# AIRFLOW WINDOW: NUMERICAL STUDY AND SENSIBILITY ANALYSIS OF THERMAL PERFORMANCES

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## ABSTRACT

In this paper, the thermal behaviour of a ventilated window is studied. The particularity of this system is the existence of an airflow passing between the glasses of a triple glazing. The aim is to preheat outdoor air entering the room both by heat loss recovery and by solar gains. To evaluate the thermal performances of the window, a 2D model simulating the heat and mass transfers within the airflow window has been developed, and validated from experimental results. The effects of several parameters on thermal performances are investigated, e.g. glass emissivity, air flow rate and convective coefficient.

## INTRODUCTION

Buildings in France account for 43 % of the total final energy consumption. To reduce energy demands, it is crucial to limit energy losses in winter and gains in summer, in particular through windows and ventilation. Windows and ventilation systems are two important building elements that generate significant heat losses (respectively 10 to 15 % and 20 to 25 % for existing buildings). This is particularly true in the case of passive buildings for which the energy consumption is essentially due to both systems previously cited (Feist et Schnieders 2009).

This can be achieved with a satisfactory indoor air quality using innovative systems, such as ventilated windows. These windows combine the function of a classical window to the one of an entering fresh air pre-heating system.

Recent studies (Wei et al. 2010; Chow et al. 2009) have shown that ventilated windows, characterized by the existence of free or forced air convection between glass layers, have promising performances on energy conservation and indoor air quality improvement. The particularity of the system is the existence of a single airflow passing through the triple glazing. Air moves first downward between the first two glass layers, then moves upward between the second and the third glass layers and finally enters the room (Fig. 1c).

The aim is to preheat outdoor air entering the room, on the one hand by heat loss recovery and, on the other hand, by solar gains. This can both reduce energy needs and improve occupants' thermal comfort. The main advantage of this system is a better thermal performance in comparison to a conventional airflow ventilation system combined to a conventional triple layers glazing window.

Some studies have pointed out the advantages of ventilated windows but these studies only deal with single air layer (Ismail et Henríquez 2005; Appelfeld et Svendsen 2011; Baker et McEvoy 2000; Carlos et al. 2010) (Fig. 1a) or two independent airflows (Gosselin et Chen 2008) (Fig. 1b) which are quite difficult to use in practice. One study consider a single airflow passing between the glasses of a triple glazing, but for building cooling applications only (Kim et Yang 2002). In this case, the entering air flows from inside to outside.

In the present study, a two dimensional model simulating the heat and mass transfers within the airflow window has been developed to evaluate its thermal performances (heat gains and loss reductions). This model, presented in the first part, has been validated from results obtained using an experimental set up which is described in the second part. Then, in the last part, the effects of several parameters on the airflow window's thermal performances have been investigated, such as window characteristics (glass emissivity and transmissivity), and ambient conditions (indoor air temperature, airflow rate and heat exchange coefficients).

## VENTILATED WINDOW DESCRIPTION

The ventilated window considered here is composed of a triple glazing (Fig. 1c). The outdoor fresh air enters the window through an opening located at the top of the outside surface of the window. Entering air flows downward in the first layer, between the two first glasses, then flows under the second glass and moves upward in the second layer. Finally, air enters the room through an opening located at the top of the inside surface of the window. Airflow through the window can be induced both by natural and forced ventilation systems. In this study, a Controlled Mechanical Ventilation, single flow, ensured the airflow.

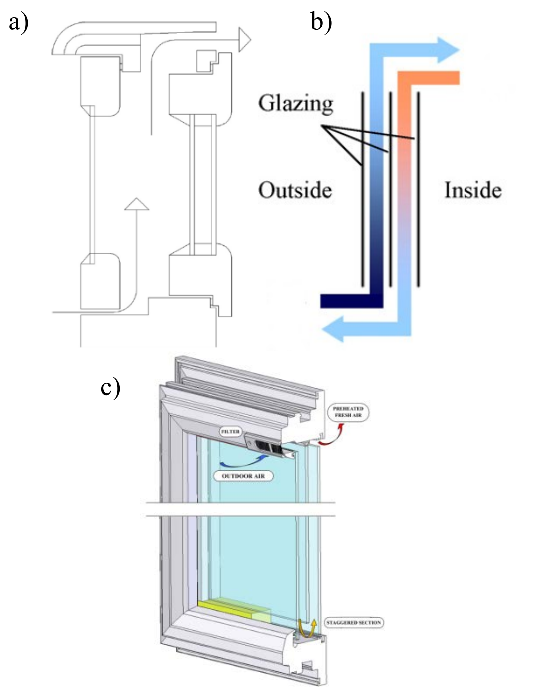


*Fig. 1 a) Glazing studied by Appelfeld et al.* (Appelfeld et Svendsen 2011)*, b) Glazing studied by Gosselin et al* (Gosselin et Chen 2008)*,*
*c) Principle of the present studied glazing.*

## MODEL

The two-dimensional scheme used to model the airflow glazing is illustrated on Fig. 2. Interfaces with walls are assumed to be adiabatic (black lines on the lower and upper parts). The glasses are colored in blue, air in white and frame in yellow. Heat transfers within the window are: conduction in glazing and frame, convection and radiation (short and long wavelengths).

In order to model heat and mass transfers through the window, a nodal approach has been adopted. It consists in dividing the window into isothermal zones as illustrated by the Fig. 2. By writing the thermal balance, we obtain the following differential equations system:

$$C.\dot{T} = A.\vec{T} + \vec{B} \qquad (1)$$

where $\boldsymbol{C}$ is the thermal capacity matrix, $\vec{T}$ the temperature vector, $\boldsymbol{A}$ the conductance matrix for the convection, conduction and radiation. $\vec{B}$ is the vector of solar heat flux absorbed by glasses. A temporal implicit scheme has been used. Moreover, radiation fluxes have been linearized and all elements have been assumed to be grey bodies.

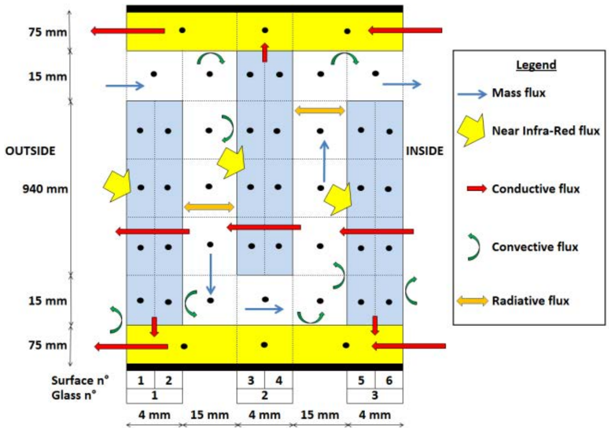


*Fig. 2 Airflow window diagram showing heat and mass transfers.*

In order to validate the developed numerical model, experiments have been conducted. The experimental set up used is presented in the next part.

## EXPERIMENTAL SET UP

The experimental set up consists in two similar cells of 6 square meters (volume of 15 $m^3$ per cell). These cells have been built within an existing hangar located in La Rochelle, France. Only one face of each cell is exposed to outdoor climatic conditions, and is equipped with a window. Two exhaust fans of 90 mm diameter circulate the air through ducts from the two experimental cells to a technical room. In this paper, we present the results obtained for the left cell only, equipped with the ventilated window (Fig. 3).

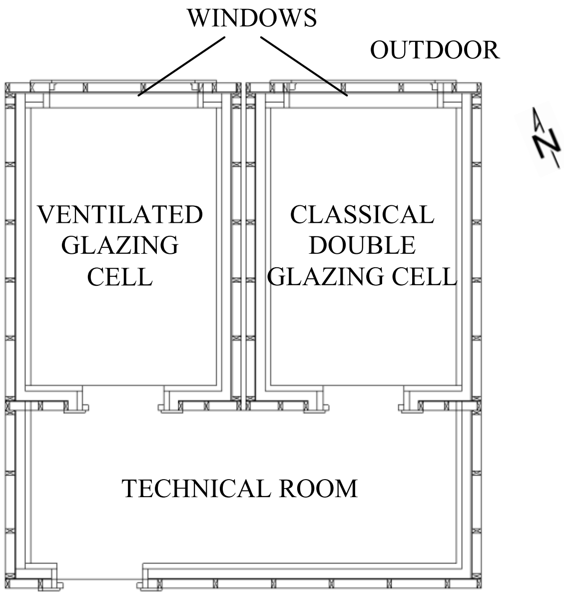


*Fig. 3 Scheme of the experimental cells.*

The walls, ceilings, as well as doors, were thermally insulated using 160 mm of polystyrene, and 100 mm for floors. In order to limit energy balance disturbances in the cells, data acquisition and command system equipments were placed within the technical room.

**Regulation and acquisition**

The indoor air temperature was maintained at 20 °C (minimum) using an electric convector (regulation with a temperature interval of ±0.1 °C). This electric convector was controlled by an angle phase controller, which was driven by a data logger Agilent 34980A. This data logger was also equipped with three 40-channel armature multiplexers with low thermal offset. The data logger for acquisition-command was controlled from a computer (Labview software), and exhaust fans by a dual DC power supply.

**Sensors**

The ventilated glazing was equipped with 22 thermocouples on glasses, and 12 in air layers: 6 in the first air layer and also 6 in the second one. At last, 2 thermocouples were placed on the bottom rail in the air layers and 2 on the top rail (Fig. 4). In order to limit solar radiation effects on thermocouples measurements, these ones are very thin: their wire diameter was 79 µm for a hot junction diameter of 150 to 200 µm. Thermocouples located on glazing surfaces, were placed using a thermal conductive and transparent glue. Three thermocouples were located within the cell, at the centre, at 20, 105 and 180 cm height. The mean indoor air temperature value was calculated from these three thermocouples and was used to monitor the indoor air temperature (wire of 300 µm diameter).

All temperature sensors have been preliminary calibrated in a thermo-regulated bath using a Pt100 as a reference temperature probe that has an uncertainty of ±0.2 °C. After the calibration process, the reading of 40 consecutive measures was checked by comparison to the reference probe value. All sensors have a maximum deviation of ±0.15 °C from the reference probe value and a standard deviation lower than 0.02 °C on the range 0 to 90 °C.

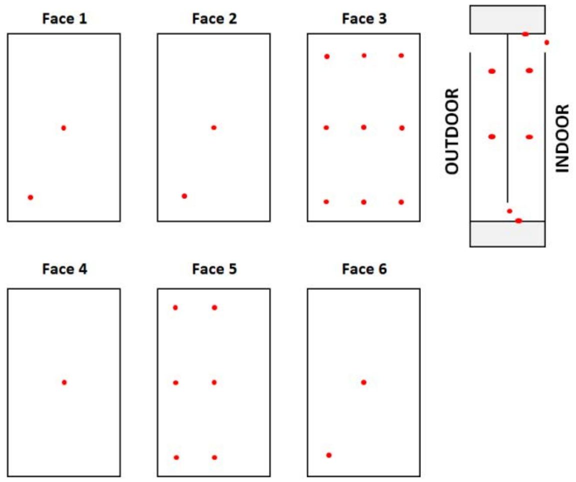


*Fig. 4 Positions of thermocouples on glasses (viewed from indoor), in air layers and on the frame.*

The exhaust airflow rate was deduced from the air speed one that was measured by a hot wire anemometer located at the centre of the circular exhaust duct of the controlled mechanical ventilation system. In order to determine the airflow rate through the ventilated glazing, this one was preliminary tested separately on a test bench. Thus, the pressure drop was measured for several tested flow rates. By this way, the relative pressure, measured by a differential micromanometer between outdoor and indoor, allows to deduce the airflow rate value. One can note that the cell is very airtight: for a pressure difference of 10 Pa, 6 $m^3.h^{-1}$ were measured from a blowerdoor test. Thus, a maximum of fresh air passes through the ventilated window.

Concerning the outdoor conditions, both a pyranometer and a pyrgeometer were placed vertically on the hangar facade in order to measure the incident solar flux on the window. In the same time, a Pt100 and a capacitive sensor measured the air temperature and the relative humidity respectively.

**Experimental conditions**

The results, presented in the following, have been obtained over a period of 27 hours, from January 10, 2013 to January 12, 2013. Every 30 seconds, the armature multiplexers scanned all the sensors. Moreover, in order to reduce measurement's fluctuations, the mean value of three consecutive measurements in a lap time of about 40 ms for each channel has been considered.

**Properties and boundary conditions**

The reference case considered in the following is defined by the Table 1. Geometrical parameters are defined on Fig. 2.

*Table 1. Geometric and physic parameters of the reference case.*

| PARAMETER | VALUE | UNIT |
|---|---|---|
| Glazing width | 0.62 | m |
| Glazing height | 0.97 | m |
| Indoor air temperature | 20.0 | °C |
| Indoor exchange coefficient | 3.0 | $W.m^{-2}.K^{-1}$ |
| Air flow rate | 9.0 | $m^3.h^{-1}$ |
| Emissivity (clear glass) | 0.89 | - |
| Emissivity (coated faces) | 0.15 | - |
| Solar transmissivity of glass 1 | 0.91 | - |
| Solar transmissivity of glasses 2 and 3 | 0.70 | - |

The first glass is clear while the two others ones have a low emissivity treatment on faces 3 and 5. The room walls temperatures are assumed to be equal to the inside air one. The internal convection coefficients have been preliminary determined from flow numerical simulations performed using Comsol Multiphysics®. The air speed was varied from 0.02 to 0.74 $m.s^{-1}$, thus the airflow was laminar (Reynolds number from 60 to 2300). The internal convection coefficient is defined for each zone as follows:

$$h_i = \frac{\Phi_i}{S(T_{ai}-T_{gi})} \quad (2)$$

where $h_i$ is the internal convection coefficient between an air zone and a glass zone [$W.m^{-2}.K^{-1}$], $S$ the contact surface between an air zone and a glass zone [$m^2$], $\Phi_i$ the heat convective flux between an air zone and a glass zone [W], $T_{ai}$ the mean air temperature of the air zone [°C] and $T_{gi}$ the mean temperature of a surface glass in contact with an air zone [°C].

The obtained mean internal convection coefficients values are for example 1.9; 4.5; 1.3 and 5.4 $W.m^{-2}.K^{-1}$ for faces 2 to 5 respectively with a flow rate of 15 $m^3.h^{-1}$. Such different values can be explained by streamlines variations within the air layers. Indeed, due to changes in the flow direction induced by the inlet, the outlet as well as the double bend between the two air layers, the analysis of streamlines shows that they are more attached on faces 3 and 5 than on opposite faces, i.e. faces 2 and 4. The effect of the airflow behavior, not presented here, is reflected in the internal convection coefficients evaluated for each air zone for airflow rates from 1 to 40 $m^3.h^{-1}$.

## RESULTS

At first, experiments results obtained for the reference case were used in order to validate the numerical model. Measured environmental conditions corresponding to the reference case are shown on the Fig. 5.

**Environmental conditions**

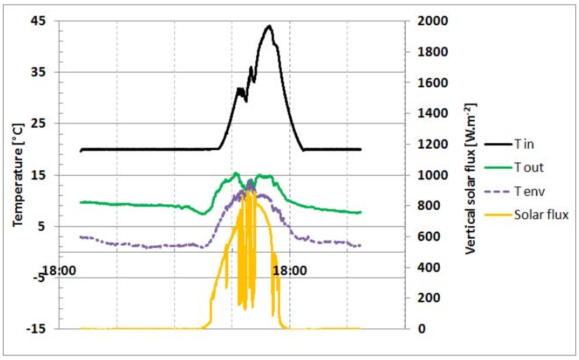


*Fig. 5 Environmental conditions.*

During the night, the indoor air temperature ($T_{in}$) was maintained at a minimum of 20 °C while, during the day, it reached a maximum of 44 °C. This high temperature level can be explained by the cell insulation quality and the high ratio of glazing. $T_{out}$ represents the outdoor air temperature at a distance of 20 cm from the facade. $T_{env}$ represents the equivalent environmental temperature front of the window. This outdoor temperature depends on the mean sky and nearby surfaces temperatures, and was calculated from pyrgeometer measurements. The day considered for the reference case is quite warm and sunny for La Rochelle (minimum temperature of 7.4 °C). The intermittent cloudy period in the middle of the day induces sudden drops of incident vertical solar flux. These conditions are particularly interesting to investigate the developed numerical model behaviour, since solar flux represents an important part of ventilated window's performances (Baker et McEvoy 2000; Gosselin et Chen 2008).

**Dynamic validation of the numerical model**

The entering air, blown from the ventilated window is a good indicator of overall performances of the window, since the preheated air is mainly responsible for the heat recovery.

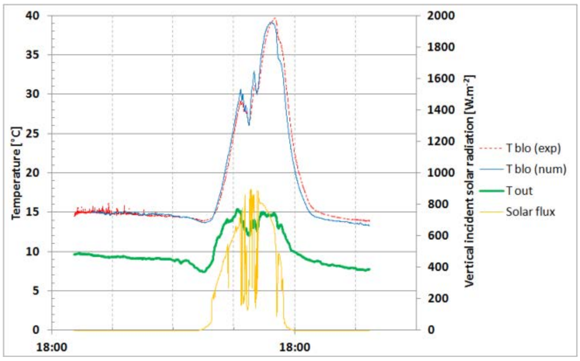


*Fig. 6 Comparison between experimental and simulated results in term of blown air temperature.*

Fig. 6 shows the evolution of blown air temperatures. Values predicted by the developed model (blue curve) are very close to the reference experimental results (red dashed curve). When there is no solar radiation, the temperature gain between outdoor and blown air temperatures is about 5.5 °C. With solar radiation, this gain reaches up to 30 °C. Such high temperature levels can be explained by the fact that the indoor air temperature is very high during the day (see Fig. 5). Subsequently, the heat recovery capacity by the entering air is important. Additionally, the direct solar absorption by glazing is also transmitted to this entering air, and thus contributes to maintain high indoor air temperature levels.

In the next part, a sensitive analysis is presented. It consists in evaluating the contribution of several parameters on the ventilated window thermal performances.

**Sensitive analysis**

There are two types of parameters that can influence the heat recovery capacity of the ventilated window, namely environmental and window parameters. The effect of some of these parameters on both the blown air temperature and the heat recovery is investigated here (Fig. 7). In order to estimate the contribution of each parameter, a Sensitive Index (SI) has been defined as follows:

$$SI = \frac{\frac{T_{ref}-T}{T_{ref}}}{\frac{P_{ref}-P}{P_{ref}}} \qquad (3)$$

where $T_{ref}$ is the reference temperature, T is the temperature when the measured value for a given parameter is P, $P_{ref}$ is the reference value of the considered parameter and P is the measured value of the parameter. Thus, the Sensitive Index defines the variation of the temperature (for example the blown air one) according to variation of a parameter. The parameters considered here are the indoor air temperature ($T_{in}$), the glazing solar transmissivity (τ), the airflow rate (AF), the convective heat exchange coefficient (h) and the glass emissivity for clear (ε) and coated faces (ε low-e).

Fig. 7 shows mean values of the Sensitive Index calculated during "night" and "day" separately. The "night" is defined as the period over which the incident solar radiation is less than 5 $W.m^{-2}$, while the "day" corresponds to the remainder period.

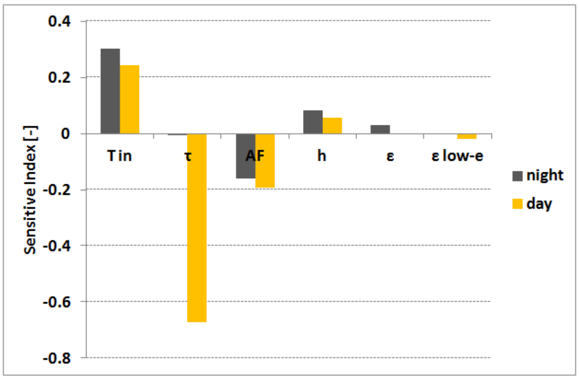


*Fig. 7 Mean Sensitive Index values of the blown air temperature during night and day.*

In Fig. 7, the parameters are classified in term of importance from left to right. SI values have been calculated for a variation of +10 % of each parameter based on the reference case presented previously on Table 1.

The effect of the indoor air temperature variation is quite similar during night and day: a variation of +10 % of the indoor air temperature induces a variation of +3 % of the blown air temperature during the night.

Obviously, the solar transmissivity has solely an impact during the day. The transmissivity significantly impacts the blown air temperature during the day. This points out that the choice of glasses is crucial in order to efficiently preheat the entering fresh air. One can note that if τ is increased, the blown air temperature decreases but, in the same time, direct solar gains in the room increase. Thereby, this aspect is investigated in the next part, which is devoted to the analysis of the global energy gain in the cell.

If the airflow rate (AF in Fig. 7) has obviously an effect on the blown air temperature, it is limited to a SI of -0.2 during both the night and the day. The convective heat exchange coefficient has even fewer effects on the blown air temperature. Now, about the emissivity values, they have globally a negligible impact on the blown air temperature. The coated faces act more during the day when the treated glasses reach a high temperature (maximum of 43.8 °C for the face 4), and thus limit losses. Conversely, an increase in the emissivity of uncoated faces slightly decreases the heat recovery by the entering air during the night.

It has been highlighted that the effect on the blown air temperature of the considered parameters is very different. In order to characterize the effect of these parameters in terms of window thermal performances, a heat balance analysis has been conducted (Appelfeld et Svendsen 2011; Baker et McEvoy 2000). The energy input is defined as follows:

$$Q_{ei} = Q_{air} + Q_{face\ 6} + Q_{frame} + Q_t \quad (4)$$

where
$Q_{ei}$ is the energy input in the cell. A negative value indicates a loss of energy by the window [W]
$Q_{air}$ is the energy needed to heat fresh air to the cell temperature [W]
$Q_{face\ 6}$ is the heat flux through the indoor glass [W]
$Q_{frame}$ is the indoor heat flux by the window frame [W]
$Q_t$ is the direct solar radiation transmitted in the cell [W]

$$Q_{air} = \dot{m}\,.\,C_p.(T_b - T_{in}) \quad (5)$$

where $\dot{m}$ is the mass air flow rate [kg.s$^{-1}$], $C_p$ the heat capacity [J.kg$^{-1}$.K$^{-1}$], $T_b$ the blown air temperature [°C] and $T_{in}$ the cell air temperature [°C].

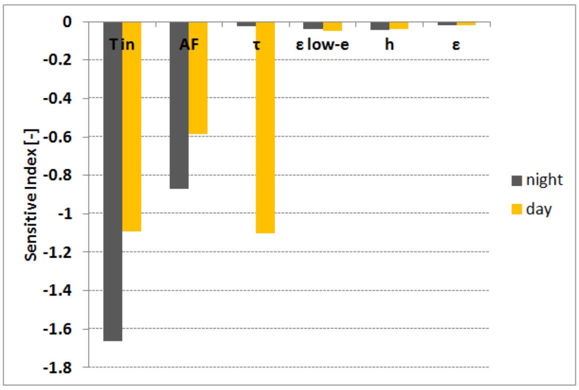


*Fig. 8 Mean Sensitive Index values of the energy input during night and day.*

Here again, parameters have been classified in terms of performance from left to right in Fig. 8. One can note that the day period being shorter than the night one, this last is predominant in the global results. It appears that the three parameters that were the most significant on the blown air temperature are here again the most significant on the energy input. Nevertheless, it can be noticed that the day and night repartition is quite different. If compared to the reference case, the 10 % increase in the indoor air temperature during the night induces a 16.6 % decrease in energy input (SI = 1.66). This result is less evident during the day period, with a SI of 1.09. Similar results can be observed for the airflow rate with a lighter impact in comparison to the indoor air temperature. Regarding the solar transmissivity, it has a great influence on the heat recovery. The three last parameters effects are negligible from a heat recovery point of view.

**Impact of the position of coated faces**

In practice, the glazing optical characteristics, i.e. τ, ε and $\varepsilon_{low\text{-}e}$, are linked together. Indeed, if τ is high, ε and $\varepsilon_{low\text{-}e}$ are high too. The choice of a glazing, ventilated or not, consists in a compromise between a good direct solar radiation transmission and a low long-wavelength emissivity (needed to reduce heat loss during the night). As the glazing considered here is composed of three separated glasses, it is possible to choose the number and the position of coated faces. Therefore, all possible combinations have been investigated, using the clear and treated glasses defined for the reference case (Table 1). Since it is not possible to use a two-sided coated glass, such configurations have been excluded. Finally, there are 27 possible glazing configurations with none, one, two or three low-e faces.

Fig. 9 shows the blown air temperature in function of the energy input in the room for the 27 glazing configurations. Four sets of glazing configurations are defined, depending on the number of coated faces. In Fig. 9, the energy input is normalized by the maximum energy input for day and night periods.

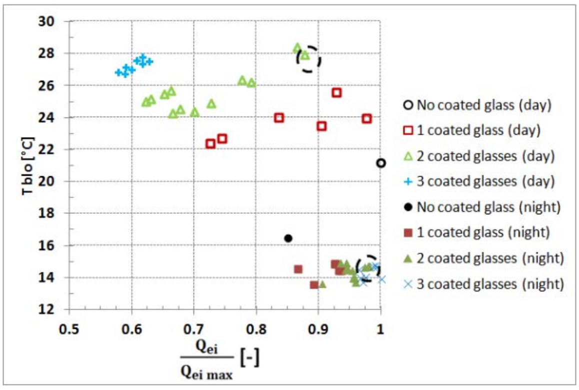


*Fig. 9 Blown air temperature as a function of the normalized energy input.*

It can be observed that the gap between the best glazing configuration and the worst one is greater during the day period. As expected, the no coated glazing transmits a maximum of direct solar radiation during the day, but it is the worst glazing during the night. Finally, note that our reference glazing, dash-circled in black in Fig. 9, seems to be a quite good compromise in terms of performances during both day and night periods.

Fig. 10 shows the blown air temperature according to mean energy input during the whole period (day and night). The reference glazing is again dash-circled in black. For a sunny day the clear glazing is good, with a mean input of 12.2 W. Indeed, this value is closed to the highest one which has been obtained with a single coated glass ($Q_{ei}$ = 13.8 W).

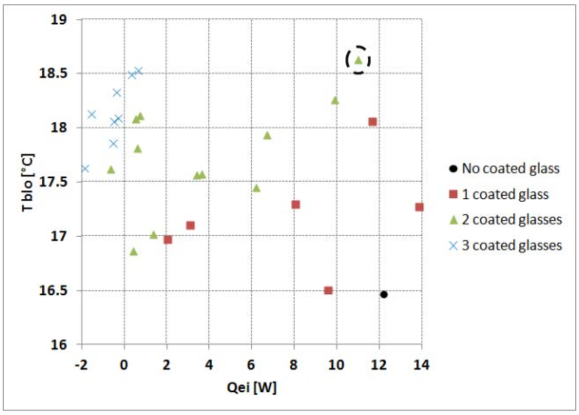


*Fig. 10 Blown air temperature in function of energy input during the whole period (day and night).*

Three coated faces glazings are adapted to a north orientation. When a glazing is near the south-facing, one can adopt another configuration, paying attention to the sequence of coated faces.

In conclusion, the best glazing configurations are the followings: a single low-e face at the 5th position, two treated faces at the 3rd and 5th positions and no treated face. One can note that the clear glazing is very sensitive to solar radiation (Fig. 9). It should be used only on sunny locations.

**Impact of indoor air temperature and airflow rate**

Fig. 11 presents the effect of indoor air temperature and airflow rate on the effective heat recovery $Q_{hr}$, which is defined as follows:

$$Q_{hr} = Q_{hr,air} + Q_{face\ 6} + Q_{frame} + Q_t \quad (6)$$

where

$$Q_{hr,air} = \dot{m} . C_p . (T_b - T_{out}) \quad (7)$$

In this equation, $Q_{hr,air}$ is the heat recovery by blown air [W]. Other parameters have already been defined. Fig. 11 points out that the more the indoor air temperature increases, the more the input energy and effective heat recovery are reduced (Fig. 11).

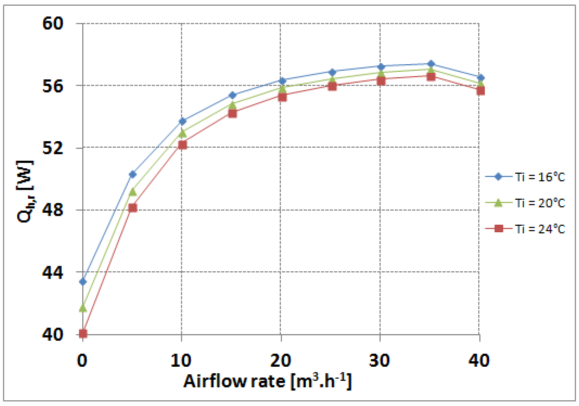


*Fig. 11 Effective heat recovery in function of airflow rate.*

This one significantly increases with the airflow rate for flow rate values lower than 10 $m^3.h^{-1}$. For larger airflow rate values, the effective heat recovery tends to a maximum of 57.4 W, reached for 35 $m^3.h^{-1}$, and finally decreases. This can be explained by an increase in losses by the window frame (Appelfeld et Svendsen 2011). If compared to the indoor air temperature, the airflow rate has a greater impact on the effective heat recovery than the indoor air temperature. This is due to the possible wide variation of the airflow rate.

## CONCLUSION

The ventilated window allows recovering losses energy and absorbed solar energy. These advantages are mainly influenced by meteorological conditions. Others parameters can be controlled like indoor air temperature, airflow rate and glazing optical properties to optimize the energy input in the room. Among them, the indoor air temperature has the most influence on energy input, in particular during the night. Concerning the glasses, a light variation of the solar transmissivity affects significantly both the direct solar energy input and the blown air temperature. Moreover, the choice of number of low-e faces and their position are crucial: the best compromise between day and night being one low-e coat on face 5 for a sunny period. However, there is no direct link between blown air temperature and energy input. Finally, the air flow rate is predominant on thermal performance. The window efficiency is higher for low airflow rates (< 10 $m^3.h^{-1}$). The fall of the energy input happens for airflow rates beyond 35 $m^3.h^{-1}$ due to the increase of frame losses and no extra heat recovery.

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