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Research Paper Engineering

Numerical Approach To Predict The Thermal Performance Of Parallel And Counter Flow Packed Bed Solar Air Heaters

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ABSTRACT

A design of parallel and counter flow solar air heaters with or without packing (porous media) has been proposed to improve the thermal performance. To check the heat transfer characteristics and thermal performance of the parallel and counter flow solar air heaters, are studied numerically. Mathematical designs to predict the heat transfer characteristics of the parallel and counter flow solar air heater based on the governing energy balance equations has been developed . To solve these models and to provide an iterative approach, forward difference method of finite difference scheme is employed. In the present work the effect of mass flow rate and varying porosity of the packed bed are studied. Also the effect of fraction of mass flow rate in parallel flow solar air heater has been studied theoretically. Results indicate that the efficiency of various models increases with increase in mass flow rates and decreases with increase in porosity. It is found that the thermal efficiency of counter flow solar air heater with porous media is greater than 10-20% from the parallel flow solar air heater with porous media, 10-30% from parallel flow and counter flow solar air heater without porous media.

Keywords : Packed Bed, Porosity, Heater, Thermal Efficiency, Parallel Flow

Introduction

Conventional type solar air collectors are designed to provide maximum amount of heat at lower cost. These types of solar air collectors collect solar energy and because of low operating and maintenance cost, they are widely used as a heating media. Useful heat energy from flat plate solar air heaters can be used in many thermal applications in drying agricultural products such as in seeds, fruits, and vegetables and residential also some time in industries and as a auxiliary heater for heating building in winter time. One of the drawbacks of these conventional types of solar air heaters is that there is heat loss accurse from the front cover. To avoid such heat loss and to increase thermal efficiency by increasing heat transfer rate to the following air, packing of a material is provided in the upper or lower cannel, because packing provides a large contact surface area and turbulence for the air flow. One more advantage of packing is that it provides a heat storage capacity.

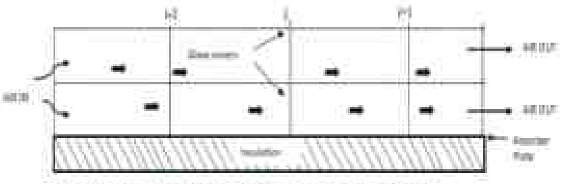
The various characteristics of solar air heaters have been widely studied by various researchers, Mohammad [1] presented an analysis for novel type solar air heater. The main idea is to minimize the heat losses from the top glass cover of the collector and maximize heat extraction from the absorber. In order to make heater more efficient, porous media is placed in the second passage. Study by Ramani, B.M. et al.[2] demonstrates that double pass counter flow solar air heater with porous material in the second air passage and discus the effects of various parameters on thermal performance and pressure drop characteristics. Thermal performance of a double-pass solar air heater with packed bad above the heater absorber plate was investigated experimentally and theoretically by Ramadan et al. [3]. Limestone and gravel were used as packed bed material and recommended to operate the system with packed bed with values of mass flow rate equal to 0.05 kg/s or lower to have a lower pressure drop across the system. The thermal performance of a double glass, double pass solar air heater with a packed bed in the lower channel was investigated experimentally and theoretically by El-Sebaii, et al. [4]. Aldabbagh et al. [5] studied the thermal performance of single and double pass solar air heaters with wire mesh layers and investigated heater performance experimentally. To study the heat transfer characteristics and performance of the double pass flat plate solar air heaters with or without porous media numerically, Naphon, P et al.[6]derived a mathematical model from the energy balance equations and to solve these equations , implicit method of finite difference scheme was employed. M.K.Mittal et al. [7] investigated the thermo hydraulic efficiency on a packed bed solar air heater having the duct packed with blackened wire screen matrices of different geometrical parameters. Omojaro et al. [8] investigated experimentally the single and two-pass (counter-flow) solar air heater with steel wire mesh layers as porous media in the lower channel without an absorber plate and having porosity more than 85%. Garg et al. [9] used an absorber with fins attached in order to improve the thermal performance of the single pass solar air collector. Thakur et al. [10] and Varshney and Saini [11] investigated the use of wire mesh screen as packing material for single pass solar air collectors and derive a correlation for j-factor. However, the theoretical work in the field of parallel and counter flow solar air heaters is still limited.

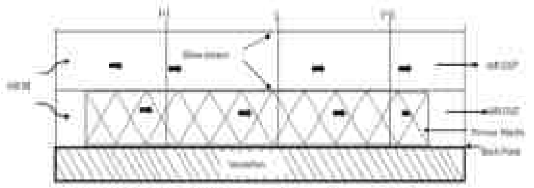
In the present work, a mathematical model capable of providing the numerical solution to predict the thermal performance of a parallel and counter flow solar air heater with or without packed bed is developed. Materials such as wire mesh screens, iron scrap are considered.

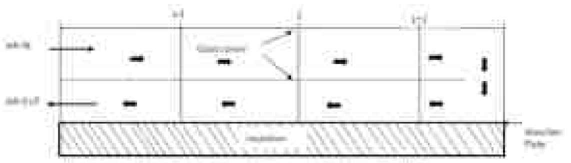
. Therefore, mathematical model is solved by use of forward difference technique of finite difference scheme and examined by using a constructed computer program that uses an iterative solution procedure.

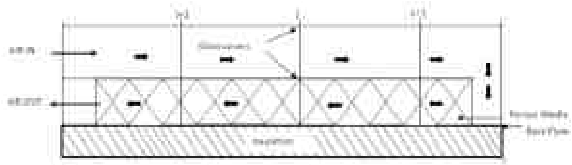
1. Mathematical model

In the present study, a mathematical model is obtained by the application of the governing conservation laws. The heat balance is accomplished across each component of a given air heater, i.e., the glass covers, the air streams in both of the upper and lower channels, for the absorber or back plate. Fig.1 shows the systematic view of four types of solar air heaters which are considered.









It is assumed that air velocity in the channel at any section is constant, the flow of heat is one-dimensional and steady, heat loss across the sides of the duct is very small and hence neglected, no conduction inside the heater, the porous absorber and the air stream are in thermal equilibrium because the value of volumetric heat transfer coefficient in the pores of the porous matrix is very high. Hence the energy balance equations for four types of solar air heater models are written as.

2.1 Energy balance equation for parallel and counter flow solar air heater without porous media.

For upper glass cover

$$\alpha_{gu} = [h_{r(gu-a)} + h_w](T_{gu,i} - T_a) + h_{r(gu-gl)}(T_{gu,i} - T_{gl,i}) + h_{c(gu-fu)}(T_{gu,i} - T_{fu,i}) \quad (1)$$

For lower glass cover

$$\alpha_{gl}\tau_{gu} = [(h_{r(gl-gu)} + h_{c(gl-gu)})](T_{gl,i} - T_{gu,i}) + [h_{r(gl-p)}(T_{gl,i} - T_{p,i})] + h_{c(gl-fu)}(T_{gl,i} - T_{fu,i}) \quad (2)$$

For flow in upper channel

$$\frac{mC_p}{w}\frac{(T_{fu,i+1} - T_{fu,i})}{\Delta x} = h_{c(gl-fu)}T_{gl,i} + h_{c(gu-fu)}T_{gu,i} - (h_{c(gl-fu)} + h_{c(gu-fu)})T_{fu,i} \quad (3)$$

For flow in lower channel

$$\frac{mC_p}{w}\frac{(T_{fl,i+1} - T_{fl,i})}{\Delta x} = h_{c(b-fl)}(T_{p,i} - T_{fl,i}) + h_{c(gl-fl)}(T_{gl,i} - T_{fl,i}) \quad (4)$$

For absorber plate

$$\alpha_p\tau_{gu}\tau_{gl} + h_{r(gl-p)}(T_{gl,i} - T_{p,i}) = h_{c(p-fl)}(T_{p,i} - T_{fl,i}) + U_p(T_{p,i} - T_a) \quad (5)$$

1.2 Energy balance equation for parallel and counter flow solar air heater with porous media in lower channel.

Equations (1) and (3) are same for parallel and counter flow solar air heater with packed bed in lower channel. So the energy balance equations for packed bed, flow in lower channel and for back plate is written as.

For lower glass cover

$$\alpha_{gl}\tau_{gu} = [(h_{r(gl-gu)} + h_{c(gl-gu)})](T_{gl,i} - T_{gu,i}) + [h_{r(gl-m)}(T_{gl,i} - T_{m,i})] + h_{c(gl-fu)}(T_{gl,i} - T_{fu,i}) \quad (6)$$

For porous matrix (packed bed)

$$\alpha_m\tau_{gl}\tau_{gu} = h_{r(m-b)}(T_m - T_b) + h_{r(m-gl)}(T_m - T_{gl}) + h_{c(m-f)}(T_m - T_{fl})A_m \quad (7)$$

For flow in lower channel

$$\frac{mC_p}{w}\frac{(T_{fl,i+1} - T_{fl,i})}{\Delta x} - k_p\delta\frac{(T_{fl,i+1} - 2T_{fl,i} + T_{fl,i-1})}{\Delta x^2} = h_{c(gl-fl)}(T_{gl,i} - T_{fl,i}) - h_{c(b-fl)}(T_{b,i} - T_{fl,i}) - h_{c(m-fl)}(T_{m,i}) \quad (8)$$

For back Plate

$$h_{r(b-m)}(T_{m,i} - T_{b,i}) = h_{c(b-fl)}(T_{b,i} - T_{fl,i}) + U_b(T_{b,i} - T_a) \quad (9)$$

Calculation methods

The above assumptions are based upon the fact that the volumetric heat transfer coefficient in solid matrix is very high. Effective thermal conductivity value changes from 5-20 times the air thermal conductivity, but the effect on the results of simulation is significant. Hence, kp is set to 0.3 W/mK (Mohamad [1]).

The meanings of all symbols and notations of various heat transfer coefficients in equations (1) (9) of the different elements of the solar air heater given; vise,

$h_{r(gu-gl)}, h_w, h_{r(gl-gu)}, \quad h_{c(gl-fu)}, h_{r(gl-m)}, \quad h_{r(p-m)}, h_{r(p-b)}, h_{c(p-fl)}$ and $h_{c(b-fl)}$

are calculated using the correlations given in literature Garg et al. [9].

The convective heat transfer coefficient for air flowing over the outside surface of the glass cover is proposed by McAdams [12] as follows:

$$h_w = 5.7 + 3.8V \quad (10)$$

The natural convective heat transfer coefficient between the upper and lower glass cover,

$h_{c(gl-gu)}$ determined from correlations provided by Hollands et al. [13]

$$Nu = 1 + 1.44\left[1 - \frac{1708}{Ra\,Cos\,\beta}\right]^+ \left[1 - \frac{1708\,(Sin\,1.8)^{1.6}}{Ra\,Cos\,\beta}\right] + \left\{\left[\frac{Ra\,Cos\,\beta}{5830}\right]^{1/3} - 1\right\}^+ \quad (11)$$

Where the + exponent implies that only positive values of the terms in the square brackets should be used.

Convective heat transfer between the air flowing in the packed upper channel and the lower glass cover,

$h_{c(gl-fu)}$

may be obtained as

$$h_{c(gl-fu)} = Nu_m k_f / \varepsilon D_{hu} \quad (12)$$

Where D_h, is the hydraulic diameter of the upper channel, and is given as

$$D_h = \frac{4A_{fu}}{p} = \frac{4(wD)}{2(w+D)} = \frac{2(wD)}{(w+D)} \quad (13)$$

Nu_m is the Nusselt number for the packed bed and is given by

$$Nu_m = 0.2Re_m^{0.8}Pr^{1/3} \quad (14)$$

Where Re_m is Reynold's number for the wire mesh packed bed channel and is calculated as Thakur et.al. [10].

$$Re_m = 4r_H G_o / \mu \quad (15)$$

$$r_H = \varphi d_w/4(1-\varphi)\,, \quad (16)$$

. r_H is hydraulic radius and is related to the packing size and the void space.

G_O is the mass velocity, kg/sm^2 is given by

$$G_0 = \dot{m}/(A_c)\varphi \qquad (17)$$

A_{fu} is the frontal area of the upper channel of the solar air heater is the porosity of the packed bed and is given by Thakur et.al. [10].

$$\varphi = \frac{p_t^2 D - \left[\frac{\pi}{2}(d_w)^2 p_t\right] n}{p_t^2 D} \qquad (18)$$

Where n is the number of wire mesh layers.

The convective heat transfer coefficient from the absorber plate to the air flowing in the upper channel $h_{c(p-fu)}$ is assumed to be equal to $h_{c(fi-fu)}$ Forson et.al. [14].

Equations (1), (2), (5), (6), (7) and (9) are solved simultaneously to give expressions for nodal temperatures $T_{gl,i}$, $T_{gu,i,}$ $T_{m,i,}$ $T_{p,i}$ and $T_{b,i}$

Equations of flow for parallel flow and counter flow without porosity are simplified as:

For upper channel

$$T_{fu,i+1} = \frac{1}{y}\left[h_{c(gl-fu)} T_{gl,i} + h_{c(gu-fu)} T_{gu,i} + T_{fu,i}\left(1 - \left(h_{c(gl-fu)} + h_{c(gu-fu)}\right)\right)\right] \qquad (19)$$

For lower channel

$$T_{fl,i+1} = \frac{1}{y}\left[h_{c(gl-fl)} T_{gl,i} + h_{c(p-fl)} T_{p,i} + T_{fl,i}\left(1 - \left(h_{c(gl-fl)} + h_{c(p-fl)}\right)\right)\right] \qquad (20)$$

Equations of flow in for parallel flow and counter flow with porosity are simplified as:

For upper channel

$$T_{fu,i+1} = \frac{1}{y}\left[h_{c(gl-fu)} T_{gl,i} + h_{c(gu-fu)} T_{gu,i} + T_{fu,i}\left(1 - \left(h_{c(gl-fu)} + h_{c(gu-fu)}\right)\right)\right] \qquad (21)$$

For lower channel

$$\begin{aligned} T_{fl,i+1} = \frac{1}{(1-A)}\Big[& A T_{fl,i-1} + s\left(h_{c(gl-fl)} T_{gl,i} + h_{c(b-fl)} T_{b,i} + h_{c(m-fl)} T_{m,i}\right) \\ & + T_{fl,i}\left[1 - 2A - s\left(h_{c(gl-fl)} + h_{c(b-fl)} + h_{c(m-fl)}\right)\right]\Big] \end{aligned} \qquad (22)$$

Guass elimination technique is used to solve the Equations for the bed temperatures $T_{gl,i}$, $T_{gu,i,}$ $T_{m,i,}$ $T_{p,i}$ and $T_{b,i}$ given in Boyce et.al. [15].

The following boundary conditions (B.C.) were applied:

B.C. for parallel flow solar air heater with or without packed bed in lower channel

$$T_{fu}\big|_{x=0} = T_a\,, \qquad T_{fl}\big|_{x=0} = T_a\,,$$

B.C. for counter flow solar air heater with or without packed bed in lower channel

$$T_{fu}\big|_{x=0} = T_a\,, \qquad T_{fl}\big|_{x=0} = T_{fu}\big|_{x=L}\,,$$

Following parameters are considered;

- length of solar air heater, l = 2.2 m
- width of solar air heater, w = 0.45 m
- depth of the upper channel, D_u = 0.025 m
- depth of the lower channel, D_l= 0.025 m
- transmissivity of the glass covers, T_{tu}, T_{gl}, = 0.92
- absorptivity of the glass covers, a_{gu}, a_{gl}, = 0.05
- absorptivity of the porous material, α_m, = 0.95
- absorptivity of the absorber plate, α_p= 0.95
- air mass flow rate, m=(0.01kg/s -0.05 kg/s)
- porosity of the porous media, φ= 0.91 to 0.96
- conductivity of the wire mesh screens, k_m= 0.3W/m-k

Iterative solution procedure

First of all equations from (1) to (8) were solved simultaneously to obtain the temperature of the glass covers, porous media, absorber plate, back plate and for flow in upper and lower channel. It was assumed that the initial temperatures of all the elements of the solar air heater are equal to the ambient temperature. The heat transfer coefficients were computed accordingly. An iterative procedure was then created and the mean temperatures for the different sections (the glass cover, porous media, absorber plate, back plate) of the solar air were computed by using guass elimination method. The newly computed temperatures and heat transfer coefficients were put into the flow equations to obtain the outlet temperatures. The process was repeated until all consecutive mean temperatures do not differ by more than 0.01 °C. The computer program is based on MATLAB and proceeds as outlined above. At the end of the program, the required temperatures averaged over the entire length of the heater are obtained in addition to the outlet temperature of the airflow and the efficiency of the collector.

Results and discussions

Thermal performance of parallel and counter flow solar air heaters with or without packing are investigated theoretically. All of the models are predicted for various mass flow rates ranges from 0.01 to 0.05 kg/s and for ranges of porosity 92-96%. Obtained results are compared with each others to check the best performance among all of the heaters models. Thermal performance for parallel flow with or without porous media is checked on

$$m_{f1} = \frac{m}{2}\,, m_{f2} = \frac{m}{2} \; and \; m = m_{f1} + m_{f2}$$

Fig.2 show a comparison of thermal efficiency between parallel flow and counter flow solar air heaters without packing. It is found that thermal efficiency of the solar air heaters increases with increase in mass flow rates and also found that thermal performance of CFSAHWOP is approximately more then 3-10% that of PFSAHWOP. Theoretical study of PFPBSAH and CFPBSAH shows that, the thermal performance of counter flow solar air heater is higher than that of parallel flow solar air heater. Fig.3 shows that at a fixed solar intensity (I=800 w/m2) and a fixed porosity value say 92%, yields approximately 7 to 20 ◦C more temperature rise for counter solar air heater then parallel flow solar air heater with porous media. This is shown is Fig.4, parallel flow and counter flow solar air heaters with porous media have higher values of thermal efficiency 10 -20 % and 15-30% then parallel and counter flow without porous media respectively. Fig.5 and Fig.6, illustrates the variations of thermal efficiency with respect to the mass flow rate for various ranges of porosity 92-96% for parallel flow and counter flow respectively. It is found that counter flow solar air heater have higher thermal efficiency then parallel flow packed bed in lower channel. Also Fig.7 ad Fig.8 show the variation of the effective thermal efficiency with respect to the mass flow rate for parallel flow and counter flow with porous media in lower channel. It is found that effective thermal efficiency increases with increasing mass flow rate for PFPBSAH and decrease for CFPBSAH as mass flow rate increase above 0.03 kg/s.

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When $m_{f1} \neq m_{f1}$, total mass flow rate is fixed and two different streams of mass flow rates are entering through upper and lower channel, Fig.9 and 10, presents the effect of fraction of mass flow rate on the thermal and effective thermal efficiency of parallel flow solar air heater. It is found that thermal and effective thermal efficiencies are increases as r increasing from 0.2 -0.8.various curves are drawn for various values of mass flow rate, m, kg/s. Maximum value of theoretical thermal and effective thermal efficiency are found to be 70% and 68% respectively at r = 0.8.

Fig.2. Effect of mass flow rates on the thermal efficiency of parallel and counter flow solar air heaters without packing.

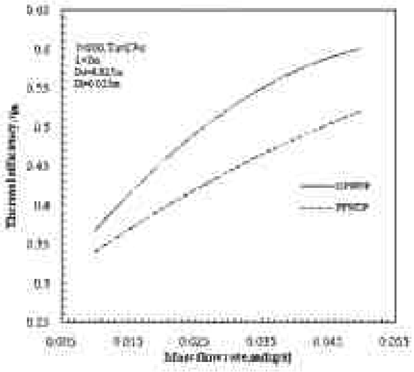


Fig.3. Effect of mass flow rate on the temperature rise in ◦C of the PFPBSAH and CFPBSAH.

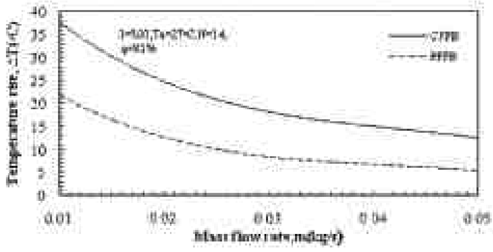


Fig.4. Effect of mass flow rates on the thermal efficiency of the parallel and counter flow solar air heaters with or without packed bed.

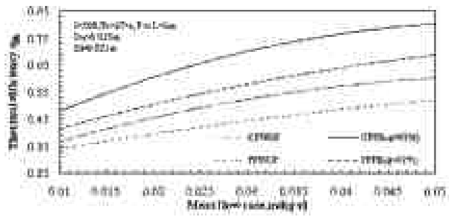


Fig.5. Effect of mass flow rate on the thermal efficiency (ηth) of the parallel flow packed bed solar air heater.

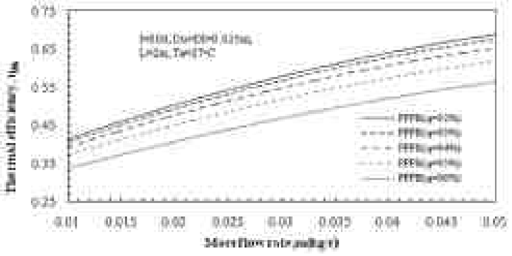


Fig.6.Effect of mass flow rate on the thermal efficiency of the counter flow packed bed solar air heater.

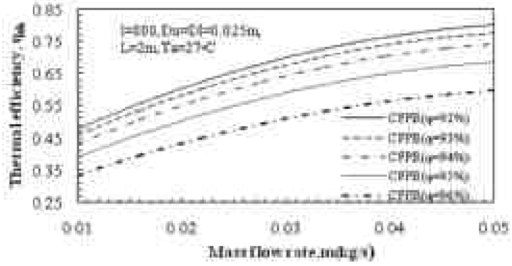


Fig.7. Effect of mass flow rate on the effective thermal efficiency of the parallel flow packed bed solar air heater.

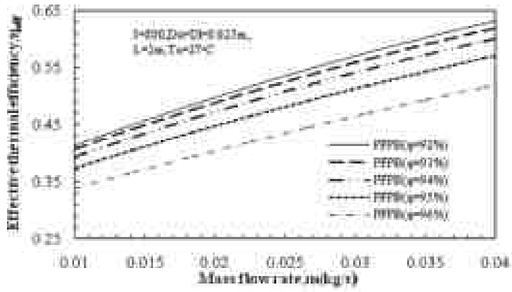


Fig.8. Effect of mass flow rate on the effective thermal efficiency of the counter flow packed bed solar air heater.

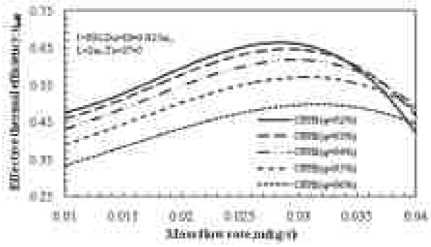


Fig.9. Effect of the fraction of mass flow rate on the thermal efficiency of parallel flow packed bed solar air heater.

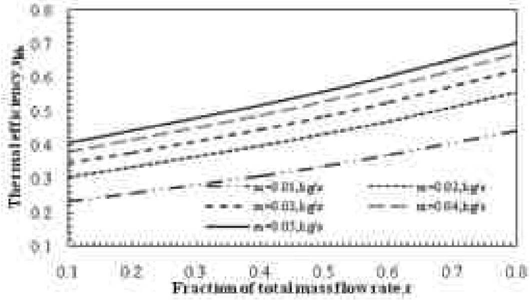
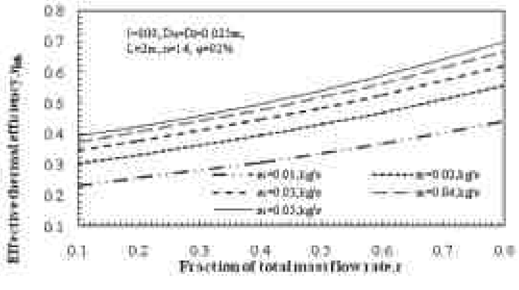


Fig.10. Effect of the fraction of mass flow rate on the effective thermal efficiency of parallel flow packed bed solar air heater.



Conclusions

In the present study, the mathematical model is presented to predict the heat transfer characteristics and the performance of the parallel and counter flow solar air heater with or without packing in its lower channel. High conductivity porous media such as iron scrap, iron wool or wire mesh screen matrix provides better thermal performance and storage capacity. The numerical solution procedure of the energy conservation equations by using a computer code to predict the various temperatures and thermal performance of the present solar air heaters are made. The models are validated by comparing their thermal performances with results obtained from parallel flow and counter flow solar air heaters with or without packing. The solar air heater with porous media gives 20-30% higher thermal efficiency than that of without porous media. The counter flow solar air heater on an average gives 35% higher thermal efficiency than that of parallel flow system for the range of parameter investigated.

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