

Supply air window PAZIAUD®: Comparison of two numerical models for integration in thermal building simulation

F. Gloriant¹, P. Tittlein², A. Joulin³, S. Lassue⁴

Abstract

The principle of a supply air window is to ensure the renewal airflow between the panes of glass before entering inside the room. The Paziaud® window is composed of three panes of glass separated by ventilated U-shaped air gaps. The goal is to create a passive system of heat recovery contributing also to the room ventilation. The window is modeled in two ways, the first one is based on a numerical model of hydrodynamics and heat transfer, while the other one is based on heat balance represented by an electric analogy. Both results are compared in terms of blowing air temperature and thermal efficiency, and provide a validation of the simplified model, which could be used later in a numerical thermal code.

Keywords: forced and natural convection, modeling, radiation, ventilated window

Introduction

Windows constitute in a building, from a thermal point of view, one of the most vulnerable part between the inside and the outside environments. Indeed, windows are integrated within the building envelope, but contrary to the others parts of the envelope, windows transmit light and provide a quite lower resistance to heat flow. On the other hand, they are able to collect solar energy to heat up when the solar provision is important and the external temperature is weak (Catalina et al. 2008).

The main difference between a standard window and a supply air window consists in the presence of natural or forced convection generated within the gaps between the panes of glass (McEvoy et al. 2003; Ismail and Henriquez 2005, Ismail et al. 2009). The objective is to create a passive system of heat recovery which also participates to the building ventilation. An airflow inside the window caused by a depressurization in the room is performed with an extraction system. Thus, the heat loss by the window is recovered providing the pre-heating of the incoming air (Carlos et al. 2010). While a heat recovery ventilation system is complex to install in a rehabilitation context, this kind of window presents an interesting alternative in return with an extra cost of 20% compared to a classical double-glazed window, but yielding a non negligible heat recovery.

In this article, we focus particularly on one kind of supply air window, the Paziaud® window, inside which the air circulates along a U-shaped cavity. In order to evaluate the

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general performance of the Paziud® window, it is necessary to implement a numerical method which able to represent the flow dynamics and heat transfer inside the window. The results of two different models are presented: the first one is based on the solving of Navier-Stokes equations by means of a CFD code, whereas the second one is based on balances at higher scales which can be shown in a form of electrical analogy. The goal of this study is to compare the results of both models, and thus to validate the simplified model which could be used in a thermal building computational code.

Moreover, another issue is encountered about the performances of the Paziud® window versus the parameters of its environment.

For classical windows, a global coefficient is defined, of thermal transmission (U-value). For the supply air window which participates to the renewal air of the building, this coefficient is not clearly defined in the literature and the rules and regulations about it are not well documented and remain empiric. As a consequence, we propose in this article to present several indicators able to evaluate thermal performances of the Paziud® window.

Formulation and studied system description

A laminar natural or mixed convection airflow is considered in a U-shaped conduct formed by three panes of glass (figure 1). Air enters within the window at external temperature of 0 °C, circulates within the U-shaped conduct heating up by means of the heat flux coming from the internal environment and enters the room at an unknown temperature. The heat flux passing through the glass “i” is called Φ_i (in W.m^{-2}). Horizontal surfaces are considered adiabatic.

The temperature difference between both environments is 20 °C. Concerning the boundary conditions of the two glazed surfaces, two heat transfer coefficients are defined, h_{int} and h_{ext} corresponding to the internal and external environments. Their respective values differ with the presence or not of radiative heat transfer (values from French thermal building regulation RT 2012). Three simulations 1, 2 and 3 are carried out and are summarized in table 1.

Table 1: Parameters retained for different simulations

	h_{int} ($\text{W.m}^{-2}\text{°C}^{-1}$)	h_{ext} ($\text{W.m}^{-2}\text{°C}^{-1}$)	Glasses surface emissivity
simulation 1	4	20	No radiation considered
simulation 2	7	25	0.9
simulation 3	7	25	0.9 except for low-e surfaces: 0.1

Simulations performed without radiative heat transfer do not correspond to any physical reality because walls delimiting airflow are made of glass and thus are emissive. However, they allow to reduce hypothesis to a pure conductive problem and to measure the radiative heat transfer influence. Moreover, let us notice that the main issue to solve in order to obtain an independent simplified model that could be used in a complete building model is the determination of convection coefficients hc_{ij} (figure 3). Concerning radiative coefficients hr , they are not very sensitive to the airflow and thus do not represent issues *a priori*.

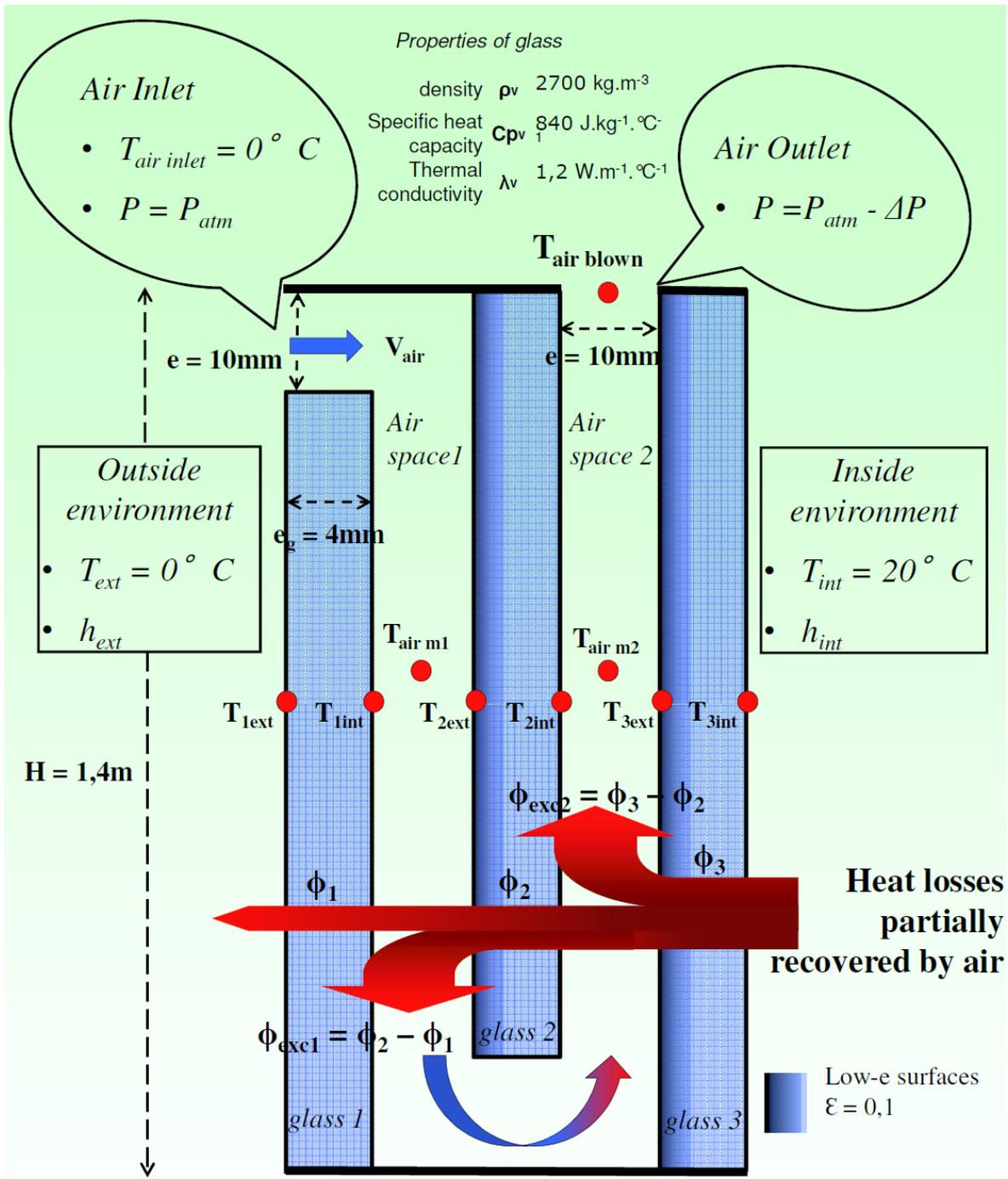


Figure 1. Schematic diagram and boundary conditions

In each test, the pressure difference varies from the inlet and the outlet, and consequences on heat fluxes Φ_i crossing the different panes of the window are determined, as well as the air temperature entering the room. The corresponding values of the Prandtl number is $Pr=0.71$, Grashof number is $Gr=3111$, and the aspect ratio is $A=140$.

Performance indicator used

In order to evaluate window performances, it is necessary to define the indicators. Those presented here are based on modification of the heat balance due to supply air window compared to a classical system with ventilation outside the window.

The heat balance in the considered control zone is:

$$q_{source} + \sum_{i=1}^n q_{wall_i} + \sum_{i=1}^m q_{win_i} + q_{ACH} = \rho_{air} \cdot V_{air} \cdot C_{P_{air}} \cdot \frac{dT_{air}}{dt} \quad [1]$$

It is considered that the window equipped with an inlet air vent participates to the zone heat balance with 2 terms:

- q_{win} that represents the deperditive heat flow through glass and frame
- q_{ACH} that represents the heat flow due to air change with :

$$q_{ACH} = \dot{m} \cdot C_{P_{air}} \cdot (T_{airblown} - T_{int}) \quad [2]$$

In the case of a classical window, air is blown at the exterior air temperature while in the case of a supply air window, air is preheated before entering the room at temperature $T_{airblown}$.

Usually, the U-value of glass is defined as the report between heat flow rate through glassed part of the window and air temperature difference between inside and outside. With a supply air window, it must be determined which heat flow rate must be considered. The heat flow is indeed different through the three glasses. Two possibilities of heat flow are considered by changing control volume for the heat balance:

- considering heat flow rate at the exterior face of exterior glass (Φ_1 , control zone 1)
- considering heat flow rate at the interior face of interior glass (Φ_2 , control zone 2)

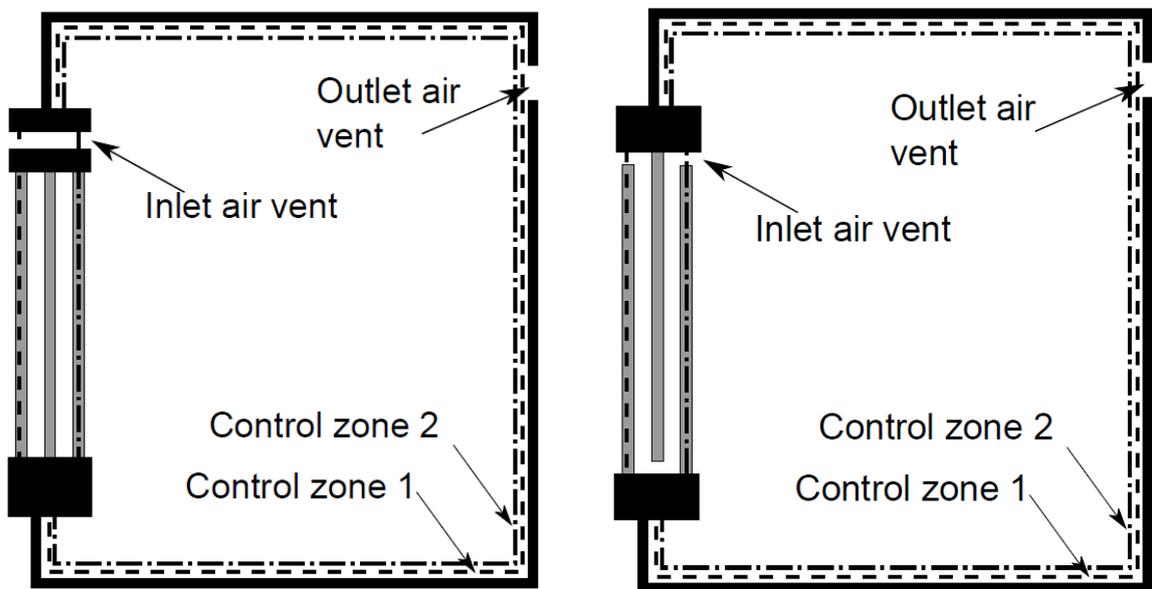


Figure 2. Choice of control zone for indicators calculation (on the left: classical window, on the right: supply air window)

If the control zone 1 is considered (see figure 2), exterior air enters with the exterior temperature and exits via the extraction vent (or another kind of outlet as, for example a door) at inside air temperature. It is the same case for a classical window and Φ_1 can directly be used to define U_e coefficient of the window. Let us note that this coefficient

takes into account the heat flow recovered by air. That is the reason why Φ_I is strongly reduced with regard to classical window:

$$U_e = \frac{F_1}{(T_{int} - T_{ext})} \quad [3]$$

But if this method is used, it must be considered in the equation [1] that no heat recovery is done on building's air renewal and that air is blown with the exterior temperature. If the control zone 2 is considered, an interesting information can be obtained on air energy recovery. In this case, air enters the control zone with a temperature that is different from the outside temperature. Another U-value can be defined (called here U_{dyn}) that should be associated with a performance indicator linked to the supplied air (called R_{dyn}):

$$U_{dyn} = \frac{F_3}{(T_{int} - T_{ext})} \quad [4]$$

$$R_{dyn} = \frac{(T_{air\ blown} - T_{ext})}{(T_{int} - T_{ext})} \quad [5]$$

These indicators can be considered as the characteristics of a classical system equivalent to a window with a U-value U_{dyn} based on real surface heat loss and a heat recovery system on extracted air with a performance value R_{dyn} .

Numerical simulations

Model A : CFD code

Steady state numerical simulations of 2D laminar mixed or natural convection airflow in a Paziud® window are performed with the commercial CFD code Fluent®. Thermophysical properties of the fluid are supposed constant except the density (Boussinesq approximation) and are evaluated at the inlet fluid temperature ($T_{air\ inlet} = 0\text{ °C}$).

Spatial discretization of the governing equations is achieved by means of the finite volume method. Conservation equations are discretized with the first-order upwind scheme for the energy and momentum equations and with the second order upwind scheme for the pressure equation. Pressure-velocity coupling is solved with the SIMPLEC (SIMPLE-Consistent) algorithm. The radiation model used to solve the radiative transfer equation RTE equation is the Discrete Ordinates radiation model (DO) (Fiveland 1984). The discrete ordinates (DO) radiation model solves the RTE for a finite number of discrete solid angles, each associated with a vector direction fixed in the global Cartesian system.

The non uniform structured grid is composed of (44x722) cells refined close to the glasses and close to the inlet and outlet. The independence of the converged numerical solution versus the grid has been previously studied.

Model B: Model adapted to the thermal building behavior calculation

The model A that uses CFD helps us to understand physical phenomena inside the window. However, it is not reasonably usable to study global thermal behavior of a building in a dynamic simulation. That is the reason why a second model was developed. In this model,

energy balances are realized at a higher level. Figure 3 represents model B with an electrical analogy.

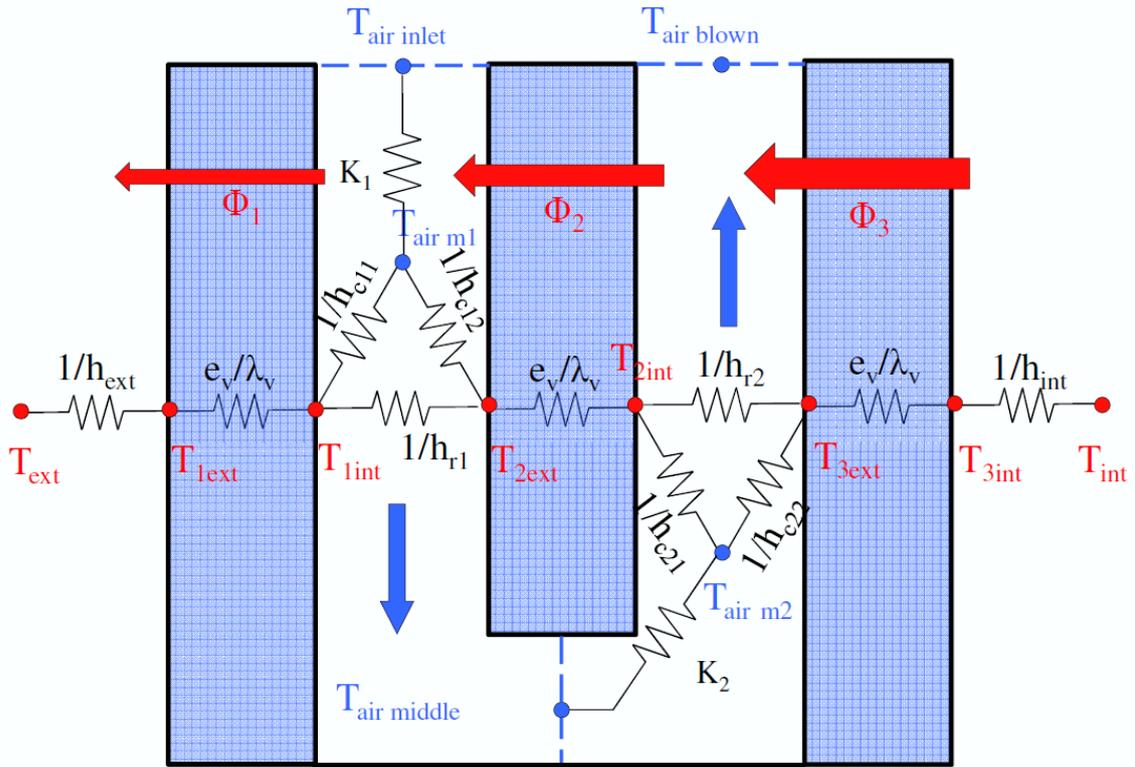


Figure 3. Schematic diagram of model B presented with electrical analogy

The representation of such an air gap is quite classical (Fraisse et al. 2006, Klein 1998). Let us note that here, the thickness of the air gap is about 1 cm and that it is not evident to use the concept of convection heat transfer coefficient at this scale because every point of the air gap belongs to the thermal boundary layer. The second strong hypothesis that will be verified using the model A is to consider that surface temperatures of glasses are uniform. With these two hypotheses, it is possible to solve analytically the thermal problem of airflow between two parallel surfaces with different temperatures (T_1 and T_2) and different heat transfer coefficients (h_1 and h_2). An enthalpy balance carried out on a thin layer of dz height and w width leads to the equation:

$$\dot{m} \cdot C_{p_{air}} \cdot \frac{dT_{air}(z)}{dz} + w \cdot (h_1 + h_2) \cdot T_{air}(z) = w \cdot (h_1 \cdot T_1 + h_2 \cdot T_2) \quad [6]$$

This is a first order differential equation with constant coefficients. The solution is:

$$T_{air}(z) = T_{\infty} \cdot \left(1 - e^{-\frac{z}{z_c}} \right) + T_{air\,inlet} \cdot e^{-\frac{z}{z_c}} \quad [7]$$

with:

$$T_{\infty} = \frac{h_1 \cdot T_1 + h_2 \cdot T_2}{h_1 + h_2} \quad \text{and} \quad z_c = \frac{\dot{m} \cdot C_p}{w \cdot (h_1 + h_2)} \quad [8]$$

T_∞ is the temperature that would be reached in an infinite long air gap and z_c is a characteristic length. If it is much greater than air gap height, temperature distribution is not far from an affine distribution. Knowing the air temperature distribution in the air gap, it is possible to determine the mean air temperature in the air space in order to evaluate heat exchange with surfaces:

$$\overline{T_{air}} = T_\infty + \frac{z_c}{z} \left(1 - e^{-\frac{z}{z_c}} \right) \cdot (T_{air\,inlet} - T_\infty) \quad [9]$$

This temperature is considered for convective exchanges with glass surfaces. The inlet air temperature of the second air space is taken equal to the outlet air temperature of the first one.

Radiative exchanges are considered between both surfaces with a view factor equal to 1 (boundary effects neglected) using radiosity method with or without linearization. As glasses have a very low inertia regarding to the rest of the building, steady state heat transfer is considered.

Results and discussion

Case studied

First of all, results are presented for two different configurations. The first one corresponds to the case where air enters the building exclusively due to the different air temperature between inside and outside (thermal draft). Pressure difference between inlet and outlet air vent is null. For the second one, a pressure difference of 4 Pa is imposed between inlet and outlet vent (reference value of French thermal regulation).

Then, for each model and each simulation, the outlet air temperature is presented for a pressure difference of 0 to 6 Pa (it corresponds to a volume flow of about 3 to 30 m³.h⁻¹).

It is important to remark that hc_{ij} coefficients of model B (see figure 3) are obtained by the results of model A. These values depend on the volume airflow and the surface considered. For example, in the simulation 3, under 4 Pa, in the second air space: $hc_{21}=2.6 \text{ W.m}^{-2}\text{.}^\circ\text{C}^{-1}$ and $hc_{22}=6.7 \text{ W.m}^{-2}\text{.}^\circ\text{C}^{-1}$.

Temperature distribution in the window

It was written above that in the model B, the surface temperature was considered as uniform (isotherm glass surface). Figure 4 shows that glass temperatures calculated by model A are not uniform at all. Temperature variation can be of 6 to 7 degrees depending on the height considered.

In the model B, the results of calculation (based on uniform surface temperatures) show a decreasing exponential of air temperature as a function of the vertical position in the gap. For example, in the case of the first air gap, it appears that, even if this shape is observed with the model A at the beginning of the gap, an important decrease of temperature occurs near the outlet. This effect is not compatible with decreasing exponential form that tends to a constant value. It is due to the fact that at the bottom of the window, air temperature is nearly identical in both air spaces. The heat flow is consequently very low across the second glass and is particularly lower than heat flow across the first glass because air temperature difference is higher.

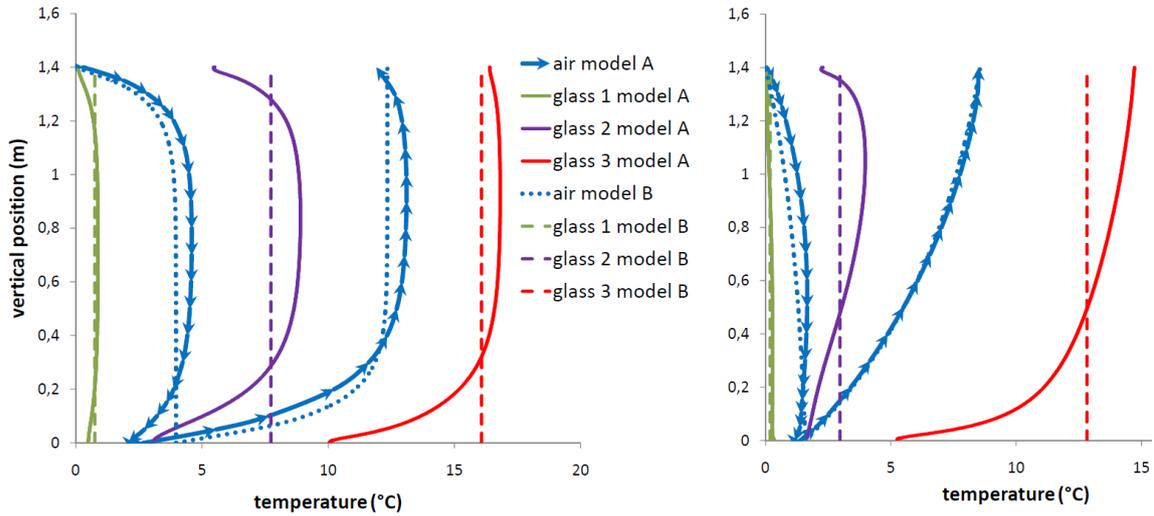


Figure 4: evolution of glasses and air temperatures as a function of height in the window (simulation 3): on the left, pressure difference is nil (volume airflow of $2.9 \text{ m}^3 \cdot \text{h}^{-1}$ due to natural convection); on the right, pressure difference: 4 Pa (volume airflow of $20.9 \text{ m}^3 \cdot \text{h}^{-1}$)

Figure 4 shows that model B does not reproduce correctly what happens physically due to the hypothesis of uniform temperature. It must be verified if this approximation has a significant effect on global energy balance and indicator defined.

Comparison of models and window performance

Figure 5 represents the outlet air temperature variation as a function of volume airflow and of the model used. It can be noticed that for simulations 1 and 3 (without radiation and low-e glasses with radiation), outlet air temperature is very close between both models (deviation about 0.2°C). For simulation 2, the deviation is greater (about 0.5°C). That is certainly due to inhomogeneity of radiation on the glass surfaces that is not at all considered in the model B. However, results have to be put in perspective because, for the moment, average convection heat transfer coefficients introduced in the model B are calculated in model A. Now, correlations have to be found to allow making calculation with model B without requiring CFD calculation.

Table 2 presents the window's performance indicators. It can be seen that the deviation between models for heat flow exchanged by the window between inside and outside environment is quite low (less than $1 \text{ W} \cdot \text{m}^{-2}$). These results comfort the idea to use model B for global simulation of buildings behavior.

The value of U_e calculated here is about $0.25 \text{ W} \cdot \text{m}^{-2} \cdot ^\circ\text{C}^{-1}$. It is three times less than a classical triple glass. Let us note that this value varies with the volume airflow across the window. If it is considered as constant during the simulation, the value of U_e can directly be used in the model in replacement of classical U-value. It can also be considered a U-value U_{dyn} and a heat recovery system on exhaust air (efficiency R_{dyn}). If volume airflow is not constant, model B has to be employed for the numerical simulations.

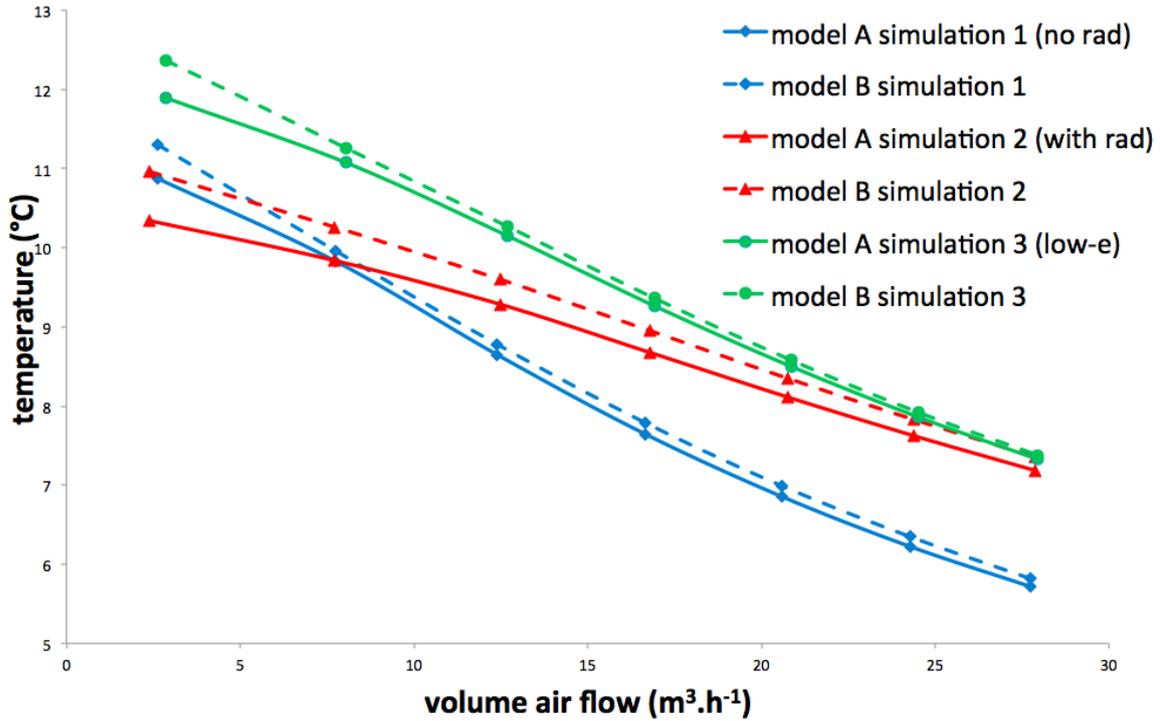


Figure 5: Outlet air temperature as a function of the volume airflow

Table 2: indicators of performance of the window for simulation 3 (low-e glasses) under 4 Pa ($20.9 \text{ m}^3 \cdot \text{h}^{-1}$)

model	$T_{air\ blown}$	Φ_1	Φ_3	U_e	U_{dyn}	R_{dyn}
A	8,50 °C	-5,2 W.m ⁻²	-49.8 W.m ⁻²	0,26 W.m ⁻² .°C ⁻¹	2,49 W.m ⁻² .°C ⁻¹	42,5 %
B	8,58 °C	-4,4 W.m ⁻²	-49,6 W.m ⁻²	0,22 W.m ⁻² .°C ⁻¹	2,48 W.m ⁻² .°C ⁻¹	42,9 %

Conclusion and perspectives

This numerical study has shown that a simple model based on global balances for a supply air window (model B) brings out similar results when using a commercial CFD code (model A) concerning magnitudes used in the thermal balance (heat fluxes across panes of glass and outlet temperature).

One needs to put these results into perspective; it is not yet possible to do without the Fluent® code (model A) because it helps us to “feed” the model B for the determination of the heat transfer convective coefficients along the panes of glass. In a symmetric heating, convective coefficients are analytically determined. In our problem, each air gap is submitted to different thermal boundary conditions (heat flux and temperature) according to the studied wall. A part of the heat flux transmitted by one of the glass surfaces is recovered by the air whereas the other part is escaping from the other glass surface. Correlations exist for the calculus of convective heat transfer coefficients in the case of an asymmetric heating, but it is very difficult to find some corresponding to this particular case. Research is achieved to solve this problem. This study has permitted to define

performance factors for the supply air window and to establish its efficiency in the heat recovering of renewal pre-heating air.

Afterwards, we have to work on the determination of convective coefficients inside air gaps without using CFD simulation results and also on the experimental validation of the numerical models. To do this, a window has been placed between two climatic cells in our laboratory; the instrumentation is achieved to verify the thermal balances and also the temperature evolutions within the air gaps. Later, the solar radiation will be taken into account on the window performances, particularly for the air pre-heating.

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