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Modeling a triple-glazed supply-air window

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Abstract

The supply-air window is a passive system of heat recovery contributing to the building ventilation. It allows the air renewal to circulate between glasses before entering the inside environment. Based on this principle, a part of heat transfer through the glasses is recovered by the airflow. In order to study the thermal performance of this window, Computational Fluid Dynamics (CFD) simulations are firstly conducted. Then, this article proposes simplified models which can be implemented easily in building simulation codes. The models are based on analytical solution of the problem of air circulating between isothermal panes differentially heated. It has been carried out in Modelica language using the Buildings library. A special attention is paid to the convective heat transfer coefficient between air space and glasses. The results obtained from these simplified models are compared to those obtained from the CFD model.

Keywords: ventilated window, performance indicators, convective coefficient, numerical modeling, radiation, building simulation

1. Introduction

From a thermal point of view, windows constitute one of the most fragile areas in buildings between the inside and the outside environments. Windows are integrated within the building envelope, but contrary to the other parts of the envelope, windows transmit light and provide a quite lower resistance to heat flux. 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 [1]. Several studies are conducted

to characterise the thermal performances of windows and their influence on thermal comfort [2]. The double-glazed window is the most commonly used system today in buildings and its energy efficiency is often improved by using low-emission coating [3] that reduces radiation heat losses. The use of absorbing gases (such as Argon) filling the gap between the panes of glasses (instead of the air) is also a technical solution present in the market [4]. However, a lot of innovative systems are under investigation and can improve the energy efficiency window. One way is to include transparent fins in the gap between the panes of double glazed window in order to limit the convective movements and thus reduce heat transfer through the window [5]. Another innovative glazing system is the incorporation of silica aerogel between the panes of glasses instead of air. This material presents better performances in terms of acoustic and thermal insulation and light transmission [6]. The ideal window should both preserve thermal comfort and take advantage of solar heat when needed or rather protect building zone from overheating. This can be achieved with thermotropic windows that can change their light transmission behavior [7] or with enclosed pleated blind integrated in the window [8]. Another improvement is to use the window for ensuring the ventilation in buildings. That is the purpose of this article.

A supply air window is different from a standard window in terms of flow convection generated within the gaps between the panes of glass. Airflow inside the window is caused by buoyancy force and eventually by a depressurization in the building performed with an air exhaust system. The convection can be natural or forced depending on the power of the ventilation system. The system is passive in this study but a fan can be integrated in the window to generate a more important airflow. The supply air window have shown great potential to perform energy efficiency in building in cold and temperate climate [9–11]. Thus, in winter, when the outdoor temperature is lower than indoor, a large part of the heat losses through the window is recovered providing the pre-heating of the incoming air. Since the installation of heat recovery ventilation system within the retrofit context is complex, the ventilated window presents an interesting alternative. It is easy to install and provides a non-negligible heat recovery. The ventilated window can also be used to extract air from the building to the outdoor space (exhaust air window). In this case, the window is interesting for hot climates by protecting the

indoor space from overheating [12]. The two modes, supply and exhaust, can also coexist simultaneously in a same window. The dual-airflow system investigated by Gosselin *et al* [13,14] works like a crossflow heat exchanger and is particularly efficient during the heating season in terms of energy efficiency and indoor air quality [15], however the presence of two fans in the window is necessary. Another interesting innovative window is proposed by Leal *et al* [16]. The system is based on the use of a rotating absorptive glazing that can be placed at the outdoor or indoor side of a conventional window. A natural airflow is thus generated and improves energy efficiency both in cooling and heating seasons. For these configurations of window, there are not air exchanges between the inside and outside environments; thus the system is not involved in the building ventilation.

Most of the building thermal performance softwares use a multizonal approach that considers one calculation node per zone for air temperature determination. Consequently, a building is represented by a few calculation nodes (less than 20 most of the time). Heat transfer in walls is often implemented with a finite difference method also with a limited number of nodes (less than 10 usually) or a method using a model reduction (i.e. z-transform) that is more efficient in terms of computing time. For long-wave radiative transfers, simplifications are often done on view factor calculation and on linearization of heat transfer equations in order to reduce the number of equations [17]. All these efforts allow performing building simulation with a standard office computer in a reasonable CPU time. For modeling an airflow window in a building simulation code, implementation of a detailed CFD model is not advisable due to the computing time. That is the reason why it is interesting to develop simpler models based on a physical approach that may be less time consuming. The problem resides in the need of parameters that may only be obtained from experiments or from CFD simulations prior to developing simple models.

In this work, CFD steady-state numerical simulations are first conducted to provide precalculation windows parameters which will be used subsequently in simpler models of building dynamic simulations. In order to take into account a ventilated window in building dynamic simulations, several phenomena have to be modelled:

- Conductive heat transfer in the panes of glass;

- Convection and long wave radiation between the inner glass and the inside environment, between the outer glass and outside environment and also between the panes of glass;
- Convective heat transfer between the panes of glass and airflow, which is the most the most difficult phenomenon to model due to the unknown convective coefficients in air space;
- Solar radiation absorbed by the panes of glass.

The existing models for ventilated windows simulate the system performance by considering an energy balance in the window [18–22]. They differ from each other in the determination of zones in which an energy balance is performed, in the method to obtain the exchange rate in the air space and in the initial choice of flow characteristics (natural, mixed or forced convection, laminar or turbulent, steady or unsteady). It is worth noting that the convection nature depends on the environment and window's configuration, however the laminar condition is proved optimal to obtain the best performance [23].

The first author who developed a numerical model is J.L. Wright [18,19], considering a laminar, steady and fully developed airflow in forced convection conditions. Glass panes temperatures were assumed isothermal. Thus, in this study, a ventilated window with 2 panes of glasses was composed of 5 nodes; where 2 of them represented the indoor and outdoor temperatures, 2 others were the panes of glass temperatures and the last one was the airflow cavity. In a similar manner, Ismail *et al* [20] proposed in their study to divide the window in the flow direction into n layers of 5 cm thickness. Consequently, they obtained n times more nodes than Wright's system. Mc Evoy *et al* [21] assumed isothermal the glazing temperatures but simulated the air cavity with 2x4 zones.

In the Wright's study [18], the heat transfer between air and glass was obtained by expressing the two dimensional temperature solution in terms of Stieljes integrals [24]. This approach consisted in defining the performance's system beforehand. Then the window could be taken into account in a building simulation for a limited number of performance criteria. Consequently, the problem solving in the building and in the window was not simultaneous and the Wright's model would be adapted to building simulation only if the defined performance criteria are constant, that is not the case for a ventilated window. A non-sequential method seems necessary but supposes to model the heat transfer

between airflow and panes of glass with a different approach. That was the choice of Carlos *et al* [22] who used correlations defined in ISO standard [25] and based on a Wright's study [26]. This model assumed that the heat received by the airflow from the "warm" pane of glass and the heat received by the "cool" pane of glass from the airflow are expressed with an identical heat transfer coefficient. This coefficient was calculated from air velocity and another convective coefficient obtained by correlations. This hypothesis was absolutely not justified in the case of asymmetrical heating in a rectangular duct.

An interesting study of the modification of this model was proposed by L.M. Raffnsøe [27]. In this article, a model that can be implemented easily in many building simulation codes is developed. The model can be used for every kind of supply-air windows and also for exhaust-air windows. The numerical results coming from this model are compared to those of a commercial CFD code. In this study, window's frame is not considered. For the sake of clarity, a precise example is presented here. It is the case of a Paziaud® window that is a triple-glazed window with a U-shaped air channel (Fig. 1). This window was designed at the beginning of the eighties by J. Paziaud and was used in many buildings in the northern of France. More than 1500 dwellings and a few tertiary buildings were equipped in the past 30 years. Few institutional studies have been achieved on triple-glazed windows. Kim and Yang [28,29] have investigated flow and heat transfer characteristics of a Paziaud window but only for the exhaust mode during the cooling season.

This paper is organized as follows. The studied Paziaud window designed for the present work is described in Section 2. Section 3 presents and comments the global thermal performance indicators used in this study. Then, Section 4 deals with both numerical approaches performed in this work. The obtained results are presented and commented in Section 5. In the last section, conclusions are drawn based on these numerical results.

2. Problem Formulation

A laminar convective airflow is considered inside the window. Air enters within the window at the external temperature set at 0 °C, circulates inside the U-shaped conduit 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 q_i ($\text{W}\cdot\text{m}^{-2}$). Horizontal surfaces are considered adiabatic (Fig.1).

The temperature difference between both environments is set at 20 °C. Concerning the boundary conditions of the external glazed surfaces, two heat transfer coefficients are defined, h_{int} and h_{ext} corresponding to the internal and external environments. Two radiative temperatures $T_{r\ int}$ and $T_{r\ ext}$ are also defined at these side walls. An incident solar radiation I_s can be also considered. The values of these parameters are defined in accordance with the ISO standard [25] and are presented in Figure 1.

In this study, the influence of pressure difference and solar radiation on heat fluxes crossing the different panes of glass as well as the air temperature entering the room, are investigated:

- The pressure difference is varied from 1 Pa to 8 Pa corresponding to an average velocity in the cavities from $0.25\ \text{m}\cdot\text{s}^{-1}$ to $1.1\ \text{m}\cdot\text{s}^{-1}$. The range is chosen so that the airflow is laminar ($\text{Re} < 2300$). The Reynolds number is based on the hydraulic diameter [30].
- The thermal behavior of the supply air window changes with the external environment. This paper focuses on winter conditions at low outside temperature. The presence or absence of solar radiation respectively to daytime and nighttime (or daytime with overcast skies) is investigated.

Let’s note that solar radiation is not directly simulated but treated from terms sources injected in each panes of glass. The absorption and transmission parameters of the global window are calculated from Window 7 software [31] and manufacturer’s parameters (Table 1).

3. Global Performance Indicators

In building simulations, two main indicators are used to characterize a window. Their definitions are given in an international standard [25] specifying the experimental conditions to obtain them. The first one, U -value, corresponds to the heat loss factor providing heat transfer through the window per square meter for a temperature difference of one degree between inside and outside temperatures. The second one, τ_s (also called g according to [32] or $SHGC$ in United States), is the solar heat gain

coefficient which indicates the rate of solar radiation impacting the heat balance indoor. It includes the direct transmission τI_s and the part of solar radiation absorbed by the window and afterwards released inward $\alpha_{int} I_s$. Solar heat gain factor τ_s is often replaced by the shading coefficient SC which is the ratio of τ_s and a reference corresponding to the solar heat gain factor of a simple clear 3 mm thick glass ($SC = 1.15 SHGC$).

These coefficients can be adapted to the case of supply air windows. The problem is that the heat fluxes across the different panes are not identical in a ventilated window. Wright [18] defined an effective U -value called U_e that can be evaluated in the same test conditions than those of the standard considering the heat flux on the exterior pane of the window q_1 :

$$U_e = \frac{q_1}{(T_{int} - T_{ext})} \quad (1)$$

Another U -value can be defined (called thereafter U_{vent}) considering the heat flux on the interior pane of the window q_3 . In this case, it should be associated with a performance indicator linked to the supplied air (called η_{vent}) :

$$U_{vent} = \frac{q_3}{(T_{int} - T_{ext})} \quad (2)$$

$$\eta_{vent} = \frac{(T_{air\ blown} - T_{ext})}{(T_{int} - T_{ext})} \quad (3)$$

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

Wright [18] also defined an effective shading coefficient SC_e . Indeed the airflow in the window increases the part of solar radiation absorbed by the window and released inward. To calculate τ_s in the case of the supply-air window, three contributions must be considered: the direct transmission τI_s , the part of solar radiation absorbed by the internal pane of glass and afterwards released inward by convection and long-wave radiation $\alpha_{int} I_s$ and the part of solar radiation recovered by air $\alpha_{air} I_s$:

$$\tau_S = \frac{\tau I_S + \alpha_{\text{int}} I_S + \alpha_{\text{air}} I_S}{I_S} \quad (4)$$

The four parameters U_e , U_{vent} , η_{vent} and τ_S constitute a set of performance indicators to characterize ventilated windows. A similar approach is employed by Carlos [33].

4. Modeling Approach

In order to solve this problem, two different approaches are considered. The window presented in this study was first modelled using CFD. All boundary conditions are specified in Figure 1. This model uses the commercial software Fluent® [34]. It allows to better understand the window dynamics and to know the air temperature variations and the velocity profiles in the window. However, this numerical approach is time consuming to be used in global building annual simulations. That is the reason why a second model approach Adapted to Building Simulation (ABS) is developed.

4.1. CFD Calculations

A computational model has been developed using the commercial CFD software. Steady state numerical simulations of a 2D laminar convective airflow in a Paziaud® window are performed.

Thermophysical properties of the fluid are supposed depending on the temperature and are in accordance with the ISO standard [25]. Fluid properties are taken as polynomial functions and the incompressible ideal-gas law is applied to account for density variation.

A non uniform structured mesh composed of 79 216 cells refined close to the glasses and close to the inlet and outlet is found to be sufficient for this study. A grid independence study has been undertaken to ensure the adequacy of this mesh density.

Spatial discretization of the governing equations is achieved by means of the finite volume method using the pressure-based solver. The equations discretization was carried out using second order schemes for pressure and momentum (upwind scheme) and first order upwind schemes for energy and radiation. Momentum and pressure-based continuity equations are solved simultaneously with the coupled algorithm. The radiation model used to solve the longwave radiative transfer is the Discrete

Ordinates radiation model (DO) [35]. Convergence criteria were set at 10^{-6} for continuity, 10^{-7} for energy and 10^{-8} velocity and radiation.

4.2. ABS Model (Adapted to Building Simulation)

The model is quite similar to the one presented in the standard [25]. It is based on a thermal resistance network and energy balances are carried out [36] (Fig. 2). It is based on two main assumptions. The first one considers that surface temperatures of glasses are uniform. The second one stands for the method in which both convective coefficients h_{cij} in each air gap are determined. Concerning this point, the model differs from the standard.

The main difficulty of this problem is taking into account the ventilated air spaces. Indeed, the modeling approach proposed in the ISO standard [25] considers one dimensional heat transfer in the glasses where temperatures are uniform. Consequently, temperature behavior in the ventilated air gap is analytically calculated considering a fixed and uniform convective heat transfer coefficient at the glass-air interface. The air temperature evolution obtained is a decreasing exponential as a function of the height in the air gap. This model was questioned by Raffnsøe [27], concerning the two convective heat transfer coefficients. In the standard, the same value h_{cv} is set on the both side of the air gap using the formula :

$$h_{cv} = 2h_{cdv} + 4V \quad (5)$$

where V is the mean air velocity and h_{cdv} is the convective heat transfer coefficient obtained from a correlation in the case of non ventilated air space [26]. Raffnsøe showed that taking the same coefficients on both sides of the air space leads to incorrect results. This has been confirmed by our CFD simulations (section 5.2).

This is why, the heat transfer coefficients h_{cij} in our problem were calculated using the results of CFD numerical simulations. Let us note that, in each air gap, 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. It is now possible to solve analytically the thermal problem of airflow between two parallel

surfaces. An enthalpy balance carried out on a thin layer of dz height and l width leads to a first order differential equation with constant coefficients. For the air gap 1, the enthalpy balance equation is :

$$\dot{m}C_p \frac{dT_{air1}(z)}{dz} + l(h_{c11} + h_{c12})T_{air1}(z) = l(h_{c11}T_{int1} + h_{c12}T_{2ext}) \quad (6)$$

The result is a decreasing exponential:

$$T_{air1}(z) = (T_{air\ inlet} - T_{\infty1})e^{-\frac{l(h_{c11}+h_{c12})z}{\dot{m}C_p}} + T_{\infty1} \quad (7)$$

where :

$$T_{\infty1} = \frac{h_{c11}T_{int1} + h_{c12}T_{2ext}}{h_{c11} + h_{c12}} \quad (8)$$

Knowing the air temperature distribution in the air gap, it is possible to determine the mean air temperature in the total air space :

$$T_{air\ m1} = \frac{\dot{m}C_p}{(h_{c11} + h_{c12})l.H} (T_{air\ inlet} - T_{\infty1})e^{-\frac{(h_{c11}+h_{c12})l.H}{\dot{m}C_p}} + T_{\infty1} \quad (9)$$

This mean temperature is used for the determination of convective heat transfer between air and glass surfaces. The inlet air temperature of the second air space is taken equal to the outlet air temperature of the first one.

The first method employed to solve this problem consists of using convective coefficients precalculated by CFD labelled as ‘‘ABS1 model’’ throughout the rest of document. The problem is that it requires using a CFD calculation for each geometrical configuration. Another way is to use heat convective coefficients issued from correlations. Despite an in-depth investigation in the literature, no correlation was found that leads to different convective coefficients in both sides. The only way found to reveal this difference is to use an analytical result presented in Shah and London [30]. They proposed a relationship between the Nusselt number and the ratio of convective heat fluxes calculated on both surfaces. It is an analytical result obtained by considering constant and uniform heat flux on air gap boundaries. For the air gap 1, we have:

$$Nu_{11} = \frac{h_{c11} \cdot 2e_1}{\lambda} = \frac{140}{26 - 9 \frac{q_2}{-q_1}} \quad (10)$$

$$Nu_{12} = \frac{h_{c12} \cdot 2e_1}{\lambda} = \frac{140}{26 - 9 \frac{q_1}{-q_2}} \quad (11)$$

The use of these equations makes the solving of the problem independent to CFD calculation. This method will be labelled “ABS2 model” throughout the rest of document.

Both models (ABS1 and ABS2) are implemented in Modelica object oriented language using Buildings library [37]. Radiation model is directly managed by this library. Long wave radiation is considered using the radiosity model with surface factors equal to 1 in the air gaps. Short wave radiation model uses Tarcog [31] algorithm that is the same than the one used in Window 7 software. The model is then divided in sub-models containing equation systems that can be connected to existing models (Fig. 3). In the case shown in this work, for example, the model of glass layer comes from buildings library and the model of ventilated air space was developed for this particular case. We can see here that Modelica allows making other models easily with only one ventilated air space just using a graphical environment (Fig. 4).

5. Results

5.1. Air Temperature Evolution in the Air Gap

First of all, the reliability of the assumptions of both ABS models is studied. To achieve this, a comparison between CFD and ABS1 models is carried out.

Recall that in the ABS approach, the surface temperature is isothermal. Figure 5 shows that the glass temperatures calculated by the CFD model are not uniform along the walls. Temperature variation can be about 6 or 7 degrees depending on the height considered. In the ABS1 model, the results of calculation (based on uniform surface temperatures) show a decreasing exponential shape of air temperature as a function of the vertical position in the gap. If the ABS1 model does not reproduce correctly what happens physically in the window due to the hypothesis of uniform temperature, the

shape of air temperature is quite close to the CFD results. Consequently, it must be verified if this approximation has a significant effect on global energy balance and on the indicators defined in this study.

Table 2 presents the window's performance indicators. It can be seen that the deviation between models for heat flux exchanged by the window between inside and outside environment is quite low (less than 1 W.m^{-2}). Moreover, the ABS1 model gives accurate results in terms of outlet air temperature and global indicators. These conclusions strengthen the idea to use the ABS approach for global simulations of building behavior.

5.2. Convective heat transfer coefficients

It is interesting to look at the results of convective heat transfer coefficients h_{cij} obtained by CFD simulations (Fig. 6).

Firstly, the CFD model used to calculate the h_{cij} coefficients provides negative values for high airflow rate. Even if it has no physical attributes to define a negative h_{cij} coefficient, it can be explained by the fact that convective heat transfer on the internal surface of the medium glass reaches a negative value due to radiative heat transfer. This phenomenon does not occur when radiative heat transfer is not taken into account. Figure 7 shows the evolution of temperature in a horizontal section of the window. We can see that the temperature profile in the second air gap decreases close to the medium glass before increasing. The decreasing shows that the medium glass loses energy by convection in the second air gap even if mean air temperature in this air gap can be higher than the surface temperature.

Secondly, Figure 6 also shows the evolution of the convective coefficients h_{cij} calculated by the model ABS2 using equations (10-11). One can see that for each air gap, a convective heat transfer coefficient increases and the other one decreases versus airflow rate. However, the values of the calculated coefficients are quite far away from CFD values. Consequently, this relation is not very satisfying to correctly calculate h_{cij} coefficients. We will study in the rest of the paper the effect on global performance results on the window.

5.3. Global indicators

Figure 8 shows global performance indicators of the window defined previously for the three models. As it can be seen previously, global indicators calculated from the CFD model and ABS1 model are very close to each other for any airflow rate. However, the differences are more important with the ABS2 model. Concerning the indicator U_e , these differences seem to be quite negligible. For low airflow rate, the difference between ABS2 and CFD values is higher than for a high airflow rate. It seems that considering that h_{cij} coefficients reach a value close to zero on the most external surface of the air gap is quite a good approximation (Fig. 6). That is to say that for high airflow rate, most of the heat losses of the window is due to long wave radiation. Concerning the parameters U_{vent} and η_{vent} , the higher the airflow rate is, the difference of the results coming from ABS2 and CFD results becomes more important. That means that for high airflow rate, the heat flux between the inside environment and the window and the heat flux exchanged by the air are underestimated by the ABS2 model. The reason can be found at the entrance of each air space. Indeed, in this zone, the airflow is not yet stabilized and the CFD model shows the presence of swirls. These structures that are amplified by the airflow, are responsible of an accentuation of the heat transfer between the glass panes and the airflow. In the model ABS1, these phenomena are taken into account through the calculation of each convective coefficient h_{cij} with the CFD model whereas in the ABS2 model, the convective coefficients h_{cij} are determined considering the airflow is yet stabilized.

Furthermore, simulations taking into account solar radiation are carried out in order to determine the differences of behavior observed between the ABS2 and the CFD models. Boundary conditions of the ISO Standard [25] are chosen, such as the temperature difference between inside and outside was set at 20°C and the normal direct radiation is set at 300 W.m⁻². For this comparison, the total solar energy transmittance τ_S was calculated as a function of the airflow rate (Fig. 9).

This figure shows that the ABS2 model reproduces the thermal behavior of the window in a better manner, even when solar radiation is considered. Note that total solar energy transmittance increases with the airflow volume. It is a good point in winter period. It is interesting to know what kind of heat

transfer brings about this increase. Figure 10 shows the repartition between different heat transfer modes. The short wave radiation remains constant because the value of solar transmission coefficient does not change with airflow. The sum of conduction and airflow specific enthalpy increases with volume airflow because the solar radiation absorbed in each glass is better recovered with an important airflow. Less energy is transmitted outside.

6. Conclusion

The CFD simulations conducted in this study show the interest of such a window where effective U -value is very low (between 0.1 and 0.5 depending on the airflow rate) in comparison to a classical triple glazing window ($U = 1.1 \text{ W.m}^{-2}.\text{°C}^{-1}$). The efficiency of the window for heat loss and solar factor increases with the airflow rate. However, these results should be put into perspective as the difference between experiments and numerical simulations for the ventilated window are often non negligible [10,13,16]. In a future investigation, it will be necessary to get experimental data to calibrate the simplified model in order to validate it. Moreover recall that the tri-dimensional geometry and the heat transfer through the frame of the window are not considered here.

There are three different methods to consider a supply air window in a thermal building simulation (Fig. 11). The first one is to consider the effective U -value and the solar factor defined by Wright in 1985. These values have to be precalculated or measured. The problem is that it can be used only for a constant airflow rate and that it leads to a high value of pane surface temperature inside that can cause problem for comfort calculation. The second one is close to the first one. It consists of considering U_{vent} and η_{vent} . This method gives a large pane surface temperature outside that can cause problem for infrared radiative calculations. The third one is to implement a simplified model with a reduced number of equations.

This paper proposes two simplified models based on global heat balance and on analytical solution including calculation of convective heat transfer coefficients in air gaps. The first one (labelled as “ABS1 model”) consists of using convective coefficients precalculated by CFD. It yields similar results when using a commercial CFD code, concerning outlet temperature and performance

indicators. These results show therefore that the adopted modeling approach is relevant. The second model (labelled as “ABS2 model”) is developed in order to avoid the dependency on CFD precalculations. The model uses correlations to determine the convective heat transfer coefficients. The results of this second model on global evaluation of the window are less satisfactory than the ABS1 model for an airflow rate higher than $15 \text{ m}^3 \cdot \text{h}^{-1}$. However, the results accuracy remains acceptable in a building simulation.

Nomenclature

C_P	specific heat capacity of the air, $\text{J} \cdot \text{kg}^{-1} \cdot ^\circ\text{C}^{-1}$
C_{Pg}	specific heat capacity of the glass, $\text{J} \cdot \text{kg}^{-1} \cdot ^\circ\text{C}^{-1}$
e_i	width of the air space i , m
e_{gi}	width of the pane of glass i , m
$h_{c\ ij}$	convective coefficient (see Fig.2), $\text{W} \cdot \text{m}^{-2} \cdot ^\circ\text{C}^{-1}$
h_{ext}	convective coefficient of the outside environment, $\text{W} \cdot \text{m}^{-2} \cdot ^\circ\text{C}^{-1}$
h_{int}	convective coefficient of the inside environment, $\text{W} \cdot \text{m}^{-2} \cdot ^\circ\text{C}^{-1}$
h_{gext}	global (radiative and convective) heat transfer coefficient of the outside environment, $\text{W} \cdot \text{m}^{-2} \cdot ^\circ\text{C}^{-1}$
h_{gint}	global (radiative and convective) heat transfer coefficient of the inside environment, $\text{W} \cdot \text{m}^{-2} \cdot ^\circ\text{C}^{-1}$
$h_{r\ i}$	radiative heat transfer coefficient between the two panes of glass in the air space i , $\text{W} \cdot \text{m}^{-2} \cdot ^\circ\text{C}^{-1}$
H	window height, m
I_s	solar radiation, $\text{W} \cdot \text{m}^{-2}$
l	window width, m
m	mass airflow rate, $\text{kg} \cdot \text{s}^{-1}$
Nu_{ij}	Nusselt number associated to the convective coefficient $h_{c\ ij}$
P_{atm}	atmospheric pressure, Pa
q_i	heat flux through the glass i , $\text{W} \cdot \text{m}^{-2}$
$q_{exh\ i}$	heat flux recovered by the airflow in the air space i , $\text{W} \cdot \text{m}^{-2}$
Re	Reynolds number
$T_{air\ i}(z)$	air bulk temperature in the air space i at the z -position, $^\circ\text{C}$
$T_{air\ mi}$	mean bulk temperature of the air in the air space i , $^\circ\text{C}$
$T_{air\ inlet}$	air temperature at the inlet of the window, $^\circ\text{C}$
$T_{air\ blown}$	air temperature at the outlet of the window, $^\circ\text{C}$

$T_{i\ ext}$	mean surface temperature of the pane of glass i oriented to the outside environment, °C
$T_{i\ int}$	mean surface temperature of the pane of glass i oriented to the inside environment, °C
T_{ext}	temperature of the outside environment, °C
T_{int}	temperature of the inside environment, °C
$T_{r\ ext}$	radiative temperature of the outside environment, °C
$T_{r\ int}$	radiative temperature of the inside environment, °C
U	heat transfer coefficient for a conventional window
U_e	heat transfer coefficient for a ventilated window (see equation 1), $\text{W}\cdot\text{m}^{-2}\cdot\text{°C}^{-1}$
U_{vent}	heat transfer coefficient for a ventilated window (see equation 2), $\text{W}\cdot\text{m}^{-2}\cdot\text{°C}^{-1}$
SC	shading coefficient for a conventional window
SC_e	shading coefficient for a ventilated window
V	mean air velocity, $\text{m}\cdot\text{s}^{-1}$

Greek symbols:

α_{air}	rate of solar radiation absorbed by the window and recovered by the airflow for a ventilated window
α_{int}	rate of solar radiation absorbed by the window and released inwards
ΔP	pressure difference between the inlet and the outlet of the window, Pa
η_{vent}	heat recovery coefficient (see equation 3)
λ	thermal conductivity of the air, $\text{W}\cdot\text{m}^{-1}\cdot\text{°C}^{-1}$
λ_g	thermal conductivity of the glass, $\text{W}\cdot\text{m}^{-1}\cdot\text{°C}^{-1}$
ρ	density of the air, $\text{kg}\cdot\text{m}^{-3}$
ρ_g	density of the glass, $\text{kg}\cdot\text{m}^{-3}$
τ	transmission coefficient of the window
τ_s	solar heat gain coefficient of the window

Subscripts / superscripts:

ABS	Adapted to Building Simulation
CFD	Computational Fluid Dynamics

Acknowledgement

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References

- [1] Catalina T, Virgone J, Roux J-J, Blanco E. Effet de l'inertie thermique, de la surface vitrée et du coefficient de forme sur les besoins en chauffage d'une habitation. Congrès IBPSA, Lyon, France: 2008.
- [2] Cappelletti F, Prada A, Romagnoni P, Gasparella A. Passive performance of glazed components in heating and cooling of an open-space office under controlled indoor thermal comfort. *Building and Environment* 2014;72:131 – 144. doi:http://dx.doi.org/10.1016/j.buildenv.2013.10.022.
- [3] Mohelnikova J. Materials for reflective coatings of window glass applications. *Construction and Building Materials* 2009;23:1993 – 1998. doi:http://dx.doi.org/10.1016/j.conbuildmat.2008.08.033.
- [4] Ismail KAR, Salinas CT, Henriquez JR. A comparative study of naturally ventilated and gas filled windows for hot climates. *Energy Conversion and Management* 2009;50:1691–703. doi:10.1016/j.enconman.2009.03.026.
- [5] Giorgi LD, Bertola V, Cafaro E. Thermal convection in double glazed windows with structured gap. *Energy and Buildings* 2011;43:2034 – 2038. doi:http://dx.doi.org/10.1016/j.enbuild.2011.03.043.
- [6] Cotana F, Pisello AL, Moretti E, Buratti C. Multipurpose characterization of glazing systems with silica aerogel: In-field experimental analysis of thermal-energy, lighting and acoustic performance. *Building and Environment* 2014;81:92–102. doi:10.1016/j.buildenv.2014.06.014.
- [7] Yao J, Zhu N. Evaluation of indoor thermal environmental, energy and daylighting performance of thermotropic windows. *Building and Environment* 2012;49:283 – 290. doi:http://dx.doi.org/10.1016/j.buildenv.2011.06.004.
- [8] Dalal R, Naylor D, Roeleveld D. A {CFD} study of convection in a double glazed window with an enclosed pleated blind. *Energy and Buildings* 2009;41:1256 – 1262. doi:http://dx.doi.org/10.1016/j.enbuild.2009.07.024.
- [9] Barakat SA. Thermal performance of a supply-air window. *Proceedings of 12th Annual Passive Solar Conference*, vol. 12, Portland, Oregon, USA: 1987, p. 152–8.
- [10] Southall RG, McEvoy ME. Investigations into the functioning of a supply air window in relation to solar energy as determined by experiment and simulation. *Solar Energy* 2006;80:512–23. doi:10.1016/j.solener.2005.04.016.
- [11] Baker PH, McEvoy ME. Test cell analysis of the use of a supply air window as a passive solar component. *Solar Energy* 2000;69:113–30. doi:10.1016/S0038-092X(00)00048-7.
- [12] Haddad J, Elmahdy AH. Comparison of the thermal performance of an exhaust-air window and a supply-air window. *ASHRAE Transactions* 1999;105:1–9.
- [13] Gosselin JR, Chen Q (Yan). A computational method for calculating heat transfer and airflow through a dual-airflow window. *Energy and Buildings* 2008;40:452–8. doi:10.1016/j.enbuild.2007.03.010.
- [14] Gosselin J., Chen Q. A dual-airflow window for indoor air quality improvement and energy conservation in buildings. *HVAC&R Research*, Submitted for Publication n.d.
- [15] Wei J, Zhao J, Chen Q. Energy performance of a dual airflow window under different climates. *Energy and Buildings* 2010;42:111–22. doi:10.1016/j.enbuild.2009.07.018.
- [16] Leal V, Maldonado E. The role of the PASLINK test cell in the modelling and integrated simulation of an innovative window. *Building and Environment* 2008;43:217–27. doi:10.1016/j.buildenv.2006.10.025.
- [17] Peuportier B. *Eco-conception des bâtiments et des quartiers*. Ecole Des Mines de Paris: Mines ParisTech Les Presses; 2008.
- [18] Wright JL. Effective U-values and shading coefficients of preheat/supply air glazing systems. *Proc. Solar Energy Society of Canada*, Winnipeg, Canada: 1986, p. 219–24.
- [19] Ferguson JE, Wright JL. *VISION: A Computer Program to Evaluate the Thermal Performance of Super Windows*. Report No Passive-10, NRC, Canada 1984.
- [20] Ismail KAR, Henriquez JR. Simplified model for a ventilated glass window under forced air flow conditions. *Applied Thermal Engineering* 2006;26:295–302. doi:10.1016/j.applthermaleng.2005.04.023.
- [21] McEvoy ME, Southall RG, Baker PH. Test cell evaluation of supply air windows to characterise their optimum performance and its verification by the use of modelling techniques. *Energy and Buildings* 2003;35:1009–20. doi:10.1016/S0378-7788(03)00042-2.
- [22] Carlos JS, Corvacho H, Silva PD, Castro-Gomes JP. Modelling and simulation of a ventilated double window. *Applied Thermal Engineering* 2011;31:93–102. doi:10.1016/j.applthermaleng.2010.08.021.
- [23] Tjelflaat PO, Bergensen B. Improved thermal insulation in windows by laminar air flows. *Thermal Performance of the Exterior Envelopes of Buildings III* 1985.
- [24] Cess RD, Shaffer EC. Laminar heat transfer between parallel plates with an unsymmetrically prescribed heat flux at the walls. *Applied Scientific Research* 1960;9:64–70.
- [25] ISO-15099. *Thermal performance of windows, doors and shading devices - detailed calculations*. 2003.
- [26] Wright JL. A Correlation to Quantify Convective Heat Transfer Between Vertical Window Glazings. *ASHRAE Transactions* 1996;102:940–6.
- [27] Raffnsøe LM. *Thermal Performance of Air Flow Windows*. Master Thesis. Danmarks Tekniske Universitet, 2007.
- [28] Kim M-H, Oh C-Y, Hwang J-H, Choi H-W, Yang W-J. Thermal performance of the exhausting and the semi-exhausting triple-glazed airflow windows. *International Journal of Energy Research* 2006;30:177–90. doi:10.1002/er.1149.
- [29] Kim M-H, Yang W-J. An optimum design on the triple-glazed exhaust airflow window. *International Journal of Energy Research* 2002;26:355–64. doi:10.1002/er.789.
- [30] Shah RK, London AL. *Laminar flow forced convection in ducts: a source book for compact heat exchanger analytical data*. Academic Press. New-York: 1978.

- [31] Tarcog (Carli, Inc.). TARCOG: Mathematical models for calculation of thermal performance of glazing systems with or without shading devices. 2006.
- [32] NF EN 410. Verre dans la construction - Détermination des caractéristiques lumineuses et solaires des vitrages. AFNOR; 2011.
- [33] Carlos JS, Corvacho H. Evaluation of the thermal performance indices of a ventilated double window through experimental and analytical procedures: U_w -values. *Renewable Energy* 2014;63:747 – 754. doi:<http://dx.doi.org/10.1016/j.renene.2013.10.031>.
- [34] Fluent. UG, Fluent Inc, Fluent 6.3 User's Guide, Lebanon, NH 2006.
- [35] Fiveland W. Discrete-ordinates solutions of the radiative transport equation for rectangular enclosures. *Journal of Heat Transfer* 1984;106:699–706.
- [36] Fraisse G, Johannes K, Trillat-Berdal V, Achard G. The use of a heavy internal wall with a ventilated air gap to store solar energy and improve summer comfort in timber frame houses. *Energy and Buildings* 2006;38:293–302. doi:10.1016/j.enbuild.2005.06.010.
- [37] Wetter M, Zuo W, Nouidui TS. Modelling of heat transfer in rooms in the Modelica “buildings” library. *Proceedings of building simulation 12th conference of international building performance simulation association*, Sydney, Australia: 2011, p. 1096–103.

Tables

Table 1. Radiation parameters of global ventilated window

	glass 1	glass 2	glass 3
manufacturer's reference	4mm Planibel Clearvision	6mm Planibel G pos.2	4mm Planibel G pos.2
emissivity(int/ext)	0.84/0.84	0.84/0.17	0.84/0.17
solar absorption	1.7%	20.8%	9.5%
solar transmission	45.75%		

Table 2. Indicators of performance of the window performed at 20.3 m³/h (4Pa)

model	$T_{air\ blown}$	q_1	q_3	U_e	U_{vent}	η_{vent}
CFD	6.32 °C	1.9 W.m ⁻²	64.6 W.m ⁻²	0.09 W.m ⁻² .°C ⁻¹	3.23 W.m ⁻² .°C ⁻¹	31.6 %
ABS1	6.36 °C	1.6 W.m ⁻²	65.6 W.m ⁻²	0.08 W.m ⁻² .°C ⁻¹	3.28 W.m ⁻² .°C ⁻¹	31.8 %

Figures

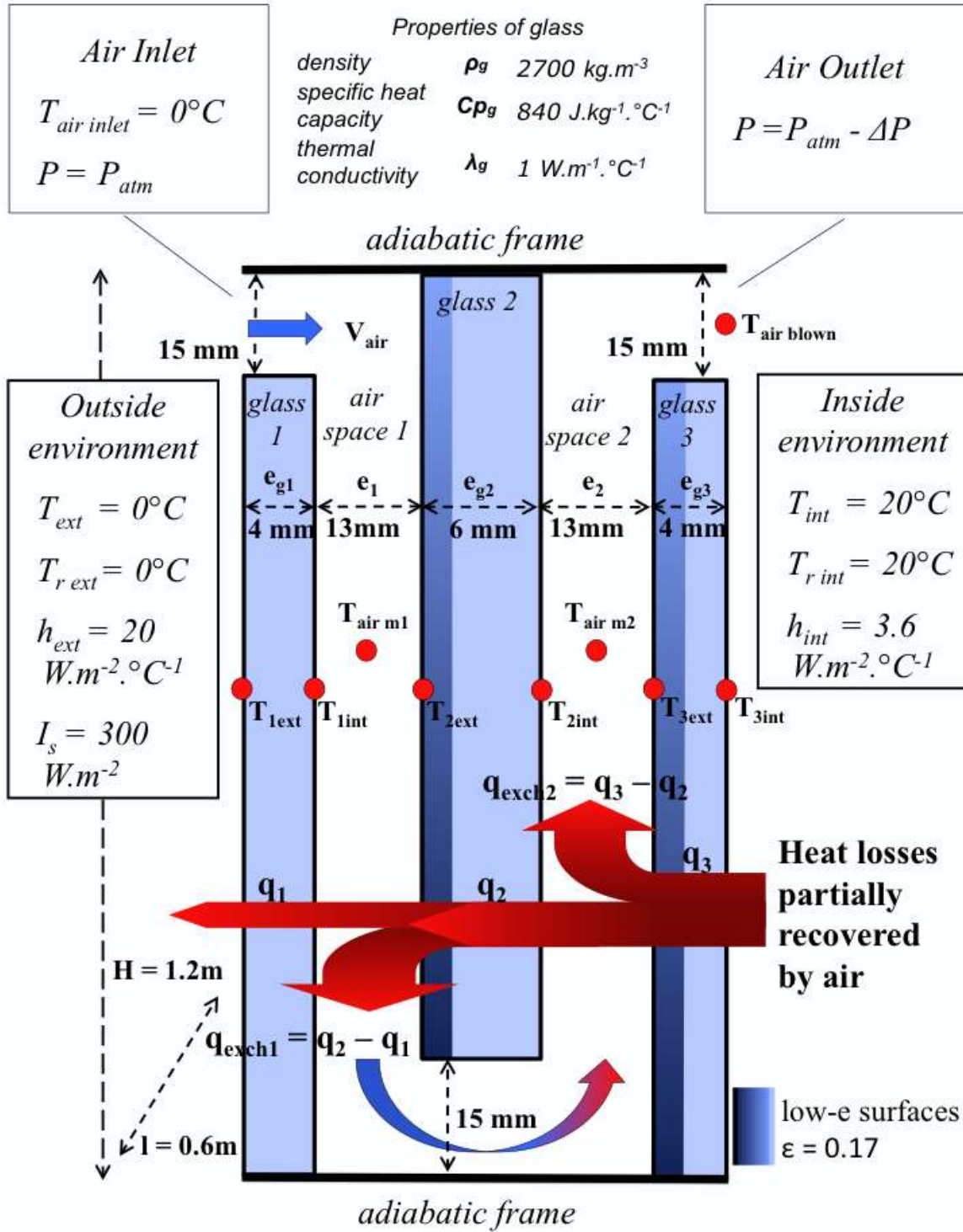


Figure. 1. Schematic diagram and boundary conditions

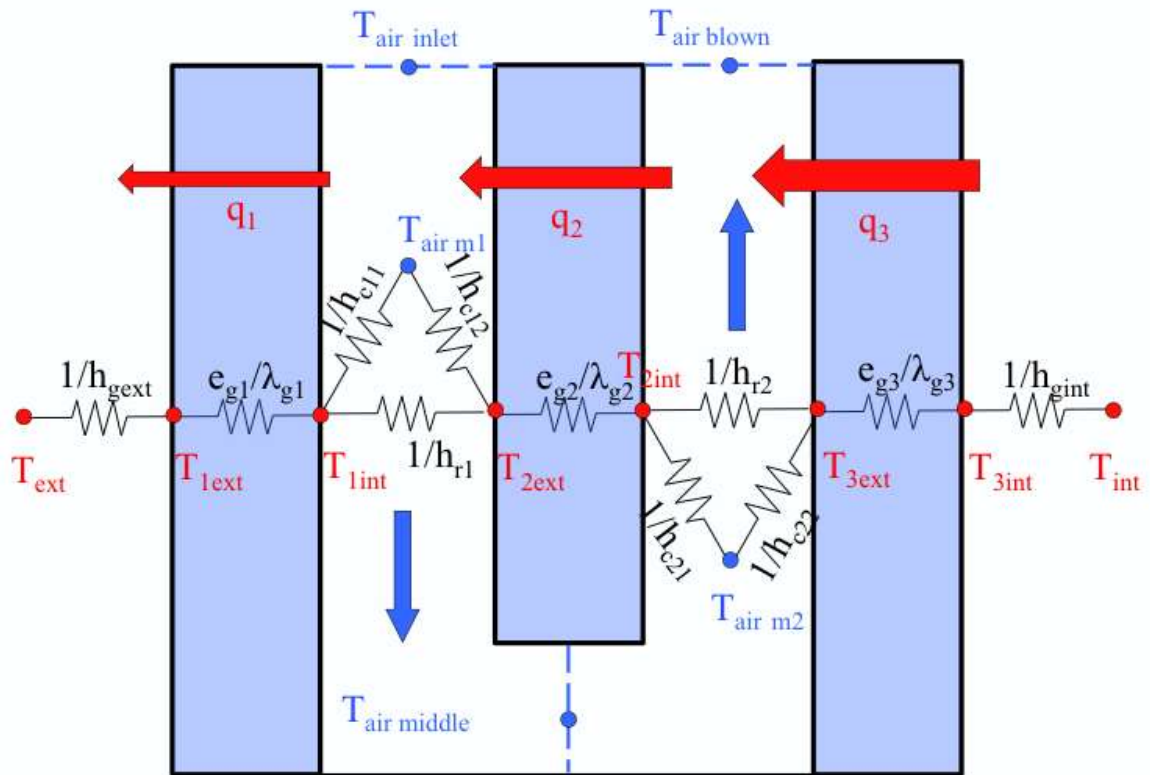


Figure 2. Schematic diagram of model ABS presented with electrical analogy without solar flux

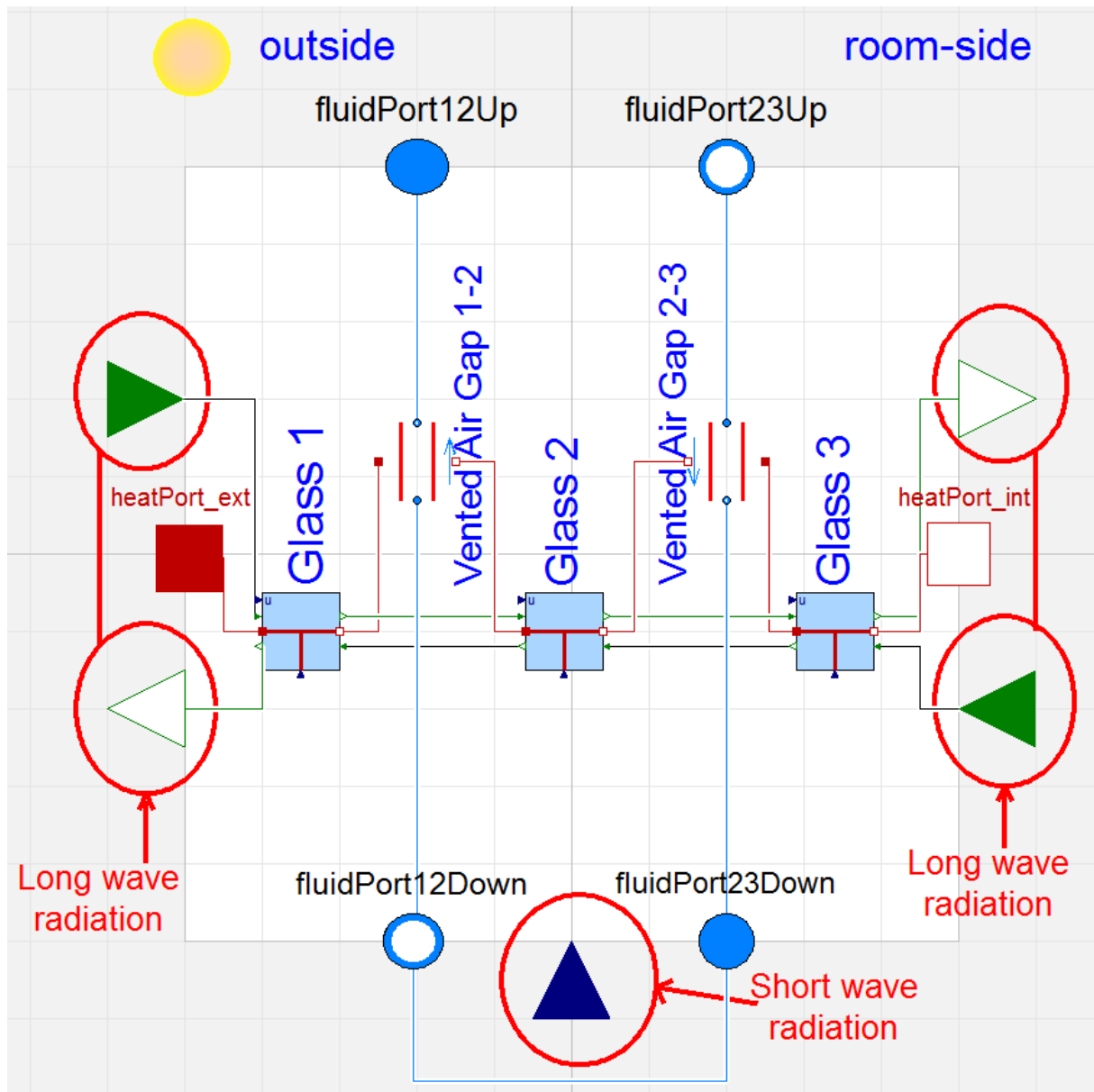


Figure 3. View of Modelica model for triple glazed supply-air window: CenterOfGlassPaziaud

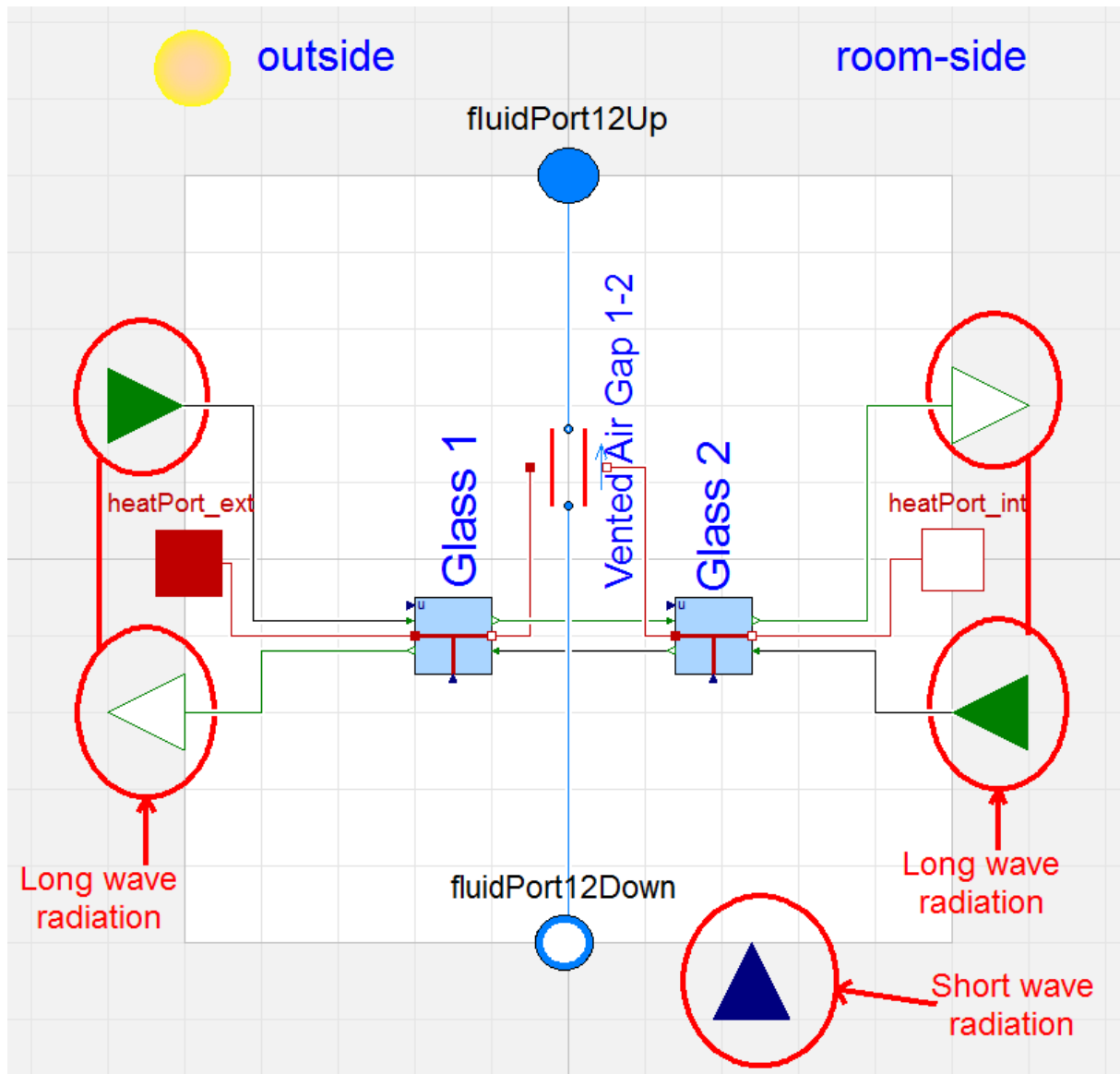


Figure 4. View of Modelica model for double glazed supply-air window

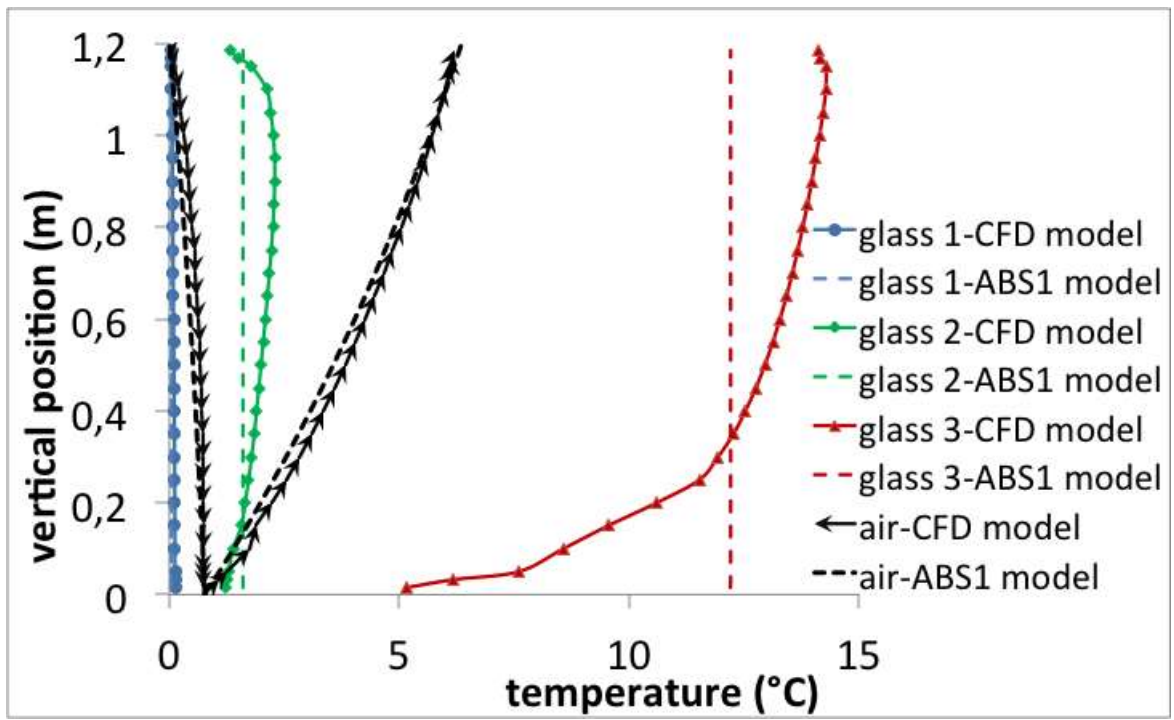


Figure 5. Evolution of glass and air temperatures versus window height performed at $20.3 \text{ m}^3 \cdot \text{h}^{-1}$ (4 Pa)

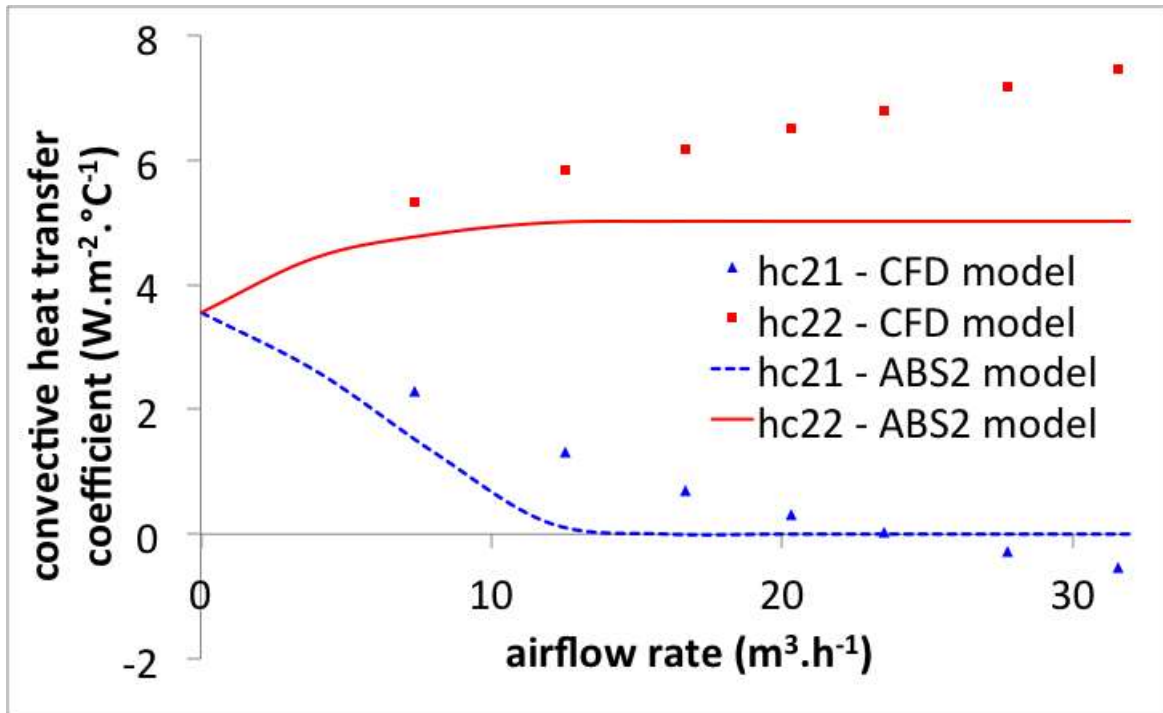


Figure 6. Evolution of convective heat transfer coefficients with airflow rate in the air gap 2

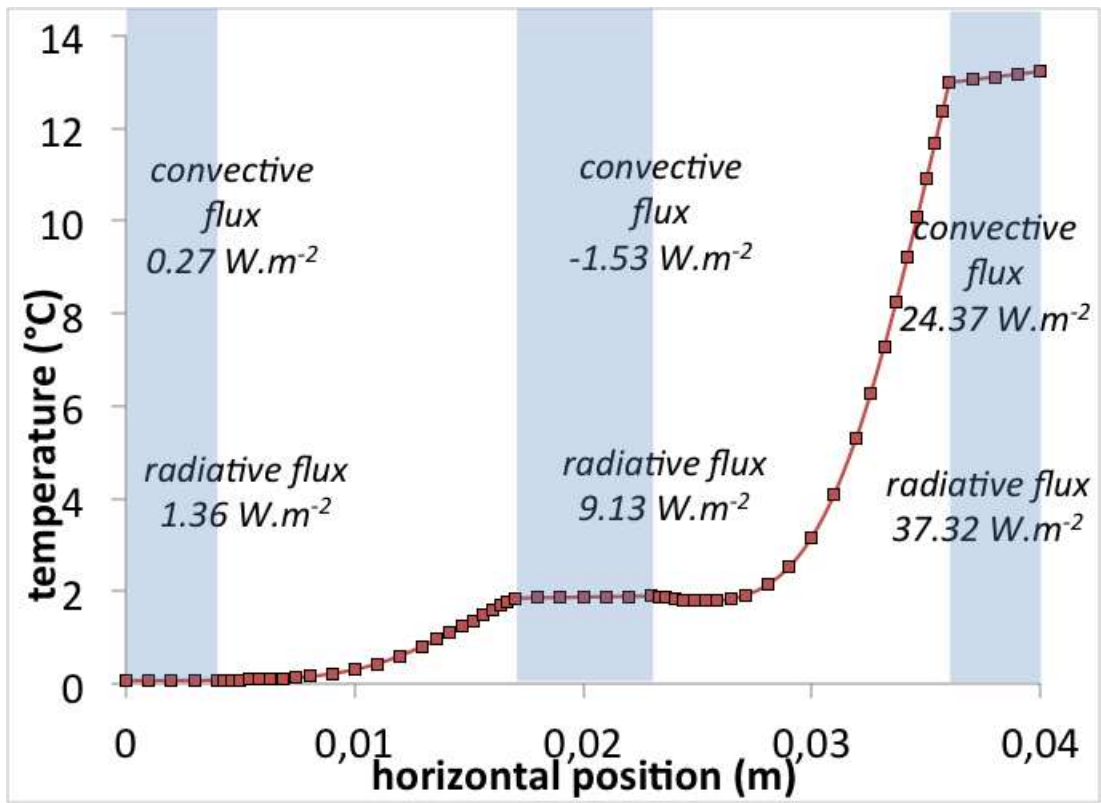


Figure 7. Evolution of the temperature and heat flux in a horizontal section ($z = 0.6\text{m}$)

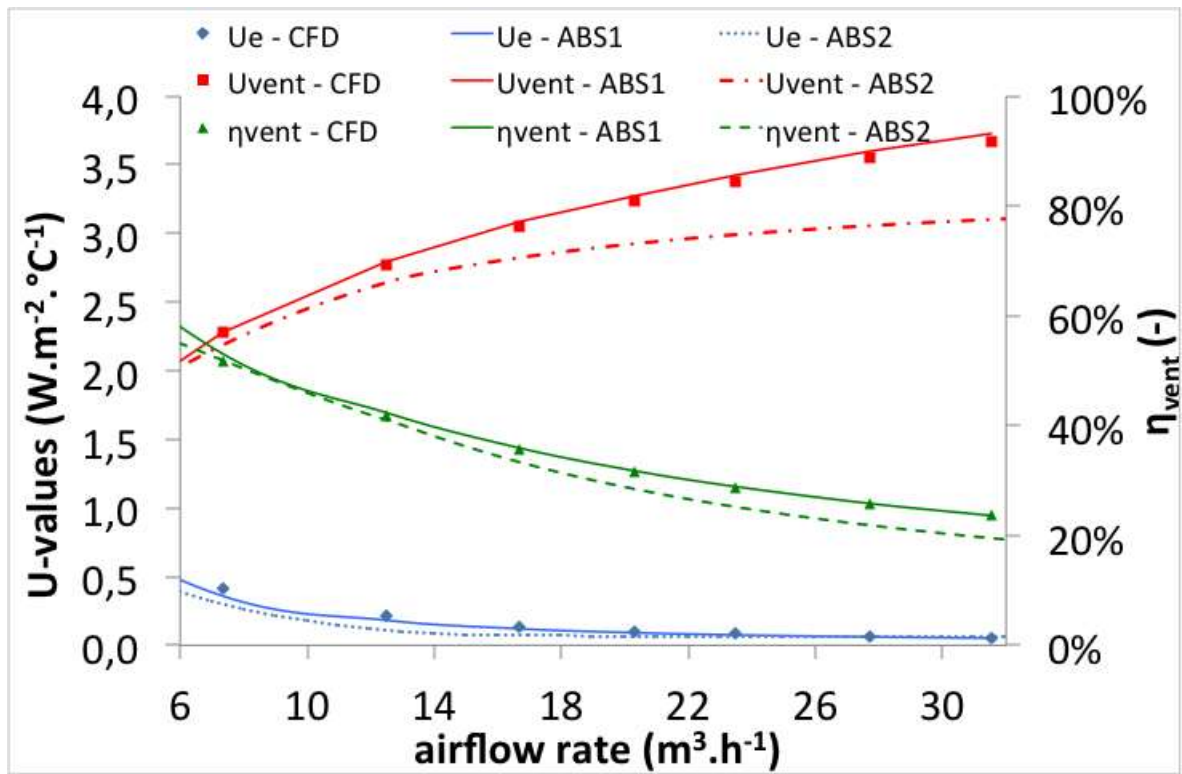


Figure 8. Evolution of U-Values and heat recovery coefficient versus airflow rate

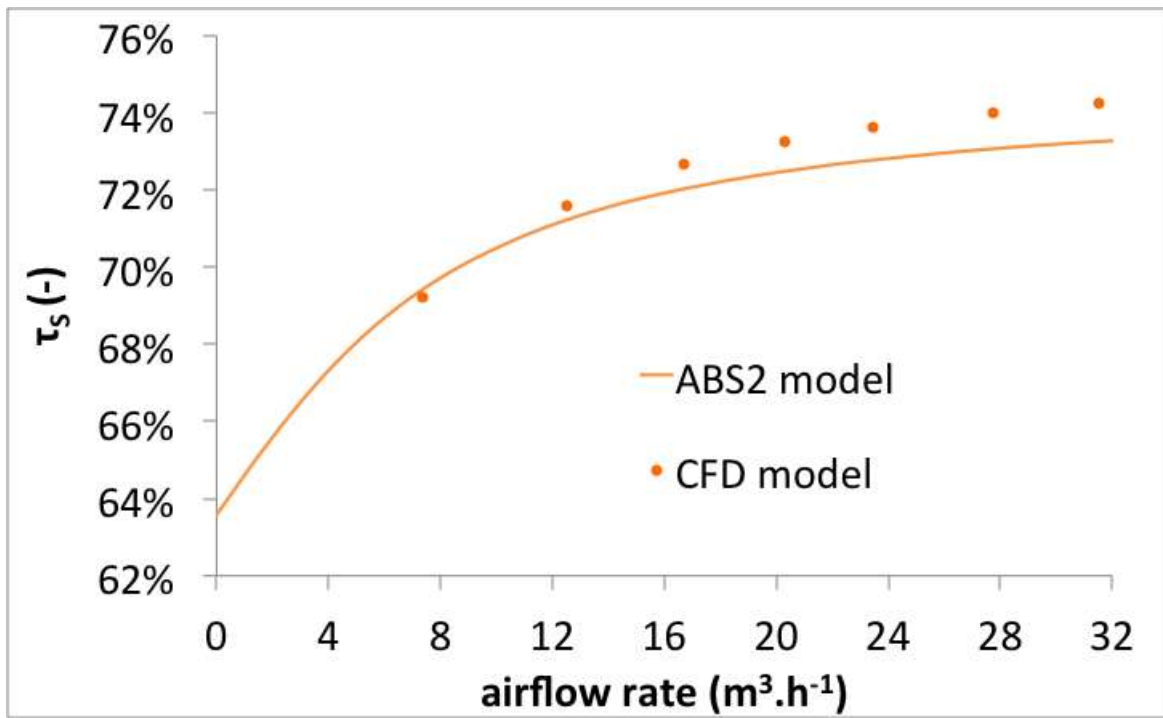


Figure 9. Evolution of total solar energy transmittance versus airflow rate

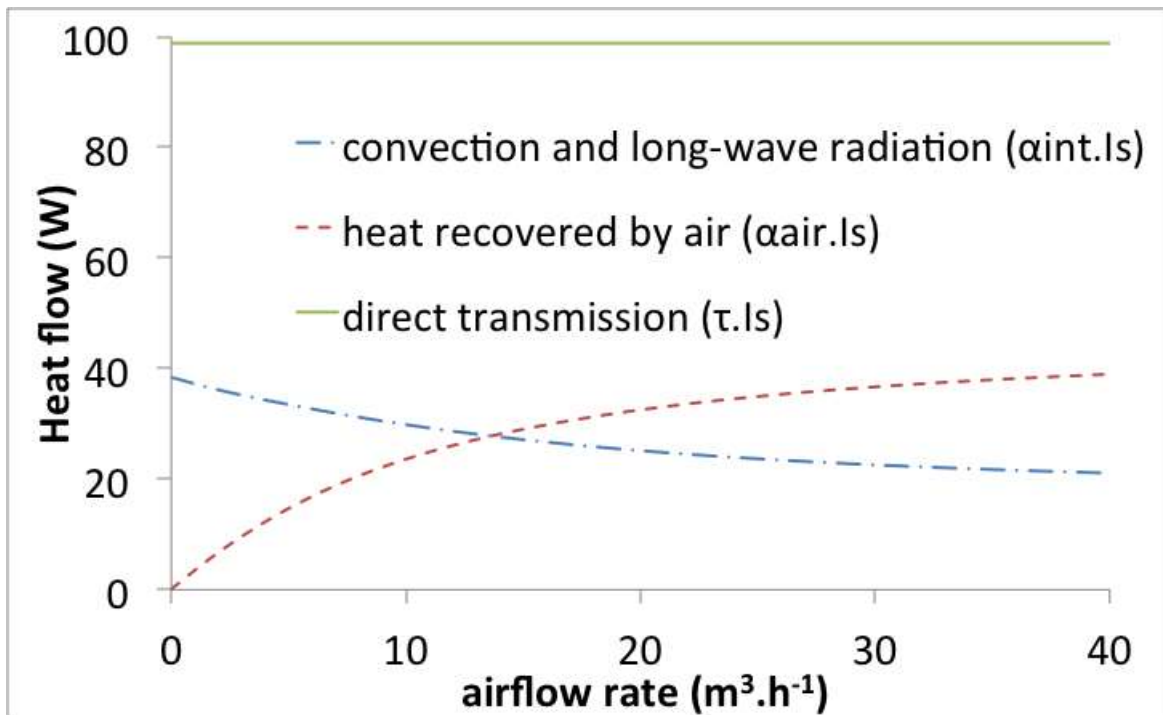


Figure 10. Evolution of different terms of the calculation of solar energy transmittance

How to model supply-air window in building simulation code?

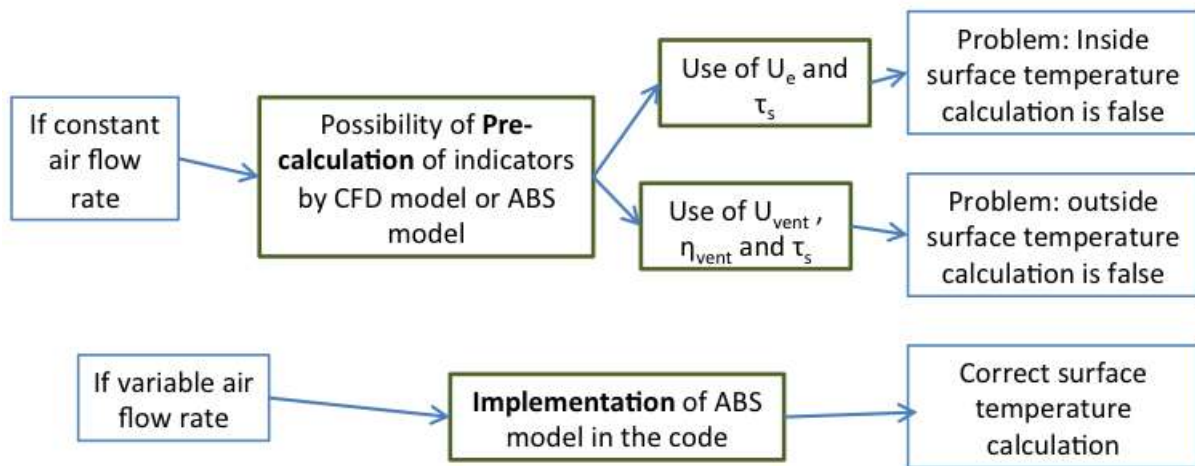


Figure 11. Three methods to consider a ventilated window in a thermal building simulation