure 8).

- Simulation of the velocity of air in the ventilated channel was very sensitive to the number of divisions used. (It must be noted that ESP-r presents as output the volumetric air flow. To convert to air velocity, a cross-section profile must be assumed. In this case a uniform profile was assumed, as results reported by Sandberg seem to indicate for this gap width [12]) A 1-zone model was unable to simulate the flow at all. The 8-zone and 16-zone models gave the best (and very similar results), closely followed by the 4-zone model. The 4×2 model performed noticeably poorer than the 8-zone model (Figure 9).

- Finally, with respect to the cooling needs of the test cell, the accuracy of the predictions generally improved with an increase in the number of
zones. The 4, 8 and 16-zone models performed significantly better than the 1 and 2-zone models, but there was little difference between them. This issue was considered decisive in this study, since the aim was to simulate the impact of the window upon building energy consumption (Figure 10).

5. HEAT TRANSFER COEFFICIENT

Since the window is being treated as a set of thermal zones, the heat transfer coefficients computed by ESP-r for the inner surfaces of the air channel are the same as for room internal surfaces. These are usually computed by Alamdari-Hammond correlations. A more specific correlation for laminar channel flow with isothermal walls, by Bar-Cohen and Rosenhow, can be found in the literature [13]:

\[
Nu_S = \left[ 576 \left( Ra_S \frac{S}{H} \right)^{2} + 2.87 \left( Ra_S \frac{S}{H} \right)^{-0.5} \right]^{-0.5} \quad (eq. 2)
\]

where S is the air gap width and H the air gap height. The resultant heat transfer coefficient, depending on the aspect ratio S/H and on the temperature difference between the surfaces and the air at the inlet is plotted in figure 11. For a window 1 metre high, with an air gap of 4.5 cm and a temperature difference (\(\Delta T\)) of nearly 20°C between the glazings and the air entering the channel, a value of \(h=3.0\) W/m\(^2\).K is representative.
In practice, however, the air flow in the air gap is not always laminar, the glass panes are not isothermal and may have different temperatures, etc. Based on experimental data from the SOLVENT window, Molina and Maestre suggested the following empirical correlation for a 4.5 cm air gap [14]:

\[ h = 3.00\Delta T^{\frac{1}{3}} \]  (eq. 3)

This correlation leads to substantially higher heat transfer coefficients. For instance, with a difference of 20 °C between the average temperature of the glazings and the temperature of the entering air, the predicted heat transfer coefficient is 8.1 W/m².K. Changing the convection coefficients within ESP-r is possible by imposing fixed \( h \) values for pre-defined periods or by selecting a pre-defined correlation that will be active whenever the HVAC system is active in the zone [15].

It is known from boundary-layer theory that heat transfer coefficients are higher at the beginning of the surface and then decrease (with the possibility of increasing again if there is a transition to turbulent flow at a certain point). This study was developed in the frame of length-averaged heat transfer coefficients, and thus a uniform value was assumed for the entire area of the surface modelled.

This study initially considered three alternatives:

i) Default correlations (Alamdari-Hammond);
ii) An imposed value of $h=3.0 \text{ W/m}^2\text{.K}$ from 8:00 to 20:00 (considered the daytime period).

iii) An imposed value of $h=8.0 \text{ W/m}^2\text{.K}$ from 8:00 to 20:00 (considered the daytime period).

At a later stage the Bar-Cohen and Rosenhow correlation (eq. 2) and the SOLVENT correlation by Molina and Maestre (eq. 3), were added to the ESP-r source code, thus allowing dynamic calculation instead of fixed convection coefficients. The alternatives with fixed coefficients are retained as user-only approaches. The number of window zones was kept at 4 and the sum of local pressure loss coefficients at $K=1.5$.

Results of measurements and simulations are presented in figures 12 to 15 and in table 2. These results show that the method of calculating the heat transfer coefficient had limited effects on the accuracy of the predictions for the temperature of the tinted glazing and for the air gap temperature. Use of the Bar-Cohen and Rosenhow correlation results in the lowest mean deviations in all system-specific variables, while the Solvent correlation and the fixed value 8 W/m$^2$.K during daytime lead to the lowest mean deviation in terms of cooling demand.

Insert Figure 12 about here

Insert Figure 13 about here

Insert Figure 14 about here
6. LOCALIZED LOSS COEFFICIENT

The construction of an air-flow network requires the input of local pressure drop coefficients. The pressure loss at an obstruction is given by:

\[ \Delta P = \frac{1}{2} K \rho V^2 \]  

(eq. 4)

The standard value of K is 0.5 for a sudden contraction (such as the channel inlet) and 1.0 for a sudden expansion (such as the channel outlet). These are, however, only recommended values, since the exact values depend strongly on the geometry of the streamlines in each particular case. For this study, three alternatives were considered:

- K=1.5, based on the standard approach;
- K=0.2, representing a scenario of low streamline dispersion;
- K=2.0, representing a scenario of higher streamline dispersion, also coincident with the value that Sandberg et al. reported as leading to good agreement with experimental results [12].

The number of window zones was kept at 4 and heat transfer coefficients kept in the default mode.
Results for the different alternatives and measurements are shown in figures 16 to 19 and in table 3. They show that the influence of the sum of the local loss coefficients upon glazing temperature and cooling needs is low. The value of $K=0.2$ gave best results in terms of air velocity, but resulted in poorer predictions of air temperature.

7. EXTERNAL HEAT TRANSFER COEFFICIENT

One possible explanation for the over-prediction of inertia observed in the simulation results could, in principle, be the calculation of the convection coefficient at the window exterior. By default, it is computed in ESP-r by the McAddams correlation based on the wind speed and direction [4]. If, for any reason, the values computed were too small, this could explain the very slow cooling of the glazing system at the end of the day. To test this possibility, an
alternative model of the window was studied in which an external convection coefficient of \( h = 17.5 \ \text{W/m}^2\text{K} \) was imposed. Results are shown in figure 20 for air gap temperature, and table 4 presents the mean deviations for tinted glazing temperature, air gap temperature, air velocity and cooling loads. The use of the imposed convection coefficient improved prediction of system temperatures at night-time, but did not affect the quality of predictions of air velocity in the ventilated channel, and actually resulted in less accurate simulation of cooling loads. Furthermore, the inertia effect was reduced only marginally.

8. USE OF ISH

ESP has an optional module for pre-calculating the distribution of the solar radiation on room surfaces, the ISH module (Insolation and SHading analysis). The impact of using this module was also assessed. This requires no change to the modelling, as it is only a matter of running the module before the global simulation. Figure 21 shows the results for the absorptive glazing temperature at mid-height, while figure 22 shows the evolution of the cooling load. Table 5 summarizes the absolute mean deviation for the three-day period. The results show that the use of ISH improved temperature predictions. The prediction of cooling needs also improved during daytime but the persistent inertia effect during night-time leads to overall poorer prediction.
9. CONCLUSIONS

The present study compared the effects of several modelling options for a ventilated window, all based on an airflow network approach. The study focussed on the influence of the following parameters: the number of zones into which the channel is divided; the method for calculating heat transfer coefficients; the values of local pressure loss coefficients. The effect of the exterior convection coefficient and of using the module for detailed radiation distribution, ISH, were also analysed.

It was found that the sensitivity of the results to the studied parameters is in the following order:

- The most important parameter is the number of zones into which the window is divided; the number of vertical divisions is especially critical, but dividing into more than 4 zones brought only marginal improvements. The model was not improved by subdividing each vertical division horizontally, too.
• The second parameter in order of importance is the heat transfer coefficient.

• The least important parameter is the local pressure loss coefficient.

The use of imposed external convection coefficient and the use of the ISH module brought some improvements to the prediction of system variables, but not to the prediction of cooling loads.

Overall, the results of this study show that the air flow network approach can provide satisfactory estimates of long-term heating and cooling loads, which could not be obtained using CFD simulation. However, simulation of specific system variables such as glazing temperature or air velocity in a ventilated glazing system such as the SOLVENT prototype could only predict the order of magnitude of the variables, and was not free from inertia effects. Accurate prediction of specific system variables may require detailed CFD modelling.

ACKNOWLEDGEMENTS

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REFERENCES


NOMENCLATURE

\( Nu_S \) – Nusselt number based on the channel with S.

\( Ra_S \) - Raleigh number based on the channel with S.
S – open air channel glass-to-glass width (m).
H – Channel height (m).
h – Convective heat transfer coefficient (W/m².K).
\( \Delta T \) - Temperature difference between the average glazing temperature (considering the two sides of the air channel) and the air entering the air channel (K).
\( \Delta P \) - Local pressure drop at a certain point of the air flow path (Pa).
K – Local pressure loss coefficient.
\( \rho \) - Air density (kg/m³).
V – Average air velocity at the location of the obstruction to the air flow (m/s).
\( X_{\text{simul},i} \) - Simulated value of quantity X at hour i.
\( X_{\text{meas},i} \) - Measured value of quantity X at hour i.
n – number of hours used for comparison between measurements and simulation.
Solar Radiation

Double Clear Glazing

Absorptive Glazing

Air channel

Indoor

Outdoor

Winter mode

Figure 1 a): SOLVENT window in winter mode

Solar Radiation

Indoor

Outdoor

Summer mode

Figure 1 b): SOLVENT window in summer mode
Figure 2: SOLVENT window air network modelling
Figure 3: Screen capture of the ESP-r model of the PASSYS test cell in Porto with the SOLVENT window integrated.
Figure 4: Prototype of the SOLVENT window (in summer mode) installed at the PASSYS test cell.
Figure 5: Location of measurement sensors.
Figure 6: Outdoor air temperature and solar radiation (global horizontal) during simulation period.
Figure 7: The effect of the number of zones in the air flow model on the prediction of the temperature of the absorptive glazing mid-height.
Figure 8: The effect of the number of zones in the air flow model on the prediction of air temperature at channel mid-height.
Figure 9: The effect of the number of zones in the air flow model on the prediction of air velocity at channel mid-height.
Figure 10: The effect of the number of zones in the air flow model on the prediction of the daily cooling needs.
Figure 11: Heat transfer coefficient in the air gap according to equation 2.
Figure 12: The effect of the channel surface convective heat exchange coefficient on the prediction of the temperature of the absorptive glazing at the middle of the pane.
Figure 13: The effect of the channel surface convective heat exchange coefficient on the prediction of the air temperature at channel mid-height.
Figure 14: The effect of the channel surface convective heat exchange coefficient on the prediction of the air velocity at channel mid-height.
Figure 15: The effect of the channel surface convective heat exchange coefficient on the prediction of the daily cooling needs.
Figure 16: The effect of the pressure loss coefficient on the prediction of the temperature of the absorptive glazing mid-height.
Figure 17: The effect of the pressure loss coefficient on the prediction of the air temperature at channel mid-height.
Figure 18: The effect of the pressure loss coefficient on the prediction of the air velocity at channel mid-height.
Figure 19: The effect of the pressure loss coefficient on the prediction of the daily cooling needs.
Figure 20: The effect of the external convective heat exchange coefficient on the prediction of the air temperature at channel mid-height.
Figure 21: The effect of using ISH on the prediction of the absorptive glazing temperature at mid-height.
Figure 22: The effect of using ISH on the prediction of the cooling load.
Table 1: Absolute Mean Deviation between simulated and measured values, varying the number of channel divisions

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<tr>
<td>Air gap temperature mid-height (°C)</td>
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<tr>
<td>Velocity in the air gap (ms⁻¹)</td>
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<td>Cooling needs (kWh day⁻¹)</td>
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Table 2: Absolute Mean Deviation between simulated and measured values, varying the convection coefficient in the air gap.

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<tr>
<td>Cooling needs (kWh day⁻¹)</td>
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Table 3: Absolute mean deviation between simulated and measured values, varying the pressure drop coefficient.

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<td>0.11</td>
</tr>
<tr>
<td>Cooling needs (kWh day(^{-1}))</td>
<td>3.4</td>
<td>0.35</td>
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Table 4: Absolute mean deviation between simulated and measured values, varying the convective heat exchange coefficient of the external surface of the window.

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<td>Air gap temperature mid-height (°C)</td>
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<tr>
<td>Velocity in the air gap (m/s)</td>
<td>0.23</td>
<td>0.11</td>
</tr>
<tr>
<td>Cooling needs (kWh/day)</td>
<td>3.4</td>
<td>0.35</td>
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Table 5: Absolute Mean Deviation between simulated and measured values, depending on the mode used to analyse insolation and shading.

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