

## Development and characterization of semitransparent double skin PV facades with heat recovering

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### ABSTRACT

This research aims at developing new standardized typologies of semitransparent double skin facades formed by an external semitransparent PV laminate, a wide air gap and a rear glass (Figure 1). There are actually many buildings in Europe which incorporate such active facades, but all them have been designed as user defined projects and very few of them accurately evaluate the feasibility of using the heat produced by the air channel. This research tries to address this situation: the Spanish company ISOFOTON, together with CIMNE and the BSC-CNS has started a collaboration to design and standardize these ventilated façades. This paper describes the results of two stages of a more wide research: an intensive evaluation of the existing heat transfer relations for buoyancy induced flows; and the programming of a dynamic transient solver able to define the thermo-electrical performance of this façades.

### 1. INTRODUCTION

Concerning to the mathematical model to define the energy performance of such façades, simplified methods have been proposed by (Eicker Ursula. 2003) and Mei (Mei Li. 2003). More sophisticated models for double skin façades are also developed by Faggembau (Faggembau D. 2003) and Saelens (Saelens Dirk. 2002). All these authors have assumed forced flow correlations for the convection heat transfer coefficient and they didn't consider the effect of the asymmetry in the boundary conditions. Saelens also made a review of several ways to model the air channel and he concluded that a single volume model with linear variation of the average temperature has the same accuracy than finite difference schemes.

### 2. MATHEMATICAL MODEL

#### 2.1 Physical model

Figure 1 shows the physical model postulated for the ventilated PV façade.

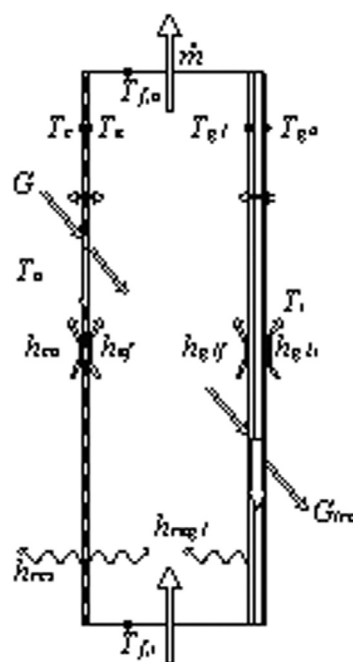


Figure 1. Physical model of the façade

Air enters the façade at the bottom opening with an average inlet temperature ( $T_{fi}$ ) which is assumed equal to the uniform exterior air temperature ( $T_o$ ). Hot air exits from the top of the chimney at outlet average temperature ( $T_{fo}$ ). The following heat transfer processes simultaneously happens within the façade: solar radiation absorbed by the solid layers, thermal radiation between them, thermal radiation between the external layer of the PV laminate and the sky and heat convection between the exterior, the interior and the façade.

### 2.2 Heat conduction transfer within the semitransparent PV laminate

The semitransparent PV laminate is formed by three layers: the exterior layer is glass, the middle layer is formed by the PV cells and EVA in the spaces in between and the third layer is Tedlar. The governing equation within each layer is:

$$\rho c_p \frac{\partial T}{\partial t} - \nabla \cdot (k \nabla T) = Q \quad (1)$$

The boundary conditions are:  $T = T_e$  in the last node and  $k \nabla T \cdot n = \bar{h}_o (T_o - T_c) + h_{rcs} (T_s - T_c)$  at  $x=0$ . Where:

$\rho$ : Density of each layer (Kg/m<sup>3</sup>)

$c_p$ : Specific heat coefficient (J/KgK)

$T$ : Temperature of the layer (K)

$k$ : Thermal conductivity (W/mK)

$Q$ : Volumetric heat source (W/m<sup>3</sup>)

$T_e$ : Temperature of the last node (K)

$T_c$ : Temperature of the first node (K)

$T_s$ : Radiative sky temperature (K)

$\bar{h}_{co}$ : Average convection heat transfer coefficient between the glass layer and the exterior (W/m<sup>2</sup>K)

$h_{rcs}$ : Radiation heat transfer coefficient (W/m<sup>2</sup>K)

Equation 1 is solved using a finite element method in one dimension. The temporary term is discretized using the trapezoidal rule. By deriving the weak form of Equation 1 and applying the Galerkin method, we can obtain a discretized system of equations defined as:

$$[A] \cdot [T^{n+\theta}] = [r^{n+\theta}] \quad (2)$$

where:

$[A]$ : Coefficients matrix

$[T^{n+\theta}]$ : Temperatures at time step  $n+\theta$

$[r^{n+\theta}]$ : Residual array

$n$ : Time step number

$\theta$ : Temporal coefficient

where  $\theta=1$  for the backward Euler scheme (first order), and  $\theta=1/2$  for the Crank-Nicolson scheme (second order). Matrix  $[A]$  is three-diagonal. The values of the terms in each array depend on the layer. The physical properties of the second layer (PV cells and EVA) are affected by the packing factor ( $P$ ), defined as the ratio between the sum of the PV cells surfaces and the overall surface of the façade.

The system of equations is solved using a direct three-diagonal algorithm (TDMA). Once the temperatures of each node are obtained, the conduction heat flux towards the air channel ( $q''_{e,cond}$ ) is computed using Fourier's law.

### 2.3 Heat conduction within the rear Glass

The governing equation which defines the heat conduction within the rear glass is Equation 1. The FEM discretization will be the same as the PV laminate. The boundary conditions will be the opposite of the PV laminate (Dirichlet at  $x=0$  and Neumann at last node). Once the temperatures of each node are obtained, the conduction heat flux towards the air channel ( $q''_{g1,cond}$ ) is also calculated.

### 2.4 Air channel mass flow rate and heat transfer

The air channel is modelled through a stationary one-dimensional volume, with a linearly temperature variation in the vertical direction. Three different working conditions are possible:  $q''_{ef} = q''_{glf}$ ;  $q''_{ef} \neq q''_{glf}$ , and adiabatic condition  $q''_{ef}; q''_{glf} = 0$ . The determination of these heat fluxes depends on the heat conduction and the heat radiation fluxes. Concerning the nature of the flow, three situations are considered: natural, forced (wind or mechanical fan) and mixed convections. In the natural convection situation, two flow regimes exist: the laminar and turbulent regimes. Besides, a distinction between thin and wide channels is considered. The governing equation, once it is integrated, can be expressed as:

$$\dot{m} c_p (T_f - T_{f,i}) = (q''_{ef} + q''_{glf}) W y \quad (3)$$

where:

$T_f$ : Average temperature at height  $y$  (K)

$y$ : Height (m)

$T_{f,i}$ : Average temperature at the inlet (K)

$\dot{m}$ : Mass flow rate (Kg/s)

$q''_{ef}; q''_{glf}$ : Convective heat sources

#### 2.4.1 Average wall and outlet temperatures

Assuming the Newton law of cooling for the heat convection sources, making some arrangements and integrating along the façade height we can obtain the average wall temperatures.

#### 2.4.2 Mass flow rate

The flow-rate is obtained by equating the sum of the pressure differences which drive the flow (wind and buoyancy) with that of those opposing it (hydraulic and friction losses) (Brinkworth B.J. 2000). The result of this methodology is expressed as:

$$A \cdot (H^+)^3 + B \cdot (H^+)^2 - \frac{1}{2} [(f_{app} Re_D) \cdot H^+ + \sum K_k] = 0 \quad (4)$$

$H^+ = \frac{H}{2Re_b}$  where:

$A = \frac{2SGr_D}{Pr}$  Dimensionless height

Buoyancy term

$$B = \frac{\Delta P_w D_h^4}{\rho v^2 H^2} : \text{Wind-induced term}$$

- $S$ : Stratification coefficient
- $K_h$ : Inlet and outlet hydraulic losses
- $f_{app}$ : Apparent friction factor

The term  $f_{app} Re$  is likewise depending on  $H^+$ , which means that an iterative process must be set up. The expression of  $f_{app} Re$  depends on the boundary conditions, on the flow nature, and on the flow situation: in laminar free convection, the correlations of Kakaç (Kakaç S. 1987) are used. In laminar free convection and asymmetric uniform heat fluxes new correlations were obtained (see section 3):

$$f_{app} Re = 0 + \frac{0.7311 \left( \frac{q_h''}{q_c''} \right)^{0.4067}}{H^+} \quad (5)$$

In turbulent free convection, the term  $f_{app} Re$  is no longer linear dependent on  $H^+$ . For this situation only correlations for forced turbulent fully developed flow exist (Filonenko G.K. 1954). In mixed flow convection, a composition of the previous correlations will be used. The pressure difference term ( $\Delta P_w$ ) is calculated using dynamical pressure expression in function of the pressure coefficients.

Once all the terms are determined, a Newton Raphson method is used to solve Equation 4, and the mass flow rate is determined in function of the  $H^+$ .

### 2.5. $\overline{Nu}$ correlations for free convection

The average convection heat transfer coefficients are derived from the Nusselt numbers ( $\overline{Nu}$ ). In laminar free convection flows the  $\overline{Nu}$  is obtained through the following correlations:

Table 1. Correlations of  $\overline{Nu}$  for free convection

Symmetric uniform heat fluxes (Rohsenow Warren M. 1998) ; c=1.15
Asymmetric uniform heat fluxes:
Hot wall ( $\overline{Nu}_h$ ), new correlation (section 3):
$\overline{Nu}_h, \overline{T}_w = \left[ \frac{12}{Ra_{qh}''} \left( \frac{q_{wc}''}{q_{wh}''} \right) + \frac{6}{Ra_{qh}''} \left( 1 - \frac{q_{wc}''}{q_{wh}''} \right) + \frac{1.88}{(Ra_{qh}'')^{0.4}} \right]^{-0.5}$
Cold wall ( $\overline{Nu}_c$ ), new correlation (section 3):
$\overline{Nu}_c, \overline{T}_w = \left[ \frac{12}{Ra_c''} \left( \frac{q_{wh}''}{q_{wc}''} \right)^{-0.167} + \frac{1.88}{(Ra_c'')^{0.37}} \right]^{-0.548}$
Uniform heat flux and adiabatic wall (Rohsenow Warren M. 1998) ; c=1.07:

In turbulent free convection flows, in the case of wide channels, the correlation of Churchill is used (Churchill S.W. 1973).

### 2.5.1 $\overline{Nu}$ correlations for forced convection

In forced convection flows, a distinction between developing and fully developed flows is made. In case of laminar developing flows, the correlations of  $\overline{Nu}$  of Nu-ajan (Bejan A. 2003) and Kays (Kays W.M. 2004) are used. In case of laminar fully developed flows, the  $\overline{Nu}$  are constants. In case of turbulent developing flows, a correlation obtained by Saelens (Saelens Dirk. 2002) is used. Concerning to the turbulent developed flows, the equations of Kays (Kays W.M. 2004) are also valid.

### 2.6 Convection heat transfer within the rear glass

In the case where the rear glass is formed by multiple glass layers, the equations and correlations obtained for air cavities will be used.

### 2.7 Convection heat transfer between the exterior, the interior and the façade

The convection heat transfer between the glass layer of the PV laminate and the exterior follows the Newton's law of cooling. In 1984, Sharples (Sharples S. 1998) concluded that the linear correlations obtained for solar collectors, are also valid for ventilated facades. The convective heat transfer between the last layer of the rear glass and the room also follows the Newton law of cooling but the convective heat transfer coefficient is affected by the HAVC system of the building.

### 2.8 Solar radiation

The semitransparent PV façade is formed by an equivalent opaque surface, formed by the sum of the PV cells and a semitransparent equivalent surface formed by the sum of the spaces in between.

### 2.8.1 Spectral and angular dependency of the opaque surface

The opaque surface is formed by an external glass layer, the EVA layer and the PV cells. The product tau-alpha ( $\tau\alpha$ ) is the optical property to be determined.

The angular dependence was deeply analysed by Parretta (Parretta Antonio. 1999) who concluded that an equivalent refractive index, higher than the glass refractive index, must be used. Since reflection at the interface EVA/PV cell is diffuse, the PV absorptance will be independent of the incidence angle and the angular dependence will only affect the transmittance of the glass and EVA joint. The incidence angle modifier ( $IAM_{PV}$ ) will be used (Duffie John A. 1991). This  $IAM$  will be obtained, for the beam radiation, by two methodologies:

using Fresnel equations of the air/glass interface or using some mathematical expressions obtained by Barker and Norton (Barker G. and Norton P. 2003).

2.8.2 Spectral and angular dependency of the semi-transparent surfaces

Concerning to the semitransparent equivalent surface, an overall hemispherical value has been obtained from the spectral dependency of each layer (glass, EVA and Tedlar). The optical properties of the rear glass are obtained from the TRNSYS database. The angular dependence must include the reflections between the semitransparent surfaces, thus, the net radiation method (Siegel Robert. 2002) will be used.

2.9 Thermal radiation

The thermal radiation between the semitransparent surface and the rear glass; between each glass layer; between the last layer of the rear glass and the room in contact as well as between the glass of the PV laminate and the sky is determined by solving for the net heat transfer between two infinite grey surfaces.

2.9 Electricity generated by the PV

The electricity generated by the PV cells must be expressed as:

$$q_e'' = (\tau\alpha)_n IAM_{PV} G_t P \eta_{PV} \quad \text{W/m}^2 \quad (6)$$

where:

$\eta_{PV}$ : PV efficiency

The PV efficiency is depending on the PV cells temperature and on the incident radiation. To consider these two factors, a linear variation mode has been adopted.

3. GLOBAL DYNAMICAL MODEL

Three TRNSYS types are used to define the façade: the PV laminate type, the air channel type, and the rear glass type. All the types are dynamically coupled by the TRNSYS sequentially solver: the heat conduction sources obtained from the first and third types are transferred to the second type as a Newmann boundary condition. Once the heat transfer coefficients, the mass flow rate and the average wall temperatures are obtained within this type, they are returned back to the first and third type. This is an iterative process which should converge at each time step.

4. NUMERICAL ANALYSIS OF THE HEAT TRANSFER COEFFICIENTS

4.1 Introduction

Many authors have made analysis of cavities or venti-

lated façades under buoyant flows. However they are all limited to laminar symmetric boundary conditions. Within this research a methodology based on Reynolds Averaged Navier Stokes (RANS) CFD simulations is used over laminar and asymmetric flows (Houzeaux G. 2006)

4.2 Validation for the laminar situations

A wide number of simulations have been carried out (256); some of them have been used to validate the CFD model and the numerical assumptions with experimental results (Salom Jaume. 1999) and previous authors (Bar-Cohen A. 1984, Bejan A. 2003, Brinkworth B.J. 2000, Olsson Carl-Olof. 2004, Ramanathan S. 1991).

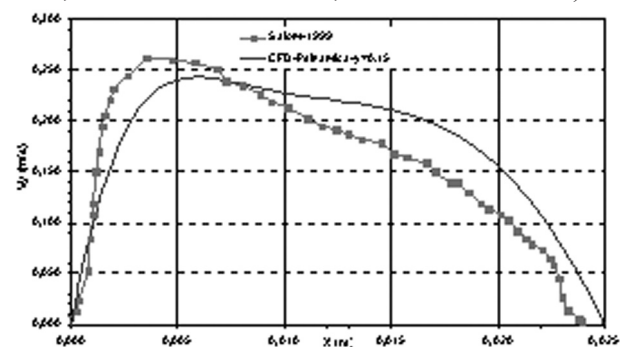


Figure 2. Velocity distribution in the section (y=0.15). uniform wall temperatures. (H/b=12;Ra'=1346.6)

In Figure 2, a comparison between some numerical results and the experimental results is showed. In Figure 3, a comparison of the numerical  $\overline{Nu}$  and the expressions of the different authors is showed

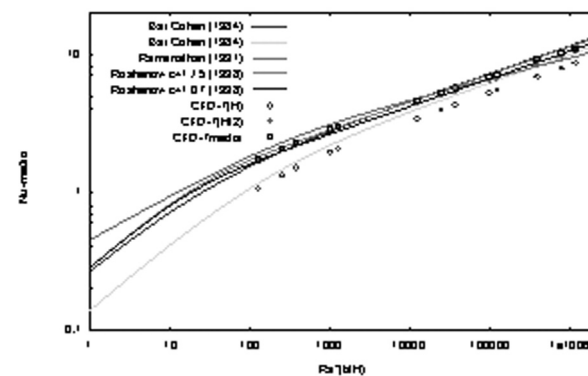


Figure3. Comparison of  $\overline{Nu}$  for symmetric uniform wall heat fluxes

4.3 New correlations for Nu and mass flow rates

As it has previously been mentioned, in the literature no correlations for asymmetric boundary conditions is found, hence, using software SPSS, new correlations have been obtained.

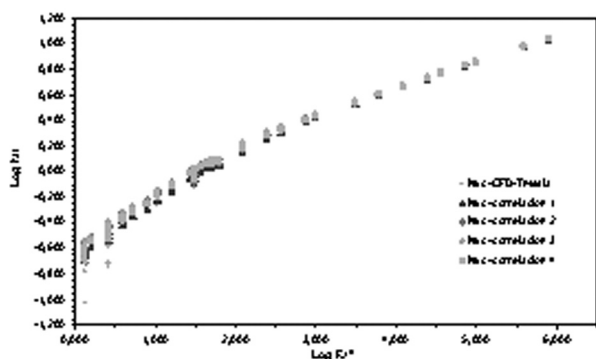


Figure 4.  $\overline{Nu}$  for asymmetric uniform wall heat fluxes

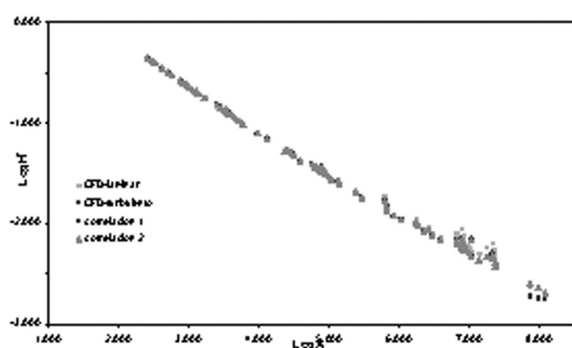


Figure 5.  $H^+$  for asymmetric uniform wall heat fluxes

In Figures 4 and 5, a comparison of these correlations with the numerically calculated results is shown.

## 5. CONCLUSIONS

A strong thermodynamic coupling exists in the double-skin PV façades and this interaction can only be predicted by sophisticated and state-of-the-art simulation techniques as was done in the current study. Three new TRNSYS types have been developed and must be further validated with experiments and real façades monitoring. This research has also clearly show that still a lot of work must be done to clearly define the performance under transient turbulent free convection. Besides, further improvement in the CFD models must be done so that the methodology defined in this research must be extended to this field

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