NUMERICAL STUDY OF A MODIFIED TROMBE WALL SOLAR COLLECTOR SYSTEM

by

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The paper presents numerical analysis of efficiency of the modified Trombe wall with forced convection. The analyzed system comprises a double glass glazing, and a massive wall with opening and central channel in it. In order to increase the efficiency, a fan is provided at the bottom vent of the wall. It is more advanced as compared with simple Trombe solar wall with a relatively low thermal resistance, which is taken as a reference in experimental analysis. The mathematical model, composed for the massive solar wall efficiency, is usually very complicated and assessment of the thermal behaviour requires the use of thermal simulation techniques. This paper presents steady-state and one-dimensional mathematical model for simplified analysis of thermal efficiency of modified Trombe solar wall. The results from presented model were analyzed to predict the effects of variations in the operational parameters on the solar wall efficiency: solar radiation intensity, air velocity in the entrance duct, and room air temperature. The results have been compared with the available experimental study, and the comparison has shown satisfactory agreement. The obtained results have be used for simple and fast running design tools that designers can use in the early phases of the design process for approximate calculations of efficiency of the passive solar heating systems.

Key words: *modified Trombe wall, solar air heating, energy efficiency*

Introduction

The passive solar systems is well-know method for use of solar energy as sources of heating in buildings. In practice buildings with passive solar system are still rare because there is the lack of available information for the efficiency of passive solar systems. The analized heating system is the modified Trombe wall system that consists of a transparent cover (two glasses) and the massive wall with central channel in it (fig. 1). At the bottom and the top of the massive wall, there are vents for allowing an air circulation between entrance duct (air gap between transparent cover and wall), central channel and room space. At the bottom vent there is a fan so the air circulates through wall by forced convection. In order to approve that analyzed type of the wall has better temperature effects in relation to the classic Trombe wall the experimental testing have been done.

Subject to the position of the valve, the modified Trombe wall ensures the following issues: A – room heating by hot air circulation from the entrance zone into the adjacent room; B – heat storage, partly by the direct absorption and conduction, and partly by hot air circulation from the entrance duct into the channel space; C – room heating and heat storage by simulta-

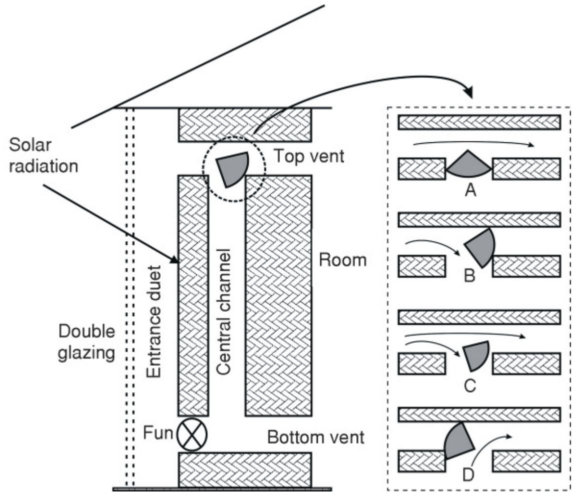

Figure 1. Schematic representation of the modified Trombe wall

neous air circulation from the entrance duct into the room and the channel space; D – room heating when the wall is not exposed to the solar radiation activity, the room being heated by both radiation and natural convection from the inside wall, and hot air circulation from the central channel.

The literature provides abundant information on numerous analyses of different types of solar wall. Smolec *et al*. [1] described two-dimensional model for calculating heat transfer in a Trombe wall and its experimental verification. Bhandari *et al*. [2] have introduced the concept of solar heat gain factor for calculating the net energy gain of passive heating elements and other components of a building due to incident solar radiation. Zrikem *et al.* [3] studied theoretically the transfer of solar radiation in a composite Trombe-Michel wall collector system. In an experimental investigation, Akbarzadeh *at el.* [4] used flow visualizations studies, which have given a deeper insight into the fundamental flow mechanisms of the Trombe wall passive test cell. Yedder *et al*. [5] have conducted a numerical study of the thermal performance of a classic Trombe wall solar collector system including solar collector and the adjacent room. Shen *et al.* [6] developed a simulation model for numerical study of classical and composite solar wall by software TRNSYS, using by the finite difference method and experimental results from recorded during several years. Skrivastava *et al*. [7] analyzed south facing wall consisting of a mass of concrete/brick the surface of which is blackened and glazed, with a network of pipes from which heat can be extracted by fluid flow in such a manner as to ensure constant retrieval of the plane temperature. Smolec *et al.* [8] studied the limitations of the various models used for the prediction of heat transfer in the Trombe wall.

In this paper, modified Trombe wall solar collector system is studied numerically. The study presents the effect of varying air velocity in entrance duct, solar radiation and room temperature on the efficiency of the heating system in such a manner as to provide changes in any of these properties as well as the improvement of the wall efficiency. The results have been compared with the available experimental study. The presented method should require minimum data input and give reliable indication of behaviour of analyzed solar wall.

The system description

The heat transfer modes and heat transfer exchanges in the system are presented in fig. 2. The external transparent cover transmits solar radiation in, but holds back heat. External wall surface is painted black and act as absorber of solar radiation. Aimed at the heat loss decrease, the appropriate solution of the solar wall is based on application of a special absorber and thermo-insulation coating over the outer wall surface. Analyzed solar wall is divided and the external part, acting as a partition, and the internal one, acting as a massive wall. The energy input to the system is the solar radiation received on the absorber surface. The air in the entrance duct is heated in contact with the external wall surface, rises and circulates towards the central channel of the wall and the room (when the vents are open). The net losses from the system are due to convection and radiation losses from the absorber surface. The wall is cooled by forced convection of the air in the gap between wall and glazing (entrance duct). At the top and bottom of the wall, vents are provided for thermo-circulation of air with constant heat flux into the adjacent room. Cooler room air, drawn through the bottom vent, is heated at it passes up the entrance duct and then delivered into the room through the top vent. This convective heat tides, where the winter sun is lower in the sky and thus falls more directly on the wall. The change in sun's trajectory during the year also enables the wall to be shaded, using overhangs or eaves, during the summer when the sun's altitude is greater, and heating is not required.

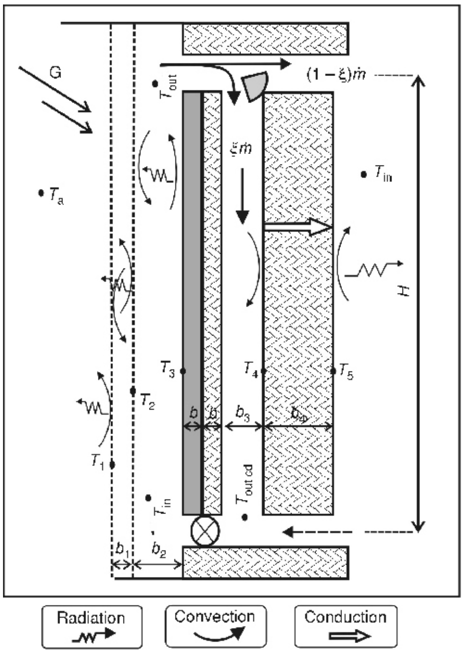

Figure 2. Definition of heat transfer modes and parameters of the wall

The geometrical, thermo-physical and operational parameters of the system are:

(1) *Double glass glazing unit*: $b_1 = 2$ cm, $b_2 = 10$ cm, $\varepsilon_{1\,2} = 0.95$, $\tau = 0.85$;

(2) *Solar wall*: $\rho = 2400$ kg/m^3, $H = 2.7$ m, $Y = 3$ m, $k_w = 0.9$ W/mK, $b = b_3 = 5$ cm, $b_4 = 10$ cm, $\varepsilon_3 = 0.95$, $\alpha = 0.95$;

(3) Outside conditions: $t_a = 0$ °C, $t_r = 20$ °C, $w = 0$ ms.

Mathematical model

All three mechanisms of heat transfer (*i. e.*, conduction, convection, and radiation) occur simultaneously during heat transfer in the solar wall. In this investigation, the following simplified assumptions have been made: the model is steady-state and heat transfer through the system is simplified to one-dimensional one by assuming the different layers of wall construction to be at uniform temperature at any given time, thermo physical properties of air and all materials involved constant and independent of temperature, the resistance of conduction heat exchange through glazing is neglected, air is considered as a non-participating medium in radiation heat exchange, the entire system is wall insulated so there are no heat lateral losses, two horizontal boundaries are adiabatic, and the temperature of the adjacent room is constant. As in all steady-state methods, the role of the storage capacity of the massive wall was not considered. According to these assumptions the mathematical model of the wall is developed.

The solar radiation transmitted through the glazing and transformed into heat by the absorber is:

$$S = \tau \alpha G \tag{1}$$

Energy losses from both glazing at temperatures T_1 and T_2 and from the external surface of the wall at temperature T_3 are:

$$q \quad h_w (T_1 - T_a) \quad F_{\varepsilon 1} \sigma (T_1^4 \quad T_a^4) \tag{2}$$

$$q \quad h_{1\,2} (T_2 - T_1) \quad F_{\varepsilon 2} \sigma (T_2^4 \quad T_1^4) \tag{3}$$

$$q \quad h_{2\,3}(T_3 - T_2) \quad F_{\varepsilon 3}\sigma(T_3^4 \quad T_2^4) \tag{4}$$

When heat effects of the sun radiation activity are considered over a sufficiently long period of time, an essential equation form is used for the obtained useful energy:

$$q_{\mathrm{f}} = S - q \tag{5}$$

The energy balances for the airflow in an entrance duct is given as:

$$q_{\mathrm{f}} \quad \frac{\rho V c_{\mathrm{p}}(T_{\mathrm{out}} \quad T_{\mathrm{in}})}{A_{\mathrm{w}}} \tag{6}$$

$$q_{\mathrm{f}} = h_{2\,3}(T_2 + T_3 - T_{\mathrm{out}} - T_{\mathrm{in}}) \tag{7}$$

Volume flow rate of air is given as $V = C_{\mathrm{d}}A_{\mathrm{v}}[2gH\beta(\mathrm{T}_{\mathrm{out}} - \mathrm{T}_{\mathrm{in}})]^{1/2}$ and discharge coefficient of two rows of vents in series with similar cross-section areas is $C_{\mathrm{d}} = 0.57$ [4].

Air temperature at the upper vent is calculated from [2]:

$$T_{\mathrm{out}} \quad T_{\mathrm{in}}\exp(\ \theta H) \quad T_3[1 \quad \exp(\ \theta H)], \quad \theta \quad \frac{h_{23}b_2}{\dot{m}c_{\mathrm{p}}} \tag{8}$$

Mass airflow through the central channel is $\xi\dot{m}\,(0 < \xi < 1)$, and the energy balances for the air is given as:

$$q_{\mathrm{f1}} \quad \frac{\xi\dot{m}c_{\mathrm{p}}(T_{\mathrm{out}} \quad T_{\mathrm{out\ cd}})}{A_{\mathrm{w}}} \tag{9}$$

$$q_{\mathrm{f1}} \quad h_{\mathrm{f4}}\ \frac{T_{\mathrm{out}} \quad T_{\mathrm{out\ cd}}}{2} \quad T_4 \tag{10}$$

Heat flow into the internal part of the wall (thickness b_5) is transferred into the room air by conduction, and subsequently by convection and radiation (see fig. 2):

$$\frac{k_{\mathrm{w}}}{b_4}(T_4 \quad T_5) \quad h_{5\,\mathrm{r}}(T_5 \quad T_{\mathrm{r}}) \quad F_{\varepsilon 5}\sigma(T_5^4 \quad T_{\mathrm{r}}^4) \tag{11}$$

The efficiency of the presented heating concepts, by solar wall system, can be defined as the relation of the useful heat delivered by the solar wall to the total energy input as follows:

$$\eta \quad \frac{q_{\mathrm{f}} \quad q_{\mathrm{f1}} \quad h_{5\,\mathrm{r}}(T_5 \quad T_{\mathrm{r}}) \quad F_{\varepsilon 5}\sigma(T_5^4 \quad T_{\mathrm{r}}^4)}{S \quad q_{\mathrm{el}}} \tag{12}$$

Electrical energy can be found if we know the kinetic energy of the entrance air and useful fan effectiveness $q_{\mathrm{el}} = v^3\rho b_2/H\eta_{\mathrm{f}}$ [9], where η_{f} is degree of the fan useful effect (we used $\eta_{\mathrm{f}} = 0.95$).

Average convection coefficients ($h_{\mathrm{i\,j}}$) used in previous equations are used from literature references. The average convection coefficient due to wind on glazing from McAdams expression is [4]:

$$h_{\mathrm{w}} = 5.7 + 3.8w \tag{13}$$

where w stands for uniform wind velocity.

The average convection coefficient between glazings was calculated from [10]:

$$\mathrm{Nu}_{1\,2} \quad 0.062\mathrm{Gr}^{0.327} \qquad h_{1\,2} \quad 0.82\frac{(T_2 \quad T_1)^{0.327}}{b_1^{0.019}}\ 1 \quad 0.018\ \frac{T_1 \quad T_2}{2} \quad 283 \tag{14}$$

Equation (14) holds in the range $1.5 \cdot 10^5 < \mathrm{Gr} < 10^7$.

The average convective coefficient in the entrance duct between the wall and the glazing for forced airflow over a vertical plane surface is [4]:

$$h_{2\,3} = 5.68 + 4.1v \tag{15}$$

where v stands for uniform air velocity in the entrance duct.

The average convective coefficient in central channel between air and the internal part of the wall is [1]:

$$\mathrm{Nu}_{\mathrm{uf}\,4} \quad 0.0155\,\mathrm{Re}^{0.83}\,\mathrm{Pr}^{0.5} \qquad h_{\mathrm{f}\,4} \quad 6.1v^{0.83}\,\frac{b_2^{0.83}}{b_3} \tag{16}$$

The average convective coefficient from the wall surface to the room air is:

$$\mathrm{Nu}_{5\,\mathrm{r}} \quad 0.15(\mathrm{Gr\,Pr})^{0.33} \qquad h_{5\,\mathrm{r}} \quad 1.124\,\frac{T_5 \quad T_\mathrm{r}}{T_\mathrm{r}}^{0.33} \tag{17}$$

Equation (17) holds in the range $\mathrm{GrPr} > 10^9$.

For two parallel surfaces as various parts of the system, emission factors are calculated from the following equations:

– for exterior glazing and ambient air: $F_{\varepsilon 1} = \varepsilon_{1\,2}$
– for exterior and interior glazing: $F_{\varepsilon 2} = 1/[(2/\varepsilon_{1\,2}) - 1]$
– for interior glazing and massive wall: $F_{\varepsilon 3} = 1/[(1/\varepsilon_{1\,2}) + (1/\varepsilon_3) - 1]$
– for wall surface and room air: $F_{\varepsilon 5} = \varepsilon_3$

System (1)-(12) is one-dimensional steady-state mathematical model of the modified Trombe wall. These equations accompanied by properly selected correlation equation for convective and radiative heat transfer coefficients have to be solved using iteration numerical technique to calculate unknown values. The input parameters considered for the analysis are: solar radiation G, air velocity in the entrance duct v, air inlet temperature in the entrance duct t_{in}. The output parameters of the model are: S, q, q_f, q_{f1}, T_1, T_2, T_3, T_4, T_5, T_{out}, $T_{\mathrm{out\,cd}}$, and η.

Experimental results

In order to analyze thermal characteristics of the presented concept of modified Trombe wall, experimental research was performed with simple models of classic vent-free Trombe wall and modified Trombe wall (fig. 3) provided with vents for the flow of hot air from the entrance zone into the central channel [9]. Testing models with equal collecting surfaces and transmittance-absorptance characteristics are placed vertically and south-oriented. Rear surfaces are open to the outside environment.

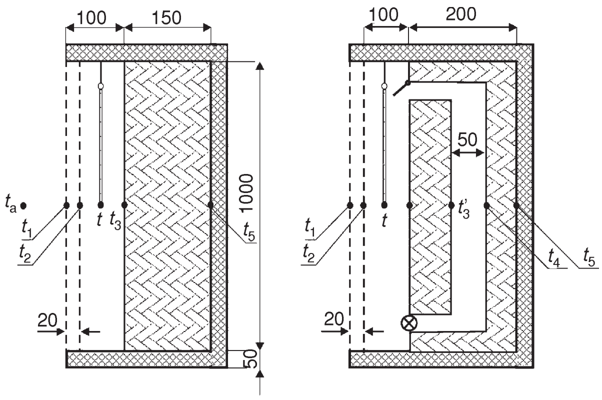

Figure 3. Scheme of experimental models of classic Trombe wall and modified Trombe wall with location of temperature measure points: t_a – ambient, t_1, t_2 – glazing glasses, t – air in the entrance zone, t_3 – absorber, t'_3, t_4, t_5 – inside wall with radial fan and central channel

The walls are constructed of concrete with absorbing surfaces painted with matt black paint. The external wall surface is shielded from the environment by two panes of 3 mm thick commercial glass. The walls are thermo-insulated from the lower side and all wall sides (except the front and rear ones) along with glass girders are covered with a mineral 50 mm thick wall of thermal conductivity 0.037 W/mK, and specific density 27 kgm. All surfaces of both models with installed thermal insulation are covered with tin housing.

Comparative testing of both massive walls included measuring of distinctive temperatures (fig. 4) at both the time when the radial fan (20 W, 1450 rpm) at modified Trombe wall was running (by day – in the sunshine) and when it was not running (when the absorbing wall surface was not exposed to sun rays – by day and at night). The moveable key was in the open position during the fan running. Temperature of ambient air and temperature of air in entrance zone were measured using mercurial thermometer. The thermometers were calibrated over the range 0-100 °C (class of accuracy 0.2 °C). All temperatures of glass and wall were measured with digital apparatus MDL-910, with temperature range from –5 to 70 °C (class of accuracy 0.1 °C). Figure 4 show measured temperatures at both types of solar wall for three of many days of testing.

Figure 4 shows comparatively characteristic temperatures at both types of walls. The experimental results infer that the temperature in the mid-zone of transparent and absorbing wall surface of the composite wall is higher than the one of the classic Trombe wall in the morning, and lower in the afternoon. The temperature of the rear surface of the composite wall during the day (in the sunshine) is slightly higher than the one of the classic wall.

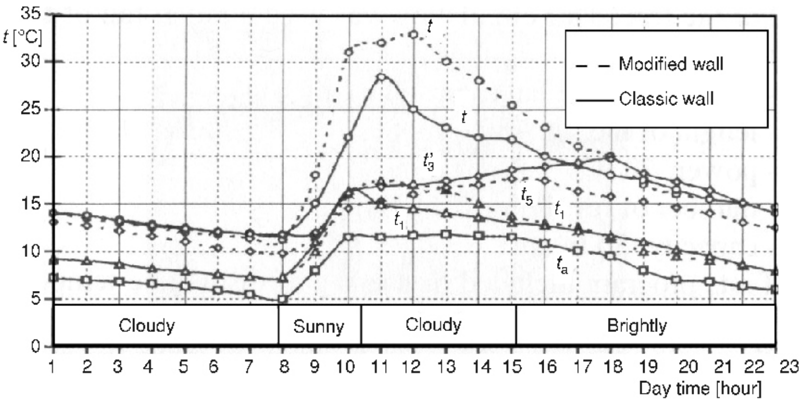

Figure 4. Comparative measured temperatures for both type of wall for Janury 2

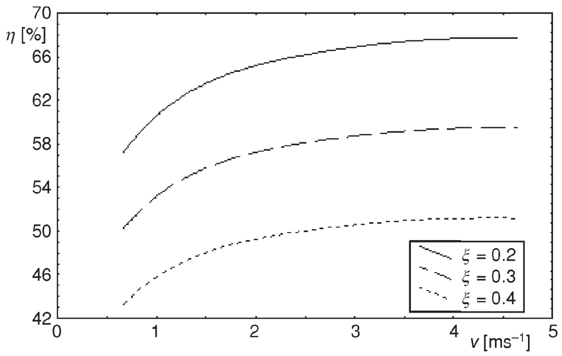

Figure 5. Variation of efficiency of the heating system with air velocity in the entrance duct

Results and discussion

The numerical analysis was carried out using values of the geometrical and thermo-physical parameters of the system given in Section *The system description*. Series of numerical calculations have been done for the evaluation of the efficiency of the analyzed heating system.

Results obtained at various air velocities in the entrance duct, *i. e.* three mass airflow coefficients through central channel for G = 1000 W/m^2 and t_{in} = 26 °C, are shown in fig. 5. The correlation between the efficiency of the system and air velocity exhibit the same trend for all mass airflow coefficients. When air velocity increases at lower airflow coefficient (which means greater mass flow of heat air into the room), the efficiency of the system is notably enhanced. For example, at 2 m/s air velocity in the entrance duct 0.3 mass airflow coefficient gives 57.39% system efficiency, whereas 0.2 mass airflow coefficient gives 65.29%. The system efficiency is the highest when air velocity attains 4 m/s.

Figure 6 show the variations of efficiency of the system as a function of the solar radiation for both different inlet air temperatures in the entrance duct and mass airflow coefficients through central channel, for v = 1 m/s. As system efficiency of the system is the function of the solar radiation, which occurs on the vertical surface of the solar wall, the higher the solar radiation the higher efficiency of the system. It should be noted that for the given solar radiation, the efficiency of the heating system is higher at lower inlet air temperature in the entrance duct. The entrance duct is the part of the massive wall where heat losses are very intensive. Therefore, if inlet air temperature in the entrance duct is lower, heat losses are lower while efficiency is higher. For example, when temperature in the entrance duct decreases from 28 to 24 °C energy efficiency increases for about 18%. For example, at 700 W/m^2 solar radiation and 26 °C inlet air temperature in the entrance duct mass airflow coefficients 0.2, 0.3, and 0.4 result in 55.28, 48.58, and 41.98% of system efficiency, respectively.

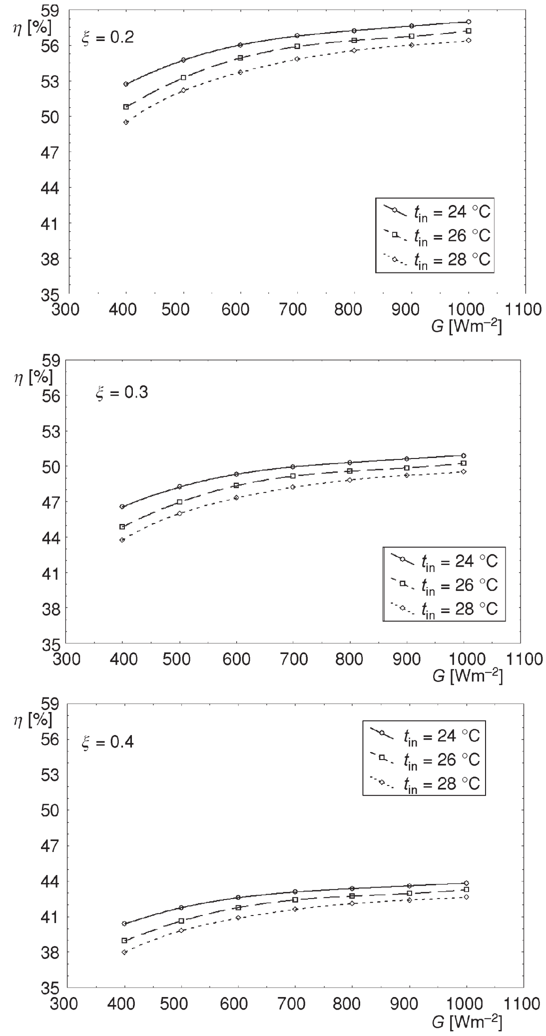

Figure 6. Variation of efficiency of the heating system with solar radiation

For practical purposes, energy efficiency of the presented system, expressed as a function of the solar radiation, can be calculated by the function given in tab. 1.

The effect of the air inlet temperature in the entrance duct on the system efficiency, for three different mass airflow coefficients through the central channel for G = 1000 W/m^2, is illustrated in fig. 7. The results indicate that system efficiency decreases with the increase of inlet air temperature in the entrance duct. The increase of inlet air temperature in the entrance duct for about 17% brings about decrease in the efficiency of the heating system for about 4%. For a given air inlet temperature in the entrance duct, the efficiency of the heating system is higher for lower mass airflow coefficients through central channel. Practi-

Table 1. Efficiency of the air heating system in the function of solar radiation

$\eta \quad \frac{P_1 G}{P_2 \quad G}$ [%]						
t_{in} [°C]	$\xi = 0.2$		$\xi = 0.3$		$\xi = 0.4$	
	P_1	P_2	P_1	P_2	P_1	P_2
24	62.21	69.65	54.37	64.87	46.53	58.66
26	62.38	89.87	54.46	84.65	46.64	77.69
28	62.43	100.93	54.52	95.07	46.73	87.78

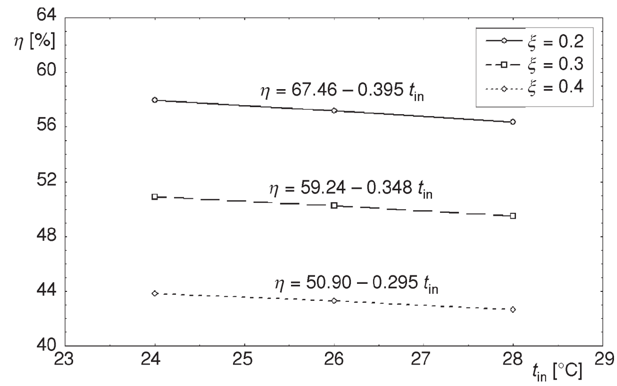

Figure 7. Variation of efficiency of the heating system with air inlet temperature in the entrance duct

cally, efficiency of the system is a linear function of inlet temperature in the entrance duct, and it can be calculated by functions given in fig. 7.

Conclusions

This study presents a simple method for the evaluation of energy efficiency of the modified Trombe solar wall. The effect of varying air velocity in entrance duct, solar radiation, and ambient temperature on efficiency of the heating system are presented. The numerical results have indicated that the efficiency of the heating system by solar wall is enhanced with the increase in both air velocity in the entrance duct and solar radiation. Efficiency of the system decreases with the rise in air inlet temperature in the entrance duct. At constant solar radiation, with increasing air velocity in the entrance duct, solar wall efficiency increases. This simplified method enables users to evaluate the efficiency of the heating system of a number of options and to make comparison and predictions of thermal behavior under operating conditions.

Nomenclature

A_v – vent cross-section area, [m^2]
A_w – massive wall area, [m^2]
b – thickness of insolation on the left surfaces, external wall thicknesses, [m]
b_1 – distance between glazing, [m]
b_2 – distance between glazing and massive wall, [m]
b_3 – central channel thickness, [m]
b_4 – internal wall thickness, [m]
c_p – specific heat of air at constant pressure, – [Jkg^{-1}K^{-1}]
$F_{e\,ij}$ – emission factor between surfaces i and j, [–]
G – solar radiation incident on vertical surface, [Wm^{-2}]
Gr – Grashoff number [= $g\beta(T_2 - T_1)b_1^3/\nu^2$] [–]
g – gravity acceleration, [ms^{-2}]
H – wall height, [m]
h_{12} – average convection coefficient between i and j, [Wm^{-1}K^{-1}]
h_{f4} – average convection coefficient in central channel between air and the wall, [Wm^{-1}K^{-1}]
h_{5r} – average convection coefficient from the wall to the air in the room, [Wm^{-1}K^{-1}]

h_w – average wind convection coefficient, [$Wm^{-1}K^{-1}$]
k_w – thermal conductivity of the wall, [$Wm^{-1}K^{-1}$]
L – characteristic length (= $2b_2$), [–]
L_1 – characteristic length (= $2b_3$), [–]
m – mass flow rate, [m^3s^{-1}]
Nu_{12} – local Nusselt number (= $h_{12}L/k$), [–]
Nu_{f4} – local Nusselt number (= $h_{f4}L/k$), [–]
Nu_{5r} – local Nusselt number (= $h_{5r}L/k$), [–]
Pr – Prandtl number (= 0.7), [–]
q – heat losses, [Wm^{-2}]
q_f – useful heat energy in the entrance duct, [Wm^{-2}]
q_{f1} – useful heat energy in the central channel, [Wm^{-2}]
q_{el} – fan electric energy, [Wm^{-2}]
Re – Reynolds number (= vL/ν), [–]
S – solar radiation transmitted through the glazing, [Wm^{-2}]
T_a – ambient temperature, [K]
T_{in} – inlet air temperature in the entrance duct, [K]
T_{out} – outlet air temperature from the entrance zone, [K]
$T_{out\,cd}$ – outlet air temperature from the central channel, [K]
T_r – room air temperature, [K]
T_1, T_2 – average temperatures of glazing, [K]
T_3 – average temperature of absorbing surface of the wall, [K]
T_4, T_5 – interior and exterior average temperature of inner part of the wall, [K]
v – air velocity in the entrance duct, [ms^{-1}]
w – wind velocity, [ms^{-1}]

Greek letters

α – absorptivity of the wall surface, [–]
β – volume coefficient of expansion of air, [K^{-1}]
ε_{12} – emissivity of glazing, [–]
ε_3 – emissivity of wall, [–]
η – efficiency of the heating system, [–]
η_f – degree of the fan useful effect, [–]
ν – kinematic viscosity of the air, [m^2s^{-1}]
ξ – mass airflow coefficient through the central channel, [–]
ρ – air density, [kgm^{-3}]
σ – Stefan-Boltzmann constant, [$Wm^{-2}K^{-4}$]
τ – glass transmittance, [–]

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