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Performance analysis for flat plate collector with and without porous media

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Abstract

The present work involves a theoretical study to investigate the effect of mass flow rate, flow channel depth and collector length on the system thermal performance and pressure drop through the collector with and without porous medium. The solution procedure is performed for flat plate collector in single and double flow mode. The analysis of the results at the same configuration and parameters shows that the system thermal efficiency increases by 10-12% in double flow mode than single flow due to the increased of heat removal, and increase by 8% after using porous medium in the lower channel as a result of the increase of heat transfer area. At the same time the pressure drop will be increased. All collectors show improved efficiency obtained when the collector operates at relatively high flow rates, and at relatively low collector temperature rise since the collector losses will be less in low temperature difference.

Keywords: Single and double flow solar air heater; porous medium; thermal performance; pressure drop; flow channel depth

Notations

A	Area of collector that absorb solar radiation, m^2
A_e	Area base on the perimeter of the collector
C_p	Specific heat of working fluid, J/kg K
D	Collector width, m
D_h	Hydraulic diameter, m
f	Friction factor
Fr	Heat removal factor
H	Collector height, m
h	Convective heat transfer coefficient, W/m^2 K
h_r	Radiation heat transfer coefficient, W/m^2 K
h_a	Convective heat transfer coefficient between the glass and the ambient air, W/m^{2}K
I	Solar radiation, W/m^2
k	Fluid thermal conductivity, W/m K
k_b	Back insulation conductivity, W/m K
k_e	Edge insulation conductivity, W/m K
L	Collector length, m
$\dot{m}$	Collector flow rate, kg/s
N	Number of glass cover
Nu	Nusselt number
P	Pressure drop across the duct, Pa
Q_u	Rate of useful energy gain, W
Re	Reynolds number
T_a	Ambient air temperature, K
T_c	Cover temperature, K
T_f	Fluid temperature, K
T_i	Fluid inlet temperature, K
T_p	Absorber plate surface temperature, K
T_{pr}	Porous media temperature, K
T_{base}	Bottom plate temperature, K
U	Overall heat loss coefficient, W/m^2 K
U_b	Back loss coefficient, W/m^2 K
U_e	Edge loss coefficient, W/m^2 K
U_t	Top loss coefficient, W/m^2 K
V	Air velocity, m/s
W	Collector width, m
x_b	Back insulation thickness, m
x_e	Edge insulation thickness, m

Greek symbols

ε_p	Emittance of absorber plate
ε_c	Emittance of glass cover
ρ	Air density, kg/m^3
σ	Stephen-Boltzman constant
τ	Solar transmittance of glazing
α	Solar absorptance of collector plate
μ	Dynamic viscosity, Pa.s
η	Efficiency

Subscripts

a	ambient
b	back
c	cover
f	fluid
h	hydraulic

p plate
u useful

1. Introduction

Energy is a vital need in all aspects and due to the increasing demand for energy coupled with its inefficient consumption, the environment has been polluted either directly or indirectly. To prevent this from becoming a global disaster, it is inevitable to strengthen efforts of energy generation and utilization using sustainable means and progressively substituting the fossil fuels for renewable sources of energy. The solar radiation level in Malaysia is high, ranging from 6.6kWh/m^2 in January to 6.0kWh/m^2 in August, which is ideal for several solar energy applications (Mohd *et al*., 1999).

Extensive investigations have been carried out on the optimum design of conventional and modified solar air heaters, in order to search for efficient and inexpensive designs suitable for mass production for different practical applications. The researchers have given their attention to the effects of design and operational parameters, type of flow passes, number of glazing and type of absorber flat, corrugated or finned, on the thermal performance of solar air heaters (Ratna *et al.,* 1991; Ratna *et al.,* 1992; Choudhury *et al.,* 1995; Karim and Hawlader, 2004).

Ratna *et al.* (1991) has presented theoretical parametric analysis of a corrugated solar air heater with and without cover, where they obtained the optimum flow channel depth, for maximum heat at lowest collector cost. Ratna *et al.* (1992) has found that there exists an optimum mass flow rate corresponding to an optimum flow channel depth. This result has been concluded after conducting a study on 10 different designs of solar air heaters. Choudhury *et al*. (1995) has calculated the ratio of the annual cost and the annual energy gain for two-pass solar air heaters with single and double covers above the absorber. They concluded that for shorter duct lengths and lower air mass flow rates, the performance of the two pass air heaters with a single cover is most cost-effective as compared to the other designs.

Karim and Hawlader (2004) have performed an experimental study on three types of solar air collectors, i.e. flat plate, finned and corrugated absorbers. They reported that the V-corrugated is the most efficient collector and the flat plate collector is the least efficient.

In spite of this concern on improving the performance of solar air heaters, little has been published on the effect of the air flow passage dimension on the efficiency and pressure drop and hence the cost-effectiveness of the system. Bashria *et al.,* (2004a) and Bashria *et al*., (2004b) presented a developed internet based mathematical simulation to predict the thermal performance for different designs of solar air heaters.

The study presented in this article uses the aforementioned developed program to find the influence of different parameters, such as mass flow rate, flow channel depth and collector length on the system thermal performance and pressure drop across the collector, for flat plate collector in single and double passes with and without using a porous media.

2. Theoretical analysis

Figure 1 shows schematically the cross sectional views and the thermal network of the solar air heaters investigated in the present work. In all types, the air heaters are composed of three plates, i.e. the cover, the absorber and the rear or back plate. The air flows in the upper channel depth between the cover and the absorber plate in type 1(i.e. Single flow), but it is turned to continue flowing in the lower duct between the absorber and the rear plate in type 2, 3 (i.e. Double flow). The lower duct has been packed with glass wool as a porous medium of 0.80 porosity in type 3.

The following analysis is based on energy balance at various components of the collector models, along with the different heat transfer coefficients at their surfaces. The assumptions made are:

i) Heat transfer is steady and one dimensional
ii) The temperatures of the glass, absorber and bottom plates vary only along the x-direction of the air flow
iii) There is no leakage from the smooth flow channels
iv) The absorption of solar radiation in the cover is neglected insofar as it affects loss from the collector
v) Heat losses through the front and back of collector are to the same ambient temperature

At some location a long the flow direction, the absorbed solar energy heats up the plate to a temperature T_p.

In single pass flat plate collector energy is transferred from the plate to the ambient air at T_a through the back loss coefficient U_b, to the fluid at a temperatureT_a through the convection heat transfer coefficient h_2, and to the bottom of the cover through the radiation heat transfer coefficient h_r. Energy is transferred from fluid to the cover through the convection heat transfer coefficient h_1. Finally energy is lost to the ambient air through the combined convection and radiation coefficient U_t. The steady-state energy equations yield the following equations:

Collector cover

$$h_1(T_f - T_c) + h_r(T_p - T_c) = U_t(T_c - T_a) \quad (1)$$

Flat plate absorber

$$U_b(T_p - T_a) + h_2(T_p - T_f) + h_r(T_p - T_c) = I\tau\alpha \quad (2)$$

Fluid medium

$$h_1(T_c - T_f) + h_2(T_p - T_f) = (\frac{\dot{m}C_p}{W})(\frac{dT_f}{dx}) \quad (3)$$

Where

h_1 = Convection heat transfer coefficient from fluid to cover, W/m^2 K

h_2 = Convection heat transfer coefficient from plate to fluid, W/m^2 K

x = Axial coordinate along the flow direction

In double pass double duct flat plate collector energy is transferred from the plate at T_P to the cover through the radiation heat transfer coefficient h_{r1}, and to the fluid in the upper duct flowing between the glass cover and the plate at a temperature T_{f1} through the convection heat transfer coefficient h_2, and to the fluid in the lower duct flowing between the absorber plate and the base plate at a temperature T_{f2} through the convection heat transfer coefficient h_3. Also energy is transferred to the base plate through the radiation heat transfer coefficient h_{r2}. Energy is transferred from the fluid flowing in the upper duct to the cover glass through the convection heat transfer coefficient h_1 and from the fluid in the lower duct at T_{f2} to the base plate through convection heat transfer coefficient h_4. Finally energy is lost to the ambient air through the combined convection and radiation coefficient U_t through the cover glass. The steady-state energy balance on the cover, the plate and the fluid in the upper and lower ducts gives the following equations:

Collector cover

$$h_1(T_{f1} - T_c) + h_{r1}(T_p - T_c) = U_t(T_c - T_a) \quad (4)$$

Fluid 1

$$h_2(T_p - T_{f1}) = (\frac{\dot{m}C_{p1}}{W})(\frac{dT_{f1}}{dx}) + h_1(T_{f1} - T_c) \quad (5)$$

Absorber

$$I\tau\alpha = h_{r1}(T_p - T_c) + h_2(T_p - T_{f1}) + h_3(T_p - T_{f2}) + h_{r2}(T_p - T_r) \quad (6)$$

Fluid 2

$$h_3(T_p - T_{f2}) = (\frac{\dot{m}C_{p2}}{W})(\frac{dT_{f2}}{dx}) + h_4(T_{f2} - T_r) \quad (7)$$

Bottom plate

$$h_4(T_{f2} - T_r) + h_{r2}(T_p - T_r) = U_b(T_r - T_a) \quad (8)$$

where

h_3 = Convection heat transfer coefficient from plate to fluid in the lower duct, W/m^{2}K

h_4 Convection heat transfer coefficient from fluid in the lower duct to the base plate, W/m^2 K

h_{r2} Radiation heat transfer coefficient from plate to the base plate, W/m^2 K

T_r Bottom temperature, K

For Double Pass Double Duct Flat Plate Collector with Porous Medium in the Lower Duct energy is transferred from the plate at T_P to the cover through the radiation heat transfer coefficient h_{r1}, to the fluid in the upper duct through the convection heat transfer coefficient h_2, and to the fluid in the lower duct through the convection heat transfer coefficient h_3. Energy is also transferred to the porous media through the radiation heat transfer coefficient h_{r2}. Energy is transferred from the porous media to the fluid in the lower duct through the convective heat transfer coefficient h_4. Energy is transferred from the fluid flowing in the upper duct at T_{f1} to the cover through the convection heat transfer coefficient h_1, and from the fluid in the lower duct at T_{f2} to the base plate through convection heat transfer coefficient h_5. Energy also is transferred from the porous media to the bottom plate through the convection heat transfer coefficient heat transfer coefficient h_6. Finally energy is lost to the ambient air through the combined convection and radiation coefficient U_t through the cover glass. The steady-state energy balance gives the following equations:

Collector cover

$$h_1(T_{f1} - T_c) = h_{r1}(Tp - T_c) = U_t(T_c - T_a) \quad (9)$$

Fluid 1

$$h_2(T_p - T_{f1}) = (\frac{\dot{m}C_{p1}}{W})(\frac{dT_{f1}}{dx}) + h_1(T_{f1} - T_c) \quad (10)$$

Absorber

$$I\tau\alpha = h_{r1}(T_p - T_c) + h_2(T_p - T_{f1}) + h_3(T_p - T_{f2}) + h_{r2}(T_p - T_{pr}) \quad (11)$$

Fluid 2

$$h_3(T_p - T_{f2}) = (\frac{\dot{m}C_{p2}}{W})(\frac{dT_{f2}}{dx}) + h_4(T_{pr} - T_{f2}) + h_5(T_{f2} - T_r) \quad (12)$$

Porous medium

$$H_{r2}(T_p - T_{pr}) + h_6(T_{pr} - T_r) = h_4(T_{pr} - T_{f2}) \quad (13)$$

Bottom plate

$$h_5(T_{f2} - T_r) + h_6(T_{pr} - T_r) = U_b(T_r - T_a) \quad (14)$$

Where

h_4 = Convection heat transfer coefficient from porous media to fluid in the lower duct, W/m^{2}K

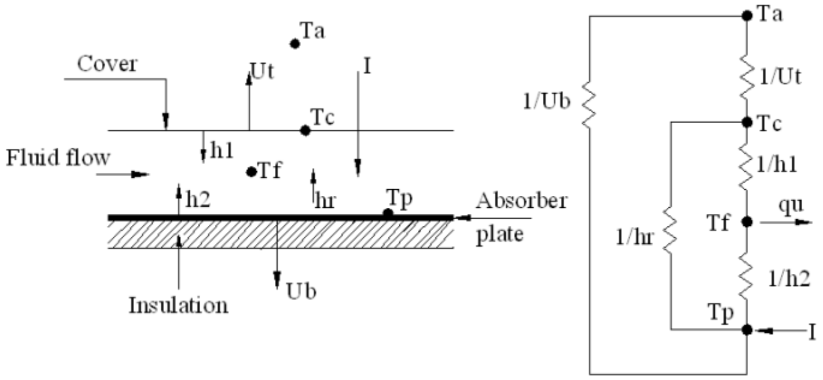
h_5 = Convection heat transfer coefficient from fluid in the lower duct to the base plate, W/m^2 K

h_6 = Convection heat transfer coefficient from porous media to the base plate, W/m^{2}K

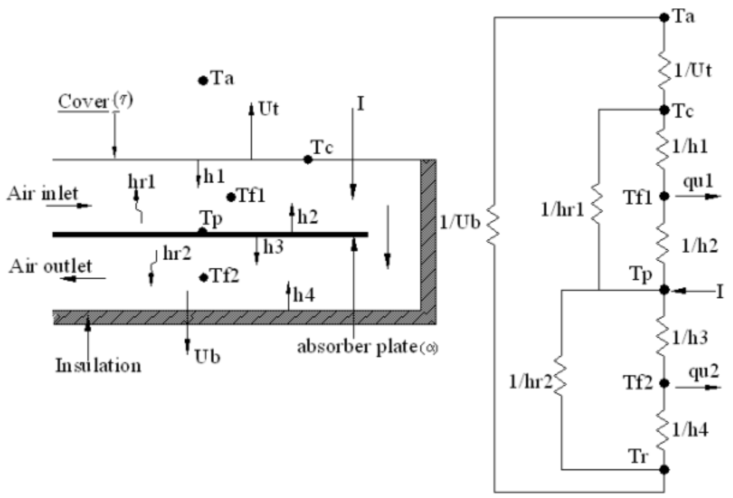
h_{r2} = Radiation heat transfer coefficient from plate to the base plate, W/m^2 K

The various heat transfer coefficients at different components of the air heaters are illustrated in the thermal network in Figure 1.

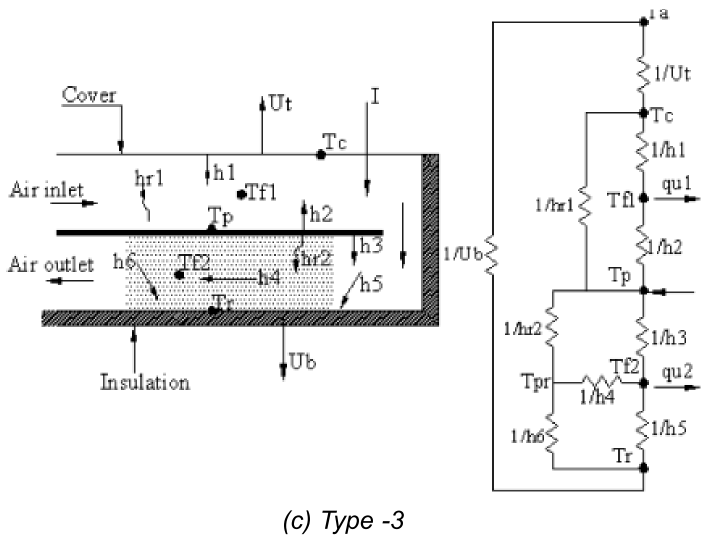
The thermal efficiency which is defined as the ratio of the useful energy to the total incident solar radiation is expressed by the Hottel-Whillier-Bliss equation (Duffie and Beckman, 1991)



(a) Type -1



(b) Type -2



(c) Type -3

Figure 1: Schematic diagram of the solar air heaters with thermal network

$$\eta = \frac{Q_u}{A\,I} = F_R(\tau\alpha) - F_R U \frac{(T_i - T_a)}{I} \quad (15)$$

2.1 Heat transfer coefficients

The convective heat transfer coefficient to the ambient air (Mc Adams, 1954)

$$h_a = 5.7 + 3.8V \quad (16)$$

The radiation heat transfer coefficient from the absorber to the glass cover can be stated as Duffie and Beckman (1991) and Ratna *et al.* (1992):

$$h_r = \frac{\sigma(\overline{T}_p^2 + \overline{T}_c^2)(T_p + T_c)}{\frac{1}{\varepsilon_p} + \frac{1}{\varepsilon_c} - 1} \quad (17)$$

The convective heat transfer coefficients are calculated using the following relations (Kays and Crawford, 1980):

$$h = \frac{Nuk}{D_h} \quad (18)$$

$$Nu = 0.0158(\mathrm{Re})^{0.8} \quad (19)$$

$$\mathrm{Re} = \frac{\dot{m}D_h}{A_f\,\mu} \quad (20)$$

$$D_h = 4\frac{\text{Cross sectional area of the flow}}{\text{wetted perimeter}} \quad (21)$$

The overall heat transfer loss from the collector is the summation of three separate components, the top loss coefficient, the bottom loss coefficient, and the edge loss coefficient. The empirical relations for these coefficients are mentioned by Duffie and Beckman (1991) and Al-Ajlan *et al.* (2003) as follows:

$$U_t = \left\{ \frac{N}{\frac{C}{T_P}\left[\frac{T_P - T_a}{N + f}\right]^e} + \frac{1}{h_a} \right\}^{-1} + \frac{\sigma(T_P + T_a)(T_P^2 + T_a^2)}{(\varepsilon_P + 0.00591Nh_a)^{-1} + \frac{2N + f - 1 + 0.13\varepsilon_P}{\varepsilon_c} - N} \quad (22)$$

$$U_b = \frac{k_b}{x_b} \quad (23)$$

$$U_e = \frac{U_{e'}A_e}{A} = \frac{k_e}{x_e}\left[\frac{2(L + W)H}{LW}\right] \quad (24)$$

$$U = U_t + U_b + U_e \quad (25)$$

Where

C = 520x(1– 0.000051 β^2)

f = (1+0.089 h_a – 0.1166 h_a ε_p)x(1+0.07866 N)

e = 0.43x(1 – (100/Tpm))

2.2 Pressure drop

When the air flows through the channel in the air heater, due to friction the air pressure drops along the length of the flow channel. This pressure drop across the flow duct is given by the following expression (Ratna *et al.*, 1992):

$$p = f\left(\frac{m^2}{\rho}\right)\left(\frac{L}{D}\right)^3 \quad (26)$$

$$f = f_o + y\left(\frac{D}{L}\right) \quad (27)$$

The values of f_o and y are

f_o = 24 / Re, y = 0.9 for laminar flow (Re<2550)

f_o = 0.0094, y = 2.92 $\mathrm{Re}^{-0.15}$ for transitional flow (2550<Re<10^4)

f_o = 0.059 $\mathrm{Re}^{-0.2}$, y = 0.73 for turbulent flow (10^4< Re<10^5)

3. Results and discussion

Numerical values of different parameters such as outlet temperature, efficiency and the pressure drop were computed corresponding to an ambient temperature 33°C, solar radiation 500W/m^2, air velocity 1.5 m/s, inlet temperature of 35°C and different values of mass flow rate. The detailed specifications of the data for all types of solar air heaters considered in this study are given in Table 1 overleaf.

3.1 Single pass flat plate collector (Type 1)

The computed values of the efficiency, outlet temperature and pressure drop for the single pass flat plate collector plotted against different values of mass flow rate for two flow channel lengths are given in Figures 2 to 4. It is shown that the efficiency increased with increasing mass flow rate and that due to the increase of heat supplied, while the outlet temperature decreased with increasing mass flow rate.

It can be said that the collectors will be more efficient operating at relatively high flow rates, and at relatively low collector temperature rise. The reason for this is that the higher the collector temperature, the more heat will be lost out the glazing, heat lost out the glazing reduces the heat output of the collector. In addition, the efficiency decreased with increasing the collector length from 1.5m to 2.4m (Figures 2 and 3).

Figure 4 illustrates the variation of the pressure drop with the mass flow rate for collector flow length of 1.5m and 2.4m. It is shown that the pressure drop is a function of mass flow rate and the collector length. Pressure drop increases with increas-

Table 1: Specification for solar air heaters

Collector tilt angle (degree)	10
Collector length (m)	2.4 and 1.5
Collector width (m)	1.2
Depth (m)	0.01, 0.015, 0.02 and 0.035 for type 1
Upper depth (m)	0.035 for type 2 and 3
Lower depth (m)	0.03, 0.045, 0,06 and 0.075 for type 2 and 3
Plate type	Flat plate
Absorber material	Black steel, α = 0.9 and ε = 0.85
Cover material	Ordinary clear glass, τ = 0.85
Number of cover	1
Insulation material	Fiber glass , k = 0.045 W/m.k
Back insulation thickness (m)	0.05
Edge insulation thickness (m)	0.05
Porous medium	Glass-wool of 0.8 porosity for type 3

ing mass flow rate and collector length which is due to the increase of pumping expand in the collector.

The variation in efficiency and pressure drop with flow channel depth for different collector lengths 2.4m and 1.5m are displayed in Figures 5 and 6 respectively at fixed mass flow rate of 0.04 kg/sec. Figure 5 indicates that at fixed mass flow rate the efficiency is decreased with the increase of flow channel depth, and this effect is more predominant for longer flow channel and that due to increase of the collector losses. Figure 6 illustrate that the pressure drop increase with the decrease in the flow depth, and this increase is more for longer channel flow.

The variation in outlet temperature with the flow channel depth is displayed in Figure 7, which indicates that the outlet temperature can be increased with decreasing flow depth.

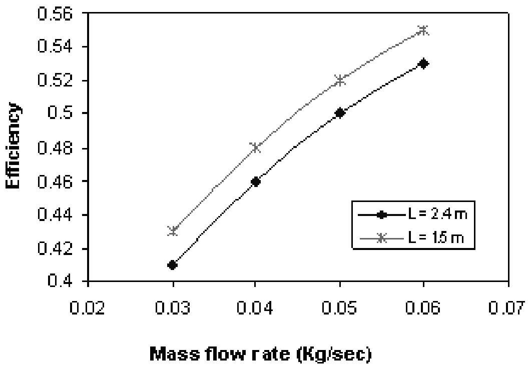


Figure 2: Variation of efficiency with mass flow rate (type 1)

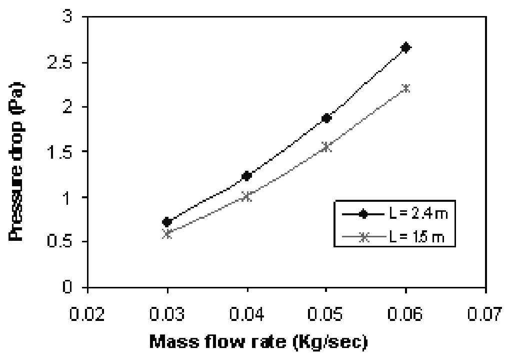


Figure 4: Variation of pressure drop with mass flow rate (type 1)

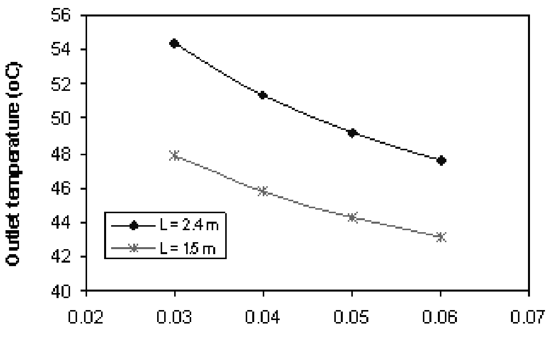


Figure 3: Variation of outlet temperature with mass flow rate (type 1)

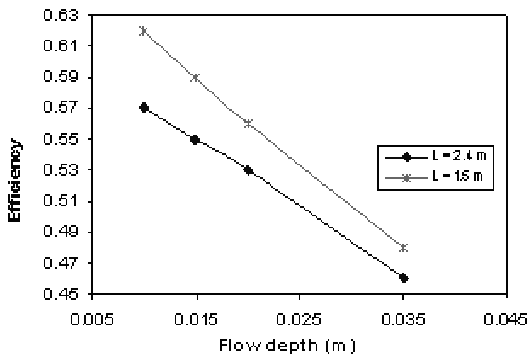


Figure 5: Variation of efficiency with channel flow depth (type 1)

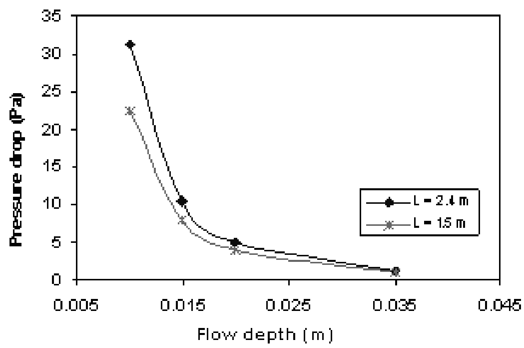


Figure 6: Variation of pressure drop with channel flow depth (type 1)

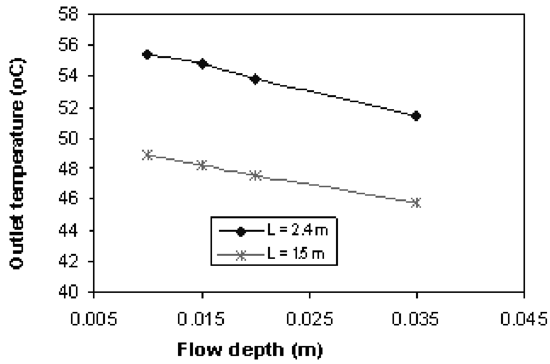
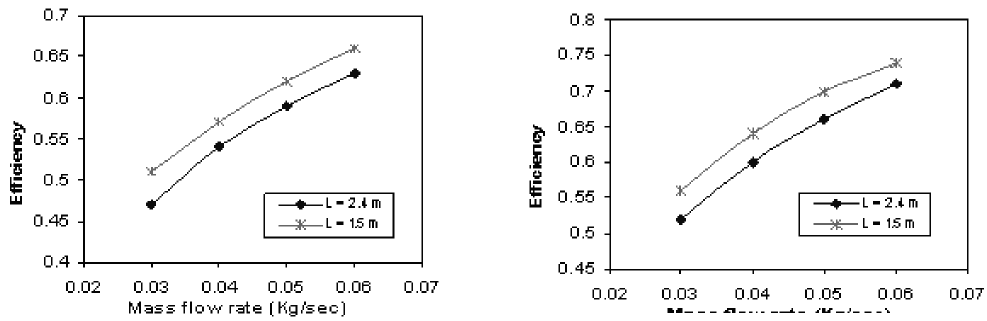


Figure 7: Variation of outlet temperature with channel flow depth

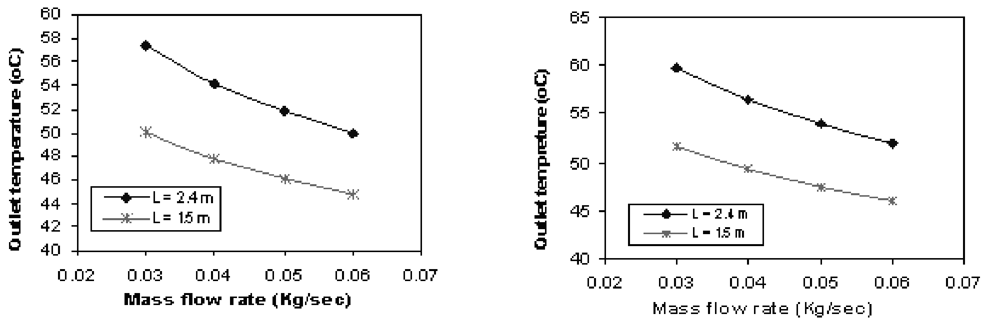
3.2 Double pass flat plate collector with and without porous media (type 2 and 3)

The variation of efficiency, outlet temperature and pressure drop with different mass flow for type 2 and 3 are shown in Figures 8-10. With the increase in mass flow rate the thermal efficiency of the system increases**.** On the other hand the outlet temperature is decreased with the increase of mass flow



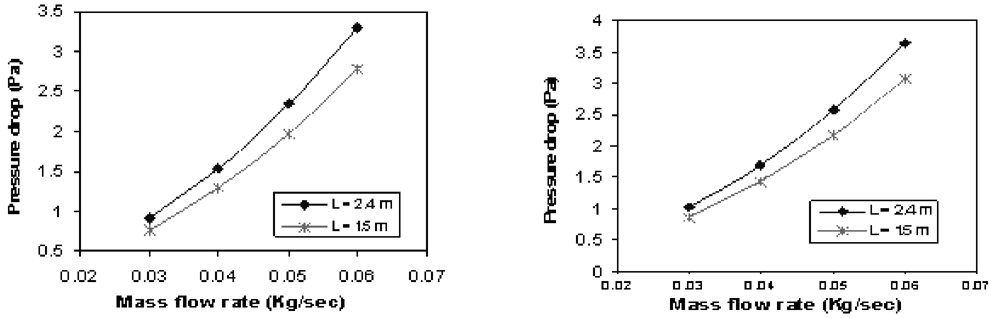
Type- 2 *Type -3*

Figure 8: Variation of efficiency with mass flow rate



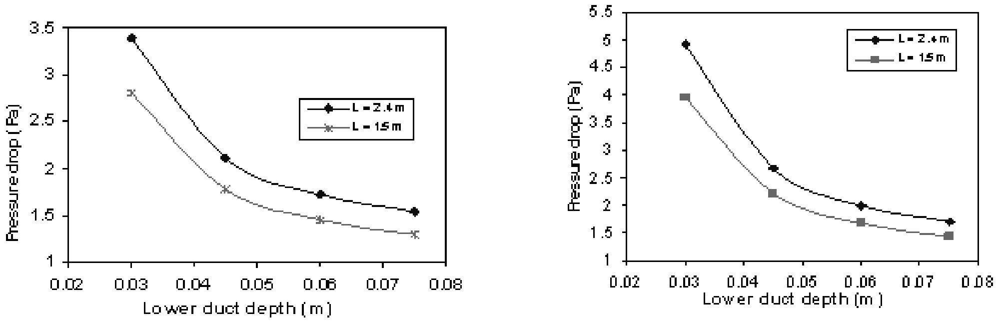
Type- 2 *Type -3*

Figure 9: Variation of outlet temperature with mass flow rate



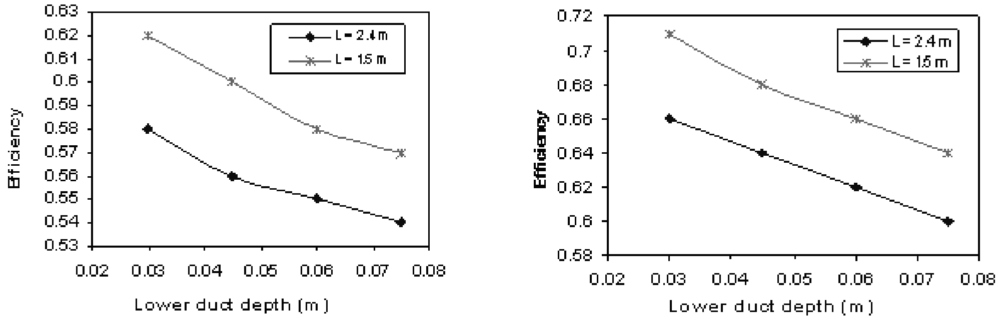
Type- 2 *Type -3*

Figure 10: Variation of pressure drop with mass flow rate



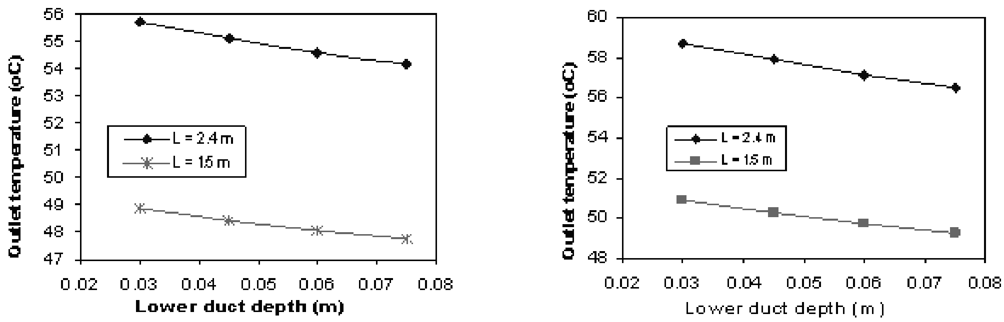
Type-2 *Type- 3*

Figure 11: Variation of pressure drop with lower channel depth



Type -2 *Type- 3*

Figure 12: Variation of efficiency drop with lower channel depth



Type -2 *Type- 3*

Figure 13: Variation of outlet temperature with lower channel depth

rate. By comparing the values of the efficiency and outlet temperature for type 2 and 3, it is found that the use of porous media in type 3 increases the system efficiency by 5-8 % and the outlet temperature from 3-7°C. This increase cause an increase in the pressure drop for type 3, which means increasing of the cost of the pumping power expanded in the collector as shown in Figure 10.

The effect of lower channel depth on the pressure drop, efficiency and outlet temperature for types 2 and 3 is conducted and displayed on Figures 11-13 for fixed mass flow rate of 0.04kg/s and two different flow channel lengths of 2.4m and 1.5m. The graphs illustrate that, with the increase of lower channel depth the pressure drop decreased as well as the efficiency and the outlet temperature also decreased.

The short channel length 1.5m has an effect on the pressure drop and system efficiency, since it increases the system efficiency at the same time it decreases the pressure drop. But the outlet temperature is higher while long channel length 2.4m than the short one 1.5m.

An analysis of the values in different Figures for type 1-3 at the same configuration and parameters; mass flow rate 0.04kg/s, channel length 2.4m, flow depth 0.035m and 0.045m lower channel depth in double flow (type 2 and 3). It is found that the system thermal efficiency increases by 10-12% in double flow mode type 2 than single flow type 1 and increase by 8% after using porous media in the lower channel type 3 than the double flow without porous media type 2. This increase in efficiency in double pass mode due to the increased heat removal for two channels compared to one flow channel in single pass operation, on the other hand the use of porous medium increases the heat transfer area which contributed to the higher efficiency.

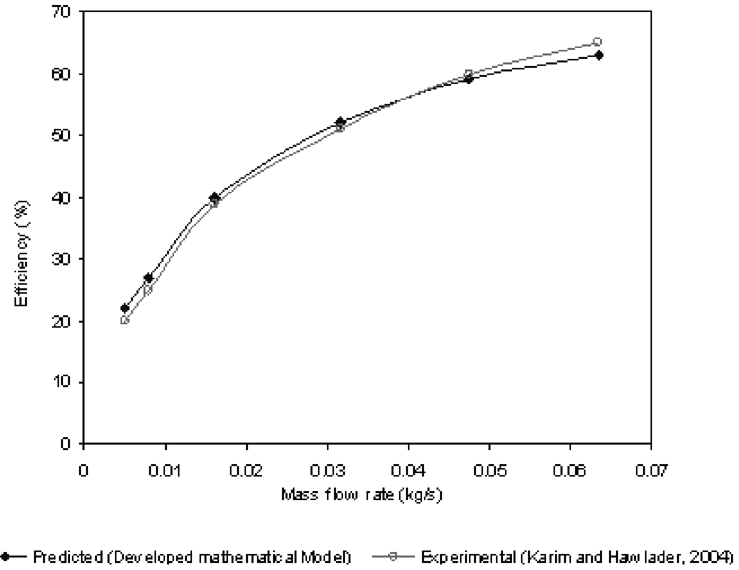


Figure 14: Efficiency variation with mass flow rate

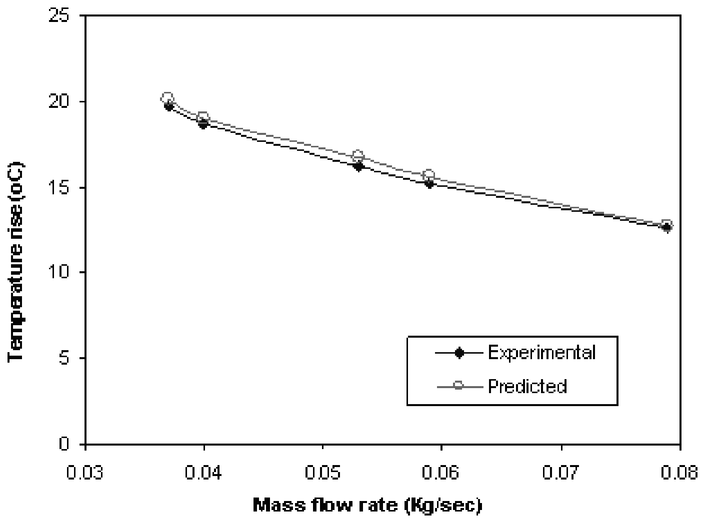


Figure 15: Effect of mass flow rate on temperature rise of double pass

Therefore the system efficiency increases by 18% after using porous media in the lower channel type 3 than the single flow pass type 1. At the same time the pressure drop will be increased, due to the increase of pumping power expand in the collector as a result of using the porous medium.

4. Validation

Validation and verification focus on the comparison between the predicted output and the experimental results for each solar air heater type. The experimental results has been done by Karim and Hawlader, (2004) for flat plate collector in single and double pass mode, while El-Radi, (2001) has done an experiment on the double pass flat plate collector with porous media in the lower duct.

The output from the mathematical simulation has been run according to same configuration and parameters done in experimental work by Karim and Hawlader, (2004) and Elradi, (2001).

In Single Pass Flat Plate Collector Figure 14 shows predicted and the experimental output results of the variation of efficiency with flow rate for flat plate collector in single mode. The efficiency is a function of mass flow rate hence, increasing the mass flow rate increases efficiency. The same result has been obtained by experimental results done by Karim and Hawlader, (2004). The correlation between experimental and predicted efficiencies (R^2= 99.57%, $P<0.001$) for single flow mode is shown in Figure 14, where the predicted output and the experimental results shows a close agreement with error difference between 1% to 5%.

For Double Pass Flat Plate Collector El-Radi (2001) found the effect of mass flow rates on temperature rise of double pass flat solar collector at 540W/m^2 solar radiation and 33°C ambient temperature. Under the same conditions the effect of mass flow rates on temperature rise of double pass flat solar collector has been predicted by the mathematical model. The results are given in Figure 15. By comparing the experimental and predicted results it is found that the temperature rise is decreased with the increase of mass flow rate and a close correlation has occurred between the experimental and the predicted results (R^2= 99.76%, $P<0.001$).

Karim and Hawlader (2004) performed experimental work on the double pass flat plate collector, Figure 16 shows predicted and experimental output results of the variation of efficiency and outlet temperature with flow rate for double pass flat plate collector. It is shown that the efficiency increased by increasing the mass flow rate while the outlet temperature decreased by increasing the mass flow rate. This result is similar to that obtained by the experimental work done by Karim and Hawlader (2004). A good correlation (R^2=99.7%, $P<0.001$) and (R^2=98.5%, $P<0.001$) has been found between the experimental and predicted efficiencies and outlet temperature respectively as shown in Figure 16.

For Double Pass Flat Plate Collector with Porous Medium complete measured experimental results from El-Radi (2001) were chosen for comparison with the developed program output. The experimental result was measured during the determination of thermal performance of double pass solar air collector with and without porous media. In this project the solar incidence was varied between 400 and 620 W/m^2 and the mass flow rate was varied between 0.04 to 0.09 kg/s. The average measured temperature distribution at solar radiation 500 W/m^2 with different values of mass flow rate is shown in Table 2 from El-Radi, (2001).

The predicted output was calculated by using the developed program, the entered data of solar radiation, tilt angle, ambient and inlet temperature

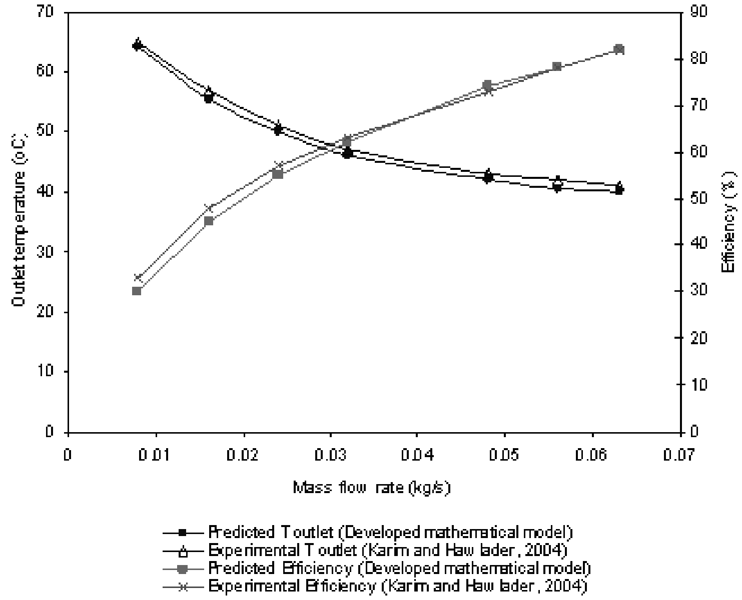


Figure 16: Variation of efficiency and outlet temperature with flow rate

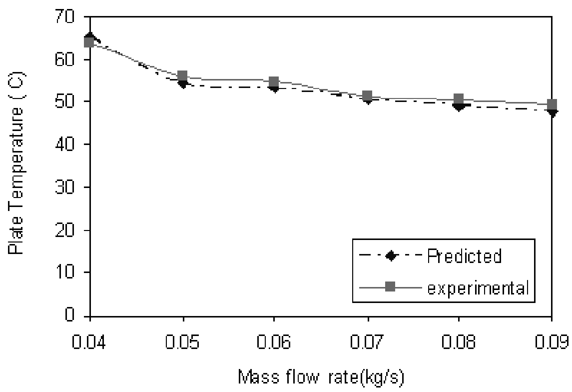


Figure 17: Effect of mass flow rate on plate temperature

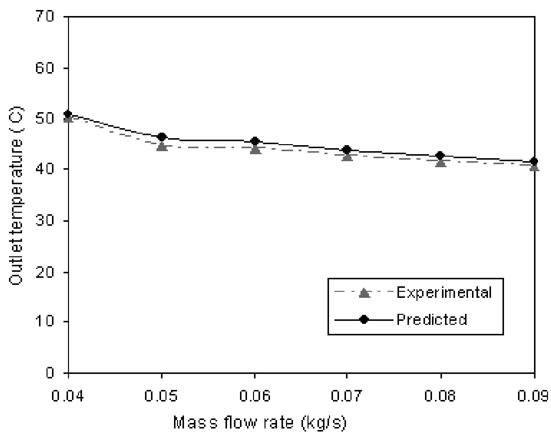


Figure 18: Effect of mass flow rate on outlet temperature

and collector specification were typical to that mentioned in El-Radi (2001). Close agreement has been achieved between the experimental results by El-Radi (2001), and the predicted output as shown in Figures 17, 18 and 19.

6. Conclusion

A mathematical simulation to predict the effect of different parameters on system thermal performance and pressure drop, for flat plate collector in single and double flow mode with and without using a porous media have been conducted. It is found that increasing the mass flow rate through the air heaters results in higher efficiency but also it is increased pressure drop. On the other hand decreasing the channel flow depth results in increasing the system efficiency and outlet temperature at the same time it increased the pressure drop.

Table 2: Average temperature distribution in the collector at solar radiation 500 W/m^2,
Source: El-Radi (2001)

Mass (kg/s)	0.04	0.05	0.06	0.07	0.08	0.09
T ambient	33.7	33.5	33.8	33.6	33.1	33.8
T inlet	38.68	35.28	35.55	35.02	34.64	34.12
T outlet	50.48	44.72	43.97	42.62	41.38	40.66
T plate	65.37	54.3	53.38	50.43	49.1	47.7
T fluid	44.02	39.92	42.2	38.5	37.62	36.98

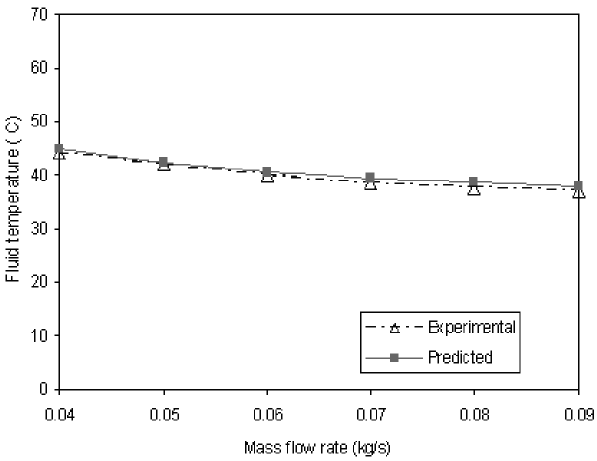


Figure 19: Effect of mass flow rate on fluid temperature

The channel length also has an effect on the thermal efficiency hence the system efficiency is increased more for short channel length than the long one, while the pressure drop is less than the pressure drop in long channel length.

The double flow is more efficient than the single flow mode due to the increased heat removal for two channels compared to one flow channel, and the using of porous media increase the system efficiency and the outlet temperature hence the use of porous media increases the heat transfer area. This increment will result in the increase of the pressure drop thus increasing the pumping power expanded in the collector.

References

Al-Ajlan, S.A., Al-Faris, H., and Khonkar, H. A. Simulation modeling for optimization of flat plate collector design in Riyadh, Saudi Arabia, *Renewable Energy*, 28 (2003), 1325-1339

Bashria A. A, N. M. Adam, S. M. Sapuan, M. Daud, H. Omar, H. M. and F Abas. Prediction of Thermal Performance of Solar Air Heaters by Internet-Based Simulation, *Part A: Journal of Power and Energy, Proceedings of the Institution of Mechanical Engineers* 218 (2004a), pp. 579-587.

Bashria A. A. Yousef, N. M. Adam, M. Daud, H. Omar, M. H. M. Ahmed and Elradi A. Musa, Expert System Based Prediction of the Performance of Solar Air Collectors in Malaysia, *Journal of Energy and Environment*, 3 (2004b), pp. 91-101

Choudhury. C, CHauhan P. M. and Garg H.P. , Performance and Cost Analysis of Two-Pass Solar Air Heaters, *Heat Recovery System and CHP*, 15 (1995) No. 8, pp. 755-773.

Duffie, J.A. and Beckman, W.A., Solar Engineering of Thermal Processes, (Toronto, John Wiley 1991.)

El-Radi A. M. Thermal performance of the double pass solar air collector with and without porous media. M.Sc. Thesis, (Faculty of Engineering, Universiti Kebangsaan Malaysia. Bangi, Malaysia 2001)

Karim M.A., Hawlader M.N.A. Development of Solar Air Collectors for Drying Applications. *Energy Conversion and Management,* 45 (2004) pp 329-344.

McAdams, W.H., Heat transmission, (McGraw-Hill, New York, 3rd ed., 1954)

Mohd, Y.H.; Baharudin, Y.; Daud, W.R.W. Concept Model of A 100 KWP PV Village in Malaysia, *Commercialization of Malaysian RandD, Kementeriam Sams, Teknologi dam Alam Sekitar*, 1 (1999) pp. 206- 211.

Ratna Verma, Ram Chandra and H. P. Garg, Parametric Studies on the Corrugated Solar Air Heaters With and Without Cover, *Renewable Energy*, 1 No.3/4 (1991) pp. 361-371

Ratna Verma, Ram Chandra and H. P. Garg. Technical note- Optimization of Solar Air Heaters of Different Designs, *Renewable Energy,* 2 (1992) No.4/5, pp. 521-531.

W.M. Kays and M.E. Crawford, Convective Heat and Mass Transfer. McGraw – Hill, New York 2nd edition (1980), p. 213.

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