PERFORMANCE ANALYSIS FOR V-GROOVE ABSORBER

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Abstract

This paper concerns 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, on V-groove absorber at single and double flow mode. This study has been conducted by using a developed internet based mathematical simulation. It concludes that in V-groove absorbers types, the double flow mode is 4 - 5% more efficient than the single mode. On the other hand the use of porous media in double flow increases the air heater efficiency to be 7% more efficient than an air heater in single mode, and 2 - 3% more efficient in double flow mode without porous media. The analysis and the graphs obtained in this paper can be a helpful tool for a design engineer to construct economical and efficient solar air heaters with technical dimensions.

Keywords: Single and double flow V-groove absorber, porous media, thermal performance, pressure drop, flow channel depth

Introduction

Solar air heaters form the major component of solar energy utilization systems which absorb the incoming solar radiation, converting it into thermal energy at the absorbing surface, and transferring the energy to a fluid flowing through the collector. Solar air heaters, because of their inherent simplicity, are cheap and the most widely used collection device (Momin *et al*., 2002). It has a wide range of applications in drying agricultural products, industrial process heating in textile and papers, space heating and greenhouse heating. Several designs of solar air collectors have been studied over the years, hence the design of suitable air collectors is one of the most important factors controlling the economics of solar drying. Therefore 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. Previous researchers have focused on the effects of design and operational parameters, type of flow passes, number of glazing and types of absorbers flat, corrugated or finned, on the thermal performance of solar air heaters (Verma *et al*., 1991; Verma *et al*., 1992; Choudhury *et al*., 1995 and Karim *et al*., 2004). Sopian Supranto *et al*. (1999) studied the use of porous media in doublepass solar air heaters and

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found that the use of porous media in the second channel of solar air heaters increases the heat transfer area. In addition, this type of air heater has a higher thermal performance compared to the conventional single-pass solar air heater. An internet based mathematical simulation has been conducted and developed by Bashria *et al.* (2004a, 2004b) to predict the thermal performance for different designs of solar air heaters with flat or V-groove absorbers, single, double or triple glass covers, in single pass or double pass double ducts with or without porous media.

This study uses the aforementioned developed program by Bashria *et al*. (2004a, 2004b) to find the effect of different parameters, such as mass flow rate, flow channel depth and collector length on the system thermal performance and pressure drop through the collector, for V-groove absorbers in single and double passes with and without using a porous media.

Theoretical Analysis

To model the solar air types and obtain their relative equations, a number of simplifying assumptions has been made to lay the foundation without obscuring the basic physical situation. These assumptions are as follows:

1. Performance is steady state.
2. There is no absorption of solar energy by a cover insofar as it affects losses from the collector.
3. Heat transfer fluid is considered a non-participating medium.
4. The radiation coefficient between the two air duct surfaces is found by assuming a mean radiant temperature equal to the mean fluid temperature.
5. Loss through front and back are to the same ambient temperature.

In single pass V-groove absorber, energy is transferred from the plate to the bottom of the cover through both the radiation heat transfer coefficient h_r and the convection heat transfer coefficient h_1, and to the fluid at a temperature at T_f through the convection heat transfer coefficient h_2. Energy is transferred from the fluid to the ambient air at T_a through the back loss coefficient U_b. Energy is lost to the ambient air through the combined convection and radiation coefficient U_t. Schematic drawings for single pass V-groove absorber and thermal net work are shown in Figure 1(a). The steady-state energy equations yield the following equations (Bashria *et al*., 2004a):

Collector cover

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

V-groove absorber

$$h_2 (T_p - T_f) + h_1 (T_p - T_c) + h_r (T_p - T_c) = I\tau\alpha \quad (2)$$

Fluid medium

$$h_2 (T_p - T_f) + U_b (T_a - T_f) = \left(\frac{mC_p}{W}\right)\left(\frac{dT_f}{dx}\right) \quad (3)$$

In double pass double duct energy is transferred from the plate at T_P to the cover through the radiation heat transfer coefficient h_{r1} and the convection heat transfer h_1, and to the fluid in the upper duct flowing through the V-groove 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 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. Figure 1(b) illustrates the double pass double duct V-groove absorber and the thermal net work. The steady-state energy balance on the cover, the plate and the fluid in the upper and lower ducts (with the assumption that the heat transfer fluid is a non participating medium) gives the following equations:

Collector cover

$$h_1 (T_p - T_c) + h_r (T_p - T_c) = U_t (T_c - T_a) \quad (4)$$

V-groove absorber

$$h_2 (T_p - T_{f1}) + h_1 (T_p - T_c) + h_r (T_p - T_c) + h_3 (T_p - T_{f2}) + h_{r2} (T_p - T_r) = I\tau\alpha \quad (5)$$

Fluid medium in the upper passage

$$h_2 (T_p - T_{f1}) = \left(\frac{mC_{p1}}{W}\right)\left(\frac{dT_{f1}}{dx}\right) \quad (6)$$

Fluid medium in the lower duct

$$h_3 (T_p - T_{f2}) = \left(\frac{mC_{p2}}{W}\right)\left(\frac{dT_{f2}}{dx}\right) + h_4 (T_{f2} - T_r) \quad (7)$$

Bottom plate

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

In case of double flow double duct packed with porous media (glass-wool of 0.8 porosity) energy is transferred from the plate at T_p 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 and from the fluid at T_{f2} to the bottom plate through convection heat transfer coefficient h_5. Also energy is transferred from the porous media to the bottom plate through the convection heat transfer coefficient heat transfer coefficient h_6. Figure 1(c) illustrates the double pass double duct V-groove absorber with porous media in the lower duct and the thermal net work. The steady-state energy balance gives the following equations

Collector cover

$$h_1 (T_p - T_c) + h_r (T_p - T_c) = U_t (T_c - T_a) \quad (9)$$

V-groove absorber

$$h_2 (T_p - T_{f1}) + h_1 (T_p - T_c) + h_r (T_p - T_c) + h_3 (T_p - T_{f2}) + h_{r2} (T_p - T_{pr}) = I\tau\alpha \quad (10)$$

Fluid medium in the upper passage

$$h_2 (T_p - T_{f1}) = \left(\frac{mC_{p1}}{W}\right)\left(\frac{dT_{f1}}{dx}\right) \quad (11)$$

Fluid medium in the lower duct

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

Porous media

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

Bottom plate

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

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).

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

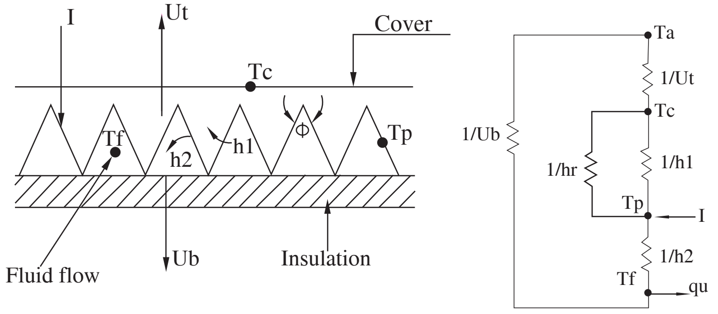
Heat Transfer Coefficients

In order to solve the models, the convective heat transfer coefficient for air flowing over the outside surface of the top glass cover and inside channel are needed. The following correlation proposed by McAdams (1954) for air flowing over the outside surface of the glass cover is used to predict the convective heat transfer coefficient

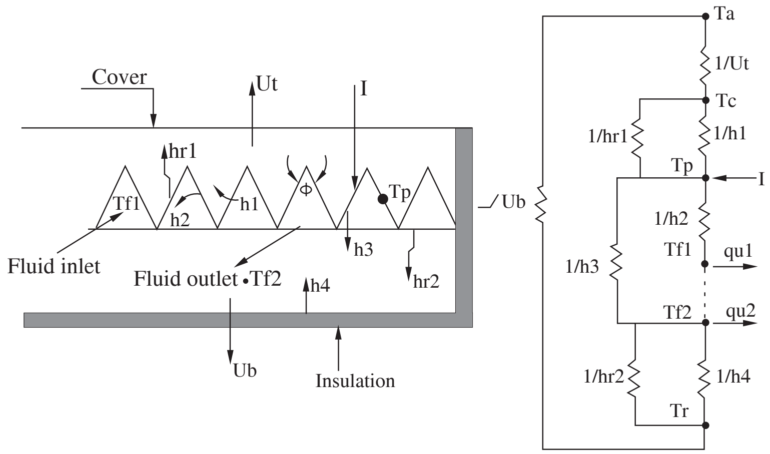
$$h_a = 5.7 + 3.8V \quad (16)$$

where h_a is the convective heat transfer coefficient, and V is the wind velocity.

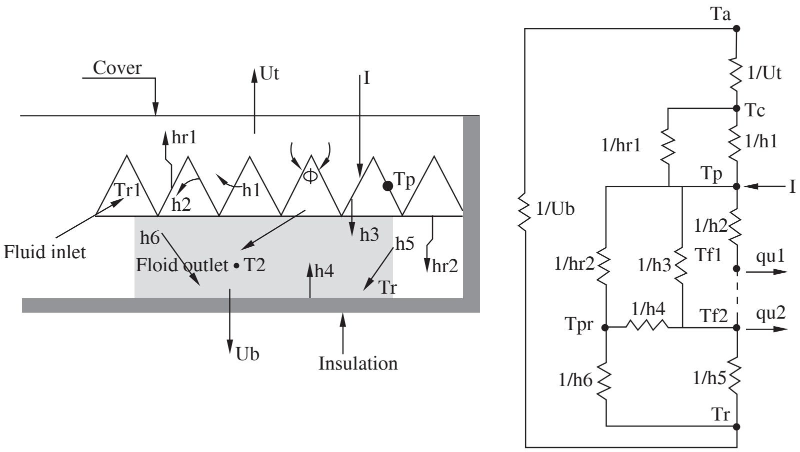
The radiation heat transfer coefficient from the absorber to the glass cover can be stated as follows using Duffie and Beckman, (1991) and Verma *et al*. (1992)



(a) Type -1



(b) Type -2



(c) Type -3

Figure 1. Schematic diagram of V-groove solar air heaters with thermal network

$$h_r = \frac{\sigma(\overline{T}_p^{\,2} + \overline{T}_c^{\,2})(T_p + T_c)}{\frac{1}{\varepsilon_{p'}} + \frac{1}{\varepsilon_c} - 1} \quad (17)$$

In the V-groove the radiation is reflected several times in the grooves, each time absorbing a fraction of the beam. This multiple absorption gives an increase in the solar absorptance but at the same time increases the long-wavelength emittance (Ruslan *et al*., 2003).

$$\varepsilon_{p'} = \frac{2\varepsilon_p}{1+\varepsilon_p} \quad (18)$$

The convective heat transfer coefficients are calculated using the following relations

$$h = \frac{k}{D_h} Nu \quad (19)$$

where Nu is Nusselt number, k is the air thermal conductivity and D_h is the equivalence diameter of the channel. Nusselt number for laminar flow region (Re < 2300), transition flow region (2300 < Re < 6000), and turbulent flow region respectively as mentioned by (Naphon, 2005) are:

$$Nu = 5.4 + \frac{0.00190(\mathrm{Re}\,\mathrm{Pr}(\frac{D_h}{L}))^{1.71}}{1 + 0.00563(\mathrm{Re}\,\mathrm{Pr}(\frac{D_h}{L}))^{1.17}} \quad (20)$$

$$Nu = 0.116(\mathrm{Re}^{2/3} - 125)\,\mathrm{Pr}^{1/3} \left(1 + (\frac{D_h}{L})^{2/3}\right)(\frac{\mu}{\mu_w})^{0.14} \quad (21)$$

$$Nu = 0.018\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.4} \quad (22)$$

where Re is the Reynolds number, Pr is Prandtl Number

$$\mathrm{Re} = \frac{\dot{m} D_h}{A_f \mu} \quad (23)$$

$$D_h = 4\frac{\text{Cross sectional area of the flow}}{\text{wetted perimeter}} \quad (24)$$

For the V-groove absorber it is found that the hydraulic diameter equals (Bashria *et al.*, 2004a):

$$D_h = \frac{2H_v \sin(\phi/2)}{1 + \sin(\phi/2)} \quad (25)$$

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 (Verma *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

$f_o = 24/\mathrm{Re}$, $y = 0.9$
for laminar flow (Re < 2550)
$f_o = 0.0094$, $y = 2.92\mathrm{Re}^{-0.15}$
for transitional flow ($2550 < \mathrm{Re} < 10^4$)
$f_o = 0.059\mathrm{Re}^{-0.2}$, $y = 0.73$
for turbulent flow ($10^4 < \mathrm{Re} < 10^5$)

Simulation Procedure

A prototype internet-based computer program was developed to be used as a tool to support the design of solar air heaters used for drying agricultural products in Malaysia. Dreamweaver™ combined with the Active Server Pages

(ASP) as an extension environment, have been chosen as a development software tool. The programmatic code was written by VBScript as scripting language to execute commands on a computer (Bashria *et al.,* 2004a). Rules have been used to specify a set of actions performed for a given situation. The required data are classified into three groups, the general input data concerning metrological conditions, the collector characteristics data that contain all specific manufacturing attributes related to the collector and the energy characteristics data which contain measured data about inlet temperature and mass flow rate related to the transfer media inside the collector (Bashria *et al*., 2004b). The aforementioned internet based computer program is in the process of continual development but it can be accessed through the website http://www.eng.upm.edu.my/home/bashria/public_html/default.asp

Initial work for simulation was based on determining the types and components of solar air heaters, its use, advantages and disadvantages. In developing the numerical procedure, several steps and processes are involved. The primary task can be categorized into five steps, as follows: Identification and characterization of the problem, conceptualization, mathematical model, implementation and testing. The flow cart of such development processes is shown in Figure 2. The numerical algorithm and mathematical model that used to calculate the thermal performance were based on Hottel-Wrillier-Bliss equation and on the developed energy balance equations at various components of the collector models, along with the different heat transfer coefficients at their surfaces which have mentioned previously in the theoretical analysis section. These equations have been transferred to syntax code and implanted into the developed computer program.

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 500 W/m^2, air velocity 1.5 m/sec, inlet temperature of 35°C and different values of mass flow rate.

Validation

Validation and verification focus on the com-parison between the predicted output and the experimental results for each type of solar air heater. The experimental results have been carried out by Karim *et al*. (2004), for V-groove absorber in single and double pass mode. The output from the mathematical simulation has been run according to the same configuration and parameters used in experimental work done by Karim *et al*. (2004). A great correlation has occurred between the experimental and the predicted efficiencies ($R^2 = 99.5\%$, $P < 0.001$) for single flow mode as shown in Figure 3. Also a good correlation ($R^2 = 98.7\%$, $P < 0.001$) and ($R^2 = 97.5\%$, $P < 0.001$) has been found between the experimental and predicted efficiencies and outlet temperature respectively as shown in Figure 4.

This study of double pass V-groove absorber with porous media has not been conducted before either theoretically or experi-mentally. The theoretical study has been conducted by using the program developed in this study.

Results and Discussion

Figure 5 shows the variation of efficiency with mass flow rate for V-groove absorber in single pass, double pass and double pass mode with porous media. From the figures, it can be seen that the efficiency of the air heater is strongly dependent on the air flow rate. The efficiencies of all three air heaters increased constantly, then tended to approach a constant value. This figure clearly shows that the double flow mode is 4 - 7% more efficient than the single flow mode. This increase in efficiency in double pass mode is due to the increased heat removal from two flow channels compared to one flow channel in single pass operation. On the other hand, the use of porous media in double flow increases the air heater efficiency to 7% more efficiency than the air heater in single mode and more than 2 - 3% efficiency in double flow mode without porous media. Hence, the use of porous media increases the heat transfer area which contributed to

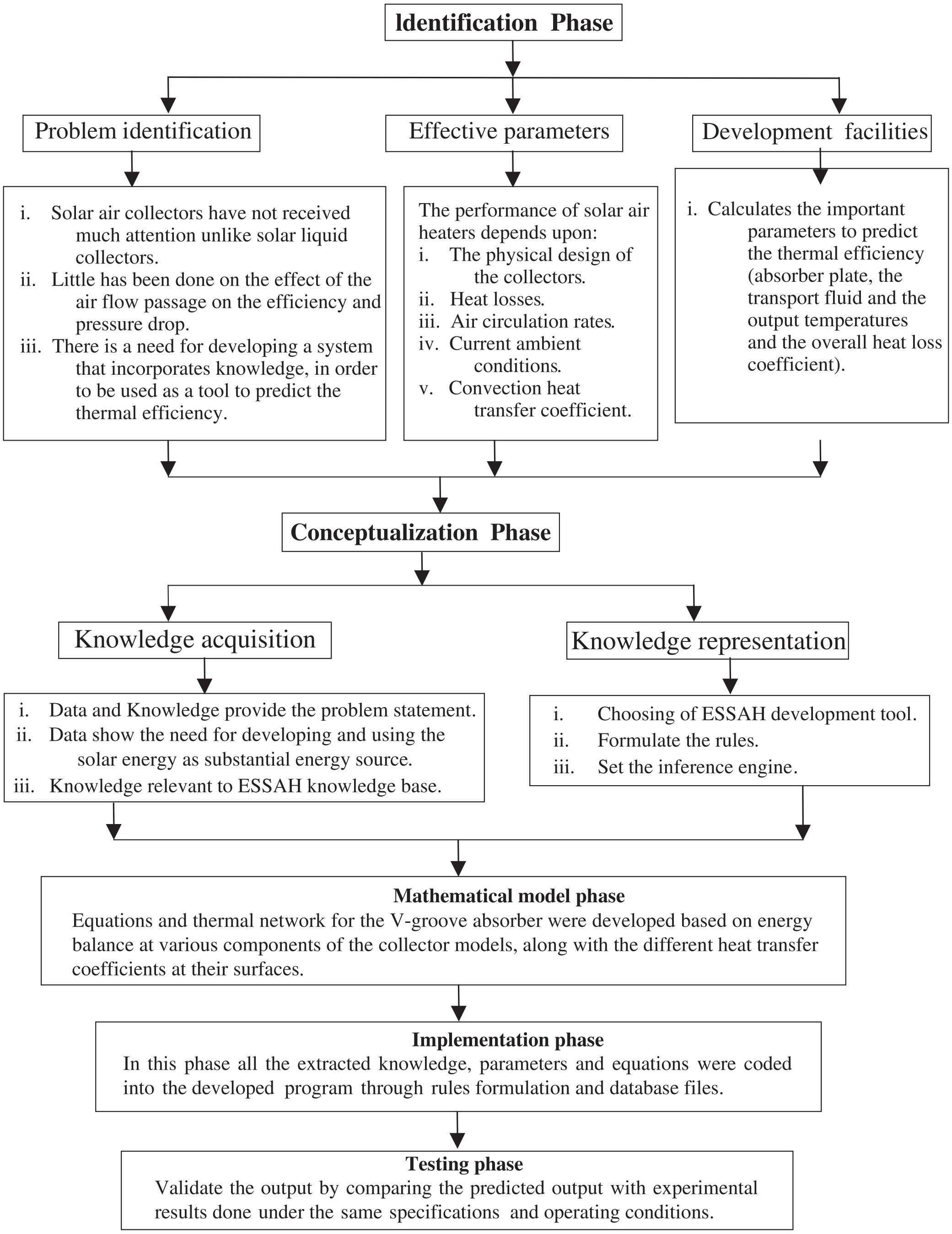


Figure 2. Simulation procedure flow chart

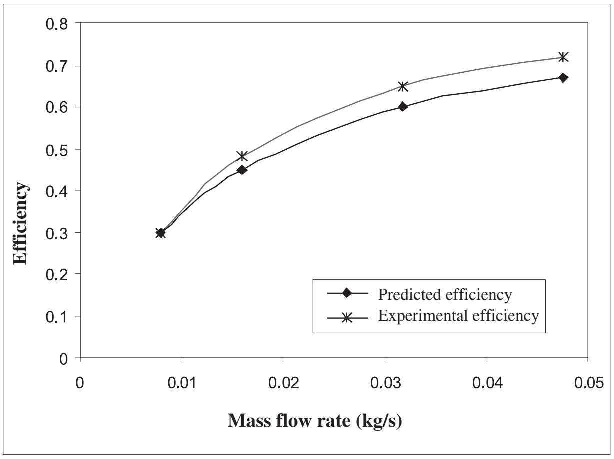


Figure 3. Experimental and predicted output for single flow mode

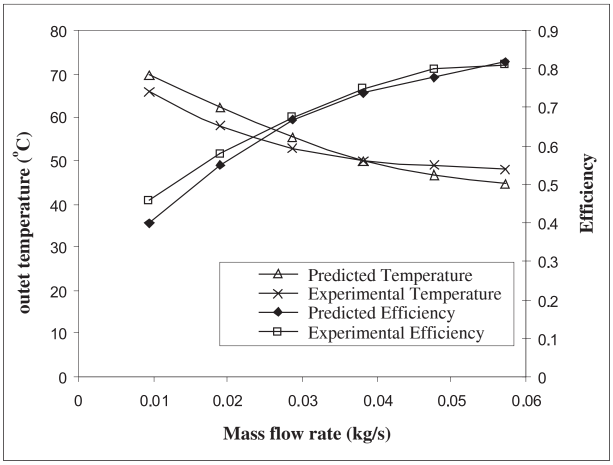


Figure 4. Experimental and predicted output for double flow mode

the higher efficiency.

Hence, the outlet temperature is an important parameter for drying applications; the outlet temperature was investigated for a wide range of flow rates. Figure 6 shows the variation of outlet temperature with flow rate. As expected the outlet temperature of the flowing air through the collector decreased with increased flow rate, but after a flow rate of about 0.05 kg/sec for single flow mode, and a flow of about 0.065 kg/sec for double flow mode the rate of the temperature drop became lower. Figure 7 illustrates the variation of the pressure drop with mass flow rate; it shows that the pressure drop is a function of the mass flow rate; hence it is increased by increasing the mass flow rate. The figure clearly shows that the value of the pressure drop in double flow mode almost doubles the value of the pressure drop in single flow. At the same time the use of porous media in the double flow V-groove absorber increases the pressure drop from 3 to 25 Pa more than the pressure drop in the double flow V-groove absorber without the porous media.

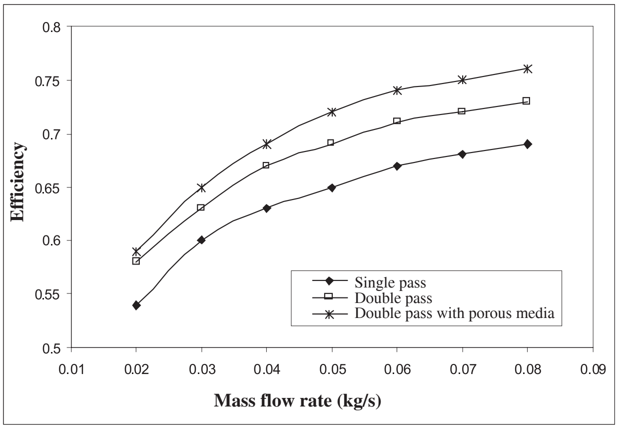


Figure 5. Efficiency variation with mass flow rate for single pass, double pass and double pass with porous media in V-groove absorber

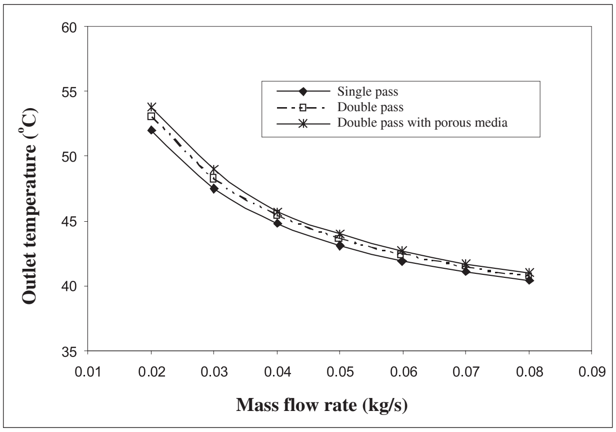


Figure 6. Outlet temperature variation with mass flow rate for single pass, double pass and double pass with porous media in V-groove absorber

The variation in efficiency and pressure drop with flow channel depth for single pass mode in V-groove absorber at different collector lengths (1.8 m, 1.5 m, and 1 m.) is displayed in Figures 8 - 9 respectively at a fixed mass flow rate of 0.034 kg/sec. Figure 8 indicates that at a fixed mass flow rate the efficiency decreases with the increase of the flow channel depth. Science the deep air channel depth reduces the heat transfer to the flowing air which resulting in low efficiency of the system, and this effect is more predominant for a longer flow channel. Figure 9 illustrates that the pressure drop increases with a decrease in the flow depth, and this increase is greater for a longer channel flow. The variation in outlet temperature with the flow channel depth is displayed in Figure 10, which indicates that the outlet temperature increases with the decrease in the flow depth.

The effect of a different upper channel depth on the pressure drop, efficiency and the outlet temperature for double pass mode with and without porous media is conducted, for a fixed mass flow rate of 0.03 kg/sec, a constant lower

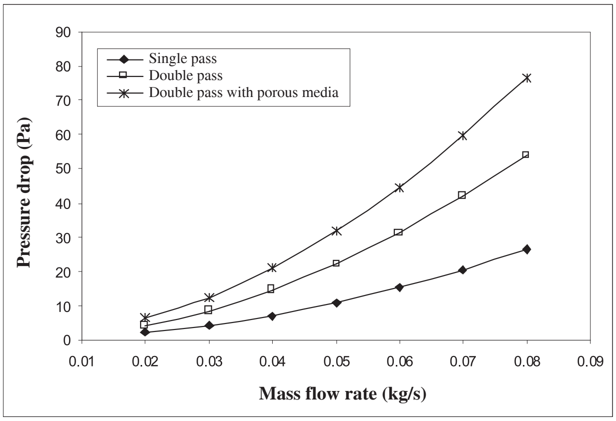


Figure 7. Pressure drop variation with mass flow rate for single pass, double pass and double pass with porous media in V-groove absorber

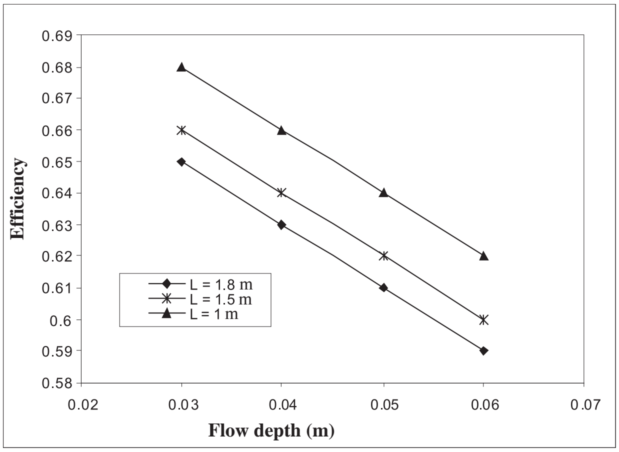


Figure 8. Efficiency variation with flow channel depth for single pass in V-groove absorber

flow depth of 0.025 m channel and two different channel lengths of 1.0 m and 1.5 m. It is found that with the increase of the flow depth the pressure drop decreases and also the efficiency and the outlet temperature decreases and by increasing the duct length, the efficiency decreases but the outlet temperature and the pressure drop increases as illustrated in Figures 11 - 13. It appears from Figures 11 - 13 that the use of the porous media increases the system efficiency by 3 - 7%, but the rise in outlet temperature is faintly low at 2 - 3^{o}C.

Conclusion

A mathematical simulation to predict the effect of different parameters on system thermal performance and pressure drop, for V-groove absorbers in single and double flow modes 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 pressure drop is increased. On the other hand, decreasing the channel flow depth results in increasing the system efficiency

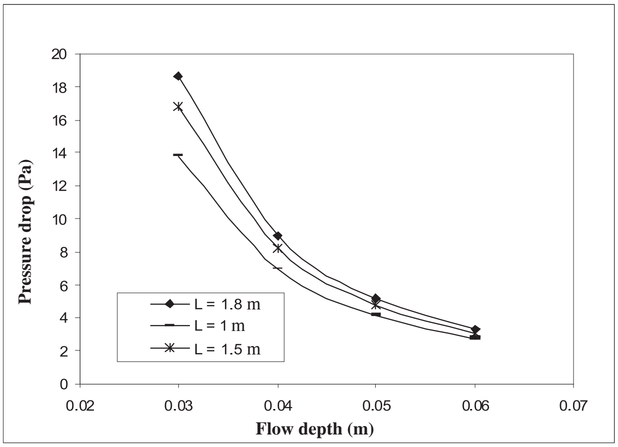


Figure 9. Pressure drop variation with flow channel depth for single pass in V-groove absorber

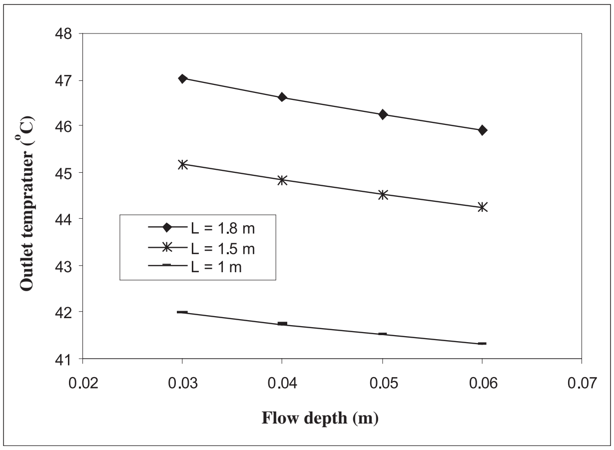


Figure 10. Outlet temperature drop variation with flow channel depth for single pass in V-groove absorber

and outlet temperature at the same time as it increases the pressure drop.

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

The double flow is more efficient than the single flow mode and the use of porous media increases the system efficiency and the outlet temperature. This increment will result in an increase in the pressure drop thus increasing the pumping power expanded in the collector.

Finally the adoption of the aforementioned mathematical simulation that has been developed to predict the thermal performance of solar air heaters, and to find the influence of different parameters seems to be promising, as it is capable of predictive a reasonable results according to chosen parameters with design rules that incorporate human expertise in this field.

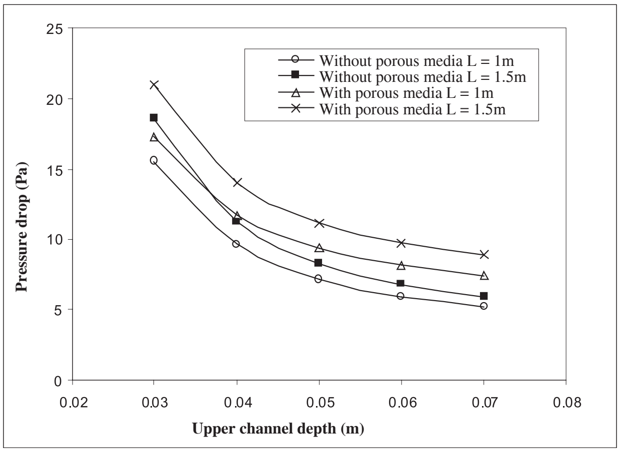


Figure 11. The variation of pressure drop with upper channel depth

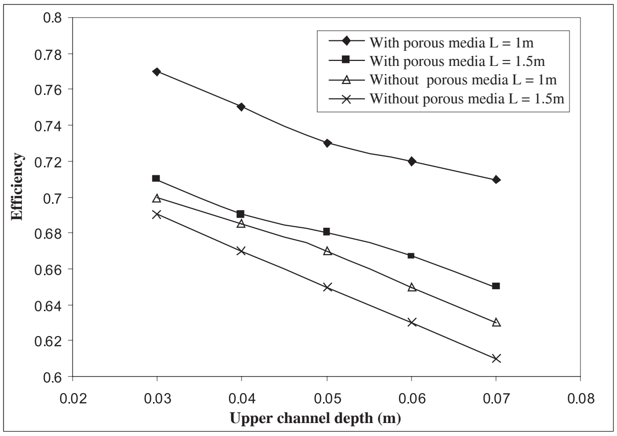


Figure 12. The variation of efficiency with upper channel depth

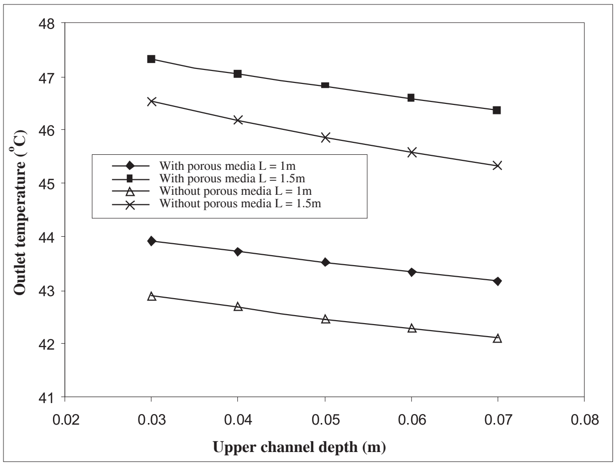


Figure 13. The variation of outlet temperature with upper channel depth

Furthermore, the use of the internet will help in sharing and distributing this knowledge.

This process should be of interest to designers and engineers who would like to compare the cost effectiveness of solar air heaters.

Notations

A	Area of collector that absorb solar radiation, m^2
C_P	Specific heat of working fluid, J/kg K
D	Collector Width, m
D_h	Hydraulic diameter, m
f	Friction factor
F_R	Heat removal factor
h	Fluid heat transfer coefficient, W/m^2 K
h_r	Radiation heat transfer coefficient, W/m^2 K
H_V	V-groove height, m
I	Solar radiation, W/m^2
L	Collector length, m
$\dot{m}$	Collector flow rate, kg/sec
N_u	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_r	Bottom plate temperature, K
U	Overall heat loss coefficient, W/m^2 K
U_b	Back loss coefficient, W/m^2 K
U_t	Top loss coefficient, W/m^2 K
W	Collector width, 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
ϕ	V-groove angle
μ	Dynamic viscosity, Pa.sec

Subscripts

a	ambient
b	back
c	cover
f	fluid
h	hydraulic
p	plate
u	useful

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