AN EXPERIMENTAL STUDY OF A NATURALLY VENTILATED CAVITY WALL

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ABSTRACT

Cavity wall is often proposed in the building envelope design as a solution for improving the thermal comfort of the inhabitants and reducing the adverse condensation effects on the building fabric. In order to evaluate the thermal effect of ventilated air gaps on building energy demand and comfort, an experimental ventilated cavity wall has been built and tested. The cavity wall separates two ambients at different temperatures that are assumed to be constant over the time required to perform the experimental analysis. The experiments were carried out for Rayleigh numbers between $2x10^2$ and $2x10^4$ and the results have been compared with the results of a proposed CFD code. Furthermore, an experimental analysis on the ventilated cavity wall is performed against the reference case of closed cavity.

KEYWORDS

Natural convection, cavity wall, CFD model

INTRODUCTION

The applications for which the inner space of a vertical channel is preferred to be ventilated are practically non-limited. From building envelopes to electronic cooling applications, ventilation by free convection is quite used for its efficiency and low implementation cost. Ventilated cavity walls consisting of two walls separated by a continuous vertical air space are widely used in building envelope design. Even though their advantages are obvious in which regards the reduction of the thermal loads during the hot season, a proper decision on the use of ventilated cavity walls as external building envelopes should be made after carefully analyzing their behavior under most likely to occur climate conditions.

Various authors have been studied the phenomenon of natural convection between two vertical flat plates. Among the studies on this subject we mention here those performed by Bodoia et al (1962), Akbary et al (1983) and Aung et al (1972). Closer related to our investigation are the studies performed on natural convection between finitely conducting isothermal vertical plates (Burch et al (1985) and Kim et al (1990)). However these studies are very specific and do not provide information on the heat passing through both vertical walls composing the system, fact that is relevant when performing a building thermal analysis.

In order to assess the performance of the naturally ventilated cavity wall, an experimental model has been built and tested. At the same time, a numerical model that accounts for the convection-conduction conjugate-problem has been developed for comparison and validation purposes. Here are reported some of the results obtained for two gap widths and various temperature differences resulting in Rayleigh numbers ranging from $2x10^2$ to $2x10^4$. In addition to this, the performance of the ventilated cavity wall has been studied against the performance of the closed cavity wall, by experimental means.

EXPERIMENTAL STUDY

In view of studying the thermal behavior of a naturally ventilated cavity wall, a simple model has been built and tested at the laboratory of Civil Engineering Department at Instituto Superior Técnico of Lisbon. The model consists of a gypsum board of 2,4 x 0,7 x 0,0125 m on whose exterior side was built a cavity by simply gluing polystyrene boards onto narrow vertical strips also made of polystyrene. It has been obtained a cavity enclosed between a wall made of gypsum boards, and a wall made of polystyrene boards. The width of the gap is given by the thickness of the vertical strips, as seen in Figure 1. Thin aluminum foil has been added to the cavity facing sides of the walls in order to minimize the radiation effects. The wall has been tested utilizing a climatic chamber of 3x2x3 m. The tested model had one side facing the laboratory indoor and the other one facing the climatic chamber interior. Thus, selecting the climatic chamber temperature relatively to the laboratory indoor temperature induced the temperature difference across the cavity wall. Temperature readings have been performed using K-type thermocouples placed on all surfaces to be monitored, at four different heights. The temperature of the air gap formed between the two vertical walls was also measured utilizing K-type thermocouples. In addition, flux meters were placed on the exterior sides of the tested model. The thermocouples and the flux meters were controlled with a data-logging device (Campbell Scientific CR10X model). Figure 1 shows the cross-sectional set-up and the position of the sensors.

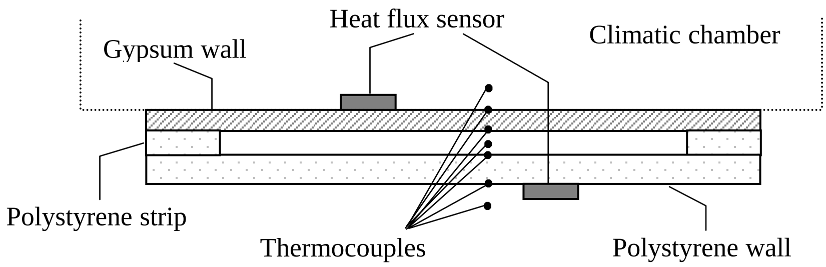


Figure 1: The cross-sectional set-up and the position of the sensors

The model has been monitored for two different gap widths, b equal to 0.03 m and 0.06 m respectively, and three different temperatures of the climatic chamber, T_{in} approximately equal to 30, 40, and 50°C. The data-logging device was set-up to take readings every minute over a period of 10 hours, for each case of study.

NUMERICAL STUDY

The physical model employed for the numerical study is described below. The heat transferred into the gap from the warmer wall cause on the internal fluid density differences, from which result buoyancy forces that induce upward flow in the channel. The fluid driven by the buoyancy forces enters the channel at the ambient conditions. The geometrical configuration of the channel under investigation is shown in Figure 2.

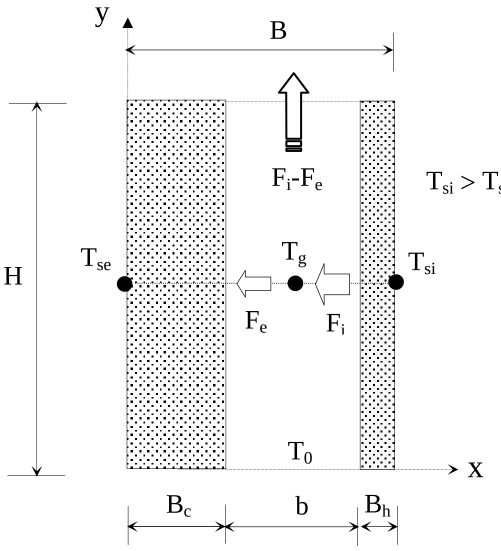


Figure 2: Geometry of the thermal flow problem

In order to obtain detailed information on the velocity and temperature fields on the cavity wall system, a CFD computer code that accounts for the convection-conduction conjugate problem has been developed. The flow is assumed to be nearly compressible, steady, laminar and two-dimensional. The program solves the transport equations for momentum and enthalpy. These equations can be written in a generalized form for the dimensionless variable Φ, as in Patankar (1980):

$$\frac{\partial}{\partial x}(\rho u \Phi) + \frac{\partial}{\partial y}(\rho v \Phi) = \frac{\partial}{\partial x}\left(\Gamma_{\Phi}\frac{\partial \Phi}{\partial x}\right) + \frac{\partial}{\partial y}\left(\Gamma_{\Phi}\frac{\partial \Phi}{\partial y}\right) + S_{\Phi} \tag{1}$$

The associated boundary conditions and other constraints can be stated as follows:

$x = 0;\ 0 \le y \le H:\ u = v = 0,\ T = T_{se}$
$x = B;\ 0 \le y \le H:\ u = v = 0,\ T = T_{si}$
$0 \le x \le B_c;\ y = 0, H:\ \partial T/\partial y = 0$
$B - B_h \le x \le B;\ y = 0, H:\ \partial T/\partial y = 0$
$B_c < x < B_c + b;\ y = 0:\ T = T_0,\ p = p_0,\ \partial[u, v]/\partial y = 0$
$B_c < x < B_c + b;\ y = H:\ p = p_0 - \rho_0 g H,\ \partial[u, v, T]/\partial y = 0$

$x = B_c, B - B_h:\ k_s\,[\partial T/\partial x]_s = k_f\,[\partial T/\partial x]_f$
$0 < x \le B_c;\ 0 \le y \le H:\ u = v = 0$
$B - B_h \le x < B;\ 0 \le y \le H:\ u = v = 0$

To solve the non-linear second order partial differential equations on the form given by Eqn. (1), they were discretized on a 50 x 50 grid according to a control volume based finite difference method. The employed grid is not uniform over the gap. The grid is getting finer as closer it gets to the wall. This approach ensured enough detail on the flow and temperature fields in the different regions of interest of the computational domain. The pressure-velocity linkage that ensures the conservation of mass has been solved by adopting staggered grids as suggested by Patankar (1980).

For solving the discretized equations for all the dependent variables (Φ), a line-by-line iterative procedure (TDMA) is used. To enhance the stability of the numerical solution under-relaxation techniques are applied to all equations.

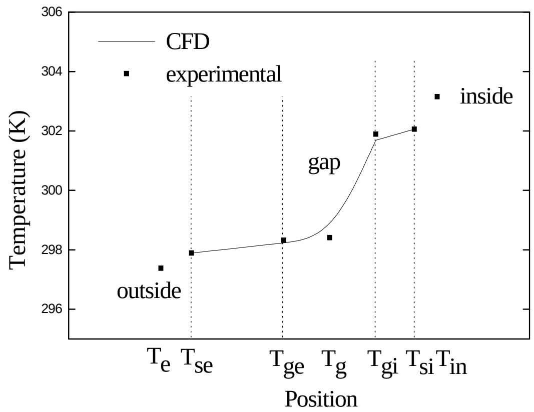
RESULTS

In a first step, the wall component has been monitored and tested for three temperature differences and two distinct gap widths. The temperatures and widths were chosen in such a way to assure a wide range of values of Rayleigh number. The temperature inside the laboratory was almost constant during the experimental work, so that, for small time intervals, it may be assumed that steady state conditions occurred. The temperature readings of the thermocouples placed on the exterior surfaces of the walls, T_{se} and T_{si}, averaged over one representative hour, were then used as input data for the numerical model. These values, together with the resulting Rayleigh numbers and the laboratory indoor and climatic chamber correspondent temperatures, are given in Table 1.

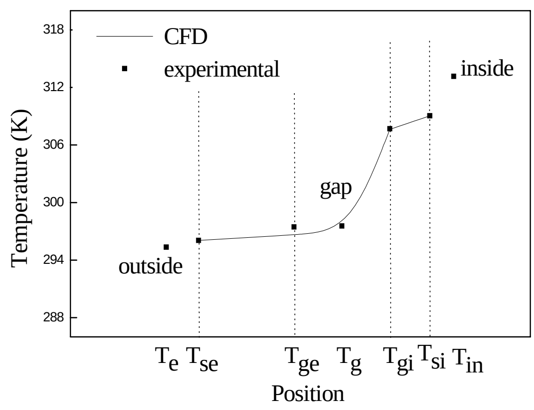
TABLE 1
Data input used to run the numerical model

b = 0,03 m					b = 0,06 m				
T_e	T_{se}	T_{in}	T_{si}	Ra_b	T_e	T_{se}	T_{in}	T_{si}	Ra_b
24,2 ºC	24,7 ºC	30 ºC	28,9 ºC	215	22,1 ºC	22,3 ºC	30 ºC	28,81 ºC	$4{,}98 \times 10^{3}$
22,6 ºC	22,9 ºC	38,3 ºC	35,9 ºC	615	22,1 ºC	22,17 ºC	39 ºC	35,9 ºC	$1{,}023 \times 10^{4}$
22,8 ºC	23,9 ºC	48,6 ºC	43,8 ºC	972	23,8 ºC	24 ºC	48,6 ºC	43,5 ºC	$1{,}45 \times 10^{4}$

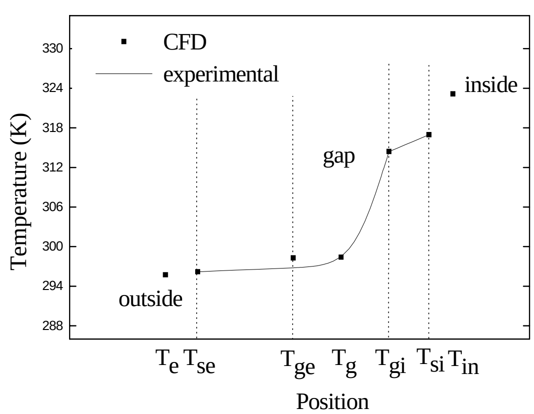
Simulations have been performed for each set of T_{se} and T_{si} shown in Table 1, and the calculated transversal temperature profiles near the outlet are plotted against the correspondent experimental results in Figure 3.



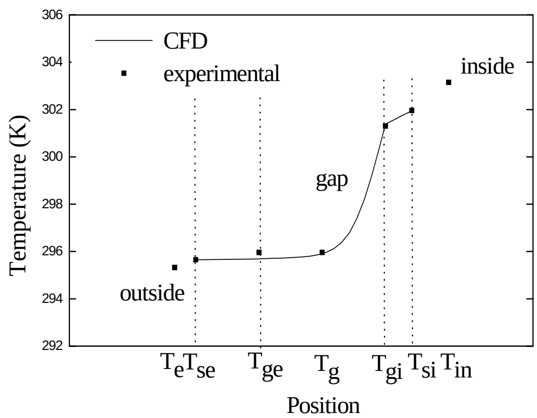
$Ra_b = 215$



$Ra_b = 615$



$Ra_b = 972$



$Ra_b = 4{,}98 \times 10^{3}$

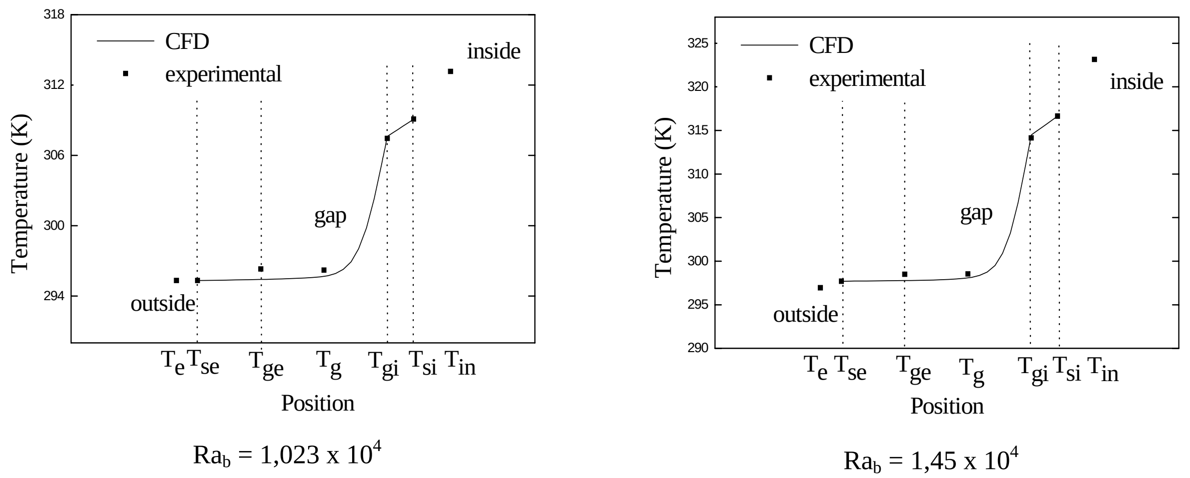


Figure 3: Experimental versus numerical results

A second approach consisted in studying experimentally the performance of the ventilated cavity wall against the performance of the closed cavity wall that was obtained by simply covering the inlet and the outlet of the vertical cavity under investigation. Temperatures and thermal fluxes were monitored for a single gap width b equal to 0.03 m, when the model was subjected to three temperature differences. The temperatures and the thermal fluxes measured for each particular case emerged here are given in Table 2.

TABLE 2
Temperatures and thermal flux measurements resulted from the experimental analysis: ventilated versus non-ventilated cavity

		Naturally ventilated cavity				Closed cavity			
T_e	T_{in}	T_{se}	T_{si}	F_e	F_i	T_{se}	T_{si}	F_e	F_i
16,5 ºC	30 ºC	17,3 ºC	26,9 ºC	0,82 W/m^2	18,56 W/m^2	17 ºC	28,3 ºC	6,34 W/m^2	6,5 W/m^2
17,5 ºC	40 ºC	18,9 ºC	34,4 ºC	3,21 W/m^2	37,14 W/m^2	20 ºC	37,3 ºC	10,25 W/m^2	10,5 W/m^2
17,5 ºC	50 ºC	19,2 ºC	41,61 ºC	4,9 W/m^2	62,9 W/m^2	20,4 ºC	45,9 ºC	16,9 W/m^2	16,9 W/m^2

Figure 4 shows the thermal performance of the ventilated cavity wall against the reference-closed cavity wall. N represents the number of times the ventilated cavity wall fluxes (F_i, F_e) exceed the values taken as reference.

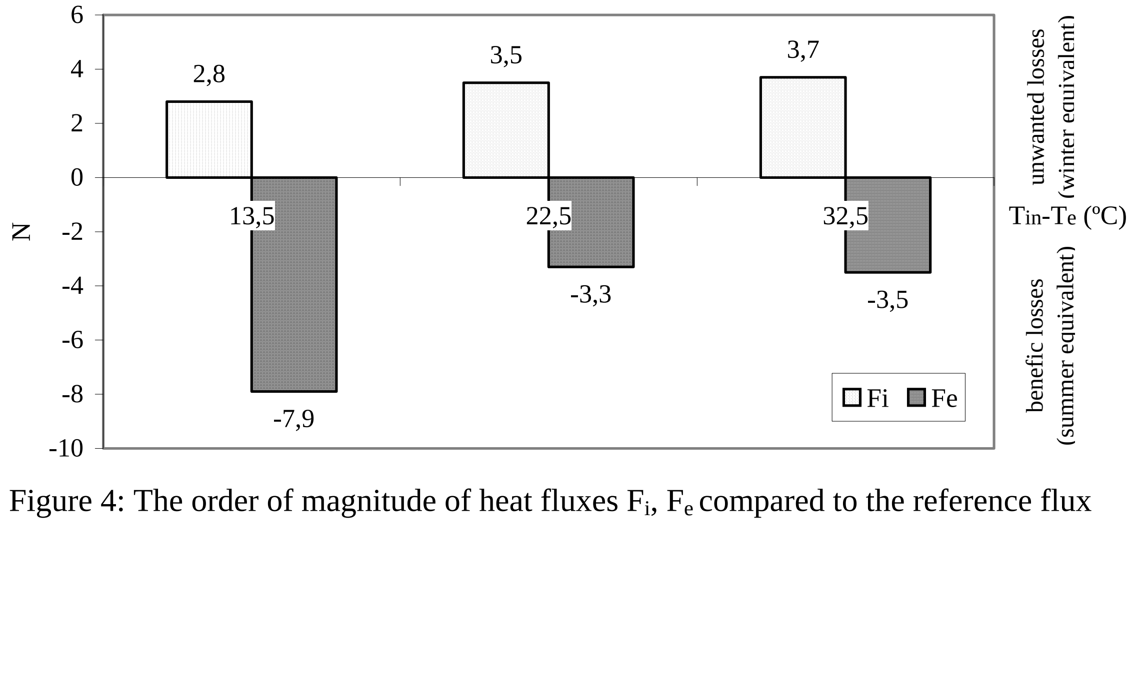


Figure 4: The order of magnitude of heat fluxes F_i, F_e compared to the reference flux

DISCUSSION AND CONCLUSIONS

An experimental model has been built and tested in order to assess the performance of the naturally ventilated cavity wall. At the same time, a numerical model that accounts for the convection-conduction conjugate-problem is proposed. As it can be seen in Figure 3, the comparison between the numerical and experimental results shows a reasonably good agreement, the noticed dissimilarities being within what would be expectable in view of the complexity of phenomena involved. Furthermore, the numerical results emphasize that the greater the Rayleigh number, the higher the temperature drop across the gap.

Looking now at the heat fluxes F_i and F_e given in Table 2 for the closed cavity case, one may notice that, for each set of T_{in} and T_e, their values are equal (or almost), fact that confirms our expectations.

The present experimental work has been carried out having always set-up the temperature of the climatic chamber higher than that of the laboratory ambient. In reality, such a situation would correspond to wintertime. In this case the heat flux crossing the warmer wall (F_i) should be used to assess the performance of the cavity wall. Nevertheless, the experiments may also provide useful information about those limited situations occurring in summer when the presence of solar radiation warms-up the external envelope (seen as T_{si} in this experimental work) and the indoor and outdoor temperatures are almost equal. For this case, the heat flux crossing the colder wall (F_e) in the present experiments turns out to be a heat gain and should be used to assess the performance of the ventilated cavity wall in summer. Figure 4 shows that ventilated cavity walls can be advantageously used in summer conditions. For wintertime, however, the use of the same system can lead to great heat losses when compared to the traditional cavity wall.

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NOMENCLATURE

B, b	wall component width, gap width
H	height
T	temperature
F	heat flux
u, v	velocity components
p, ρ, μ, k	pressure, density, viscosity and thermal conductivity
g	gravitational acceleration
β	coeff. of volumetric thermal expansion; $\beta = 1/T_0$
α	thermal diffusivity
Γ	general diffusion coefficient
Ra_b	reduced Raleigh no.; $Ra_b = \rho g \beta b^4 (T_{si}-T_0)/(H \mu \alpha)$

Subscripts

i, in	interior
e	exterior
si	surface facing interior
se	surface facing exterior
c	cold
h	hot
g	gap
ge	exterior wall surface facing the gap
gi	interior wall surface facing the gap
s, f	solid, fluid
0	inlet