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Thermal performance study on parallel flow packed bed solar air heater

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ABSTRACT

A comprehensive parametric study has been carried out in the present study on the thermal performance of parallel flow solar air heater is to study the behavior of parallel flow solar air heater under different set of conditions, obtained by changing the governing parameters like air mass flow rate and porosity, at study state conditions. Parallel flow solar air heater with and without porous media have been taken into consideration. The problem has been solved by the Finite Difference Method. It is found that the thermal efficiency increases by 78.8 % in parallel flow mode with porous media than parallel flow without porous media at the mass flow rate of 0.05 kg/s at the bed porosity of 92%.

Keywords : Packed bed; Porosity; Heat Transfer; Thermal efficiency; double flow solar air heater

Nomenclature

C_p	specific heat (KJ/KgK)
d_w	wire diameter of screen (m)
h	heat transfer coefficient (W/m² K)
I	global solar radiation (W/m²)
k	thermal conductivity (W/mK)
L	length of the heater (m)
M	mass (kg)
m	mass flow rate (Kg/s)
Q_u	thermal power output (W)
T	temperature(°C)
V	Volume (m³)
U_b	back loss coefficient (W/m² K)
v	velocity (m/s)
b	width of the collector (m)
PFPBSAH	parallel flow packed bed solar air heater
PFSAHWOP	parallel flow solar air heater without packing

Subscripts

a	ambient
c	convective
eff	effective
f	fluid
g	glass
i	inlet, grid point
m	packed bed material
o	outlet
p	absorber plate
b	back plate
r	radiative
w	wind
u	upper duct
l	lower duct

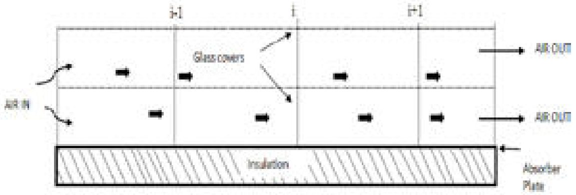
1. Introduction

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. 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. Study by Ramani, B.M. et al. [2] discussed the effects of various parameters on thermal performance of the double pass counter flow solar air heater with porous material in the second air passage 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. However, the theoretical work in the field of parallel flow solar air heaters is still limited.

2. Mathematical model

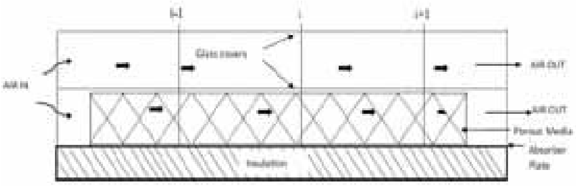
In the present work parallel flow with or without packing solar air heaters are discussed. Packing is considered in the lower channel. Models consist of two glass covers and an insulated absorber or back plate. Double flow passage is made through between two glass covers (upper and lower) and between lower glass covers and absorber as shown in fig.1. A mathematical model is first obtained by the application of the governing conservation laws. The heat balance is accomplished across each component of the given air heaters, i.e., the glass covers, the air streams in both of the channels (upper and lower), absorber plate and packed bed. 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.

(A)



Model of parallel flow without packing solar air heater

(B)



Model of parallel flow packed bed solar air heater

Fig.1 (A) Model of parallel flow without packing solar air heater (B) Model of parallel flow packed bed solar air heater.
Hence the energy balance equations for solar air heater models are written as.
2.1 Energy balance equation for PHWOPSAH.

For upper glass cover

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

For lower glass cover

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

For flow in upper channel

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

For flow in lower channel

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

For absorber plate

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

2.1 Energy balance equation for PFPBSAH.
Equations (1) and (3) are same for PFPBSAH. So the energy balance equations for lower glass cover, packed bed, flow in lower channel and absorber plate are written as.

For lower glass cover

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

For porous matrix (packed bed)

$$I\alpha_m\tau_g\tau_{gl} = h_{r(m-p)}(T_m - T_p) + h_{r(m-gl)}(T_m - T_{gl}) + h_{c(m-fl)}(T_m - T_{fl}) \tag{7}$$

For flow in lower channel

$$\frac{mC_p}{w}\frac{\left(T_{fl,i+1} - T_{fl,i}\right)}{\Delta x} = k_p\delta_p\frac{\left(T_{fl,i+1} - 2T_{fl,i} + T_{fl,i-1}\right)}{\Delta x^2} + h_{c(gl-fl)}\left(T_{gl,i} - T_{fl,i}\right) + h_{c(p-fl)}\left(T_{p,i} - T_{fl,i}\right) + h_{c(m-fl)}(T_{m,i} - T_{fl,i}) \tag{8}$$

For absorber Plate

$$h_{r(p-m)}\left(T_{m,i} - T_{p,i}\right) = h_{c(p-fl)}\left(T_{p,i} - T_{fl,i}\right) + U_p\left(T_{p,i} - T_a\right) \tag{9}$$

3. 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 (k_p) value changes from 5-20 times the air thermal conductivity, but the effect on the results of simulation is significant. Hence, k_p is set to 0.3 (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 as: $h_{r(gu-gl)}$,h_w, $h_{r(gl-gu)}$,$h_{c(gl-fu)}$,$h_{r(gl-m)}$,$h_{r(p-m)}$,$h_{r(p-b)}$,$h_{c(p-fl)}$ and $h_{c(b-fl)}$, and calculated using the correlations given in literature Garg et al. [7].

The convective heat transfer coefficient for air flowing over the outside surface of the glass cover is proposed by McAdams [8] as follows:

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

Equations (1), (2), (5), (6), (7) and (9) are solved simultaneously to give the expressions for nodal temperatures, $T_{gl,i}$,$T_{gu,i}$,$T_{m,i}$, and $T_{p,i}$ respectively. Guass elimination technique is used to solve the Equations for $T_{gu,i}$,$T_{gl,i}$,$T_{m,i}$ and $T_{p,i}$ as given in Boyce et.al. [9]. Following boundary conditions (B.C.) were applied:

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

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

Following parameters are considered;

length of solar air heater, L = 2.2 m,

width of solar air heater, = 0.45 m

depth of the upper channel, = 0.025 m

depth of the lower channel, = 0.025 m

transmissivity of the glass covers, τ_{gu},τ_{gl} = 0.92

absorptivity of the glass covers, α_{gu},α_{gl} = 0.05

absorptivity of the porous material, a_m= 0.95

absorptivity of the absorber plate, a_p= 0.95

Total air mass flow rate, = (0.02 kg/s - 0.1 kg/s)

porosity of the porous media, ϕ = 0.92 to 0.96

conductivity of the wire mesh screens, k_m= 0.3W/m-k

4. Results and discussions

Thermal performance of parallel flow solar air heaters with or without packing are investigated theoretically. All of the models are predicted for various mass flow rates ranging from 0.01 to 0.05 kg/s and for various ranges of porosity 92- 96%. Obtained results then 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_{fu}=m/2,m_{fl}=m/2 and m=m_{fu}+m_{fl}.

The effect of mass flow rate on the thermal performance of PFPBSAH for different bed porosity is presented in Fig. 2.The predicted thermal efficiency of PFPBSAH is 77.6%, 75.3%, 73.08%, 69.11% and 62.75% at a porosity of 92%, 93%, 94%, 95% and 96% respectively, whereas the predicted thermal efficiency of the PFWOPSAH is 43.4%, at a mass flow rate of 0.05 kg/s. The percentage enhancement in the thermal efficiency of PFPBSAH is found be 78.8% in comparison to PFWOPSAH at a mass flow rate of 0.05 kg/s. It has been observed that the thermal efficiency criteria does not account for pumping power losses, therefore, it is necessary to consider a thermo hydraulic or effective efficiency criteria for the optimum design.

The maximum effective thermal efficiency of PFPBSAH is predicted to be 64.5% at the mass flow rate of 0.03 kg/s (total mass flow rate of 0.06 kg/s) and the bed porosity is 92% as shown in Fig. 3. Figs 4 and Fig.5 depict the temperature variation of air in the upper and lower channel along the length of collector for different depth ratios and bed porosity of 92%. It is observed that there is a substantial rise in air temperature in the collector. It is also seen that the temperature rise of air in the upper channel is smaller than that of lower channel of the parallel flow packed bed solar air heater. The effective thermal efficiency of PFWOPSAH for the same operating conditions is predicted as 41.7%. Rise in air temperature in the lower channel is obtained maximum at mass flow rate equals to 0.01kg/s as shown in Fig.5.

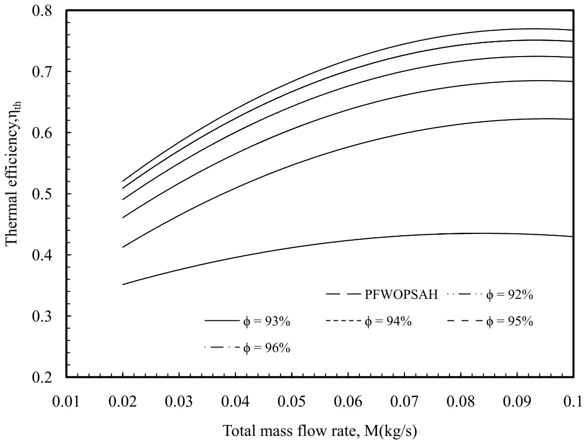


Fig. 2 Effect of mass flow rate on thermal efficiency of PFPBSAH and PFWOPSAH for different porosity.

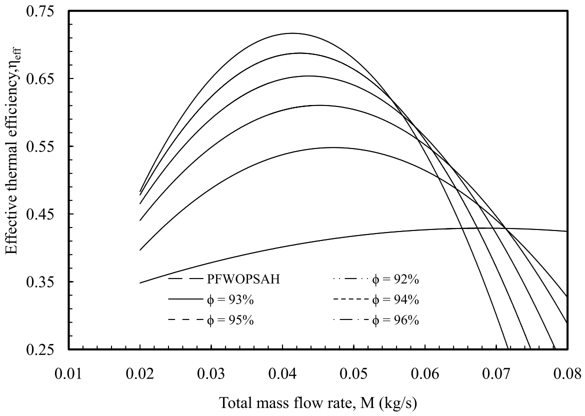


Fig.3 Effect of mass flow rate on the effective thermal efficiency of PFPBSAH and PFWOPSAH for different porosity.

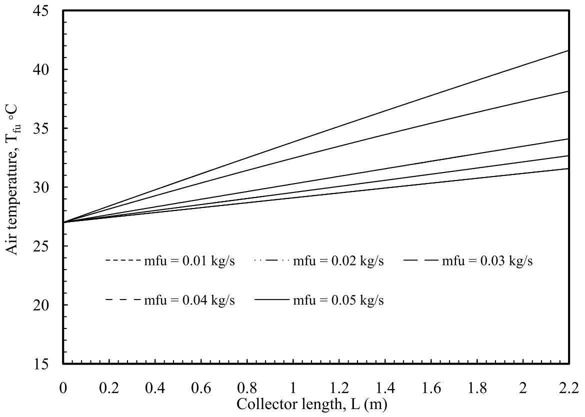


Fig.4 Temperature variations along the length of collector in the upper channel of the PFPBSAH (φ = 92%).

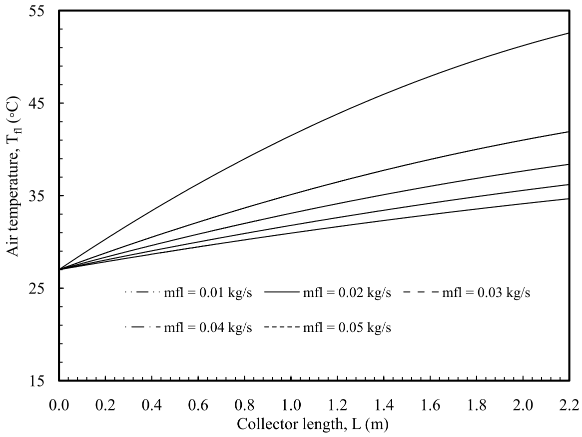


Fig.5 Temperature variations along the length of collector in the lower channel of the PFPBSAH (φ = 92%).

5. Conclusion

High conductivity porous media such as wire mesh screen matrix provides better thermal performance and storage capacity. Thermal efficiency of PFPBSAH is about 78.8% higher than that of PFWOPSAH at the mass flow rate of 0.05 kg/s, bed porosity of 92% for the range of parameter investigated. Effective thermal efficiency of PFPBSAH is obtained maximum at the mass flow rate 0.02kg/s (total mass flow rate of 0.04 kg/s) and starts decreasing beyond this mass flow rate. It is recommended to operate the system with and without packed bed with values of mass flow rate of 0.02kg/s or lower to have a lower pressure drop across the system and therefore, a reasonably high effective thermal efficiency.

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