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A Study of Heat Transfer in a Composite Wall Collector System with Porous Absorber

Wei Chen
Associate Professor Ph.D.
School of Merchant Marine, Shanghai Maritime University
200135, P.R. China
Email: weichen96@sina.com

Abstract: In this paper, heat transfer and flow in a composite solar wall with porous absorber has been studied. The unsteady numerical simulation is employed to analyze the performance of the flow and temperature field in the composite solar wall. The excess heat is stored in the porous absorber and wall by the incident solar radiation and there is a temperature gradient in the porous layer. Therefore, the porous absorber works as thermal insulator in a degree when no solar shining is available. The influence of the porosity within the porous absorber on the air flow in the porous absorber is significant. The results show that all these factors should be taken into account for a better design of a heating system.

Key words: solar energy; porous medial; simulation

1. INTRODUCTION

Passive solar heating is often applicable in cold climates. The storage wall has been extensive use since the works of Trombe [1] were published. The standard Trombe wall has the drawback of low thermal resistance, which leads to significant losses at night-time or during periods with no sun. Furthermore, part of the heat supply cannot be controlled (risk of overheating). A composite solar wall enables these drawbacks to be overcome.

In this study, a new kind of composite wall with convective porous absorber shown schematically in Fig.1 has been investigated. The numerical simulation was carried out to analyze the variation of the flow and temperature field in the composite solar wall. Major objective of the present study focuses on the selecting efficient strategies for the composite solar wall system of the heating building.

2. SYSTEM DESCRIPTION AND MATHEMATICAL ANALYSIS

2.1 System Description

A composite wall collector with porous absorber under investigation is shown schematically in Fig.1. The composite wall collector, locates on the south side of the heating room, consists of a glazing, a massive thermal storage wall and a porous-layer that is used between the glazing and the massive thermal storage wall. The porous layer is heated by solar radiation, a part of which is absorber by the porous layer, the rest transmitted through the porous layer received by the air of the heating room and the massive thermal storage wall. The massive thermal storage wall has a top and a bottom vent to facilitate the convection between the porous-layer and the air of the heating room.

2.2 Theoretical Modeling

The flow is assumed to be laminar and two-dimensional. The Boussinesq approximation is used to account for the density variation. While the flow in the composite wall except the porous absorber

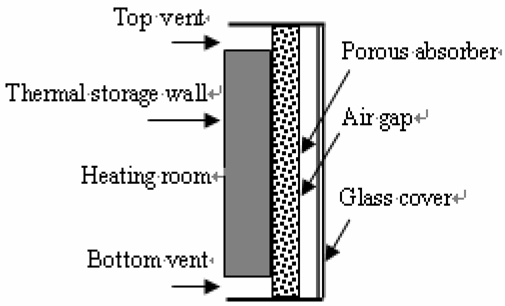

Fig1. Schematic of composite solar wall collector with porous

Navier-Stokes equations, flow in the porous absorber is governed by Brinkman ForchHeimer

Extended Darcy model [2]. The mathematical model is described below.

For the flow in the composite wall (except the convective porous layer), the governing equations can be written as:

Continuity equations

$$\frac{\partial(\rho u)}{\partial x}+\frac{\partial(\rho v)}{\partial y}=0 \qquad (1)$$

Momentum equations

$$\frac{\partial(\rho u)}{\partial \tau}+\frac{\partial(\rho uu)}{\partial x}+\frac{\partial(\rho vu)}{\partial y}=-\frac{\partial p}{\partial x}+\frac{\partial}{\partial x}(\mu\frac{\partial u}{\partial x})+\frac{\partial}{\partial y}(\mu\frac{\partial u}{\partial y}) \qquad (2)$$

$$\frac{\partial(\rho v)}{\partial \tau}+\frac{\partial(\rho uv)}{\partial x}+\frac{\partial(\rho vv)}{\partial y}=-\frac{\partial p}{\partial y}+\frac{\partial}{\partial x}(\mu\frac{\partial v}{\partial x})+\frac{\partial}{\partial y}(\mu\frac{\partial v}{\partial y})+\rho g\beta(T-T_C) \qquad (3)$$

Energy equation

$$\rho c\frac{\partial T}{\partial \tau}+\rho c(u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y})=\frac{\partial}{\partial x}(k\frac{\partial T}{\partial x})+\frac{\partial}{\partial y}(k\frac{\partial T}{\partial y}) \qquad (4)$$

For flow and heat transfer in the convective porous layer, the governing equations can be written as [2]:

Continuity equation

$$\frac{\partial(\rho u_d)}{\partial x}+\frac{\partial(\rho v_d)}{\partial y}=0 \qquad (5)$$

Momentum equations

$$\frac{\rho}{\theta}\frac{\partial u_d}{\partial \tau}+\frac{\rho}{\theta}(u_d\frac{\partial u_d}{\partial x}+v_d\frac{\partial u_d}{\partial y})=-\frac{\partial p}{\partial x}+\frac{\partial}{\partial x}(\mu_{eff}\frac{\partial u_d}{\partial x})+\frac{\partial}{\partial y}(\mu_{eff}\frac{\partial u_d}{\partial y})-\frac{\mu u_d}{K}+\frac{\rho C\theta}{\sqrt{K}}|\vec{v}_d|u_d \qquad (6)$$

$$\frac{\rho}{\theta}\frac{\partial v_d}{\partial \tau}+\frac{\rho}{\theta}(u_d\frac{\partial v_d}{\partial x}+v_d\frac{\partial v_d}{\partial y})=-\frac{\partial p}{\partial x}+\frac{\partial}{\partial x}(\mu_{eff}\frac{\partial v_d}{\partial x})+\frac{\partial}{\partial y}(\mu_{eff}\frac{\partial v_d}{\partial y})-\frac{\mu v_d}{K}+\frac{\rho C\theta}{\sqrt{K}}|\vec{v}_d|v_d+\rho g\beta(T-T_c) \qquad (7)$$

Energy equation

$$(\rho c)_m\frac{\partial T}{\partial \tau}+\rho c(u_d\frac{\partial T}{\partial x}+v_d\frac{\partial T}{\partial y})=\frac{\partial}{\partial x}(k_{eff}\frac{\partial T}{\partial x})+\frac{\partial}{\partial y}(k_{eff}\frac{\partial T}{\partial y}) \qquad (8)$$

where all constants and variables are defined in the nomenclature. Eqs.(5)-(8) form the full set of equations used to model convective flows in porous media. Eqs.(6) or (7) contains the usual balance of forces between viscosity and pressure gradient known as Darcy's law, which is extended by the further inclusion of terms modeling in turn advective inertia, boundary effects, form-drag and Eq.(7) contain buoyancy.The values for the permeability K and the inertia coefficient C in the momentum equations for the porous layer are given by Ergun(1952)[3][4]. For the porous layer of particle diameters d_r and porosity θ

$$K=\frac{d_r^2\theta^3}{175(1-\theta)^2} \qquad (9)$$

$$C=\frac{1.75}{\sqrt{175}}\theta^{-3/2} \qquad (10)$$

In addition, models for the effective properties (μ_{eff} and k_m) of the porous medium are needed. It has been found that taking $\mu_{eff}=\mu_f$ in Brinkman's extension provides good agreement with experimental data (Neale and Nader, 1974)[5] and is adopted in the present work. In present study, the conductivity of the medium k_m and $(\rho c)_m$ is calculated by [6]

$$k_m=\theta k_f+(1-\theta)k_s\ ,\ (\rho c)_m=\theta(\rho c)_f+(1-\theta)(\rho c)_s$$

2.3 Boundary Conditions and Initial Conditions

Numerical simulations were performed for sunny and control based on air temperature and operative temperature was considered. A typical cold day was considered for Shanghai, China, in November with the outdoor temperature and solar irradiance variation [7], given by equations (11) and (12).

$$T_{ao}(\tau)=\overline{T_{ao}}+T_{ar}\cos(\frac{\pi}{12}(\tau-14)) \qquad (11)$$

$$G_{sun}(\tau)=\hat{G}_{sun}\sin(\frac{\tau-a}{b-a}\pi),\quad a<\tau<b \qquad (12)$$

where $\overline{T_{ao}}=$ average outside temperature of 15 ; $T_{ar}=$ amplitude of 6 ; $\hat{G}_{sun}=$ maximum solar irradiance of 400 W/m^2 ; $a=$ sunrise hour of 6 o'clock in the morning; $b=$ sunset hour of 18 o'clock in the afternoon; τ :time ,hours

The fluid in the solar heating system is initially stagnant and at a uniform temperature which is the same as the ambient temperature. In terms of mathematical expressions, the Initial and boundary conditions drawn from Energy-balance equations are given below

For the glass cover of the composite wall:

$$\eta_g G_{sun} + Q_{gsky} + Q_{gh} + Q_{ai} + Q_{ao} = 0 \; ; \; u = 0, v = 0$$

For the porous absorber outside surface

$$x = h_{po} \; , \; K_m A_p \frac{dT_p}{dx} = \delta \eta_p G_{sun} A_p + Q_{ph} + Q_{apo}$$

$$u\Big|_{y=h_{po}^-} = u\Big|_{y=h_{po}^+}, v\Big|_{y=h_{po}^-} = v\Big|_{y=h_{po}^+}, \quad p\Big|_{x=h_{po}^-} = p\Big|_{z=h_{po}^+},$$

$$\mu_m \left(\frac{\partial u_d}{\partial y} + \frac{\partial v_d}{\partial x}\right)\Big|_{x=h_{po}^-} = \mu\left(\frac{\partial u}{\partial y} + \frac{\partial v}{\partial x}\right)\Big|_{x=h_{po}^+}$$

For the thermal storage wall inside surface

$$k \frac{\partial T_{wall}}{\partial x} = h_{wi} A (T_{room} - T_{wall}), u = 0, v = 0$$

For the heating room

$$(\rho c V) \frac{\partial T_{room}}{\partial \tau} = Q_{rin} + Q_{rout} + h_{wi} A (T_{wall} - T_{room}) + K_{wall} A (T_{ROOM} - T_{ao})$$

3. NUMERICAL PROCEDURE.

For the present study, the governing equations (1-10) together with the boundary conditions mentioned above were solved with the SIMPLER method. The computer code based on the mathematical formulation discussed earlier and the SIMPLER method is validated for various cases publish in the literature [8]. The control-volume formulation utilized in this method ensures continuity of the convective and diffusive fluxes as well as overall momentum and energy conservation. The harmonic mean formulation was used to handle abrupt variations in the thermal physical properties, such as the permeability and thermal conductivity, across the interface, for example, at the porous /fluid layer interface. This ensured the continuity of convective and diffusive flues across the interface without requiring the use of an excessively fine grid structure. .

As far as the unsteady-state numerical calculations, the time step was concerned and several values of $\Delta\tau$ had been examined for the grid chosen. For example, It had been found that the maximum deviation between the results using $\Delta\tau$ =30s and $\Delta\tau$ =45s was only 1.5%. Hence, the time step of $\Delta\tau$ =45s together with the grid size of 108×88 were used for the unsteady-state numerical calculations performed in this study.

The porous layer made of Quartzite and the thermal storage wall made of Concrete were chosen in the simulation

Quartzite [9]:

$\rho = 2635 \, kg/m^3$, $c = 0.732 \, kJ/(kg \cdot K)$,

$k_p = 5.17 W/(m \cdot K)$

Concrete [9]:

$\rho = 2243 \, kg/m^3$, $c = 0.837 \, kJ/(kg \cdot K)$,

$k_s = 1.173 W/(m \cdot K)$

The overall composite solar wall dimensions are 0.2m wide, 1.8m high. The width of the porous layer is 0.05m and that of air gap about 0.06m.

4 RESULTS AND DISCUSSION

To analyze the effects of the porous absorber on the temperature distribution and gas flow in the composite solar wall, and to discuss the the influences of the particle size and the porosity within the porous absorber on the heating room temperature, we made the numerical calculation for the present model. The temperature unit in the Isotherm figures is .The results are analyzed as follows:

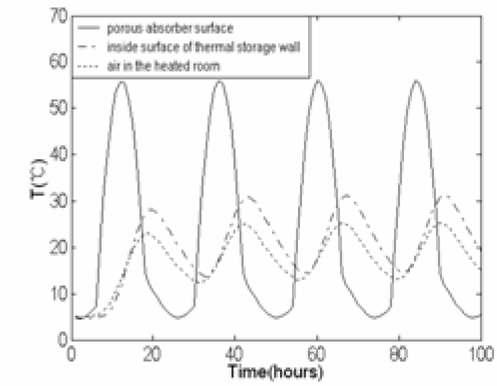

Fig.2. Comparison among predicted values of temperature for porous absorber surface, air in the heating room and south wall surface inside the heating room

Observing Figs.2, we can find that during the heating time, the surface of the porous absorber has higher temperature comparatively and the solar energy absorbed by the porous layer in the heating system was used to increase the temperature of the air in the

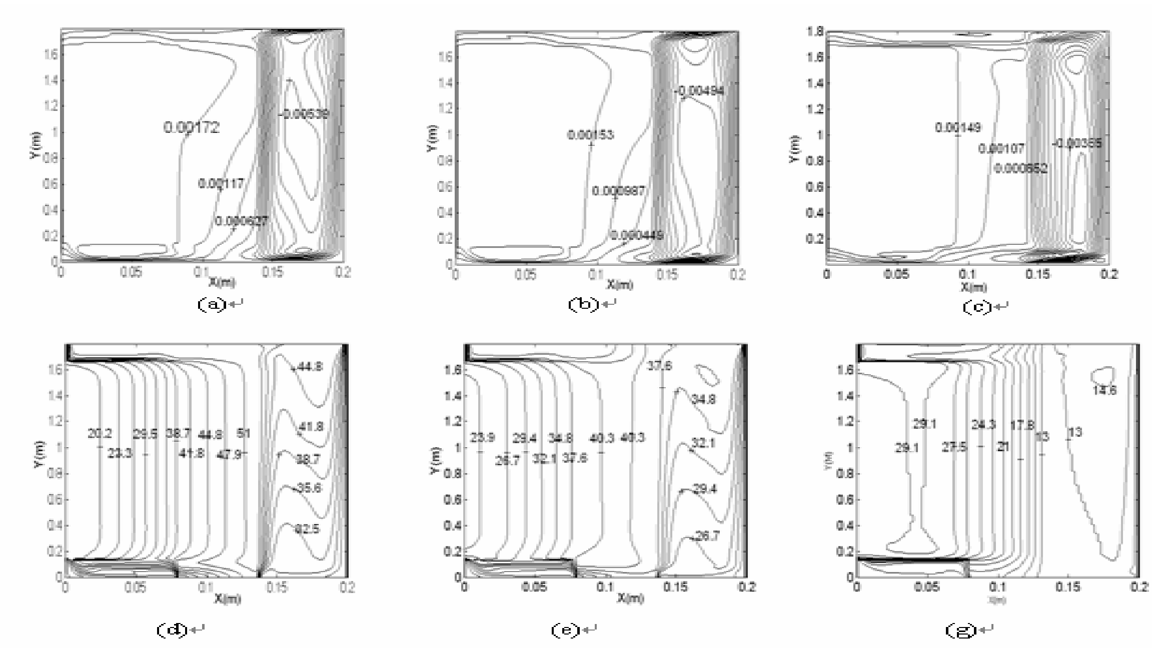

Fig.3. Development of flow and temperature field with time for the composite solar wall with porous absorber porous(d=1cm,θ=0.5) (a) streamlines for 14 hours in a day (b) streamlines for 16 hours in a day (c) streamlines for 19 hours in a day (d) isotherms for 14 hours in a day (e) isotherms for 16 hours in a day (f) isotherms for19 hours in a day

heating room and thermal storage wall, In contrast, when the sun is no shining, the temperature of the thermal storage wall is higher and heat stored in the massive wall releases to raise the air temperature in the heating room.

As shown in Figs.3, the air temperature near the bottom vent inside the composite solar wall is lower, in contrast, the air temperature near the up vent of the heating is higher. The temperature gets to rise from the bottom to the top inside the composite solar wall. The cooler room air is drawn through the bottom vent and into the composite solar wall, and the convective heat exchange among the room air, the air in the gap and the porous absorber occurs, so the air in the composite solar is heated, it rises upward and out through the top vent and into the heating room. It produces an air cycle in the heating system. So, the heat exchanges between the air in the room and the outside in the thermal conduction of the wall and the convective heat transfer. The comparison of the flow and temperature field inside the composite solar wall made among at 14, at 16 and at 19 in the afternoon can show that the peak temperature within the porous absorber moves in versus time. During the solar radiation such as at 14 and at 16 in the afternoon, the convective heat exchange among the room air, the air in the gap and the porous absorber is intensified comparatively. In contrast, when the sun is not shining such as at 19 in the evening, the convection is thus strongly reduced and the outside surface temperature of the porous layer is much more below the inside surface temperature of the porous layer, so the convective heat exchange and the radiation exchange between composite solar wall and ambient is decreased and the heat loses of the heating system is strongly reduced; as the excess heat is stored within the porous absorber and the south storage wall during solar radiation and there is temperature gradient in the porous layer and the thermal storage wall, the inside surface temperature of the porous absorber is higher comparatively. Therefore, when the sun is no shining, the porous layer servers as semi-thermal insulator. The amount of heat losing from the room to the outside can be reduced.

5. CONCLUSIONS

A numerical model has been developed to study the effects of the porous absorber on the temperature distribute and airflow in the composite solar wall.

From above discussion, we can conclude that the occurrence of the overheating during the solar radiation in the heating system and the effects of the ambient during no solar shining on the heating room temperature decreases. The excess heat is stored within the porous absorber and thermal storage wall in the period of the solar radiation. The influence permeability within the porous absorber on the heating room temperature is significant. Therefore, the particle sizes and porosity within the porous absorber should be chosen properly avoid the occurrence of the overheating and the no reaching the heating requirements when the solar shining is available. So, in a passive-solar heated building, the composite wall with a porous absorber under investigation can work better in saving energy than the Trombe wall in winter.

NOMENCLATURE

A :area, m^2 ; c : specific heat, $J/kg \cdot K$

C :inertia coefficient, d :diameter, m

g **:**gravitational acceleration vector m/s^2

G_{sun} :rate of solar flux incident on glass cover,

h_{po} :horizontal coordinate of porous absorber outside surface , m ;

h_{pi} :horizontal coordinate of porous absorber outside surface , m

k_f :fluid thermal conductivity, W/(m·K)

k_m : apparent thermal conductivity, W/(m·K)

k_s :soild thermal conductivity, W/(m·K)

k_w : thermal conductivity for surface of partition wall, W/(m·K)

K :permeability of porous absorber, m^2

P: pressure, *Pa*

Q_{ai} : convective heat exchange between air inside gap and the glass cover, *J*

Q_{apo} :convective heat exchange between air inside gap and the outer surface of porous

Q_{gp} : thermal radiation exchange between glass cover and the outside surface of porous

Q_{sky} : thermal radiation exchange between the glass cover and the sky, *J*

Q_{inr} : heat flux from the composite wall to the heated room through the top vent, *J* ;

Q_{outr} :heat flux from the heated room to the composite wall through the bottom vent, *J* ;

T*:* temperature; T_{ao} :ambient temperature， , $|\vec{v}_d|$ mean velocity(= $\sqrt{u_d{}^2 + v_d{}^2}$) m/s

Greek symbols

β :thermal expansion coefficient, *1/K;*

τ :time, s; μ :dynamic viscosity, Kg/(m·s)

ρ :density, Kg/m³; θ *:*porosity of the porous

η :absorptivity; δ : transmissivity;

Subscripts

C: cold wall; d :Darcy ; *eff:* effective;

m*:* apparent mean; p: porous

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