Simulation of a full scale Trombe wall system

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Abstract

Two dimensional transient modelling of the flow and heat transfer in an idealized Trombe wall system was performed for a fixed set of physical dimensions. The governing equations were solved using multiphysics software. Buoyancy and thermal accumulation in a brick wall are the main effects considered. During operation, the solid and fluid temperatures constantly increase, while circulation flow rates, which are significant from the start, show a slow decrease. The room is heated by the parallel mechanisms of buoyancy flow from the collector space and directly by the hot wall, but the latter effect becomes progressively more important. Heat losses through the glazed surfaces of room and collector are of similar magnitude. However, when the wall clearances are closed the collector loss shows an important increase. The role of the typical Trombe wall circulation in the control of heat losses is discussed.
Keywords: Trombe wall system, natural convection, numerical models.

1 Introduction

In this paper a two dimensional simulation of a conventional Trombe wall system is described. The importance of passive solar heating systems in latitudes that favour winter solar collection on vertical walls has increased in recent years. These systems are known to depend on buoyancy as the main driving force for air flow, and on the accumulation of energy by a massive wall for thermal comfort in periods of time without solar energy input [1, 2]. Given the low flow rates that occur in such devices, the Trombe wall system is a very interesting subject for numerical simulation, especially because the experimentation in such systems is still relatively expensive.

The thermal performance of Trombe wall systems has been studied by experimental and numerical methods. Numerical predictions of the thermal

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behaviour of Trombe wall systems are of two kinds: either macroscopic approaches using overall energy balances and heat transfer correlations, or solutions of the system of differential equations: Navier - Stokes, continuity and energy. Most of the published simulations are based on the first approach [e.g. 3, 4, 5], often coupled with one dimensional transient simulations of heat transfer through the wall. Although relatively simple and manageable, macroscopic models do not supply much physical information for the description of the many thermal and fluid flow phenomena occurring in the system.

The differential approach can provide more physical information on flow rates and heat transfer for design, and can also reveal unexpected phenomena that may occur, but it still poses considerable difficulties because one has to deal with time dependent, three dimensional, full scale models with high thermal and velocity gradients. An early attempt to numerically predict the thermal behaviour of Trombe wall systems under the assumption of laminar regime was presented by Ormiston *et al*. [6]. A two dimensional region comprising a collection space, wall and heated space was considered. The wall face adjacent to the collection space was at imposed temperature while the other face was insulated. The thermal capacity of the wall was not taken into account. The Rayleigh number for the collection space was 10^7, based on the width of that space. This justified the laminar flow assumption. Other simulations have considered turbulence models: In [7], ventilation rate predicted using a k-ε turbulence model was found to increase with the wall temperature and heat gain. The effects of the gap between wall and glazing, wall height, glazing type and wall insulation were also investigated. In [8], the interaction of natural convection and radiation in a channel was investigated by means of a differential model.

The objective of this work is to describe quantitatively the fluid-thermal behaviour of a full scale Trombe wall system. Specifically we are interested in describing the flow patterns and heat transfer mechanisms occurring in this system, and their evolution in time. Some idealizations are made in order to make the problem tractable: a) Two dimensional flow is considered, b) Buoyancy and thermal accumulation in the wall are considered the main physical effects, and thermal radiation is excluded, except for the solar input localized on the massive wall. For ease of computations, laminar flow is assumed. This assumption is realistic even for very high Rayleigh numbers if the heated and cooled walls are vertical.

The heat input to the wall is considered constant. This differs from a real situation in which the solar irradiation will vary along the day. However, the purpose here is to obtain some operational characteristics under relatively stable conditions. Main results are the air flow rates moved by the system; the heat transfer rate to the heated space, the heat losses though windows and the accumulation of energy by the wall. The variations of all these quantities are determined in time. Normally Trombe wall systems never reach a steady state, but simulations have to cover at least one full cycle of heating and cooling. It would be necessary to simulate the Trombe wall operation for several days, so the initial conditions, which always are somehow arbitrary, are 'forgotten'. The

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problem as formulated here admits a steady state which was, however, not reached in a total simulation time of several hours.

2 Formulation and numerical method

A two-dimensional, laminar, time-dependent model was prepared using finite element multiphysics software, Comsol 3.5a. The region of analysis (Fig. 1) is a rectangular air-filled medium, 2.4 m high by 2 m wide, with a massive brick wall (2.1 m high and 0.15 m wide) dividing the fluid region into a collecting narrow slot with a gap e= 0.1 m at left and a space to be heated (room) at the right. Clearances of 0.15 m width are placed above and below the wall to allow fluid interchange between both sides of the wall. Physical properties of the wall are:

$$\rho = 1800\ kg/m^3,\ C_p = 840\ J/kgK,\ k = 0.81\ W/mK.$$

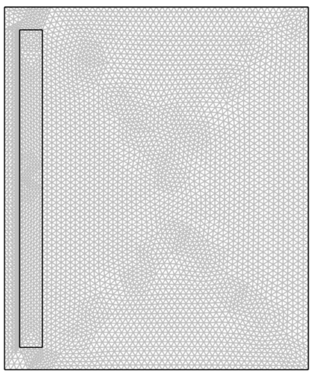

Figure 1: Region of analysis showing fluid and solid (wall) regions, and mesh.

The dimensions are similar to the ones found in real applications, except for the width of the heated space (1.75 m), which is made small because of computational limitations. An initial uniform temperature of 283.15 K is imposed. Uniform heating is applied to the left face of the wall from t=0. This single heat source, intended to represent solar input, is taken as constant in time. Two levels of heating, q= 250 and 500 W/m^2, are used in different runs. The latter irradiation rate is at or near the upper limit to be found in Trombe wall systems. The vertical boundaries of the region are glazed surfaces which behave as cold sources assumed to take a uniform, constant external ambient temperature of 283.15 K. The horizontal region boundaries are insulated. With these boundary conditions the system could reach a steady state.

The dimensional system of equations consists of the fully coupled, time dependent compressible Navier Stokes, continuity and energy equations. The non slip condition is imposed to velocities on all internal solid boundaries of the region. The buoyancy term in the momentum equations is modelled by making the density to depend on temperature and pressure as in an ideal gas:

$$F_y = -\rho g = -\frac{g\,p}{R_s T}, \qquad R_s = 287\,J/kgK$$

This approach is preferred to the Boussinesq approximation which requires reference temperatures to be chosen somewhat arbitrarily, and their choice may affect the solution. In the ideal gas assumption the density is determined from field values of temperature and pressure. The laminar flow assumption is justified by noting that the Rayleigh number for the collection space, in terms of the imposed heat flux (q) and the corresponding gap (e) is written as:

$$Ra = \frac{g\beta q e^4}{k\nu\alpha}$$

The thermal expansion coefficient is taken as the reciprocal of the initial air temperature. The resulting Rayleigh numbers are 9.33×10^7 and 1.86×10^8 for q= 250 and 500 W/m^2 respectively. These values are still in the range of laminar regime in differentially heated cavities with vertical heat sources.

The region of analysis is divided into 9666 triangular elements (Fig.1), 8630 in the fluid region and 1036 in the solid. To stabilize the system, streamline diffusion and crosswind diffusion are introduced. The Petrov-Galerkin method is found to stabilize computations effectively. A robust Backward Differentiation Formula (BDF) method is used, which selects the appropriate time step automatically. Convergence is controlled by adjusting the number of iterations in every time step, and by selecting appropriate values for absolute and relative tolerances in the different dependent variables separately. The time- dependent model was run for periods of at least 6 hours (physical time).

3 Results and discussion

3.1 Thermal and fluid flow characteristics

The time evolution of temperature field is described. A circulation pattern of warm air moving from the collection space to the room and back is observed, as expected. Heat conduction through the massive wall and convective flow through the clearances are the dominant heat transfer mechanisms to the room.

After imposing the heat flux condition on an enclosure of uniform initial temperature, the imposed heat flux causes rapid increase in the temperature of the left side of the wall. The temperatures in the whole system, which are initially uniform and equal to the temperature of the cold sources, start to increase. The average solid temperature increases much quicker than the fluid one. Therefore the accumulation of sensible heat at the wall is considerable; suggesting that heating of the room could be sustained for some time after suppressing the heat source. In spite of this, heating of the wall is slow because of its high density and specific heat. It takes some time for the effect of the imposed heat flux to be noticed across the wall. Therefore, it takes more than two hours before the wall starts supplying heat to the room. The temperature distribution in the walls is characterized by nearly vertical isotherms from the start. Isotherms propagate across the wall as heating progresses (Fig.2).

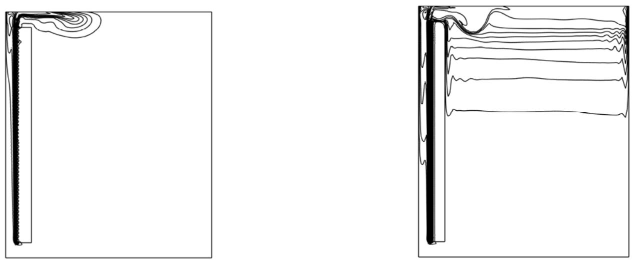

Figure 2: Isotherms at t=50 sec (left) and t=600 sec, q=250 W/m^2.

The inception of the flow for q=250 W/m^2 is shown in Figure 2 by means of isotherm diagrams. At very short times after the imposition of the heat source (t= 50 sec), a buoyancy driven stream has been generated, ascends in the collection space and is discharged at the top of the room. At t=600 sec. nearly a half of the height of the room has been affected by convective heating, although the wall temperature in most of the room side is still at the initial value. As a result of the early convective effect that crosses the upper part of the room, heat loses through the glass window on the right of the heated space start very early.

Heating and circulation have nearly reached the whole space by t=1800. A stratified temperature distribution is formed, with high temperatures above, although the temperature does not vary linearly with the vertical coordinate, as in a differentially heated cavity. Circulation has become general by t=3000 sec. The flow pattern in the collecting space is more complex than in the room, but essentially consists of vertical flow, upwards on the hot wall and downwards near the glazed surface.

The final flow pattern generated at the room consists of upward flow at the wall and downward flow at the window, with an essentially stagnant core and a stratified temperature distribution. This is reminiscent of the flow seen in a differentially heated cavity, except for the flow from the collection space to the heated space through the upper clearance and back to the collection space at the lower clearance. At short times, however, flow is downwards also near the wall, because of its high thermal inertia.

The volume flow rates moved by the wall take significant values (of order of magnitude 5-10 m^3/hour per meter depth) almost immediately after the imposition of heating to the wall. These flow rates are determined by integrating the horizontal velocity at the clearances. Motion is generated by buoyancy forces (natural convection). In long time intervals a continuous tendency to increase in flow rate is not observed, but rather a diminution (Fig. 3). This is probably caused by a progressive reduction in buoyancy forces, caused in turn by the constant increase in fluid temperature. The flow from the room to the collection space through the lower clearance is shown with negative values in Fig. 3. The magnitudes of flow rates of both streams are not equal at every instant, as the system never reached a steady state in the present simulations.

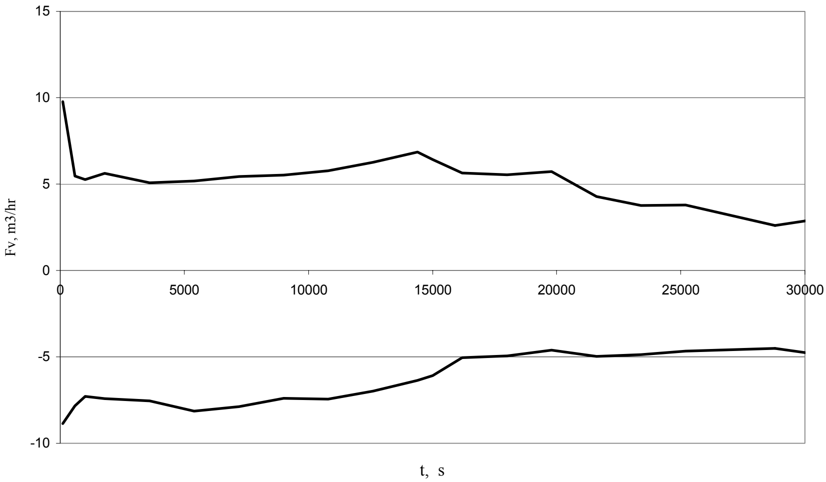

Figure 3: Time evolution of flow rates through clearances, q=250 W/m^2. Upper curve: Flow from collector to room.

Circulation consists of warm air discharged on the room, as already described. As this movement starts early, with a relatively high flow rate from the outset, warm air crosses the upper part of the room reaching the window on the right. Propagation of warm air downwards takes some time, as it has no tendency to go down. However, by t =1800 sec, a temperature higher than the initial temperature is found in the whole room.

3.2 Heat transfer

From the above description, it is evident that heat losses through the glazed surfaces are observed very early. They grow continuously in time, as the difference between the temperatures of the fluid and of the external environment increases due to air heating. The losses from the two glazed surfaces (room and collection space) are essentially of equal magnitude during the simulations (Figure 4 and Table 1).

The heat transferred to the room air is shown in Figure 4. It is composed by the heat flow from the wall face adjacent to the room, q_w, plus the convective flow. Figure 4 shows that q_w is initially negative because the temperature of the air that passes to the room from the collector is higher than the temperature of the wall face, which is still the initial temperature. The room starts to receive heat from the wall only after the heat from the source has propagated through the wall. A time t=8660 seconds is necessary for heat transfer to the room when the heating rate is q= 250 W/m^2.

Table 1: Some flow and heat transfer results.

	q=250 W/m^2	q=500 W/m^2	q=250 * W/m^2
Maximum wall temperature, K	314.3	341.4	317.3
Maximum air velocity, m/s	0.482	0.668	0.316
Conductive heat through the wall, q_w	34.82	77.11	41.09
Heat loss through collector glazing	19.72	33.93	52.07
Heat loss through room glazing	19.57	33.59	15.23

*: Trombe wall without clearances

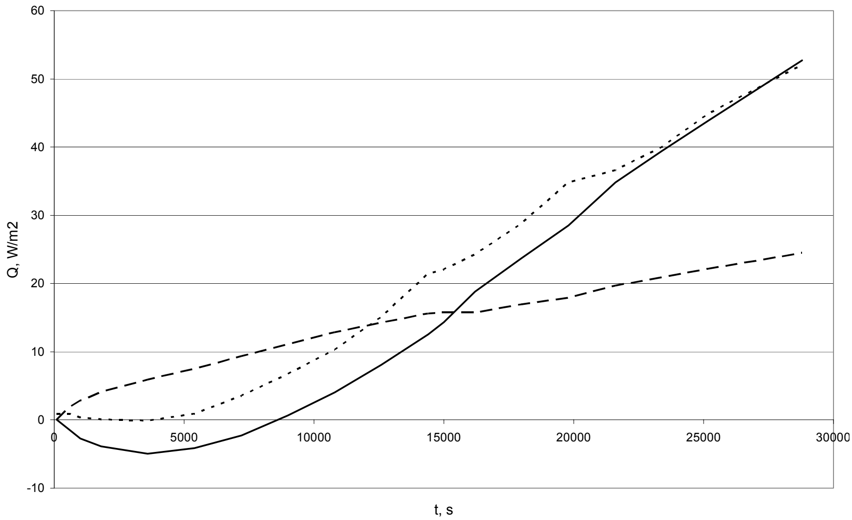

Figure 4: Evolution of heat transfer to the room. Bold line: Wall heat transfer. Dotted line: Heat transfer through the wall plus convective circulation. Dashed line: Heat transfer through the room window.

The total heat transferred to the room air increases with time, as shown by the dotted line in Figure 4. The contribution of convective flow to room heating is initially very important, but for long times the conductive heat transfer through the wall prevails, as seen in Fig. 4. This is associated with the progressive heating of room air, which lowers the available buoyancy force.

The effect of heating rate is now considered. An increase in the imposed the wall heat flux to 500 W/m^2 causes moderate increases in solid and fluid temperatures, flow rates, heat transferred to the room and heat losses through the glazed surfaces. These changes are illustrated in Table 1, with results taken at the instant t=21600 sec.

The maximum temperature in the solid occurs at the upper left corner of the wall and increases significantly with the heating rate, as shown in Table 1.

Also the heat flow through the wall, the thermal losses through the room and collector windows, as well as the heat delivered to the room, increase markedly with q.

The last column in Table 1 shows results for q= 250 W/m^2 for a variant of the Trombe wall system in which the flow clearances have been suppressed by increasing the wall height from 2.1 to 2.4 m. In this case the room is heated only by conduction through the wall. The maximum solid temperature is higher and the maximum flow velocity is lower than in the wall with clearances. The heat delivered to the room and the heat loss through the room window (Fig. 5) take more than one hour to become significant, in absence of convective flow from the collector. Although this loss is significantly lower than in the conventional disposition, the losses through the glazing in the collector greatly increase without the use of clearances as seen in Table 1 and Figure 5. This is due to the fact that, without clearances, the air warmed up by the heated side of the wall has to turn toward the cold glazed surface instead of entering the room, and this contact greatly increases the heat loss.

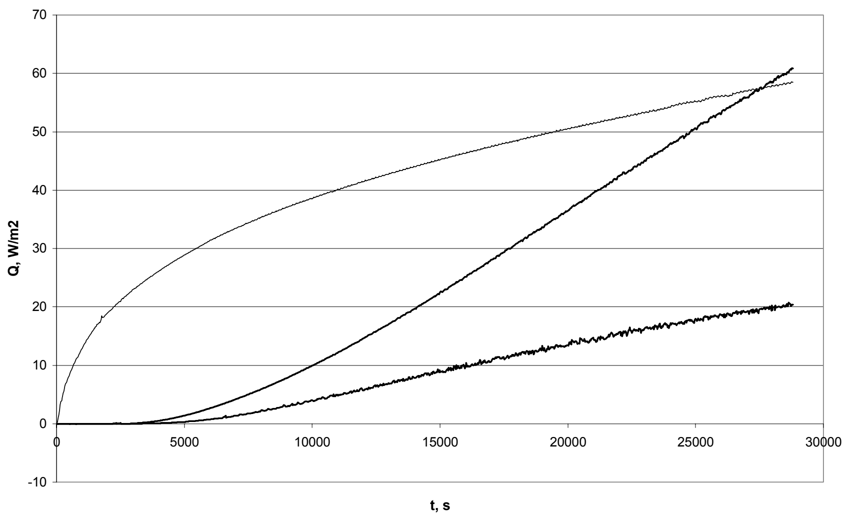

Figure 5: Trombe wall without clearances. Upper curve: collector heat loss. Middle curve: Heat flow to the room. Lower curve: Room heat loss.

4 Conclusion

A time dependent numerical model built on the software Comsol 3.5a provided results on flow rates and heat transfer in an idealized 2D, laminar, full scale Trombe wall system with a imposed heat flux source. The flow patterns (here described by isotherm lines) show that motion generation occurs early in the collector space. Motion is propagated to the room, establishing a circulation pattern similar to the one found in a differentially heated cavity, which coexists with the characteristic Trombe wall circulation from the collector space to the room and back.

Circulation rates in the range of 5 - 10 m^3/hr are obtained. They increase with the heating rate, but decay slowly in time as the fluid temperature increases. Conversely, the heat losses though the glazed surfaces of the collector and the room increase in time. The losses from both glazed surfaces are very similar. Although, due to the thermal inertia of the wall, heat transfer to the room initially depends mainly on the Trombe wall circulation, for long times it is more dependent on the heat delivered by conduction through the wall. This does not mean that the characteristic circulation between both fluid zones is unimportant: In a test with a Trombe wall without clearances, the heat losses through the collector glazed surface were found to increase significantly, greatly exceeding the losses from the room window. This means that the circulation, although having a secondary role in the heating of the room, is important as an appropriate means of controlling collector heat losses.

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