Assessment of Various Trombe Wall Geometries with CFD Study

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Abstract: An investigation of the optimal geometric configuration of a Trombe wall is presented with simulation of the transfer phenomena, that take place during its operation, using computational fluid dynamics. A numerical model is developed for a 2D steady-state simulation of a Trombe wall cross-section operation, and it is validated against an energy balance model’s results. Then the developed model is used for the evaluation of 10 different geometrical configurations examining various air gap widths, storage wall thicknesses, ventilation slots distances, and ventilation slots diameters. The examined geometries were evaluated with respect to the achieved temperature at the air gap exit and at the room facing storage wall surface, the achieved mass air flow in the air gap, and the ability of warm air stream from the gap to enter the test room. The aim was to ventilate the whole space without leaving large areas where the air just recirculates unaffected by the Trombe wall operation. According to the above-described criteria, optimum solution is an air gap width of 5 to 8 cm with increased distance between ventilation slots and a configuration of upper ventilation slot with an inclination of 30 degrees.

Keywords: Trombe wall; CFD; air gap width; ventilation slots distance; ventilation slots configuration

1. Introduction

Passive heating systems such as the Trombe wall stand out as essential building blocks on the road to zero energy buildings. Quasi-steady models [1,2], which allow relatively fast calculations for the initial design, are forced into simplifications by adopting general values for the heat transfer coefficients. Additionally, they cannot offer information about design details that could improve the passive element’s energy performance. Energy balance models [3–5] and building energy simulation (BES) models [6,7] consider one-dimensional heat transfer, make general assumptions about the flow pattern and the heat transfer within the air gap while using coefficients for taking into account the temperature gradient along the air gap [2].

Although several researchers are considering several modified forms of the Trombe wall [8], issues regarding the optimal value of key geometric features of conventional Trombe walls as presented in an extensive literature review [9] have not yet been resolved. For example, the optimal width of the air gap is the 1/10 of its height according to [10]. Thus, the optimal width is between 0.2 and 0.3 m [10]. However, other researchers propose that the width of the gap must be limited from 2 to 5 cm in order to create a small space for air [6], or even 6 cm [11]. In terms of the size of the ventilation slots, the relevant research focuses mainly on the estimation of their sizes as a percentage of the Trombe wall surface [10], while it has not been examined whether their configuration can affect the performance of the wall. The optimal thickness of the Trombe wall depends on the climatic conditions and according to [12] the optimum thickness is between 15 to 40 cm. Finally, the effect that the distance between the ventilation slots has on the operation of the Trombe wall does not appear to have been investigated in detail. Therefore, there are still gaps in research on the basic geometric features of the conventional Trombe wall.
Computational Fluid Dynamic (CFD), although a time consuming and computationally demanding method for designing and evaluating the operation of a Trombe wall, can guide the design of details (shape of Trombe wall absorption surface, opening geometry, etc.). Furthermore, CFD can offer insight into the flow inside the air gap and around ventilation slots. Consequently, the results can feed quasi-steady models with tested heat transfer coefficients and predict the flow pattern, allowing energy balance models to calculate these coefficients correctly. In addition, CFD can be used to assess and/or feed with info and/or boundary conditions energy balance and BES models [13].

The majority of CFD studies in recent years use the Finite Volume method for solving the partial differentiate RANS equations [13–18], while in older works, researchers also use the Finite Difference Method [4,19]. The majority of the published research papers use steady-state calculations [13,15,17,19], and only a few of them, due to high computational requirements, use unsteady calculations [14,20,21]. The high computational cost, as well as the particular characteristics of the Trombe wall operation to which each researcher wishes to focus his attention, lead several researchers to 2D calculations [16–18,20,22]. Nevertheless, in recent years, research with 3D CFD simulations of the Trombe wall are also available [13–15,19]. Some CFD studies simulate the operation of solar air heaters in conjunction with the space they serve [13,14,17,21]. Other CFD studies focus on the operation and optimization of specific parts of the passive system, very often focusing on the air gap without simulating the solid parts [15,16,18,19].

What is of a major importance is the way researchers treat the radiation (solar and thermal). Several published research works do not simulate all the incident solar radiation, considering known values of radiation or heat fluxes on the storage wall [15,16,18,19,21,22]. Among them, some focus on the heat transfer with convection and conduction while others simulate the heat transfer of thermal radiation with models like S2S [16,18]. Nevertheless, there are works that include in the computational domain the transparent cover and the storage wall and they simulate their operation considering as boundary conditions directly the solar incident radiation, using for the simulation the Monte Carlo [20] method or more often the DO radiation model [11,14,17]. The basic common characteristic of these works is that they consider the transparent cover as gray material with average optical properties along the whole spectrum. This approach cannot simulate accurately the operation of the transparent cover. Another drawback is the fact that they simulate the storage wall as an ‘opaque’ element, which in the DO model means that the participation of this component to heat exchange through the specular component of reflected radiation is limited. In addition, there is no possibility to set different values for absorption and emission coefficients. According to the extended review presented in [8] the majority of CFD studies use the k-ε model in order to simulate the turbulence effect in the transport phenomena, using grids that vary from 4500 to 1,500,000 computational cells according to the specific needs, the computational domain size and the characteristics of each study.

Solar air heaters are a practical system for producing warm air using solar energy and can be considered as aerothermal systems. The principle of their operation is similar to that of solar thermal panels. From numerous studies concerning solar collectors, information can be collected and assessed in relation to the increase of their efficiency. This can be realized by increasing heat transfer through the proper treatment of the absorption surface (roughness, fins, etc.) [23]. In [24] the heat transfer enhancement is studied using fins and rods on the collector surface. Kabeel et al. [25] examined the use of fins, the use of multiple air paths, and the use of wires perpendicular to the air flow in order to increase the turbulence and thus the transfer coefficient as well as the heat exchange surface.

Studies of solar chimneys could be useful in understanding the simulation of the Trombe wall operation. In [26] a solar chimney system with earth tubes is studied where the CFD model is validated with the equivalent experimental configuration. A similar study of a solar chimney is made in [27].

In the present work, a 2D CFD model is developed for simulation of Trombe wall operation serving a test room. The transport phenomena are modeled with the finite volume
method while the radiation transport is modeled with the DO model. Special attention has been given on radiation heat transfer considering non-gray cover material with detailed calculation of optical properties of the composite cover in four wavelength bands (Ultra Violet, Visual, Near Infrared and Infrared). This, simulates accurately the transportation of the radiation. Additionally, we treat the storage wall as a ‘semi-transparent’ material improving its participation in the heat transfer calculation. To our knowledge this treatment of radiation heat transfer has not been used in the past by other researchers for the simulation of Trombe wall operation. After the validation of a default geometry, other 9 Trombe wall configurations are examined investigating the influence of air gap width, storage wall width, ventilation slots size, ventilation slots distance, and of the ventilation slots geometry on the Trombe wall basic characteristics (flow pattern/flow rate inside the air gap and heat transfer coefficients). Thus, the present study contributes, through the development of the CFD model, to the current understanding of how the basic geometric features of the conventional Trombe can affect its performance.

2. Materials and Methods

2.1. Mathematical Model

The transport phenomena developed in the solar air heater system are described with the Reynolds Averaged Navier-Stokes equations which are solved with the finite volume method [28,29]. Specifically, the equations of continuity, conservation of momentum in two directions (x, y), energy, transport radiation, and the equations corresponding to the turbulence model are solved in a steady-state regime. The simplified approach of a 2D model is considered since in a cross-section including the ventilation slots with buoyancy driven flow, the, buoyancy forces in the horizontal direction may be neglected. This is a common approach adopted by many researchers [22]. The flow is considered incompressible since, in case of natural convection, the expected value of velocities is not expected to reach a Mach number greater than 0.1. For the modeling of thermal buoyancy, we used the Boussinesq approximation since it allows us to consider the density constant in all the solved equations except for the buoyancy term in the momentum equations. Thus, the computational effort is reduced while a faster convergence is achieved. For the simulation of radiative transfer, the Discrete Ordinate (DO) model was used [30,31] with angular discretization $4 \times 4$ and pixelation $3 \times 3$. Radiation transport equations are solved in two directions for four wave bands: (a) Ultra-violet radiation–UV ($\lambda = 0.1–0.39 \, \mu m$), (b) Visual radiation -VIS ($\lambda = 0.38–0.74 \, \mu m$), (c) Near Infrared radiation–NIR ($\lambda = 0.74–1.2 \, \mu m$), and (d) Infrared radiation–IR ($\lambda = 1.2–100 \, \mu m$). Energy and radiation equations are also solved for the solid parts of the computational domain. The ability of the transparent cover to allow the propagation of shortwave solar radiation and not the longwave thermal radiation due to the correct modeling of its spectral optical properties is crucial for the analysis of Trombe wall performance.

The flow was considered turbulent because the expected velocities within the air gap are expected to lead to values greater than $10^5$, as evidenced by the presented results. The turbulence effect is modeled through the high Reynolds k-ε model [32]. The standard high Re k-ε turbulence model was selected because it offers numerical stability, quick convergence, and, using wall functions, does not require very dense space discretization close to the solid boundaries (as in the case of low Reynolds models). At the same time, it offers satisfactory accuracy for calculations in large spaces [33]. This is the reason that it is used extensively in calculations that include a Trombe wall coupled with the room they serve [8]. The limited requirements of k-ε model in the number of computational cells were crucial for its selection since a very detailed discretization for the radiation model can be applied. Although this increases the computational cost, it offers an accurate solution of the radiation transmission which is the most important factor for the specific calculations.
2.2. Numerical Model

A grid with 22,000 rectangular computational cells was used. A growth ratio of 1.025 was used for the grid thickening, securing that the dimensionless distance $y+$ of the first computational cell from the wall boundaries lies between 10 and 40 since wall functions are used. This decision was based on a grid independence test which was carried out using three grids of 14,640, 22,000, and 34,200 computational cells, ensuring that in all grids the requirement regarding the value of $y+$ will be met. The value of air mass flow inside the air gap had a 3% difference between the grid of 14,640 computational cells and the grid of 22,000. Refining further the grid to 34,200 computational cells the change of air mass flow was lower than 1%. Thus, a grid of 22,000 computational cells with adequate density in critical areas was selected for our calculations.

For the pressure/momentum coupling the SIMPLEC algorithm was used. The discretization for convection terms of momentum and turbulence quantities ($k$ and $\varepsilon$) is implemented using a 2nd Order Upwind (SOU) scheme, which takes into account the values of upstream cells, providing second order accuracy. It is considered adequate for parabolic flows, as is a flow driven by thermal buoyancy. For the convection terms of energy equation, the 3rd order Monotonic Upstream-centered Scheme for Conservation Laws (MUSCL) is used, combining central difference and upwind schemes characteristics. Finally, for Radiation Transport Equations (RTEs) a 1st Order Upwind scheme (FOU) is selected since it provides adequate accuracy along with fast convergence. All the diffusive terms are discretized with a central difference scheme which is a second order of accuracy considering the values contribution of all neighbor cells as required by the nature of diffusive terms. The calculations were performed by the ANSYS Fluent software V14.5. The convergence criterion for all the parameters except energy and radiation was set to $10^{-4}$, while for energy was set to $10^{-6}$ and for radiation to $10^{-5}$. In the 1st geometry, the appropriateness of the convergence criterion was tested for the continuity equation, by setting its convergence criterion to $10^{-6}$ and monitoring as a control parameter the mass flow rate inside the air gap. The convergence criterion refinement modified the calculated mass flow rate by only 0.44%, while the execution elongation roughly doubled, without any significant improvement to the accuracy of the calculations. For that reason, the continuity criterion convergence was preserved in the initial value for all the other simulations.

2.3. Case Study and Parametric Study

The examined Trombe wall consists of a heat storage wall with a high absorbing solar radiation coefficient which is covered by a transparent cover. In the gap between them, air can circulate. This air enters the gap through ventilation slots in the lower part of the storage wall and leaves the gap through the upper ventilation slots returning to the room.

2.3.1. Studied Geometry

The 2D model simulates the operation of a Trombe wall cross-section which includes the computational domains of the transparent cover, the 7.6 cm width air gap, the storage/collection wall having a 10 cm thickness, the lower and upper ventilation slots which have a 12.4 cm diameter and the room coupled with the Trombe wall. The room has dimensions of 3 m (length) by 2.8 m (height) as presented in Figure 1a, while in Figure 1b the used computational grid is presented. The examined cross section is in the North-South axes, with the Trombe wall covering the south wall.
Figure 1. (a). Default Trombe wall cross-section geometry–grid 1. (b). Default Trombe wall geometry computational grid.
2.3.2. Boundary Conditions

The transparent cover is modeled as a ‘sandwich’ consisting of an outer surface, an inner solid semi-transparent element, and an inner surface. The outer surface (computational domain boundary) is a semi-transparent wall where a mixed thermal boundary condition is applied (combined convection and radiation heat transfer with environment and conduction with the solid material). In the cover outer surface are defined: (a) the external air temperature, (b) the convection heat transfer coefficient with the external air and (c) the equivalent sky temperature for the calculation of the thermal radiation exchange with the external environment. Additionally, on the outer surface, the solar incident radiation (beam and diffuse) is defined in four wavelength bands. The inner surface of the cover is a semi-transparent wall thermically and optically coupled with the solid material of the cover and the air in the gap between the cover and the storage wall restoring a conjugated heat transfer treatment. The storage wall is also modeled as a ‘sandwich’ consisting of an inner (gap-facing surface) surface, an interior solid semi-transparent material, and an outer (room-facing surface). Both storage wall surfaces are thermically and optically coupled with the interior solid material and with the air in the airgap and the room. The solid material of the storage wall has an absorption coefficient high enough to ensure the extinguish of incident radiation in the first computational cell. The energy and radiation equations are solved both in the fluid and solid zones. All the other room internal surfaces are considered adiabatic.

2.3.3. Optical Properties

The transparent cover consists of two panes of glass with an internal air gap. However, in the simulation, it is treated as a single material with the same thickness and equivalent thermal and optical properties.

Given the transmittance, \( \tau_{ti} \) and reflectance, \( \rho_{ti} \) of the individual glass panes \([34–40]\), the transmittance, \( \tau_t \), absorptance, \( \alpha_t \), and reflectance, \( \rho_t \), of the entire cover (two glazing system and air gap) for vertical incident radiation \([34]\) are calculated as follows:

\[
\tau_t = \frac{\tau_{t1} \cdot \tau_{t2}}{1 - \rho_{t1} \cdot \rho_{t2}}
\]

\[
\rho_t = \rho_{t1} + \frac{\tau_{t1} \cdot \rho_{t2} \cdot \tau_{t1}}{\tau_{t2}}
\]

\[
\alpha_t = 1 - \rho_t - \tau_t
\]

Then, with the following relations, the equivalent refractive index, \( n \), and the equivalent extinction coefficient, \( a_s \), for the whole structure are calculated.

Total transmittance

\[
\tau_t = \tau_a \cdot \left( \frac{1 - r}{1 + r} \right) \cdot \left( \frac{1 - r^2}{1 - (r \cdot \tau_a)^2} \right)
\]

Total reflectance

\[
\rho_t = r(1 + \tau_a \cdot \tau_t)
\]

Total absorptance

\[
\alpha_t = (1 - \tau_a) \cdot \left( \frac{1 - r}{1 - r \cdot \tau_a} \right)
\]

where, \( r \), the cover reflection, and \( \tau_a \) the transmittance due to absorption losses.

\[
r = \left( \frac{n - 1}{n + 1} \right)^2
\]

\[
a_s = \frac{1}{d} \cdot \ln \left( \frac{1}{1 - a} \right)
\]

\[
\tau_a = 1 - \alpha
\]

where \( \alpha \), the cover absorptance, and \( d \), the cover thickness
These calculations are made for the optical properties of the cover system in each of the examined wavelength bands. According to the above relationships, the simulation of the double-glazed cover 4-15-5 with air gap and low-e surface is materialized with an optically equivalent cover with optical properties given in Table 1.

Table 1. Spectral optical properties of the cover as it is considered as a single material.

| Optical Property | UV   | VIS  | NIR  | IR  |
|------------------|------|------|------|-----|
| \(\alpha_s\)    | 38.53| 9.49 | 26.51| 77.4|
| \(n\)            | 1.846| 1.822| 1.852| 1.325|

For the storage wall the extinction coefficient \(\alpha_s\) is taken equal to 1000 and the refractive index, \(n\), equal to 1.418 in the whole examined wavelength range.

2.3.4. Thermal Properties

The following Table 2 gives the thermal properties of the materials used.

Table 2. Material thermal properties.

| Material       | Density [kg/m\(^3\)] | Thermal Conductivity [W/mK] | Specific Heat Capacity [J/kgK] |
|----------------|------------------------|-----------------------------|--------------------------------|
| Transparent cover | 938.26                 | 0.057                       | 750.21                         |
| Storage wall   | 2000                   | 0.65                        | 1000                           |

2.3.5. Climatic Conditions

The external temperature was taken 14.97 °C. The wind velocity 2.3 m/s. The incident radiation normal to the transparent cover, beam, and diffusive, are given in the following Table 3 for each considered wavelength band.

Table 3. Incident radiation normal to the cover.

| Band/Radiation | UV   | VIS  | NIR  | IR  |
|----------------|------|------|------|-----|
| Beam radiation [W/m\(^2\)] | 41.93| 153.76| 108.33| 48.92|
| Diffusive radiation [W/m\(^2\)] | 11.44| 41.96| 29.57| 13.35|

2.3.6. Parametric Study

Apart from the default geometry, presented in the previous paragraphs, henceforth it will be referred to as grid 1, nine (9) more geometries were examined in order to study the Trombe wall geometry effect on the wall performance. The nine geometries’ characteristics are summarized in the following Table 4. In detail, the cases that have been examined are the following: (a) One case with smaller air gap width and 8 cases with bigger width, (b) Three cases with different storage wall thicknesses, (c) Two cases with different ventilation slots’ distances, (d) Two cases with different ventilation slots’ diameter and (e) A case with different upper ventilation slot geometry.
Table 4. Cases examined during the parametric study.

| Geometry | Air Gap Width, $d_g$ [cm] | Storage Wall Thickness, $d_w$ [cm] | Distance between Ventilation Slots, $L$ [m] | Ventilation Slot Diameter, $d_s$ [cm] | Upper Ventilation Slot Angle, $a$, [deg] |
|----------|---------------------------|----------------------------------|--------------------------------------------|-------------------------------------|----------------------------------------|
| Grid 1   | 7.6                       | 10                               | 2.1                                        | 12.4                                | 0                                      |
| Grid 2   | 5                         | 10                               | 2.1                                        | 12.4                                | 0                                      |
| Grid 3   | 10                        | 15                               | 2.1                                        | 12.4                                | 0                                      |
| Grid 4   | 10                        | 20                               | 2.1                                        | 12.4                                | 0                                      |
| Grid 5   | 10                        | 30                               | 2.1                                        | 12.4                                | 0                                      |
| Grid 6   | 10                        | 10                               | 1.8                                        | 12.4                                | 0                                      |
| Grid 7   | 10                        | 10                               | 2.4                                        | 12.4                                | 0                                      |
| Grid 8   | 10                        | 10                               | 2.1                                        | 10                                  | 0                                      |
| Grid 9   | 10                        | 10                               | 2.1                                        | 15                                  | 0                                      |
| Grid 10  | 10                        | 10                               | 2.1                                        | 12.4                                | 30                                     |

3. Results

3.1. Default Geometry Energy Performance–Validation

First the results of the simulation concerning the default geometry (grid 1) will be presented.

3.1.1. Radiation

In the following Figure 2 the radiation isocontours in the VIS wave band are presented. The VIS radiation enters from the transparent cover and is transported up to the storage wall where is fully absorbed.

![VIS Radiation Isocontours](image-url)
3.1.2. Radiation

In Figure 3 the velocity isocontours [m/s] and the streamlines describe the flow field as this was developed inside the Trombe wall air gap and the coupled room. The air enters the air gap through the lower ventilation slots, it is accelerated due to thermal buoyancy along the air gap and it leaves the air gap through the upper ventilation slots entering the room. Inside the room, a big recirculation appears covering and ventilating the biggest part of the room. Smaller recirculation close to the room floor, in the room back wall, and on the roof’s left side indicate areas that are not ventilated with this arrangement. The flow through the air gap predicted by CFD is 0.0249 kg/s which is in very good agreement with the flow predicted by the energy balance model, described in [41], which is 0.02448 kg/s.

![Flow field of the default geometry.](image)

**Figure 3.** Flow field of the default geometry.

In Figure 4 flow field details in the upper ventilation slots (air gap exit), Figure 4a, and in the lower ventilation slots (air gap inlet), Figure 4b, are given. According to Figure 4b air enters smoothly the air gap from the lower ventilation slot and accelerates inside the air-gap due to thermal buoyancy. However, according to Figure 4a in the upper ventilation slot, from where air leaves the air gap, a recirculation, covering almost the one fourth of the opening, allows air to return and contract the available exit cross section. The reason is that due to the steep change of direction, the flow detaches. In the area below the detached flow, the pressure decreases resulting in the air from the room, where the pressure is higher, being directed to the gap. Therefore, a secondary flow develops there, where the colder air recirculates removing energy from the warm air stream. This on the one hand reduces the available cross-section for warm air to escape from the gap to the room and on the other
hand, introduces an additional pressure drop in its path making it difficult for it to travel and thus affecting the operation of the Trombe wall.

![Figure 4. Flow field details: (a) upper (exit) ventilation slots; (b) lower ventilation slots.](image)

In Figure 5 velocity distributions on characteristic sections of the examined geometry are provided. In Figure 5a the velocity $u_x$ distribution in the air gap inlet is given, while in Figure 5b the velocity $u_x$ distribution across the air gap exit is presented. In Figure 5c the velocity $u_y$ distribution across the air gap at wall mid height is given. Finally, in Figure 5d variation of velocity $u_y$ along the air gap (in a middle line) is presented.

![Figure 5. Velocity distributions: (a) Air gap inlet; (b) Air gap exit; (c) Across air gap; (d) Along air gap.](image)

Air velocity in the inlet gap and inside the air gap is turbulent and almost fully developed. While at the exit, a flow separation is detected with the flow returning back to the air gap for almost the one fourth of the slot cross-section. Air velocity in the middle air gap line reaches a maximum value close to the inlet slots where a flow narrowing due to
direction change leads to a small cross section and then remains almost constant with a slow reduction due to an increase of the developed thermal boundary layers.

3.1.3. Temperature

In Figure 6a the temperature isocontours in [K] are given for the whole computational domain. Figure 6b gives the detail of the temperature distribution in the area of the upper ventilation slot with the warm air stream penetrating the room. Finally, Figure 6b shows the way in which the thermal boundary layers develop on both sides of the gap, until from a certain height, and then, they cover the entire gap, while at the same time the heat is transmitted in the storage wall.

The storage wall and the air gap temperature increase transporting heat to the coupled room through the storage wall by convection-conduction and radiation and through the gap air by ventilation.

In Figure 7 temperature and Ra number distribution in the characteristic surface of the computational domain are presented. Specifically are presented: (a) in Figure 7a the temperature distribution along the internal transparent cover surface, $\theta_{se}$, (the surface adjunct to the air gap fluid), (b) in Figure 7b the vertical temperature distribution along the air gap center line, $\theta_{int}$, (c) in Figure 7c the temperature distribution along the internal storage wall surface, $\theta_{si}$ (facing the air gap) and along the storage wall surface adjunct to the room, $\theta_{opid}$, (d) in Figure 7d the Ra number distribution along the cover internal surface, $Ra_c$, and along the internal storage wall surface, $Ra_{sw}$, (e) in Figure 7e the temperature distribution along a horizontal line in the middle of the air gap, $\theta_{int}$, and (f) in Figure 7f the temperature distribution in the air gap exit. In the same graphs the values predicted from the energy balance models [41] are presented for comparison with the name $\theta_{eb}$. The CFD distributions are presented with dashed red and green lines while the energy balance predicted constant values are presented with continuous black and blue lines.
Figure 7. Temperature and Ra number distributions: (a) temperature along the cover internal surface, $\theta_{se}$; (b) temperature along air gap in mid distance, $\theta_{int}$; (c) temperature along storage wall internal surface, $\theta_{si}$ and room facing surface, $\theta_{opi}$; (d) Ra number along cover and storage wall internal surfaces, $Ra_c$; (e) temperature along air gap width in the middle height, $\theta_{int}$; (f) temperature along the air gap exit, $\theta_{out}$. 


In Figure 7a the variation of surface temperature, $\theta_{se}$, along the transparent cover internal surface is presented. An increase in temperature is observed, which is expected. The temperature of this surface is a result of: (a) the absorption of the solar radiation on the external cover surface which is transferred as heat by conduction and radiation to the internal surface, (b) by the absorption of infrared radiation that exchanges both with air in the gap and the surface of the storage wall, and (c) by heat exchange by convection with air in the gap. The distribution of solar radiation on the external surface is uniform, but the temperature of the air in the gap and of the storage wall increases and so does the temperature on the cover surface. The temperature decreases near the ventilation slots because in these areas opposite the cover surface there is no storage wall, but also because in these areas the temperature of the neighbor storage wall is lower. The temperature returns to high values in areas where the surface of the cover comes in contact with the storage wall and the heat is transferred directly from the cover with conductivity. According to Figure 7a the 1D energy balance model cannot predict the variation of transparent cover surface temperature, $\theta_{se}$, in the areas of the ventilation slots where the convection seems to be more intensive and so overpredict the temperature. Nevertheless, the average CFD predicted value for $\theta_{se}$ is 28.13 °C with a maximum value 32.76 °C. The energy balance predicted value for $\theta_{se}$ is 32.4 °C a difference of the order of 13% and it is justified.

In Figure 7b the variation of air temperature, $\theta_{int}$, along a vertical line in the middle of the air gap is presented. The temperature of the air moving upwards is constantly increasing since in a given mass there is an accumulation of heat transferred by convection and radiation. This increase cannot be predicted by the 1D energy balance model. Nevertheless, the average value of the CFD simulation of temperature $\theta_{int}$ is 21.27 °C which presents a percentage deviation of 5% with respect to the energy balance model value of 22.27 °C. This deviation is mostly attributed to the fact that the vertical line in the middle of the gap does not represent the average value of the whole air gap.

In Figure 7c the temperature along the storage wall surface facing the air gap, $\theta_{si}$ and the temperature along the storage wall surface facing the room, $\theta_{opi}$ are given. The temperature of the air-gap facing storage wall surface, $\theta_{si}$, is due to the the absorption of incident solar radiation (directly incident or reflected from the inner surface of the cover), the infrared radiation exchanged with the inner surface of the cover and the air as well as with heat transfer by convection with the air in the gap and by conduction through the storage wall. Thus, its increase is due to the increase of the air temperature in the gap. The temperature of the room-facing storage wall surface, $\theta_{opi}$, is regulated due to heat transfer by conduction through the wall itself and so it follows the variation of $\theta_{si}$. In the CFD distributions, there is a linear increase along the height, as it was expected, but close to the ventilation slots the temperature decrease according to the thermal and flow field developed there. At the lower ventilation slots air enters at a low temperature and is expected to reduce the wall temperature through convection. However, in the upper ventilation slot from where the air exits the gap, the recirculation that is created, Figure 4a, brings the wall in contact with cold air in the room, having as a result a reduction in temperature. The CFD predicted average value of $\theta_{si}$ is 25.85 °C which is 3% higher than the constant value of 25.05 °C predicted by the energy balance model. The CFD predicted average value of $\theta_{opi}$ is 23.84 °C 7.8% higher than the 22 °C predicted by energy balance model.

In Figure 7e the air temperature, $\theta_{int}$, distribution across the air gap at the wall mid height is given as predicted by CFD. The air temperature near the surfaces surrounding the gap is higher with values corresponding to the surfaces with which they are in contact. It appears that the two thermal boundary layers developing on these surfaces have begun to converge, at the mid height, having as a result the temperature at the center of the gap to be higher than the temperature at which the air entered the gap. The average value of $\theta_{int}$ at this point is 22.1 °C, which makes this section quite representative since it differs only by 0.77% from the constant predicted value by the energy balance model of 22.27 °C.
Finally, in Figure 7f the temperature distribution across the upper ventilation slot is given. The given profile corresponds to the middle of the ventilation slot. The spatial distribution is quite different from the one in the middle of airgap presented in Figure 7e due to the presence of the recirculation close to the storage wall. This recirculation brings cold air from the room and for this reason, low-temperature values are located near the lower limit of the cross section and at the same time pushes upwards the warm air flow coming out of the gap (with a profile similar to that of Figure 7e) resulting in higher temperatures towards the center of the cross section, as well as near its upper limit. This recirculation cannot be predicted by the energy balance model. Thus, the average temperature, $\theta_{out}$, predicted by the CFD model in the air gap exit is 22.51 °C which is 8.3% lower than the value of 24.54 °C predicted by the energy balance model.

3.2. Comparison with Other Geometries
3.2.1. Thermal and Flow Field

In the Figure 8a–j the flow streamlines on a temperature isocontours background are presented for the 10 examined geometries.

![Figure 8](image-url)
In all examined geometries there are recirculation areas mainly at the back of the room. That is, areas where the air circulates without being affected by the warm current coming out of the gap of the Trombe wall. As the thickness of the air gap increases, these areas decrease in size. For a gap of 5 cm a large recirculation is formed at the back and bottom of the room which covers almost 1/3 of it. As the gap widens this recirculation initially breaks in two with one recirculation loop at the back of the room and another one near

Figure 8. Flow streamlines and temperature isocontours: (a) grid 1; (b) grid 2; (c) grid 3; (d) grid 4; (e) grid 5; (f) grid 6; (g) grid 7; (h) grid 8; (i) grid 9; (j) grid 10.
the floor while when the gap reaches 10 cm the bottom recirculation disappears and the recirculation at the back is compressed. This is because increasing the gap, increases the air mass flow and decreases the air temperature, having as a result the warm air stream coming out of the upper ventilation slot to have a smaller Archimedes number which in turn leads to higher penetration depths. It should be noted that the warm stream angle remains almost the same due to the small recirculation loop established in the upper slot and described in Figure 4a.

The room ventilation is further improved with the increase of the thickness of the storage wall. For a wall thickness of 30 cm, the recirculation almost disappears and the whole room ventilation pattern is affected by the warm air current. The reason is that as the air moves through the upper ventilation slot, the length of which increases as the wall thickness increases, it is tunnelled so that it enters the room in an almost horizontal direction.

The ventilation of the room also seems to be favored by the increase of the distance between the ventilation slots, thus improving the efficiency of the Trombe wall. In this case too, increasing the distance between the ventilation slots leads to a flow increase and a reduction of the temperature difference, thus reducing the Archimedes number and increasing the penetration length.

Finally, the change in the diameter of the ventilation slots does not seem to have a significant effect on the flow field, since an increase in diameter size simultaneously leads to a decrease in temperature resulting in the almost constant Archimedes number.

Of all cases examined, except for the last one, a small recirculation loop is created in the exit slot. As a result, air from the room returns to the gap, reducing the available cross-section for the warm air flow coming out of it, decreasing its energy, as it was explained in the discussion of Figure 4a. This recirculation introduces an additional local pressure drop in the warm air path making it difficult for the Trombe wall to function effectively. This problem seems to be successfully addressed in the 10th grid in which the upper ventilation slot has an inclination of 30 degrees upwards. In this case, detail of which is presented in Figure 9, the aforementioned recirculation at the exit ventilation slot disappears. As it comes out from Figure 8j this configuration achieves the best possible ventilation of the room with warm air having the smallest possible recirculation loop without the need for increasing the storage wall thickness. Furthermore, it provides the higher air mass flow among the cases for air gap width of 10 cm and storage wall thickness of 10 cm. In any case, the efficient operation of the Trombe wall, with regard to the flow field, is a result of the combination of its basic geometric characteristics and the geometry (depth, height) of the coupled room.
3.2.2. Velocity and Temperature Distributions at Air Gap Exit

In Figure 10 the velocity distribution at the middle of the exit ventilation slot is presented for the 10 examined geometries. The type of lines used hereinafter are as follows: (a) grid 1—thick black continuous line, (b) grid 2—thick green dashed-dot line, (c) grid 3—slim blue line with blue circles, (d) grid 4—slim purple continuous line with purple rectangular, (e) grid 5—slim brown continuous line with brown triangles, (f) grid 6—slim yellow continuous line with yellow rhombus, (g) grid 7—slim orange continuous line with orange x, (h) grid 8—slim continuous grey line with grey cross, (i) grid 9—blue thick dotted line, (j) grid 10—thick red hidden line.

Of all geometries examined, except for the last one, the recirculation loop area is identified with the negative velocity values, which indicate a return from the room to the air gap. This recirculation pushes up the warm air stream coming out of the gap reducing the cross section available in it and thus its profile (Figure 5c) is compressed towards the upper part of the slot. The basic pattern remains the same, having a maximum near the top of the slot which has a higher temperature than the bottom of the slot. This temperature difference might not be so pronounced if the case were examined in 3D. As the width of the gap increases the recirculation area shrinks and therefore the local pressure drop also decreases. This together with the increase in air mass flow results in an increase in the maximum velocity. Virtually anything that increases the air mass flow (increase of the storage wall thickness, increase of the distance between the ventilation slots) increases the maximum values of the exit velocity. Increasing the diameter of the ventilation slot modifies the distribution, because it changes the local pressure drop, but without significantly affecting the maximum velocity, as long as it does not significantly change the flow. What really modifies the velocity distribution is the slope of upper ventilation slot, of the 10th geometry, due to which the recirculation disappears (Figure 9).

In this case the distribution of air velocity in the exit slot is similar to that in the gap with maxima depending on the temperatures of the wall surrounding the slot. The velocity distribution at the exit slot as well as the maximum velocity value determine the efficiency with which the Trombe wall ventilates the coupled room as shown when Figures 8 and 10 are examined together.

In the Figure 11 the temperature distribution at the middle of the exit ventilation slot is presented for the 10 examined geometries.
The increase in temperature at which air exits the gap to heat up the room ranges from 2 to 8 degrees. The distribution at the exit is much more uniform than the middle of the gap as the thermal boundary layers in the gap before the exit have fully converged. The recirculation area with the lowest temperatures near the bottom of the exit slot is identified again, except for the 10th geometry. The maximum increase is achieved with geometry 2 which corresponds to the smaller gap. Increasing the thickness of the gap leads to a decrease in the exit temperature, since the same amount of radiation is used to heat larger amounts of air (higher air mass flow). On the contrary, increasing the thickness of the storage wall, although it leads to higher flow rates, results in higher maximum temperatures, near the upper part of the slot, on the one hand because it reduces the effect of recirculation and on the other hand because the air continues to be heated from the upper part of the storage wall. Air gap with width significantly smaller than the diameter of the exit slot gives a more uneven temperature distribution since due to the cross section change it is impossible to maintain the uniform distribution that had been achieved within the gap. However, this uniform distribution of temperature is maintained in the 10th geometry where the inclination of the slot allows the smooth change of direction of the flow.

3.2.3. Flow and Average Temperatures

In Table 5 average temperatures achieved at characteristic surfaces of Trombe wall components as well as at the upper ventilation slot are presented.

Table 5. Average Temperatures.

| Geometry | Average Temperature at Storage Wall Internal Surface, $\theta_{si}$ [°C] | Average Temperature at Room Facing Surface, $\theta_{opi}$ [°C] | Average Temperature at the Air Gap Exit, $\theta_{out}$ [°C] |
|----------|-------------------------------------------------|-------------------------------------------------|-------------------------------------------------|
| Grid 1   | 26                                              | 24.94                                           | 22.66                                           |
| Grid 2   | 26.47                                           | 22.21                                           | 24.45                                           |
| Grid 3   | 22.41                                           | 21.13                                           | 21.7                                            |
| Grid 4   | 22.35                                           | 20.21                                           | 22.16                                           |
| Grid 5   | 22.26                                           | 20.15                                           | 23                                              |
| Grid 6   | 24.69                                           | 21.83                                           | 22.39                                           |
| Grid 7   | 26.17                                           | 22.28                                           | 22.5                                            |
| Grid 8   | 25.45                                           | 21.98                                           | 21.51                                           |
| Grid 9   | 25.42                                           | 22.05                                           | 21.53                                           |
| Grid 10  | 24.25                                           | 22.67                                           | 22.64                                           |
In general, increasing the width of the air gap leads to a decrease in the temperature of the inner surface of the storage wall because most of the air that enters between the transparent cover and the storage wall absorbs heat, either directly by radiation or indirectly by convection. Nevertheless, the air gap width does not seem to be the determining factor for its configuration. On the other hand, increasing the thickness of the storage wall leads to a decrease in the temperature of the inner surface as the heat received is stored in a larger wall volume. But again, the differentiation is small as the increase in thickness also leads to an increase in the heat transfer resistance with conductivity. Increasing the distance between the ventilation slots leads to an increase in the wall temperature as the available surface of the wall for storing sunlight increases. Finally, increasing the diameter of the ventilation slots does not significantly affect the temperature of the inner surface of the storage wall.

When commenting on the temperature of the room-facing storage wall surface we must take into account the fact that the simulation is steady-state. The final value of this temperature depends on a combination of factors that determine the air flow and the temperature of the air in the gap and cannot be attributed solely to the width of the air gap. However, it is clear that the optimal value is achieved with a net air gap of 7.6 cm. Increasing the thickness of the storage wall results in, in the case of steady-state simulation, a reduction of the temperature of the room-facing storage wall surface. Changing the distance between the ventilation slots does not seem to significantly affect this temperature.

Increasing the air gap width leads to a decrease in the exit temperature of the air from the gap, as expected since the same amount of radiation is used to heat up a smaller amount of air. An increase in wall thickness would be expected to lead to a decrease in exit temperature since more heat is stored in a larger wall. However, this is not the case. As can be seen from Figures 10 and 11 in the exit slot a recirculation loop of air is created from the room returning to the gap with the room temperature. As the wall thickness increases, this recirculation decreases as the air flow from the gap to the room becomes smoother. The result is that the average air temperature in the middle of the exit ventilation slot increases as the thickness of the storage wall increases. Increasing the distance between the slots leads to a decrease in the exit temperature as the temperature of the storage wall increases inversely. Finally, an increase in the diameter of the slots leads to a small decrease in the exit temperature due to an increase in flow.

At Table 6 the achieved mass air flow is presented for the 10 examined geometries.

| Geometry | Air Gap Air Mass Flow [kg/s] |
|----------|------------------------------|
| Grid 1   | 0.0249                       |
| Grid 2   | 0.021                        |
| Grid 3   | 0.0283                       |
| Grid 4   | 0.04                         |
| Grid 5   | 0.049                        |
| Grid 6   | 0.029                        |
| Grid 7   | 0.035                        |
| Grid 8   | 0.029                        |
| Grid 9   | 0.034                        |
| Grid 10  | 0.038                        |

It is clear that increasing the width of the gap leads to an increase in flow, which agrees with the findings of other researchers [42]. However, the increase in flow is not proportional to the available cross section. Also increasing the thickness of the storage wall leads to an increase in flow mainly due to the smoother path followed by the air flow to the room due to the reduced local pressure losses. The same effect, even more intense, has the increase of the flow through the improvement of the configuration (exit angle) of the upper ventilation...
slot, which leads to an increase in the flow, maintaining, however, a high exit temperature. Increasing the distance between the slots leads to an increase in flow while the same effect is observed when the diameter of the slots are increased.

4. Discussion

The purpose of the Trombe wall is to heat up a space utilizing incident sunlight. The heat transfer to the coupled room is achieved either due to convection through the warm air that enters the room from the upper ventilation slot (therefore high flow and high temperature of the exhaust air are required) and through the room-facing storage wall surface due to convection and radiation (therefore the high temperature of this surface is required). However, it is important that this heat penetrates as well as possible into the room and only the warm air stream from the upper ventilation slot can contribute to this. Therefore, in addition to high flow and temperature, equally important is the flow field that develops and the degree to which the room is actually ventilated by the air-gap without leaving large areas in which the existing air simply recirculates. The operation of the Trombe wall associated with heat storage and its time lag performance cannot be studied by steady-state simulation. The findings of the CFD simulations will be assessed with respect to the aforementioned criteria.

It is observed that a small gap width leads to a relatively small flow. Since this reduction is not proportional to the available cross-section it leads to higher air velocities inside the air gap which in turn enhance the heat transfer by convection from the surfaces of the cover and the wall. This along with the fact that the same amount of radiation is used for the heating of a smaller volume of air leads to higher exit air temperature. Inversely, the increase of the airgap width increases the mass air flow while decreasing the gap exit temperature and increasing the temperature of the room facing storage wall surface. This is due to the fact that lower velocities observed inside the gap reduce the rate of heat transfer by convection from the wall surface. A further increase in width leads to an increase in flow but not to an increase in the room facing storage wall surface temperature since larger amounts of air in contact with the warm surfaces gaining larger amounts of heat by convection. It seems that the 7.6 cm air gap can balance the heat transfer mechanisms ensuring high temperatures both at the air exit and at the surface that is in contact with the room with a relatively small flow.

Increasing the thickness of the Trombe decreases the pressure drop at the exit and inlet ventilation slots and thus for the same thermal buoyancy higher mass flow rates are achieved. At the same time increasing the thickness of the storage wall leads to lower air temperature at the exit ventilation slot because a bigger amount of air absorbs the same available solar radiation (through radiation and convection from the gap’s solid surfaces). At the same time the temperature of the gap facing storage wall surface increases because a thicker storage wall leads to the increased thermal resistance of the wall preventing heat transferred by conduction to the room facing wall surface. However, the increase of storage wall thickness allows the air from the gap to penetrate deep into the room improving thus the Trombe wall effectiveness, a conclusion consistent with the results of previous research [19].

The only significant change resulting from the variation of the diameter of the ventilation slots is related to the velocity distribution at the air exit from the gap. The small observed reduction in temperature in the exit cross-section equals the corresponding small increase in mass flow rate. The result is that there is no significant effect on the efficiency with which the Trombe wall ventilates the coupled room. Finally, the surface temperature of the storage wall does not appear to be significantly affected.

Increasing the distance between the ventilation slots both the air mass flow and the storage wall temperature increase since a larger solar collection area is available, while the storage wall temperature increase is small. Inversely, the temperature at which the air leaves the air gap decreases by a small percentage while its distribution in the exit slot cross
section is not significantly affected. What is important is that the ability of Trombe wall to ventilate the coupled room is improved.

The sloping exit slot configuration leads to higher flow rates for specific choices of air gap width and storage wall thickness since it reduces the local pressure drop due to flow direction change. This way improves the operation of Trombe wall and allows the warm air stream to penetrate deep into the coupled room without leaving large areas with recirculation loops.

5. Conclusions

A 2D CFD model was developed for the simulation of the Trombe wall cross-section operation. The developed model is able to model adequately the transparent cover behavior in different radiation wavelengths (four wavelength radiation bands were considered) which is crucial for the simulation of Trombe wall operation as well as the adequate participation of the storage wall in the heat transfer phenomena. The CFD model presentation was accompanied by the description of the calculation of equivalent cover optical properties since the latter is modeled as a single material. The model was applied to 10 geometrical configurations of a Trombe wall and the results were presented and discussed in terms of achieved air mass flow rate, temperature, velocity distributions, and flow fields including in the computational domain the room served by the examined Trombe wall.

According to the presented research findings when high air exit temperatures are required, a small gap width of 5 cm is suggested which offers increased temperature by 7% with respect to a gap width of 7.6 cm and by 11% with respect to a gap of 10 cm. When high room-facing storage wall surface temperatures are required a gap width of 7–8 cm is suggested which gives surface temperature increase by 12% with respect to the 5 cm gap.

However, these options leave a significant part of the serviced room without heating through the warm air stream and in addition lead to low mass air flow, reduction of the gap width by 50%leads to a reduction of the supply by 25%. This can be corrected by increasing the distance between the slots, which presents an increase in the air mass flow rate by 0.34%/cm, or by increasing the storage wall thickness. A storage wall with a thickness of 30 cm gives an increase of air mass flow rate by 73% with respect to a wall with a thickness of 15 cm. Finally, a much better choice is the proper shaping of the exit ventilation slot with an inclination of 30 degrees to achieve an increase of flow rate by 35% with respect to the horizontal slots and gives optimal penetration of warm air flow into the room maintaining relatively high air temperatures.

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Abbreviations

1D 1 dimension  
2D 2 dimensions  
3D 3 dimensions  
BES Building Energy Simulation  
CFD Computational Fluid Dynamics  
DO Discrete Ordinates  
EB Energy Balance  
FOU first order upwind  
IR Infrared  
MUSCL Monotonic Upstream-centered Scheme for Conservation Laws  
NIR Near Infrared  
RANS Reynolds Averaged Navier Stokes  
RTE Radiation Transfer Equation  
SOU second order upwind  
SIMPLEC Semi-Implicit Method for Pressure Linked Equations-Consistent  
UV Ultra Violet  
VIS Visual  
a upper ventilation slot angle [degree]  
d cover thickness [m]  
dg air gap width [cm]  
ds ventilation slot diameter [cm]  
dw storage wall thickness [cm]  
L distance between ventilation slots [m]  
n refractive index [-]  
r the cover reflection [-]  
Rac Rayleigh number along the cover internal surface [-]  
Rasw Rayleigh number along the internal storage wall surface [-]  
ux velocity in the horizontal direction [m/s]  
uy velocity in the vertical direction [m/s]  

Greek letters  
α cover absorptance [-]  
αs extinction coefficient [1/m]  
αt total cover absorptance [-]  
θeb energy balance model predicted temperature [°C]  
θint the air-gap air temperature [°C]  
θse transparent cover internal surface temperature [°C]  
θsi the storage wall internal surface (facing the air gap) temperature [°C]  
θopi storage wall room-facing surface temperature [°C]  
θout air-gap exit temperature [°C]  
λ wave length [μm]  
ρt total cover reflectance [-]  
ρti reflectance of the i glass pane [-]  
τα transmittance due to absorption losses [-]  
τt total cover transmittance [-]  
τti transmittance of the i glass pane [-]

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