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VENTILATION CHARACTERISTICS OF A TROMBE WALL THERMOSYPHON LOOP

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ABSTRACT

The air movement and heat transfer characteristics are investigated for a Trombe wall channel solar passive system that is coupled to an adjoining room. A finite volume numerical analysis predicts air flow and energy delivery rates for several temperature difference values applied across the parallel walls of the channel. The performance of the wall during the ventilation (open system) mode is also analyzed to examine its suitability for passive cooling applications.

KEY WORDS: Trombe Wall, Ventilation, Passive Heating

1. INTRODUCTION

Solar passive systems are receiving increasing attention for two reasons. First, they do not involve moving parts and as a result, they are less prone to failure and thus enjoy a relatively higher life time as compared to active systems. Secondly, many of the passive solar energy systems are easily integrable with buildings, and this reduces the initial cost of the system. The present work analyzes the characteristics of an advanced passive solar system, known as the Trombe wall collector-storage system (Robert et al. 1978). A thick south facing wall made of concrete, brick, adobe or stone is located directly behind a single or double glazing. The air gap between the glass and the black painted face of the wall acts as a convective channel that forms part of the closed loop of the thermo-syphon that transports solar energy from the heated wall to the room. On a sunny day, about half of the solar energy gain by the storage wall is transferred by the airstream for immediate heating of the interior space while the remainder portion of the energy is stored in the wall for supplying the night-time energy requirements.

Although natural convection phenomenon in the Trombe wall solar passive systems has been investigated by others (Akbari and Borgers 1979; Borgers and Akbari 1984; Kettleborough

1972; Nakamura et al. 1982; Tichy 1983) , most of previous studies have considered only simple geometry involving parallel wall channel. These studies have used boundary layer approximation, and have made assumptions regarding inlet and exit vent losses (Tichy 1983). Other analyses have considered parallel wall channel geometry but have solved the Navier-Stokes equations to account for inlet and exit vent losses (Nakamura et al. 1982; Kettleborough 1972). These studies predict combined inlet and exit vent losses but are limited to simple geometry of parallel wall channel.

In this work, efforts have been directed towards analyzing numerically the turbulent natural convection in the Trombe wall channel coupled to a room. A numerical code based on the "SIMPLE" algorithm, has been employed to solve the coupled momentum and energy equations that describe the thermosyphon phenomenon in the Trombe wall solar collector system. The mass flow rates induced by the chimney action of the heated wall face are calculated as a function of temperature difference applied across the channel wall. Finally, the operation of the Trombe wall collector for summer ventilation potential is also investigated.

2. PHYSICAL MODEL

The schematics of the Trombe wall channel coupled to the room is shown in Figure 1. The operation is illustrated for both heating and cooling modes. The left surface of the room is a clear glass wall that allows solar radiation in to the air gap, resulting in temperature rise of the black painted face of the storage wall. The heating of the concrete wall face and the glass window causes a buoyancy driven flow in the channel. The low pressure created inside the channel draws in cooler air from the room. The heated air is then discharged to the room to meet the heating load demands. In the cooling mode, the vent on top of the channel and the one in the room are both open during the operation. The surface of the storage wall, facing the room, is insulated. The heated air in the channel rises due to the buoyancy action, and is ejected from the channel vent. The cooler air from outside is drawn into the conditioned space from the room vent to provide the cooling and ventilation effect. It should be noted that the Trombe wall system has normally been used for meeting the heating load demand only. In present study, the potential of Trombe wall for providing ventilation and cooling effect is also investigated.

3. GOVERNING EQUATIONS AND NUMERICAL SOLUTION PROCEDURE

The governing partial differential equations for the conservation of mass, momentum, energy and turbulence quantities can be written in the following general form

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

It is noted here that the two-equation (k-ϵ) turbulence model (Launder and Spalding 1974) has been used to calculate turbulent viscosity and thermal conductivity. For example, the turbulent viscosity in this model is expressed as

$$\mu_t = \frac{\rho C_\mu K^2}{\epsilon} \tag{2}$$

The governing equations for u, v, T, K and ϵ are solved numerically by a finite volume technique. In this method, a numerical grid, as shown in Figure 2, is chosen, and the partial differential equations are integrated over elemental control volumes drawn around the grid points. This procedure yields coupled algebraic equations for physical variables at each grid point. The quadratic upwind scheme (QUICK) (Lenonard 1979) is used to interpolate the values of various variables between the grid points. This scheme combines the relatively high accuracy of central differencing scheme with the stability property of the upwind scheme. The set of simultaneous equations is solved by a semi-implicit iterative scheme known as "SIMPLE" (Patankar 1980). This algorithm involves distribution that satisfies the mass conservation and momentum equations. The boundary conditions for velocity components involves satisfying non-slip condition. For the turbulence quantities, the wall function approach is used (Launder and Spalding 1974). All horizontal surfaces are insulated and the vertical surfaces are maintained at specified temperature levels.

4. DISCUSSION OF RESULTS

Prior to obtaining detailed results, grid independence of computed results was considered for the geometry shown in Figure 1. For all results presented here, a 111×61 variable step size grid, shown in Figure 2, was used. Results were also computed with a 151×91 grid. Figure 3 presents a comparison of the local heat transfer coefficients on the hot wall, as computed from the two grids. It is noted that except for minor deviation in the channel entrance region, the heat transfer results are essentially identical for the two grids. Same comments also apply to temperature and velocity fields. This indicates that results are essentially grid independent. Figure 4 shows the temperature variation inside the system when the left and right boundaries of the channel are maintained at 300 K and 320 K respectively. The corresponding streamline patterns are shown in Figure 5. It is noted that temperature stratification in the room is quite prounced for this case.

Table 1 gives the comparison of the total energy and air delivered to the room as the cooled wall temperature is varied from 270 to 300 K. The temperature of the hot wall is maintained at 320 K. We note that for the low cooled wall temperature case, the heat losses on the cold wall are very high and consequently mass and energy delivered to the room is low. As the cold wall temperature is raised to 300 K, the mass flow rate increases by about six fold and energy delivery rate increases by about eighteen fold.

In order to economize on the computational time, the heat transfer and ventilation characteristics were also computed by using truncated geometries shown in Figures 6a and 6b. In the case of Figure 6a, a free pressure boundary condition (p = 0) at a location x = 1.25 m is applied. In Figure 6b, the free pressure boundary condition is applied at a section x = 0.6m. Table 2 shows the comparison results of these three geometrical cases shown in Figure 1 and Figure 6. It is noted that both mass and energy delivery rates are within nine percent of one another for all three cases. This implies that the simpler geometry in Figure 6b can be used provided one is willing to tolerate an uncertainty of about nine percent in results in exchange for much lower computational effort.

Figures 7 and 8 show the temperature and flow field for the Trombe wall system operating in the ventilation mode. This case would be appropriate for the summer months for providing cooling by circulating outside cooler air into an otherwise warm room. The geometric configuration for this case is given in Figure 1. The vent opening located on the top of the right most wall is about 40 cm in width. From Figure 7, one observes that the temperature in the lower portion of the room drops from initial value of 310 K to about 304 K, a drop of about 6 K.

5. CONCLUSIONS

A numerical calculation procedure has been used to determine the ventilation and heat transfer characteristics of the Trombe wall system. The results indicate that for a given geometry, the mass flow and energy delivery rates are strongly governed by the temperature difference applied across the channel vertical walls. The flow rate increases sharply as the left glass wall increases from a temperature of 270 K to 300 K. Results also show that a truncated Trombe wall geometry can be used to model the system consisting of Trombe channel coupled to the room. The truncated geometry requires substantially lower computational efforts and the mass flow and energy delivery rate results are within nine percent of the full size geometry results. Finally, the Trombe wall operation was simulated for the ventilation case, and the results show a 6 K temperature drop in the room due to fresh air entering the room through a vent located in the room.

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TABLE 1 Effect of Cooled Wall Temperature on Mass and Energy Delivery Rates

Cooled Wall Temp. (K)	Area (m^2) Cold	Hot	Mass Flow Rate (kg/sec) In	Out	Heat Flux (W/m^2) Cold	Hot	Mean Flud Temp. (K)	Net Energy (W)
270	4.0	3.8	-1.685×10^{-2}	1.703×10^{-2}	-2103	2176	291.5	25.5
280	4.0	3.8	-5.317×10^{-2}	5.227×10^{-2}	-1467	1657	292.7	143.0
290	4.0	3.8	-7.416×10^{-2}	7.435×10^{-2}	-906.1	1227	294.2	313.4
300	4.0	3.8	-9.025×10^{-2}	9.076×10^{-2}	-409.2	895.2	295.7	518.9

TABLE 2 Comparison of Different Geometries

Free Boundary Location (m) x^*	Area (m^2) Cold	Hot	Mass Flow Rate (kg/sec) In	Out	Heat Flux (W/m^2) Cold	Hot	Mean Fluid Temp. (K)	Net Energy (W)
4.0	4.0	3.8	-7.416×10^{-2}	7.435×10^{-2}	-906.1	1227	294.2	313.4
1.25	4.0	3.8	-7.858×10^{-2}	7.869×10^{-2}	-902.1	1231	294.3	339.8
0.6	4.0	3.8	-7.943×10^{-2}	7.989×10^{-2}	-890.5	1168	294.3	344.2

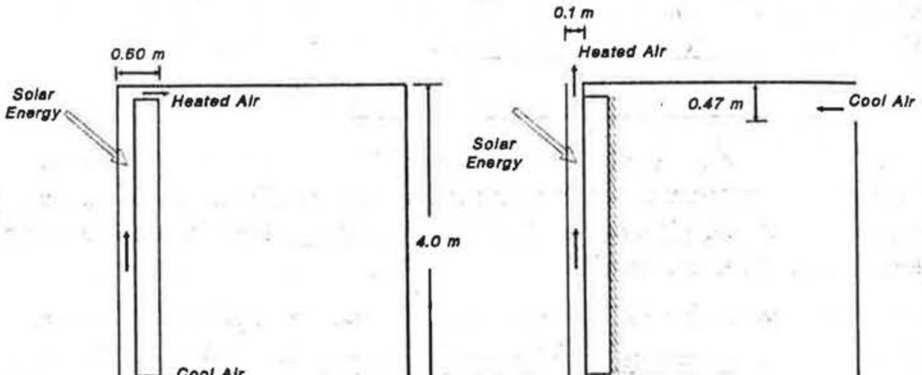

Figure 1. Schematic of Trombe wall operating in heating and cooling modes.

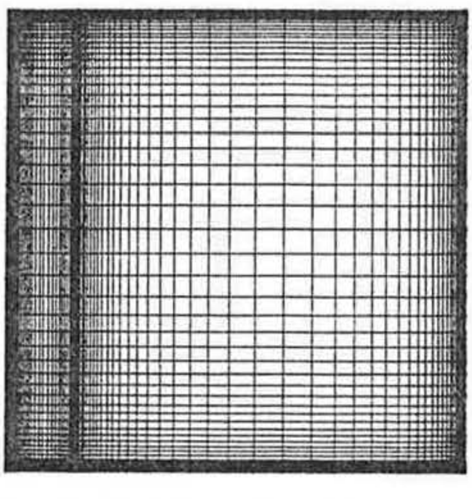

Figure 2. Grid illustration of the problem.

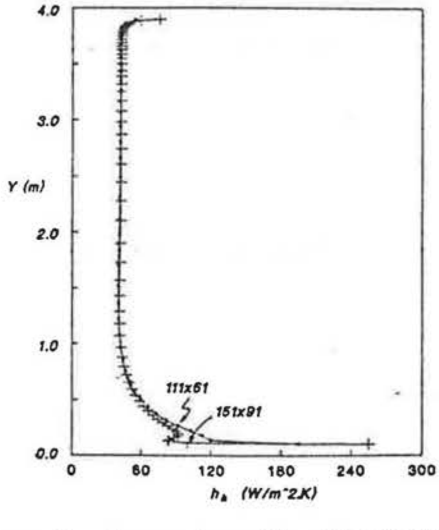

Figure 3. Comparison of local heat transfer coefficient on heated wall for 151×91 and 111×61 grids.

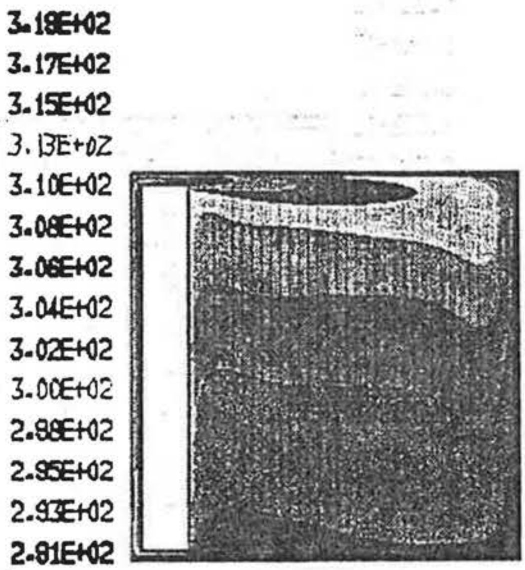

Figure 4. Temperature field for cooled wall temperature of 300 K for heating mode.

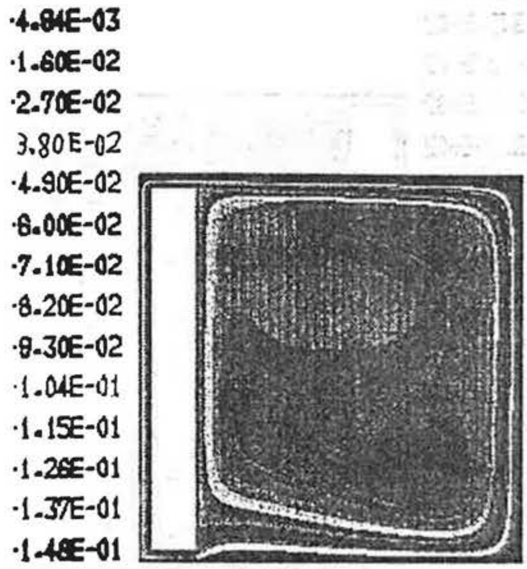

Figure 5. Flow field for cooled wall temperature of 300 K for heating mode.

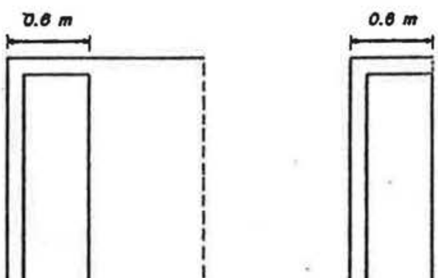

Figure 6. Schematic of truncated Trombe wall channel geometry.

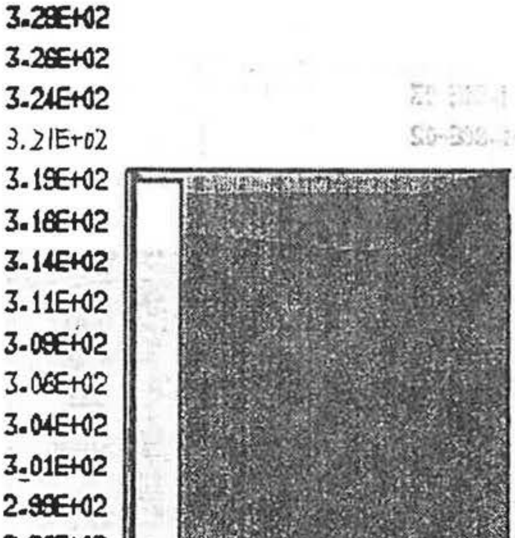

Figure 7. Temperature field for the Trombe wall channel operating in the ventilation mode.

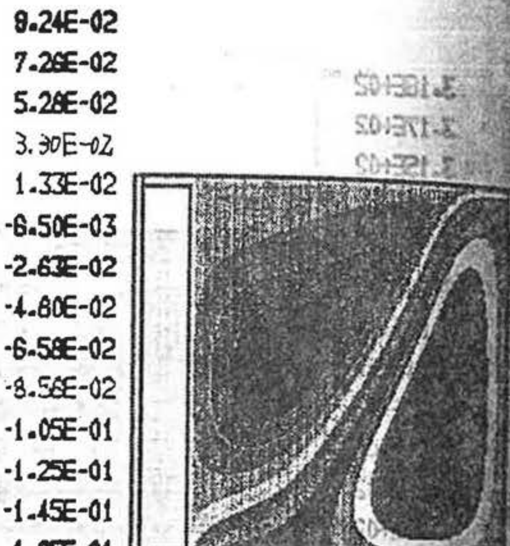

Figure 8. Flow field for the Trombe wall channel operating in the ventilation mode.

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