

CONSTRUCTAL DESIGN APPLIED TO INVESTIGATE THE INFLUENCE OF GEOMETRY ON THE MASS FLOW RATE OF AN INCLINED PASSIVE WALL SOLAR CHIMNEY ATTACHED TO A ROOM

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ABSTRACT

The present work aims to analyze the turbulent flow in an inclined passive wall solar chimney attached to a room, evaluating the influence of its geometry on the thermal performance of the building (measured by the mass flow rate in the chimney exit) by means of Constructal Design. The flow is considered turbulent, incompressible, under natural convection heat transfer, transient and in a two-dimensional domain that simulates a solar chimney attached to a room. Time-averaged conservation equations of mass, momentum, and energy are numerically solved with the finite volume method using the commercial package FLUENT. For closure modeling of turbulence, it is employed the standard $k - \varepsilon$ model. Chimney and room areas are the problem constraints. Moreover, the problem is subjected to three degrees of freedom: the ratio between the inlet opening size and chimney height (H_i/H_a) (which is maintained constant in the present investigations, $H_i/H_a = 0.05$); ratio between the width of inferior base of the chimney and its height (W_g/H_a); and the ratio between the exit air gap and the inferior base widths of the chimney (W_e/W_g). The latter two degrees of freedom are varied. Results showed that the degrees of freedom analyzed have a strong influence on the mass flow rate of the air in the building, confirming that the geometrical configuration of solar chimney can be important for the improvement of thermal conditions on the attached building.

Keywords: solar chimney; constructal design; solar energy; natural ventilation

NOMENCLATURE

A_t	Area of the chimney [m ²]
c_p	Specific heat capacity [J/(kg·K)]
g	Gravitational acceleration [m/s ²]
H	Height of the ventilated space [m]
H_a	Height of the absorber wall [m]
H_i	Size of the inlet opening to the chimney [m]
k	Turbulent kinetic energy [m ² /s ²]
L	Length of the ventilated space [m]
p	Pressure [Pa]
Pr	Prandtl number
Pr_t	Turbulent Prandtl number
T	Temperature [K]
t	Time [s]
\bar{u}	Time-averaged velocity in the x direction [m/s]
\bar{v}	Time-averaged velocity in the y direction [m/s]
W_e	Exit air gap width [m]

W_g	Width of the inferior base of chimney [m]
W_i	Size of the inlet window [m]
W_t	Thickness of the absorber wall [m]
x	Horizontal coordinate [m]
y	Vertical coordinate [m]
$^{\circ}C$	Degree Celsius

Greek symbols

α	Thermal diffusivity [m ² /s]
α_t	Turbulent thermal diffusivity [m ² /s]
β	Coefficient of thermal expansion [1/K]
Δt	Time-step [s]
ε	Dissipation rate of turbulent kinetic energy [m ² /s ³]
κ	Thermal conductivity of the fluid [W/(m·K)]
μ	Dynamic viscosity of the fluid [kg/(m·s)]
μ_t	Turbulent viscosity of de fluid [kg/(m·s)]
ρ	Density of the fluid [kg/m ³]

ν Kinematic viscosity of the fluid [m^2/s]

Subscripts/Superscripts

m	Once maximized
2m	Twice maximized
o	Once optimized
2o	Twice optimized
-	Time average operator

INTRODUCTION

Global warming is a very widespread topic, being a common sense that the temperatures in the planet is increasing each year. The 1990s and 2000s were the hottest in the last thousand years according to the website of Brazilian National Institute for Space Research (INPE – *Instituto Nacional de Pesquisas Espaciais*) (INPE, 2022). According to the same website, the projections of the Intergovernmental Panel of Changes climate (IPCC) indicate that in the next 100 years it could occur an increase on the mean global temperature in the range between 1.8°C and 4.0°C (IPCC, 2022). According to Technical Note EPE 030/2018, the increase of air conditioning thermal loads can increase not only the general demands of energy, but also can cause overloads in the generation and distribution of energy in peak hours during the days, impacting the national energy system (EPE, 2018). In addition, the increase on the demand by air conditioner devices can be related with the augmentation of environmental impact, including the emission of greenhouse gases. According to the Projeteer platform, the building sector is responsible currently by more than 40% of total consumed electricity, generating great environment impact and high energy fares for the consumers (Projeteer, 2023).

Additionally, the concern with a sustainable building until few years ago was restricted. In general, most of the residential buildings were built considering only economic and functional issues, i.e., without comprehensive awareness about energy consumption and environmental implications (Abdeen *et al.*, 2019). Gradually, this situation is changing around the world through polices and a greater awareness of engineers and architects on the projects of homes and buildings. In Brazil, the evidence of that change can be noticed in the tag *PBE Edifica*, developed as part of the Brazilian Program of Labeling (PBE – *Programa Brasileiro de Etiquetagem*). The tag concession is supplied in the project step and after the building construction and it has the purpose of encouraging the conservation and efficient use of natural resources in Brazilian buildings, reducing wastes and environmental impacts.

Thus, from aspects considered in a sustainable building, is the search for naturally ventilated environments. According to Neves and Roriz (2012), natural ventilation is an important strategy for passive cooling in buildings. In this sense, solar chimney can

provide improvements in the natural ventilation in buildings, using the solar energy as renewable and clean source of airflow momentum in the built environment (Abdeen *et al.*, 2019).

Several studies referring to solar chimney have been carried out over the years, more precisely since the 90s, seeking to provide a way to improve the mass flow rate of air in the environment connected to the chimney, as well as, to yield an adequate level of thermal comfort for the users. Khanal and Lei (2011) presented one overview about the developed studies. Authors proved that the use of solar chimney could be an excellent strategy for passive ventilation and gain in natural ventilation and thermal comfort in the building. Neves and Roriz (2012) developed theoretical predictions to present and discuss procedures for estimating the potential on the use of solar chimneys for cooling of low-rise buildings, concluding that the theoretical models have potential to be applied in the evaluation of the solar chimney performance. Khanal and Lei (2012) observed the occurrence of reverse flow at the exit of solar chimney for its conventional model of vertical wall when the thickness of thermal boundary layer is lower than the width of chimney exit, leading to the reduction of mass flow rate in the building. As a solution to reduce the reversed flow, increase the mass flow rate and, consequently, achieve a better performance in the ventilation of the environment, authors proposed a new concept of solar chimney named inclined passive wall solar chimney (IPWSC). For this, the flow inside the chimney was investigated using a numerical procedure supported by a flow visualization experiment. IPWSC was experimentally studied by Khanal and Lei (2014), where the authors demonstrated that this kind of chimney presents significant improvements in the ventilation rate compared to conventional chimneys once the airflow rate increased with the augmentation of the wall angle. Khanal and Lei (2015) also carried out a numerical investigation of turbulent flow with natural convection heat transfer in the IPWSC and proved the efficiency of this chimney model in the ventilation gain. In addition, it was demonstrated that for different heat fluxes the inclination angle of 4° led to the best ventilation of the building.

Concerning the application of Constructal Design for the solar chimney, Rajão (2016) carried out a study where it was compared the numerical methodology developed with FLUENT software with the results obtained by Khanal and Lei (2015). Results of mass flow rate of air and thermal fields were in good agreement. The work also evaluated, through the Constructal Design method, the effect of geometric ratios on the mass flow rate of air obtained in the environment to be ventilated and in the solar chimney considering a constant heat flux in the passive wall. The maximum mass flow rate was 71.6% higher than the one obtained for the worst configuration. Afterwards, Vieira *et al.* (2017) also used the

Constructal Design method, but for the numerical investigation of a different concept of solar chimney called *SCPP* (solar chimney power plant), which makes it possible to transform solar energy into electricity. The results showed that the geometric parameters of the collector and the chimney have a strong influence on the available power.

Other important studies in the area have been developed recently. For example, Serageldin *et al.* (2018) proposed an optimization of the solar chimney coupled to a soil-air heat exchanger to heat a building. Abdeen *et al.* (2019) carried out an experimental and numerical study in the hot season in Egypt with the aim of gaining thermal comfort in the coupled environment. Afterwards, Layeni *et al.* (2020) studied the possibility of applying a solar chimney with a dual purpose, that is, for ventilation and energy generation. Shi *et al.* (2020) presented a project of solar chimney to be used in a real building, considering energy savings and fire safety. Recently, Zhang *et al.* (2021) presented a literature review evidencing the solar chimney as an excellent passive ventilation strategy used to increase natural ventilation and provide thermal comfort.

Despite the various investigations and research on solar chimneys presented in the literature, few studies have been performed in the literature dealing with IPWSC, especially seeking to understand the influence of geometry on the behavior of the system. Thus, in the present work, the Constructal Design method was used for the geometric investigation of a passive inclined wall solar chimney attached to a built environment so that it is possible to investigate the effect of geometric ratios on the mass flow rate of air in the device, which is the performance parameter to be evaluated throughout the simulations.

MATHEMATICAL MODELLING

For the solution of the problem, the flow is assumed to be incompressible, turbulent, transient, with natural convection heat transfer in a two-dimensional domain. Thermal radiation is treated only indirectly, i.e., it is considered a heat flux imposed on the passive wall that simulates the effect caused by the incidence of solar radiation. The thermophysical properties are kept constant, except the density, which is varied with the Boussinesq approximation, generating the flow due to buoyance forces generated by temperature differences along the domain. For the treatment of turbulence, Reynolds Averaged Navier-Stokes (RANS) modeling is considered and the standard turbulence model $k-\varepsilon$ is adopted to close the time-averaged equations. Thus, the time-averaged governing equations of mass, momentum in the x and y directions and energy are given, respectively, by (Bejan, 2013):

$$\frac{\partial \bar{u}}{\partial x} + \frac{\partial \bar{v}}{\partial y} = 0 \quad (1)$$

$$\rho \left[\frac{\partial \bar{u}}{\partial t} + \bar{u} \frac{\partial \bar{u}}{\partial x} + \bar{v} \frac{\partial \bar{u}}{\partial y} \right] = -\frac{\partial \bar{p}}{\partial x} + (\mu + \mu_t) \left(\frac{\partial^2 \bar{u}}{\partial x^2} + \frac{\partial^2 \bar{u}}{\partial y^2} \right) \quad (2)$$

$$\rho \left[\frac{\partial \bar{v}}{\partial t} + \bar{u} \frac{\partial \bar{v}}{\partial x} + \bar{v} \frac{\partial \bar{v}}{\partial y} \right] = -\frac{\partial \bar{p}}{\partial y} + (\mu + \mu_t) \left(\frac{\partial^2 \bar{v}}{\partial x^2} + \frac{\partial^2 \bar{v}}{\partial y^2} \right) + \rho g \beta (T - T_0) \quad (3)$$

$$\left[\frac{\partial \bar{T}}{\partial t} + \bar{u} \frac{\partial \bar{T}}{\partial x} + \bar{v} \frac{\partial \bar{T}}{\partial y} \right] = (\alpha + \alpha_t) \left(\frac{\partial^2 \bar{T}}{\partial x^2} + \frac{\partial^2 \bar{T}}{\partial y^2} \right) \quad (4)$$

The equations for turbulent viscosity and thermal diffusivity, are given by (Lauder and Spalding, 1972):

$$\mu_t = \frac{\rho C_\mu k^2}{\varepsilon} \quad (5)$$

$$\alpha_t = \frac{\mu_t}{\rho Pr_t} \quad (6)$$

where α is the thermal diffusivity [m^2/s] given by $\kappa/\rho c_p$; C_μ is a constant ($C_\mu = 0.09$); Pr_t is the turbulent Prandtl number given by ν_t/α_t and ν_t is the turbulent kinematic viscosity given by μ_t/ρ [m^2/s]. The thermophysical properties used in the simulations are: $c_p = 1007 J/(kg \cdot K)$, $\alpha = 2.249 \times 10^{-5} m^2/s$, $\rho = 1.1614 kg/m^3$ (Boussinesq), $\beta = 0.0033 K^{-1}$ and $\mu = 1.85 \times 10^{-5} kg/(m \cdot s)$.

The equations for the transport of turbulent kinetic energy (k) and the dissipation rate of turbulent kinetic energy (ε), $k-\varepsilon$ model, are given by:

$$\frac{\partial k}{\partial t} + \frac{\partial (\bar{u}k)}{\partial x} + \frac{\partial (\bar{v}k)}{\partial y} = \frac{1}{\rho} \frac{\partial}{\partial x} \left[\left(\mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial x} \right] + \frac{1}{\rho} \frac{\partial}{\partial y} \left[\left(\mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial y} \right] + \frac{1}{\rho} \frac{\partial}{\partial x} [G_k + G_b] - \varepsilon \quad (7)$$

$$\frac{\partial \varepsilon}{\partial t} + \frac{\partial (\bar{u}\varepsilon)}{\partial x} + \frac{\partial (\bar{v}\varepsilon)}{\partial y} = \frac{1}{\rho} \frac{\partial}{\partial x} \left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial x} \right] + \frac{1}{\rho} \frac{\partial}{\partial y} \left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial y} \right] + C_{1\varepsilon} \frac{\varepsilon}{\rho k} (G_k + C_{3\varepsilon} G_b) - C_{2\varepsilon} \frac{\varepsilon}{k} \quad (8)$$

In Eqs. (7) and (8), G_b and G_k represent the production of the turbulent kinetic energy due to buoyancy and the mean velocity gradients, respectively, and are calculated as follows:

$$G_b = \beta g_i \frac{\mu_t}{\sigma_t} \frac{\partial \bar{T}}{\partial x_i} \quad (9)$$

$$G_k = \mu_t \left(\frac{\partial \bar{u}_i}{\partial x_j} + \frac{\partial \bar{u}_j}{\partial x_i} \right) \frac{\partial \bar{u}_i}{\partial x_j} - \frac{2}{3} \rho k \delta_{ij} \frac{\partial \bar{u}_i}{\partial x_j} \quad (10)$$

In addition to the variables previously presented, in Eqs. (7) to (10), δ_{ij} is the Kronecker delta; $C_{1\epsilon}$, $C_{2\epsilon}$ and $C_{3\epsilon}$ are constants with empirical values of 1.44, 1.92 and 0.09, respectively, and σ_k and σ_ϵ are the turbulent Prandtl numbers for k and ϵ with empirical values of 1.0 and 1.3, respectively.

PROBLEM DESCRIPTION

The system configuration and main dimensions are shown in Fig. 1, where there is an IPWSC, composed of an inclined glass wall and an absorber wall, being attached to an environment (ventilated space). The solar radiation is transmitted by the glass wall and reaches the absorber wall, which, as the name implies, absorbs a large amount of incident radiant energy, and consequently the air inside the chimney is heated, becoming lighter and leaving the chimney to the atmosphere through its top opening. This process inside the IPWSC generates an airflow in the attached room, causing the outside air to enter the room through the window, maintaining circulation of air in the environment.

The following boundary conditions were considered for the computational domain (Khanal and Lei, 2012): the walls have a non-slip and adiabatic condition, except for the absorber wall, which is subjected to a heat transfer rate per unit length in the normal direction to the constant x-y plane, $q' = 3000$ W/m. For the Inlet Window and Outlet surfaces, Fig. 1, the gauge pressure is admitted equal to zero and the temperature is 300 K (27° C). The turbulence

properties in the input window are specified by the turbulence intensity of 3% and the turbulent length scale equal to 0.0876 m.

In the present study, the dimensions of the attached environment is remained constant, with only those of the solar chimney varying.

APPLICATION OF CONSTRUCTION DESIGN IN THE SOLAR CHIMNEY

According to Bejan and Lorente (2008), the generation of the configuration of flow systems is seen as a physical phenomenon, and the principle that summarizes its universal occurrence in nature is deterministic. Constructal Theory is the mental view that the generation of flow structures seen everywhere can be based on an evolutionary principle of maximizing access to flow over time (Bejan and Lorente, 2008). This principle is the Constructal Law, which states that “For a finite-dimensional system to continue to exist (live), its configuration must evolve freely in a way that facilitates the access of the currents that flow through it” (Bejan, 2000). In this sense, engineering projects and systems can benefit from the Constructal Law in a way that makes their design evolve according to the physical principle. Several examples of application of Constructal Design in Engineering has been noticed in heat transfer, renewable energy and solid mechanics (Horbach *et al.*, 2014; Gonzales *et al.*, 2021; Teixeira *et al.*, 2021; Moreira *et al.*, 2021; Feijó *et al.*, 2022; Dos Santos *et al.*, 2017; Martins *et al.*, 2022; Lima *et al.*, 2020).

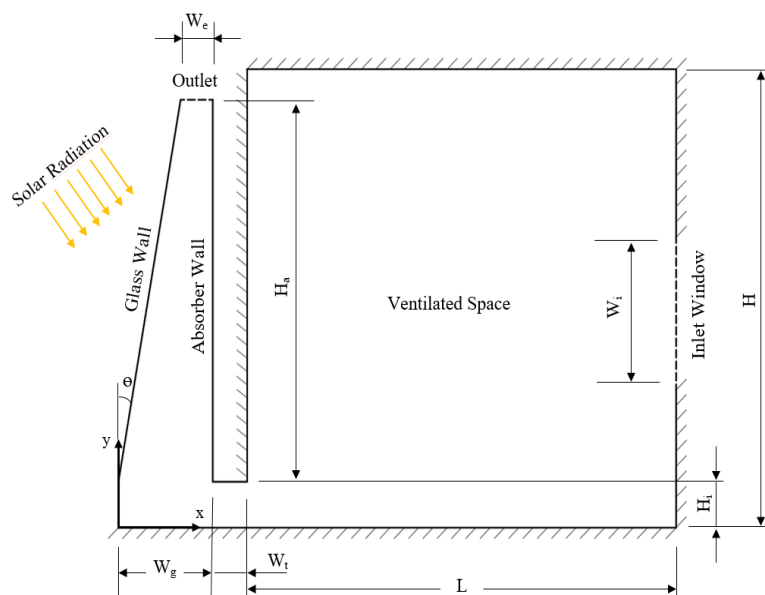


Figure 1. Configuration of a two-dimensional solar chimney (IPWSC) attached to a room.

In the present work, the Constructal Design is applied in the evaluation of the influence of the chimney geometry on the mass flow in the environment, subjecting the absorber wall to a constant heat transfer rate. The problem has as restrictions the area of the ventilated environment which is constant and equal to 9.00 m² and the chimney area which is constant and equal to 0.94 m². The area of the attached room is fixed with L = H = 3.0 m and the chimney area is calculated by:

$$A_i = \left(\frac{W_e + W_g}{2} \right) H_a + W_g \cdot H_i \quad (11)$$

Thus, as there is an area equation (geometric constraint) and 4 variables, the system is subject to 3 degrees of freedom: $H_i/H_a = 0.05$ (held constant in the present study); $W_g/H_a = 0.1, 0.3$ and 0.5 ; and W_e/W_g ranging from 0.1 to 1.0, with an increment of 0.1. Figure 2 schematically presents the set of simulations to be done, with a total of 30 numerical simulations being performed to obtain the results. In the first step, the H_i/H_a and W_g/H_a degrees of freedom are kept fixed while the W_e/W_g is varied. The largest magnitude obtained for the mass flow is the mass flow once maximized, \dot{m}_m and the ratio W_e/W_g once optimized, $(W_e/W_g)_o$. In the second step, the first step is repeated for different magnitudes of W_g/H_a . The largest magnitude of the mass flow is the twice maximized mass flow, \dot{m}_{2m} , and the corresponding optimal geometries are W_g/H_a once optimized, $(W_g/H_a)_o$, and the ratio W_e/W_g twice optimized, $(W_e/W_g)_{2o}$. In this way, it is possible to investigate the effect of geometric ratios on the mass flow in the environment.

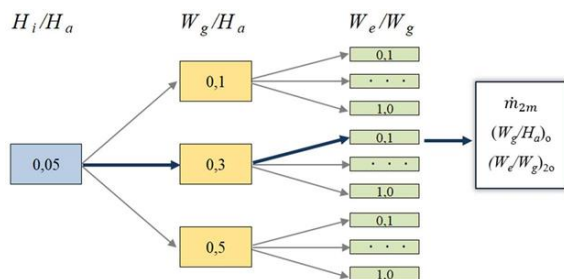


Figure 2. Proposed values for the degrees of freedom and procedure of the simulations.

NUMERICAL MODELING

Eqs. (1) - (8) are numerically solved by the finite volume method (MVF) using FLUENT software (Patankar, 1980; Versteeg and Malalasekera, 2007; ANSYS, 2011). For this study, 1,000 time steps were used where each time step was $\Delta t = 0.01$ s. For convergence, a maximum number of 300 iterations per time step is assumed. The advective terms of the momentum equations, energy and turbulence were determined using the second-order interpolation function UDS (Upstream Differencing Scheme -

Upwind) and diffusive terms were determined using the central difference scheme (CDS) (Khanal and Lei, 2015; Rajão, 2016). In all simulations, the SIMPLE (Semi-Implicit Method for Pressure Linked Equations) method was adopted for pressure-velocity coupling.

Simulations were considered converged when the residuals for mass, velocities, energy and for the turbulence model equations between two consecutive iterations were less than 10^{-6} , 10^{-6} , 10^{-8} and 10^{-6} , respectively. In addition, sub-relaxation factors of 0.7 were imposed on the conservation equations.

The verification of the computational model was previously carried out in the work by Leão et al. (2020). The results of the present computational model were compared with those obtained in the work of Khanal and Lei (2015) for a configuration like the one investigated in the present work. For the sake of simplicity, the computational model verification results are not repeated here, please see them in the work of Leão et al. (2020).

RESULTS AND DISCUSSION

Firstly, the effect of the ratio W_e/W_g over the mass flow rate in the chimney is analyzed, keeping the ratios H_i/H_a and W_g/H_a fixed, $H_i/H_a = 0.05$ and $W_g/H_a = 0.1$. Figure 3 presents the results obtained with the simulations, where it is possible to notice that there is an increase in the mass flow rate (\dot{m}) up to a certain ratio of W_e/W_g . After that, the magnitude of \dot{m} decreases smoothly, showing an almost constant magnitude. The once maximized mass flow rate is $\dot{m}_m = 0.1556$ kg/s, reached with $(W_e/W_g)_o = 0.6$. This value is 82.5% higher than the value obtained in its worst performance, obtained with the ratio $W_e/W_g = 0.1$. The worst performance is caused by the strong restriction imposed on the chimney outlet.

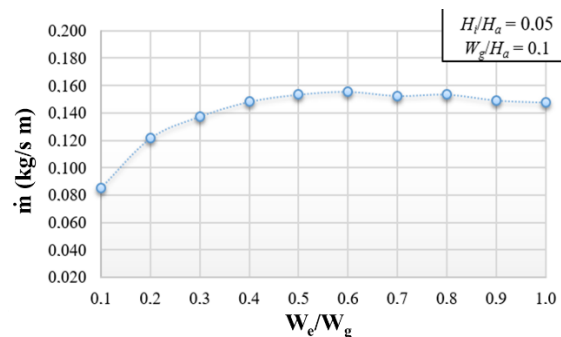


Figure 3. Effect of the ratio W_e/W_g on the mass flow rate (\dot{m}) for constant values of $H_i/H_a = 0.05$ and $W_g/H_a = 0.1$.

The following analysis consists of proceeding with the ratio $H_i/H_a = 0.05$ and varying the W_g/H_a ratio to evaluate the effect of W_e/W_g on the mass flow for different ratios W_g/H_a . The main purpose is to identify how changes in W_g/H_a influences the effect of the ratio W_e/W_g over the mass flow rate. It can be seen in Fig.

4 that for $W_g/H_a = 0.1$ the mass flow rate assumes its highest magnitude when $(W_e/W_g)_o = 0.6$, i.e., with the chimney outlet width measuring 60% of its lower chimney base width. For $W_g/H_a = 0.3$ and 0.5 , the behavior is quite different and the highest values of mass flow rate are found when the exit chimney width is narrow ($W_e/W_g = 0.2$ and 0.1 , respectively). This behavior demonstrates that changes in the ratio W_g/H_a affects the effect of W_e/W_g ratio over the performance indicator.

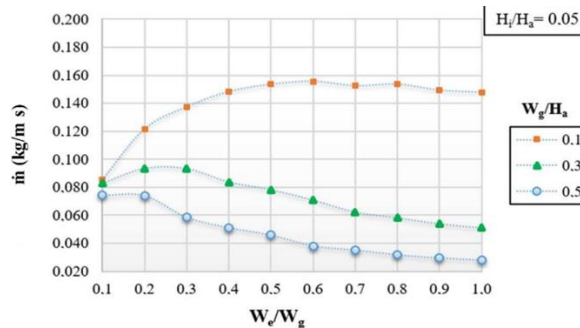


Figure 4. Effect of the ratio W_e/W_g over the mass flow rate (\dot{m}) for various ratios W_g/H_a .

In Figure 5, the optimal results obtained in Fig. 4, showing the effect of the ratio W_g/H_a on the mass flow once maximized (\dot{m}_m) and its corresponding geometry once optimized, $(W_e/W_g)_o$ are presented. The twice maximized mass flow rate (\dot{m}_{2m}) is obtained for the lowest ratio of $(W_g/H_a)_o = 0.1$, which corresponds to a magnitude 109% higher than that obtained at the ratio $W_g/H_a = 0.5$. Thus, it is possible to conclude that the ratio W_g/H_a also has an important sensitivity on the mass flow once maximized. Therefore, the results generally indicate that for the studied problem, the variation of one degree of freedom is important in the effect of other degrees of freedom on the performance. Furthermore, the variation of degrees of freedom for the second and third levels of investigation influenced the optimal configurations of degrees of freedom in previous levels.

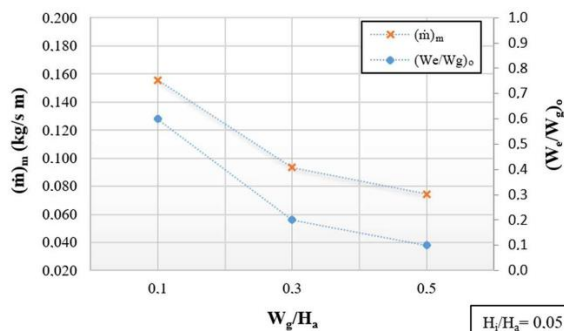


Figure 5. Effect of the ratio W_g/H_a over the once maximized mass flow rate (\dot{m}_m) and optimal ratio $(W_e/W_g)_o$ for $H_i/H_a = 0.05$.

The velocity fields for the best performance of the solar chimney, $(W_g/H_a)_o = 0.1$ and $(W_e/W_g)_{2o} = 0.6$,

and the worst once maximized case, that led to the lowest once maximized magnitude of mass flow rate, obtained for $W_g/H_a = 0.5$ and $(W_e/W_g)_o = 0.1$, are illustrated in Fig. 6. Analyzing the images, it is possible to see that the chimney configuration found for the twice maximized mass flow (\dot{m}_{2m}), presents a higher average velocity value at the exit of the chimney and a better performance for the flow in the adjacent environment when compared the worst performing configuration.

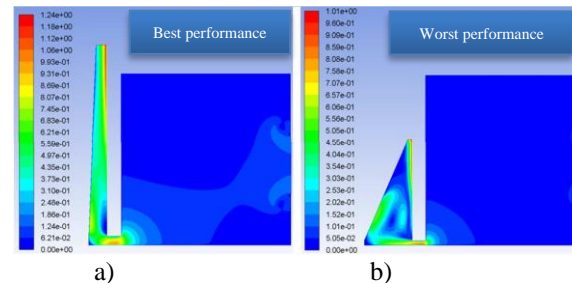


Figure 6. Velocity fields for $H_i/H_a = 0.05$ and two different configurations: (a) $(W_g/H_a)_o = 0.1$ and $(W_e/W_g)_{2o} = 0.6$; (b) $W_g/H_a = 0.5$ and $(W_e/W_g)_o = 0.1$.

CONCLUSIONS

The present work numerically analyzed the turbulent flow in an IPWSC connected to a building. The main objective was to evaluate the effect of different geometric ratios on the performance of the system by means of Constructal Design. The mass flow rate of air in the chimney exit is the performance indicator while the chimney and building areas are the problem restriction. For geometrical investigation, two degrees of freedom W_g/H_a and W_e/W_g were analyzed considering the ratio $H_i/H_a = 0.05$. Moreover, the absorber wall was subjected to a constant heat transfer rate, keeping the same amount of energy being supplied to the system. All simulations were performed in the transient regime in a two-dimensional domain, assuming turbulent flow, incompressible and with heat transfer by natural convection. Time-averaged equations of conservation of mass, balance of momentum and conservation of energy were numerically solved by the finite volume method using FLUENT software. To reproduce the turbulence of the airflow in the chimney, RANS approach was used with the standard $k - \epsilon$ turbulence model for closure of the modeling.

Results demonstrated that the mass flow rate has an important dependence of the ratio W_e/W_g . Moreover, for $H_i/H_a = 0.05$ and $W_g/H_a = 0.1$, the optimal configuration $(W_e/W_g)_o = 0.6$ led to a once maximized mass flow rate ($\dot{m}_m = 0.1556 \text{ kg/(s m)}$) which is 82.5% superior than the performance reached with the worst case. Even the first investigation showed the importance of design rationalization in this kind of problem. Afterwards, the same effect of W_e/W_g over mass flow rate (\dot{m}) is performed for different

ratios of $W_g/H_a = 0.1, 0.3$ and 0.5 , and $H_i/H_a = 0.05$. The optimal results were used to show the effect of W_g/H_a over once maximized mass flow rate (\dot{m}_m) and the corresponding optimal ratio (W_e/W_g)_o. It was possible to notice that, as the ratio W_g/H_a increased, the values found for the performance indicator decreased and the optimal ratio (W_e/W_g)_o also changed with the variation of W_g/H_a . The twice maximized mass flow rate of air (\dot{m}_{2m}) was 109% higher at $W_g/H_a = 0.1$ when compared to $W_g/H_a = 0.5$. Moreover, changes in the ratio W_g/H_a led to modifications in the behavior of the effect of W_e/W_g over \dot{m} , showing the dependence of geometrical ratios. The velocity fields were also evaluated, and it was possible to notice that the configuration that reached the highest mass flow also presented a higher velocity value at the exit of the chimney and a better flow performance in the adjacent environment. In general, results demonstrate the importance of geometric investigation in this problem, since strong differences on performance were achieved (more than 100% between the best and worst cases). Results also showed the dependence of the geometrical ratios investigated (W_e/W_g and W_g/H_a) and the influence of W_g/H_a on the effect of W_e/W_g over the system performance, which is important to comprehend the influence of the design of chimney on the system performance.

Consequently, from all the results presented, it was concluded that the Constructal Design method is an excellent tool for evaluating the sensitivity of the geometric parameters of an IPWSC and that the degrees of freedom investigated have good sensitivity on the performance of the system. Furthermore, the results indicated that it is important to evaluate the degrees of freedom in a combined way.

For future studies, it is recommended the investigation of other ratios for H_i/H_a , other thermal conditions and improvements in the modeling considering thermal radiation heat transfer in the solution of the problem.

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