VOL. 5, NO. 7, JULY 2010 ISSN 1819-6608
ARPN Journal of Engineering and Applied Sciences

www.arpnjournals.com

SIMULATION ANALYSIS OF PASSIVE SOLAR STRUCTURES USING HEAT TRANSFER EQUATIONS

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ABSTRACT

Passive solar design is an economical way of using solar energy in buildings. The thermal behavior of the buildings within its environment over a period of time can easily be predicted using accurate and simple analytical tools. This provides designers the information necessary to satisfy the occupant's needs, reduce peak cooling/heating power demands, reduce the size of air conditioning/heating equipment, and the period for which it is required. There by, increasing the possibility of a successful design. The main goal of this study is to present modeling of passive solar structures using heat transfer equations. The simulation model results correlated well with the experimental data.

Keywords: modeling, passive solar design, heat transfer equations.

INTRODUCTION

A solar energy system is a group of interacting pieces designed to collect, store and distribute the solar radiation energy as needed for some specific purpose. The performance of solar energy systems is dependent upon weather. In a solar heating/ cooling system, for example both the energy collected and the energy demand are functions of solar radiation, the ambient temperature, and meteorological variables. These forcing functions are unique in that they are neither completely random, nor deterministic; they are best described as irregular functions of time, both on a small (hourly or daily) and large (seasonally or yearly) time scale.

Although significant efforts have been directed toward energy efficiency and various passive solar energy models [1,2,3] are available. However, a great amount of energy is annually wasted in heating and cooling buildings. To produce energy efficient buildings, designers must have simple and accurate analytical tools. In order to contribute to the development of analytical tools, modeling of passive solar building based on the glass pane [4,5] and Trombe wall models [6] is presented. In this paper a practical, simple and easy to use analytical model, is developed to predict the thermal performance of a passive solar system. The model is assessed with experimental data for verification purposes.

MODELING OF PASSIVE SOLAR SYSTEMS

The basic problems in passive solar heating involve an understanding of how passive systems work, learning to design them, and calculating in what locales they will be successful. The principal elements of this approach are shown schematically in Figure-1. With a validated mathematical model, it is possible to perform parametric studies leading to improved passive systems.

A computer simulation program is designed to connect component models in a specified manner, solve the simultaneous equations of the system model, and display results. Additionally, the program is capable of simulating glass panel and Trombe wall models.

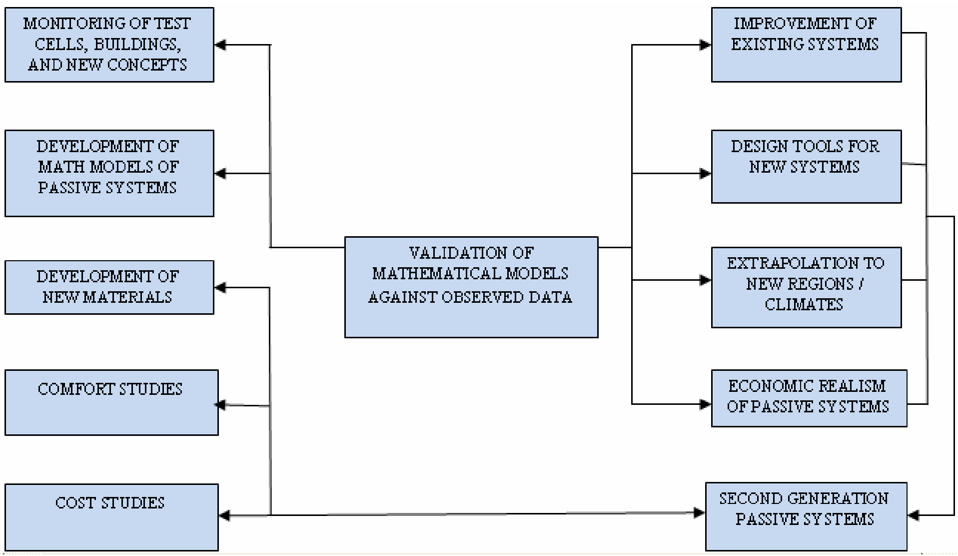

Figure-1. Components of present passive systems.

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PROPOSED MODEL

Efforts have been directed toward energy efficiency and thermal capacity of various passive solar structure models [1,2,3]. The applicability of these models is limited due to their cost and inefficient thermal performance arising from thermal lag heat losses at night. The purpose of this study is to study the thermal performance of glass pane and Trombe wall models using heat transfer equations for a passive solar system. The simulation model temperatures are compared with experimental data.

GLASS PANE MODEL

In glass pane model, solar energy passes directly into the living space through south-facing windows. The floors and /or walls provide storage for night time heating. This can be an efficient and economical project. The direct gain retrofit usually performs efficiently at the lowest cost [4,5,7,8]. Figure-2 illustrates the glass pane model geometry.

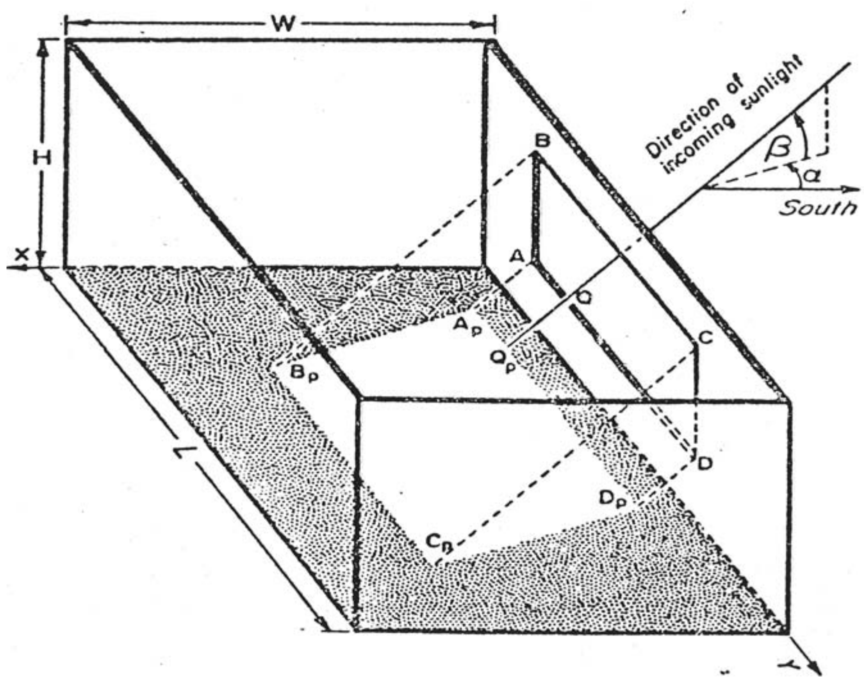

Figure-2. Glass pane model geometry.

This model is simple and easy to build. With this system, sunlight enters the house through large areas of south facing glass. The incidence angles of solar radiation depend on latitude, window orientation, day of the year, and time of day. These were calculated using relations from [9]. The fraction of insolation passing through the window, which is a function of incidence angle and number of panes, were then calculated using a method suggested by [9] for flat plat collectors. Figure-2 also shows the part of the floor directly illuminated by sunlight (area $A_P\ B_p\ C_p\ D_P$). In general, an illuminated area will be warmer than adjacent areas. Therefore, such an area was treated as an essential element to be assigned its unique temperature at any instant. The Figure shows direct illumination only on the floor. In general, parts of the floor, north wall, south wall, east wall, or west wall were illuminated at times. The illuminated area of each of these surfaces was computed at every time step. The interior surfaces were thereby divided into a total of eleven distinct areas characterized by their individual temperatures. Diffuse radiative view factors, F_{ij}, among these areas were calculated using their mathematical definitions, Simpson's rule, and view factor algebra. These view factors changed at every time step during a day, because the areas themselves change size and shape as the solar angles change.

Using these calculated quantities, solar radiation power absorbed by each area was then calculated. This absorbed energy could be stored, transported through the envelope by conduction, transported to the interior air by convection, or transported to another area by radiation (long wave). Accounting for all these possibilities resulted in the following eleven equations. The first six equations belong to glass pane model and the remaining five equations are used to simulate Trombe wall model.

$$Q_{si} + Q_{ri} - h_i A_i (Ti - T_a) - A_i (Ti - T) / R_i = (mc)_i\, dTi/dt \quad (1)$$

The index i (i = 1, 2… 11) is used to label the areas. In addition, conservation of energy for the interior air is expressed as:

$$\sum_{i=1}^{11} A_i h_i (T_i - T_a) - r\,(mc)_a (T_a - T) = (mc)_a\, dT_a/dt \quad (2)$$

Long wave radiation heating quantities in equation 1 were computed from [10] as:

$$Q_{ri} = (J_i - бT_i^{\,4})\, \varepsilon_i A_i / (1- \varepsilon i) \quad (3)$$

Where the radiosities J_i were determined from temperatures by solving:

$$бT_i^{\,4} - J_i + (1-\varepsilon_i)/(\varepsilon_i A_i)\,[\sum_{j=1}^{11} (J_j - J_i)\, A_j F_{ji}\,)] = 0 \quad (4)$$

Equation (4) was solved by Gauss elimination for each time step. Equations 1 and 2 were numerically integrated by the first order explicit Euler method:

$$T\,(t+\Delta t) \approx T\,(t) + \Delta t(dT/dt) \quad (5)$$

Time steps were limited to stability constraints which were based on the requirement that the coefficient of T in the full algebraic expression on the right hand side of equation 5 be non-negative. The full algebraic expression is obtained by substituting the expression for dT/dt in appropriate equations above. The convective film coefficients were calculated from standard heat transfer coefficients:

$$h = (k/L)\ C\ (\text{Rayleigh number})^{\,m} \quad (6)$$

C and m were obtained from heat transfer handbooks.

RESULTS OF GLASS PANE MODEL

Glass pane model was validated using experimental data from direct gain test cells at the National Center for Appropriate Technology (NCAT) located in Butte, Montana. The model was validated for a eight day period beginning November 14 and ending November 21. Figure-3 illustrates the measured and simulated test cell air temperatures along with the measured insolation and ambient temperature for the glass model. The mean deviation of the simulated temperatures from the measured temperatures was found to be -11^{o} C and the maximum deviation was 5.4^{o}C.

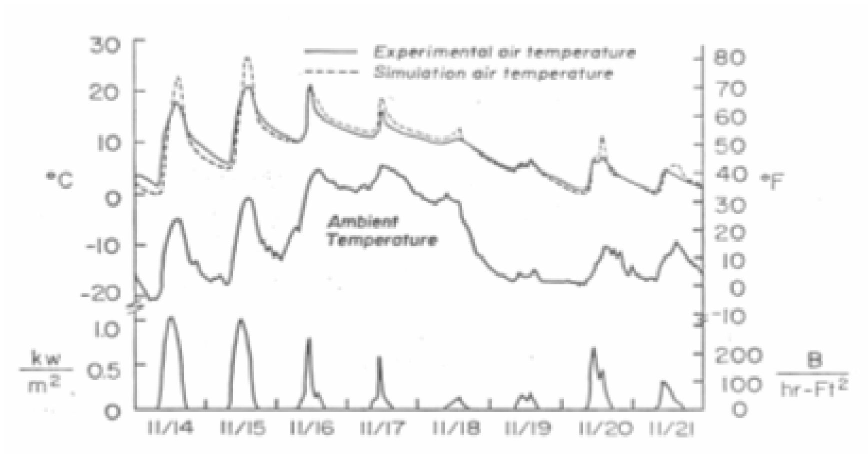

Figure-3. Measured and simulated test cell air temperatures.

TROMBE WALL MODEL

In Trombe' wall model only plane solid walls were considered. Heat transfer through the wall was by conduction with no convective mixing. The entire inside surface material of the room (excluding the storage wall) was treated as a lumped capacity body. Figure-4 shows Trombe wall model.

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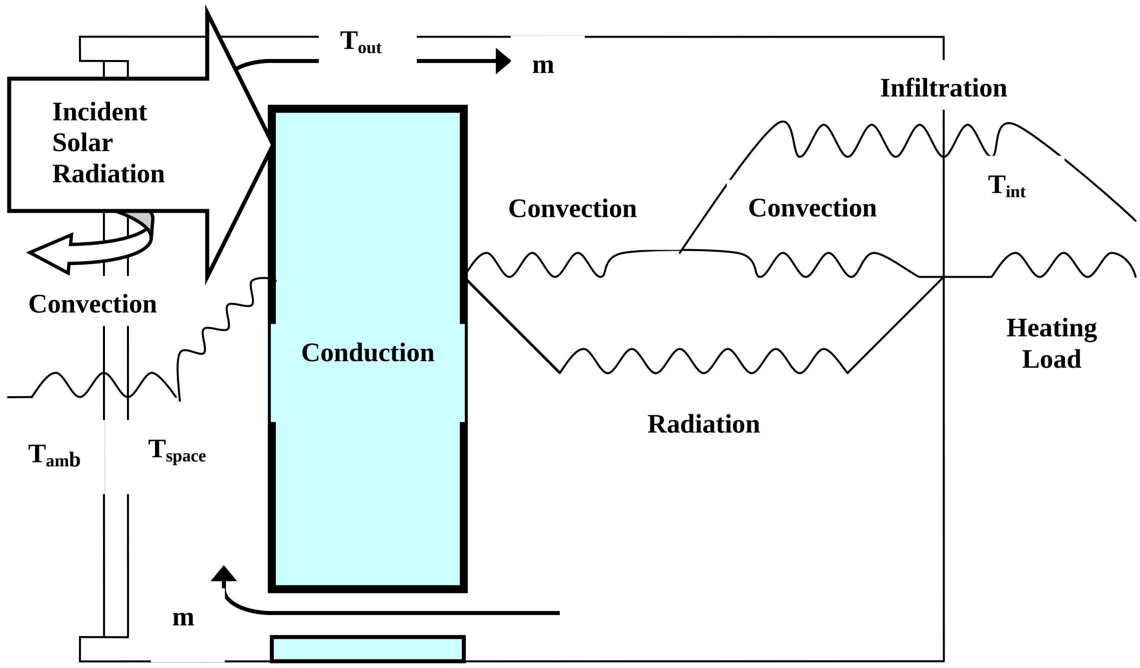

Figure-4. Trombe wall model.

$$Q_{ri} + (UA)_c (T_r - T_1) + (UA)_1 (T - T_1) = (mc)_1 \, dT_i/dt \quad (7)$$

Second and third terms on the left hand side provide convective heating by the room air and conductive heating by the ambient air respectively. Radiative heating by the storage wall is:

$$Q_{ri} = A_T \, б \, (\theta_b^4 - T_1^4) / [1/\varepsilon_T + (A_T/A_1)(1-\varepsilon_1)/\varepsilon_1] \quad (8)$$

A heat balance for the living space room air yielded:

$$h_T A_T (\theta_b - T_r) + (UA)_c (T_1 - T_r) + (mc)_r (T - T_r) + r(mc)_r (T_0 - T_r) = (mc)_r \, dT_r/dt \quad (9)$$

Where the last term on the left hand side is heating supplied by air flow from the illuminated space. The average air temperature in that space was calculated as:

$$T_s = \frac{2\, mcT_r + (UA)_w T + h_T A_T \theta_0}{2\, mc + (UA)_w + h_T A_T} \quad (10)$$

Mass flow rate was calculated from a relation reported by Balcomb, 1988. Conduction heat transfer through Trombe wall was numerically solved by an explicit finite difference model:

$$\theta_x (t + \Delta t) = \lambda \, \theta_{x - \Delta x} (t) = (1 - 2\lambda)\, \theta_x (t) + \lambda \, \theta_{x + \Delta x} (t) \quad (11)$$

Where $\lambda = \alpha \, \Delta t / \Delta x^2$. This is the discrete expression of the temperature at each time step as a polynomial in λ variable. Naturally, equation (11) was modified appropriately at the boundaries given by x = 0 and x = b. Equations (7) and (9) were numerically integrated as described in the glass pane model.

RESULTS OF TROMBE WALL MODEL

The Trombe wall model was also validated using experimental data from indirect gain test cells at NCAT. These test cells are basically the same as those in Los Alamos, New Mexico. The model was validated for a six day period beginning November 19 and ending November 24. Figure-5 illustrates the measured and simulated test cell air temperatures along with the measured insolation and ambient temperature for the test period. The mean deviation of the simulated temperatures from the measured temperatures was found to be -1.1^o C and the maximum deviation was 7.2^o C.

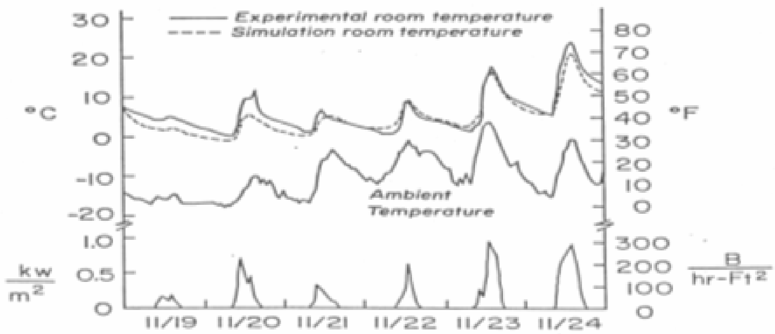

Figure-5. Measured and simulated test cell air temperature.

CONCLUSIONS

Comparison of glass pane and Trombe wall models with experimental data indicates that the models are good prediction of actual performance. This preliminary comparison of passive solar systems has shown that natural passive systems can provide a portion of space heating for a well insulated home in colder climate. The results have also shown that insolation patterns have a definite effect on the performance of the glass pane and Trombe wall systems.

Nomenclature

Q_{si}	total absorbed insolation, absorbed by area A_i
Q_{ri}	long wave solar energy absorbed by area A_i
h_i	convective film coefficient of area A_i
T_i	temperature of area A_i
T_a	interior air temperature
mc	product of mass and specific heat
R_i	thermal resistance for conduction to ambient air
T	ambient air temperature
t	time
r	air infiltration rate, air change/time
J_i	radiosities
б	Stefan Botzmann constant
ε_i	long wave emissivity of area A
F_{ij}	diffuse radiative view factor of area A_j as seen from area A_i
k	air conductivity
L	Length
U	heat transfer coefficient
A	area of the wall normal to the direction of heat transfer
T_r	room temperature
T_1	ambient temperatures
θ_x	Trombe wall temperature at location x
T_s	average air temperature

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