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COBEM-2017-1080 NUMERICAL AVALIATION OF AIRFLOW IN NATURALLY VENTILATED DSF

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Abstract. *Double skin façade (DSF) consists in a construction of an external façade to the building in order to form a layer of circulating air. DSFs have been widely used throughout the world contributing to the thermal performance of built environment. In this work, a numerical evaluation of the airflow in a DSF is made and the relation with the temperature in the faces is observed. Commercial software implements a numerical model to evaluate the heat convection in the cavity and the airflow from data previously collected in a prototype with the cavity thickness of air layer varying by 0.1 m and 0.2 m. The boundary conditions are the temperature of faces of the DSF and the wind velocity at the inlet. The results show that the thickness of air layer plays an important role since it varies the wind speed and the exchange of heat between the layers and in this case, the thickness of air layer in 0.1 m presents best results.*

Keywords: *Double skin façade, airflow, heat convection, numerical simulation*

1. INTRODUCTION

Efficient policies for energy use have been presents in Brazil since 2003 by means of the Procel Edifica program. This program develops and support projects for energy conservation in residential, commercial, and publics buildings. According to the program, the potential reduction of consumption is estimated to be around 30% with implementation of energy efficiency actions for the existing buildings, while for new buildings the potential reduction is around 50% (BRASIL, 2011).

Following this context, DSFs has become a tool in the constructive sector and consist of a building with a normal façade added an external façade usually made of glass, so the space between the layers is filled with air (Barbosa and Ip, 2014). The basic operation of a DSF is due the chimney effect that occurs when exist a difference of density between the air heated by solar radiation and the external fresh air. The heated air is expelled through the upper opening of the device (Ding, *et al.*, 2012). The entrance of air into the cavity of the façade can occur naturally or mechanically with the use of fans and exhaust fans (Nicol, *et al.*, 2012).

Therefore, a numerical study is performed considering a naturally ventilated DSF installed in a prototype with the view to evaluate the thermal behavior and airflow focusing on the temperature reduction to achieve the thermal comfort inside the prototype.

2. METHOD

In order to perform the heat transfer between the surfaces and the airflow reliably, the analyses of computational fluid dynamics simulations is an alternative to energy modelling programs. The prototype on which this study is based is built in masonry and has a DSF built in cement slabs according recent work of Souza (2017) showed in Fig.1.

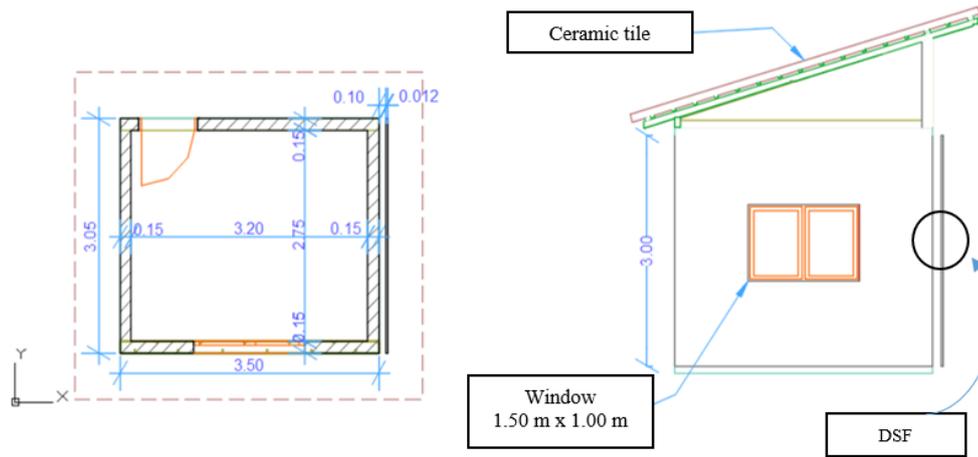


Figure 1. Prototype cell that this work is based

2.1 Numerical simulation

The simulated model is designed with focus only on the DSF as show in the Fig.2 and in the same figure is possible to note the thickness of air layer variation (0.2 m and 0.1 m).

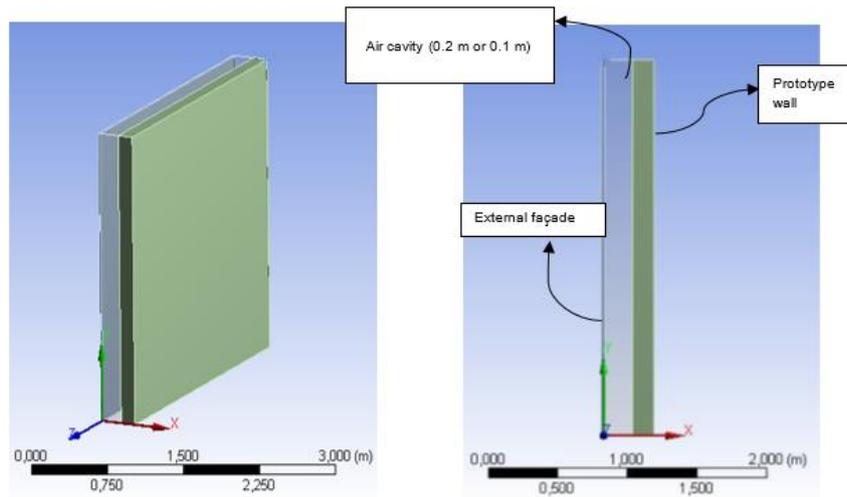


Figure 2. Simulated model

Then a reliable mesh is generated and the boundary conditions are defined. The boundary conditions are defined from the mean temperatures of the internal face of the façade and the internal face of the prototype, and the wind speed at the lower opening of DSF. These data are obtained previously by means of a experimental study of Souza (2017). The simulations are made considering an interval of every two hours for a period of one day that is from 0:00 a.m. to 10:00 p.m. This analysis is in permanent regime for each instant, and it can be denominated as pseudo-transient (Fortuna, 2000), allowing obtain results for each defined time.

The convergence criteria are defined as a standard value of 1×10^{-4} RMS value and a minimum of 1 (one) and maximum of 100 (one hundred) iterations.

2.2 Mathematical model

To the thermal performance of a DSF, Guillén *et al.* (2014) state that the solid and fluid phases need to be distinguished. So for the solid part, the temperature distribution is provided by the heat conduction equation (Eq.1).

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

Where k is the material thermal conductivity ($\text{W.m}^{-1}.\text{K}^{-1}$), T is the temperature (K), ρ is the specific mass (kg.m^{-3}), c_p is the specific heat ($\text{J.kg}^{-1}.\text{K}^{-1}$), and t is the time (s).

Considering one-dimensional conduction in a plane plate, Eq. (1) can be rewrite as Eq. (2).

$$\rho c_p \left(\frac{\partial T}{\partial t} \right) = k \frac{\partial^2 T}{\partial x^2} \quad (2)$$

The radiation from the faces of the DSF in contact with the fluid is described by the Stefan-Boltzmann equation (Eq.3).

$$E = \epsilon \sigma T^4 \quad (3)$$

Where E is the rate of radiative energy emitted per unit area (W.m^{-2}), ϵ is the surface emissivity, and σ is the Stefan-Boltzmann constant with value equals to $5.6704 \times 10^{-8} \text{W.m}^{-2}\text{K}^{-4}$.

The airflow in a DSF can be given by the continuity equation (Eq. 4), momentum equations (Eqs. 5, 6, 7) and energy equation (Eq. 8).

$$\frac{\partial \rho}{\partial t} + \frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} = 0 \quad (4)$$

$$\rho \frac{\partial(u)}{\partial t} + \rho \frac{\partial(uu)}{\partial x} + \rho \frac{\partial(vu)}{\partial y} + \rho \frac{\partial(wu)}{\partial z} = -\frac{\partial p}{\partial x} + \mu \left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2} \right) \quad (5)$$

$$\rho \frac{\partial(v)}{\partial t} + \rho \frac{\partial(uv)}{\partial x} + \rho \frac{\partial(vv)}{\partial y} + \rho \frac{\partial(wv)}{\partial z} = -\frac{\partial p}{\partial y} + \mu \left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} + \frac{\partial^2 v}{\partial z^2} \right) + g\rho\beta(T - T_{ref.}) \quad (6)$$

$$\rho \frac{\partial(w)}{\partial t} + \rho \frac{\partial(uw)}{\partial x} + \rho \frac{\partial(vw)}{\partial y} + \rho \frac{\partial(ww)}{\partial z} = -\frac{\partial p}{\partial z} + \mu \left(\frac{\partial^2 w}{\partial x^2} + \frac{\partial^2 w}{\partial y^2} + \frac{\partial^2 w}{\partial z^2} \right) \quad (7)$$

$$\frac{\partial T}{\partial t} + \frac{\partial(uT)}{\partial x} + \frac{\partial(vT)}{\partial y} + \frac{\partial(wT)}{\partial z} = -\frac{1}{\rho} \frac{\partial p}{\partial x} + \alpha \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} \right) \quad (8)$$

Where u , v , w are the velocity components (m.s^{-1}) in the directions x , y , z respectively, ρ is the specific mass of the fluid (kg.m^{-3}), α is the thermal diffusivity of the fluid ($\text{m}^2.\text{s}^{-1}$), ν is the kinematic viscosity of the fluid ($\text{m}^2.\text{s}^{-1}$) and β is the coefficient of thermal expansion of the fluid (K^{-1}).

Considering a flow with a Reynolds number above 4000, turbulence is inherent. Thus, it becomes necessary to use a turbulence model. Among the turbulence models, the k- ϵ model stands out due to its robustness, precision, and stability. The k- ϵ model is a two-equation model based on the concept of turbulent viscosity, where k represents the kinetic energy of the turbulence and ϵ is the viscous dissipation of the turbulent kinetic energy, and then the turbulent viscosity is introduced to the flow. The turbulent viscosity is given by Eq. (9) (Gabbi, 2013).

$$\mu_t = c_\mu \rho \frac{k'^2}{\epsilon} \quad (9)$$

Where μ_t is the dynamic turbulent viscosity (N.s.m^{-2}), C_μ is a constant with a defined value equal to 0.09, k' is the turbulent kinetic energy ($\text{m}^2.\text{s}^{-2}$), and ϵ is the viscous dissipation of turbulent kinetic energy ($\text{m}^2.\text{s}^{-3}$).

The values of k' and ϵ can be obtained from the differential equation of transport of the turbulent kinetic energy (Eq. 10) and the transport differential equation of the specific viscous dissipation rate (Eq. 11).

$$\rho \frac{\partial(\rho k')}{\partial t} - \nabla \cdot (\rho \mathbf{V} k') = \nabla \cdot \left[\left(\mu + \frac{\mu_t}{\sigma_k} \right) \nabla k' \right] + P_k - \rho \epsilon \quad (10)$$

$$\rho \frac{\partial(\rho \epsilon)}{\partial t} - \nabla \cdot (\rho \mathbf{V} \epsilon) = \nabla \cdot \left[\left(\mu + \frac{\mu_t}{\sigma_\epsilon} \right) \nabla \epsilon \right] + \frac{\epsilon}{k} (C_{\epsilon 1} P_k - C_{\epsilon 2} \rho \epsilon) \quad (11)$$

Where μ_t is the turbulent dynamic viscosity (N.s.m^{-2}), P_k is the production of turbulence due to viscous and buoyancy forces ($\text{kg.m}^{-1}.\text{s}^{-3}$), σ_k is a constant equal to 1.0, σ_ϵ is a constant equal to 1.3, $C_{\epsilon 1}$ is a constant equal to 1.44, and $C_{\epsilon 2}$ is a constant equal to 1.92.

3. RESULTS AND DISCUSSION

For the cavity configuration thickness in 0.2 m, the beginning of the heating of the external façade occurs around 8:00 a.m. and the peak temperature of the façade around 4:00 p.m. Figure 3 shows the variation of the temperature in a DSF with cavity thickness in 0.2 m at 4:00 p.m.

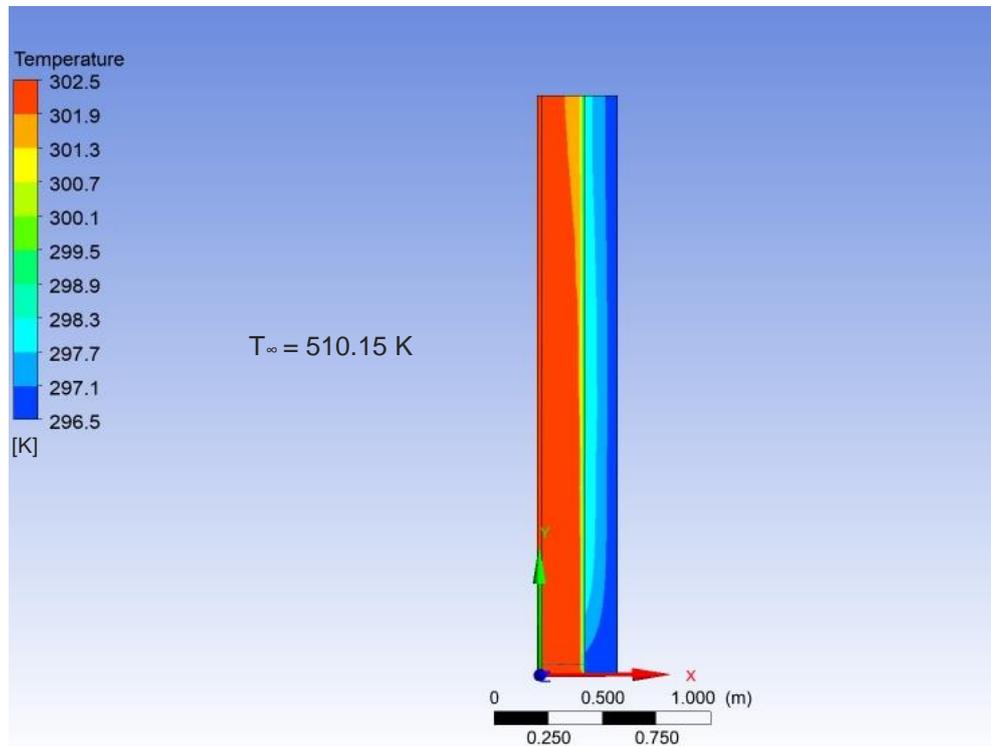


Figure 3. Temperature variation in a DSF at 4:00 p.m. (thickness of air layer in 0.2 m)

In relation to the airflow, the velocity values at the inlet and outlet of the cavity are presented in Fig. 4.

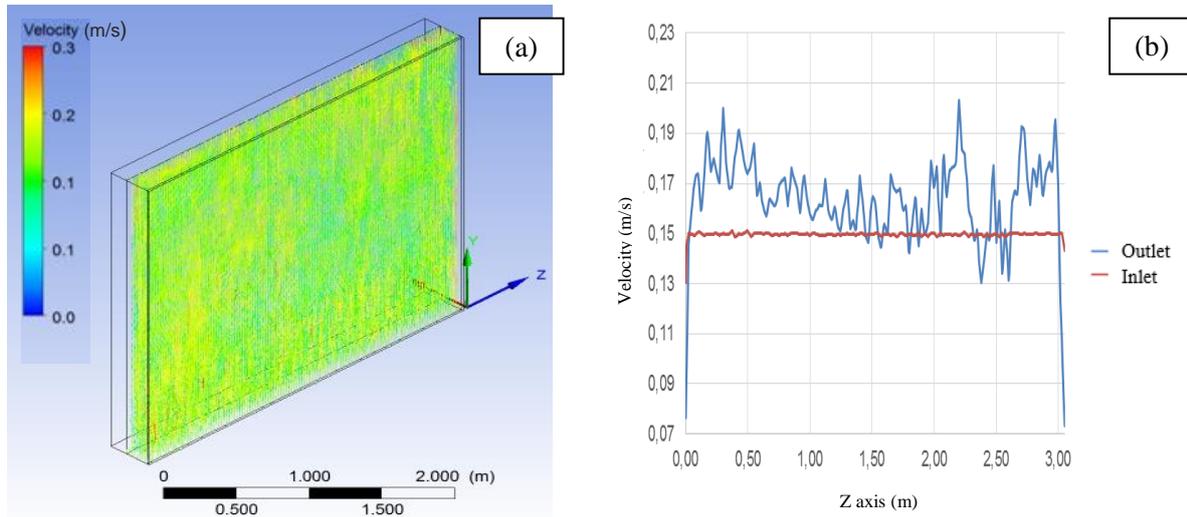


Figure 4. Airflow in a DSF with thickness of air layer in 0.2 m: (a) Velocity vectors; (b) Velocity values at the inlet and outlet.

Figure 4 shows that the air velocity increase in the outlet but with a great buoyancy. In a similar way, for the thickness of air layer at 0.1 m, the temperature variation at 4:00 p.m. is showed in the Figure 5.

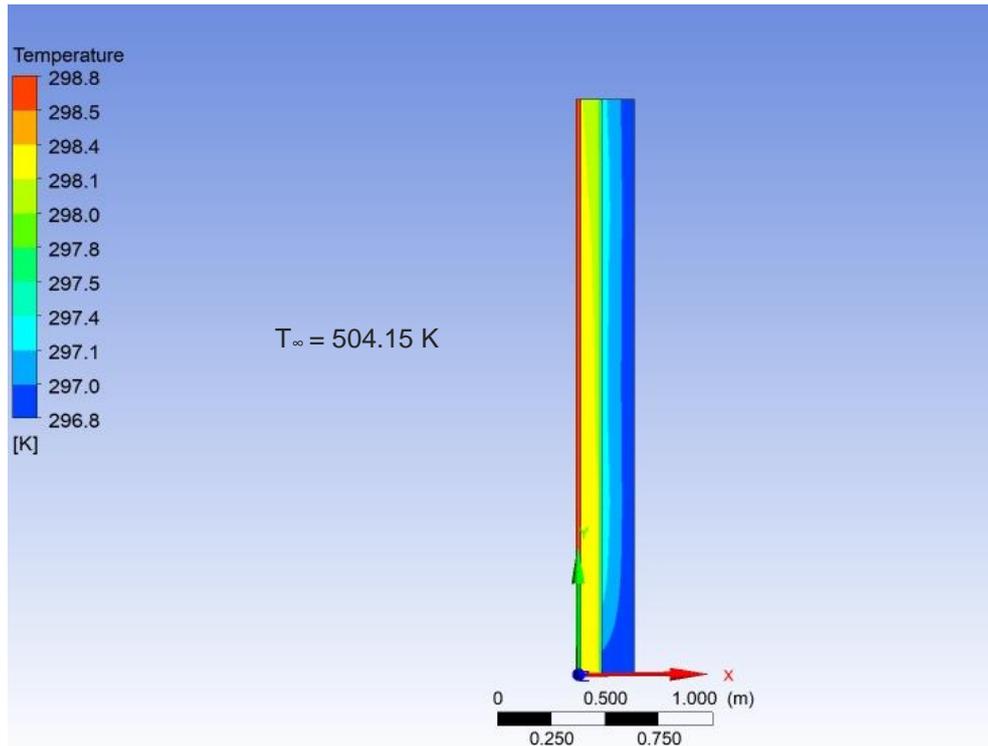


Figure 5. Temperature variation in a DSF at 4:00 p.m. (thickness of air layer in 0.1 m)

The airflow to the thickness of air layer in 0.1 m is showed in Figure 6 where is possible to see that in this case the air velocity at the inlet and outlet are larger than the thickness of air layer in 0.2 m that is explained due to difference of thickness of air layer.

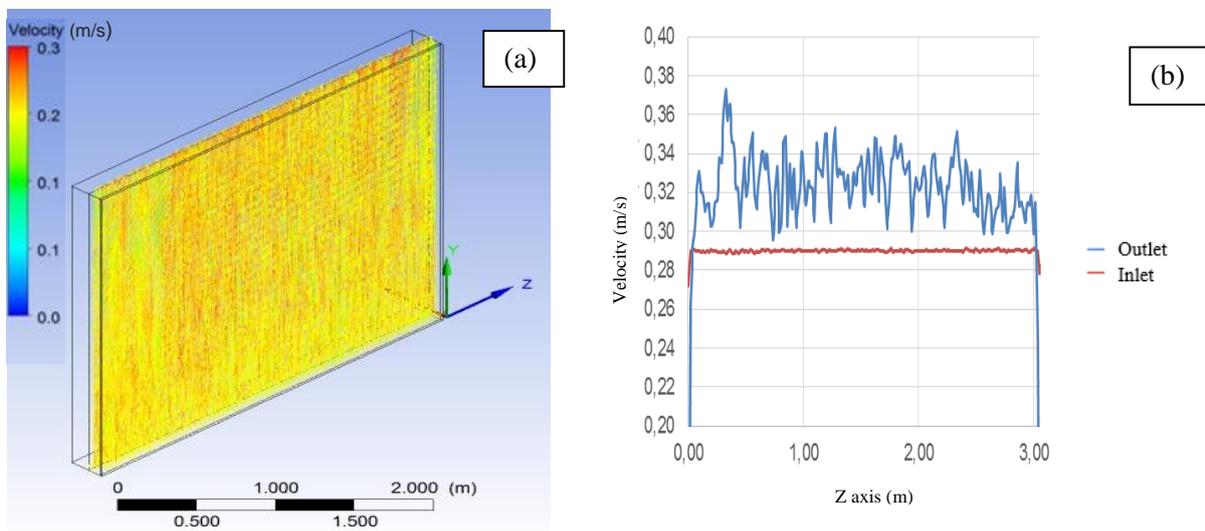


Figure 6. Airflow in a DSF with thickness of air layer in 0.1 m: (a) Velocity vectors; (b) Velocity values at the inlet and outlet.

Comparing the Fig. 4 and the Fig. 6, it is possible to note that the air inside the cavity shows greater values of air velocity to the thickness of air layer in 0.1 m, which implies in higher or lower temperatures in the faces in contact.

4. CONCLUSIONS

It is possible to concluded that a DSF is effective concerning to the thermal performance in a building by reducing the temperature on the inner face of the wall. The results shows that the thickness of air layer plays an important role in

the heat exchange, once the inlet velocity is given as a function of the thickness of air layer. For these simulations, the thickness of air layer in 0.1 m presents better results related to the decrease temperature in the DSF.

However, one must be aware of the fact that an optimum thickness of air layer is dependent on factors such as local climatic conditions beyond the characteristics of the DSF such as material and size.

5. ACKNOWLEDGEMENTS

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