Analysis of Single and Double Passes V-Grooves Solar Collector With and Without Porous Media

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

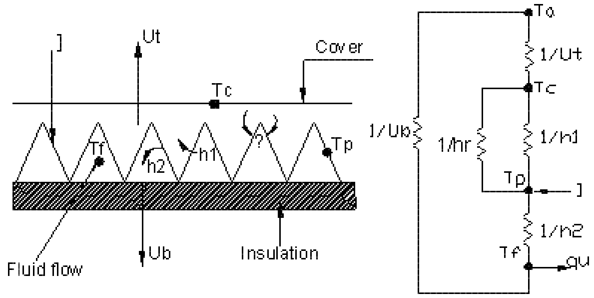
Key-Words - Single & double flow V-groove absorber; Porous media; Thermal performance; Pressure drop; Flow channel depth

1. INTRODUCTION

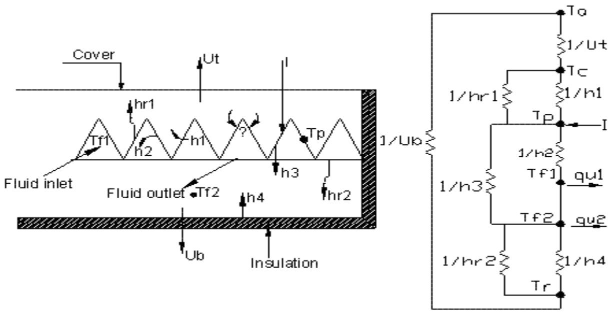
SOLAR air heaters are cheap and most widely used collection devices because of their inherent simplicity [1]. It has a wide range of applications in drying agricultural produces, industrial process heating as textile and papers, space heating and greenhouse heating. Extensive investigations have been carried out on the optimum design of conventional and modified solar air heaters. The researchers have give 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 [2,3,4,5]. The use of porous packing material in the collector duct is also one of the methods that been proposed for the enhancement of the collector's efficiency [6]. An internet based mathematical simulation has been conducted and developed to predict the thermal performance for different designs of solar air heaters [7,8]. This study uses the aforementioned developed program in [7,8] 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 thought the collector, for V-groove absorber in single and double passes with and without using a porous media.

2. THEORETICAL ANALYSIS

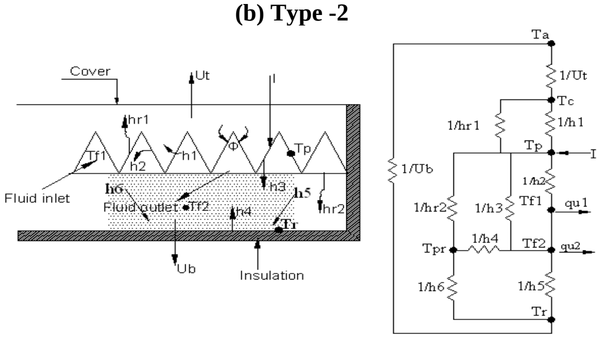
Figure 1 shows schematically the cross sectional views and the thermal net work of the solar air heaters investigated in the present work. In all types, the air heaters are composed of three plates, the cover, the V-groove absorber and the rear or back plate. The air is flowing in the upper channel depth through the V-groove absorber in type 1, and then it is turned to continue flowing in the lower duct between the absorber and the rear plate in type 2, 3. The lower duct has been packed with a porous media of 0.80 porosity in type 3. The various heat transfer coefficients at different components of the air heaters are illustrated in the thermal network in Figure 1. To model the solar air types and obtain there relative equations, a number of simplifying assumptions has been made. These assumptions are as follows (a) Performance is steady state (b) There is no absorption of solar energy by a cover insofar as it affects losses from the collector (c) Heat transfer fluid is considered a non-participating medium (d) The radiation coefficient between the two air duct surfaces is found by assuming a mean radiant temperature equal to the mean fluid temperature and (e) Loss through front and back are to the same ambient temperature.Therefore the energy balance equations for the plates and the flowing air yields the following equations:



(a) Type -1



(b) Type -2



(c) Type -3

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

a) Type -1

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

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

b) Type -2

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

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

Fluid medium in the lower duct

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

Bottom plate

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

(c) Type -3

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

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

Fluid medium in the lower duct

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

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 defined as the ratio of the useful energy to the total incident solar radiation is expressed by the Hottel-Whillier-Bliss equation [9].

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

Heat transfer coefficients

The following correlation proposed by Mc Adams [10] for air flowing over the outside surface of the glass cover is used to predict the convective heat transfer coefficient

$$h_a = 5.7 + 3.8V \tag{16}$$

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 following [3, 9]

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

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 emmitance [11].

$$\varepsilon_{p'} = \frac{2\varepsilon_p}{1 + \varepsilon_p} \tag{18}$$

The convective heat transfer coefficients are calculated using the following relations

$$h = \frac{k}{D_h} Nu \tag{19}$$

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 are [12]:

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

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

$$Nu = 0.018\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.4} \tag{22}$$

where Re is the Reynolds number, Pr is Prandtl

$$\mathrm{Re} = \frac{\dot{m} D_h}{A_f \mu} \tag{23}$$

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

For the V-groove absorber it is found the hydraulic diameter equals to [7]:

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

And the convective heat transfer coefficient in the cavity to the cover is adjusted by the ratio of the heat transfer area to the collector aperture area.

$$h = \frac{h}{\sin(\phi/2)} \tag{26}$$

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 flowing expression [3]:

$$p = f\left(\frac{m^2}{\rho}\right)\left(\frac{L}{D}\right)^3 \tag{27}$$

$$f = f_o + y\left(\frac{D}{L}\right) \tag{28}$$

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 ($10^4< \mathrm{Re}<10^5$)

The 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/sec, inlet temperature of 35°C and different values of mass flow rate.

3. RESULTS AND DISCUSSION

Figure 3 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 up to 0.06 Kg/sec in single flow mode and up to 0.07 Kg/sec in double flow mode, 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 due to the increased heat removal from two flow channels compared to one flow channel in single pass operation. On the other hand the using of porous media in double flow increase the air heater efficiency to be more 7% efficient than the air heater in single mode and more 2-3% in double flow mode without porous media. Hence, the use of porous media increases the heat transfer area which contributed to the higher efficiency. Because, the outlet temperature is an important parameter for drying applications; the outlet temperature was investigated for a wide range of flow rates. Figure 4 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/esc for double flow mode the rate of temperature drop become lower. Figure 5 illustrates the variation of pressure drop with mass flow rate; it is show that the pressure drop is a function of 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 is almost doubled 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 increase the pressure drop from 3 to 25 Pa more than the pressure drop in double flow V-groove absorber without the porous media. The variation in efficiency and pressure drop with flow channel depth for single pass mode in V-groove absorber at different collector lengths (1.8m, 1.5m & 1m) is displayed in Figure 6& 7 respectively at fixed mass flow rate of 0.034 kg/sec. Figure 6 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. Figure 7 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 8, which indicates that the outlet temperature can be increased with the decreased of flow depth. The effect of different upper channel depth on pressure drop, efficiency and outlet temperature for double pass mode with and without porous media is conducted, for fixed mass flow rate of 0.03 kg/sec, constant lower flow depth of 0.025 m channel and two different channel lengths of 1m and 1.5 m. It is found that with the increase of the flow depth the pressure drop decreased as well as the efficiency and the outlet temperature decreased and by increasing the duct length, the efficiency is decreased but the outlet temperature and the pressure drop is increased as illustrated in Figure 9-11. It is appeared from Figures 9-11 that the use of the porous media increase the system efficiency by 3-7 %, but the rise in outlet temperature is slightly small about 2-3^{o}C.

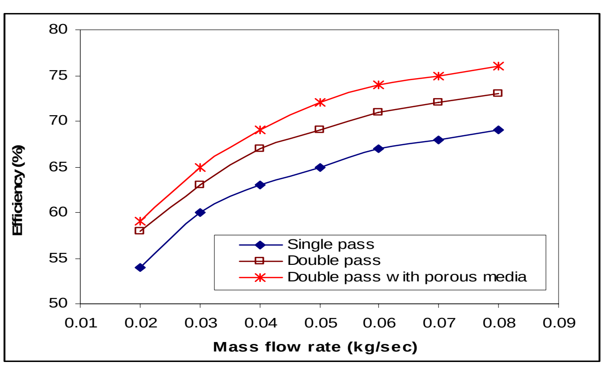


Figure 3: Efficiency variation with mass flow rate

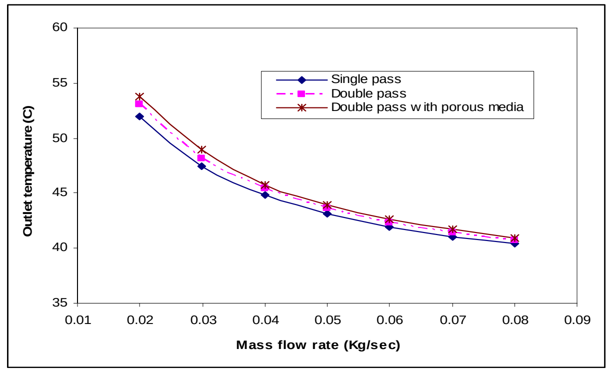


Figure 4: Outlet temperature variation with mass flow rate

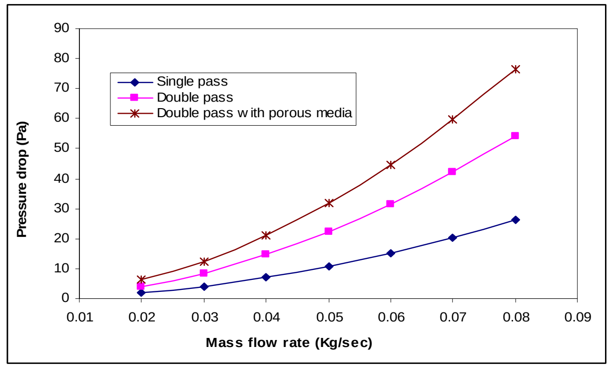


Figure 5: Pressure drop variation with mass flow rate

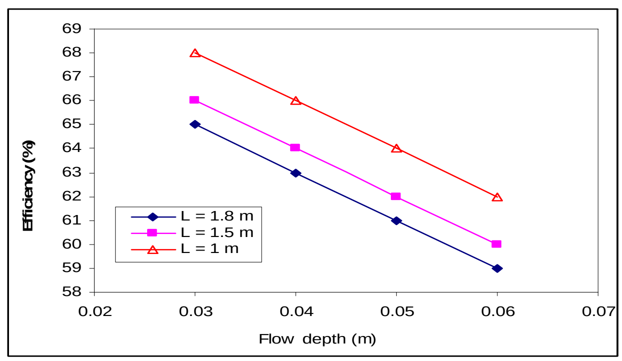


Figure 6: Efficiency variation with flow channel depth for single pass mode

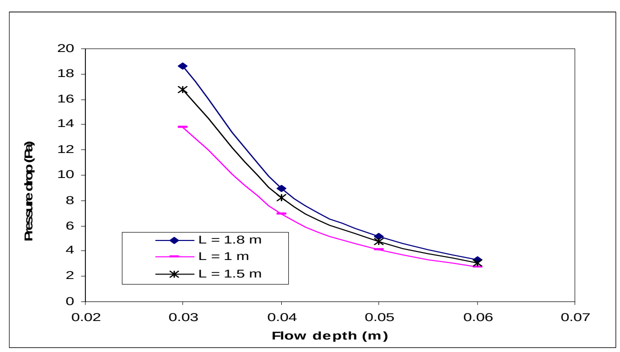


Figure 7: Pressure drop variation with flow channel depth for single pass mode

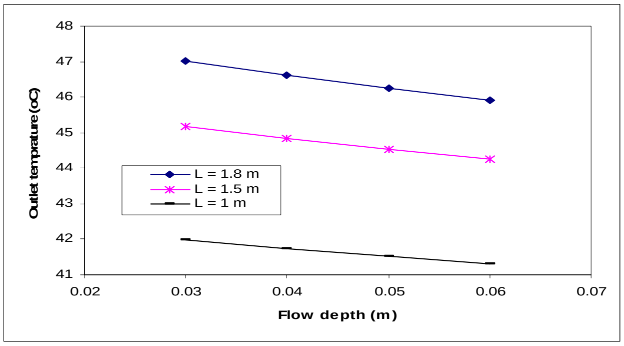
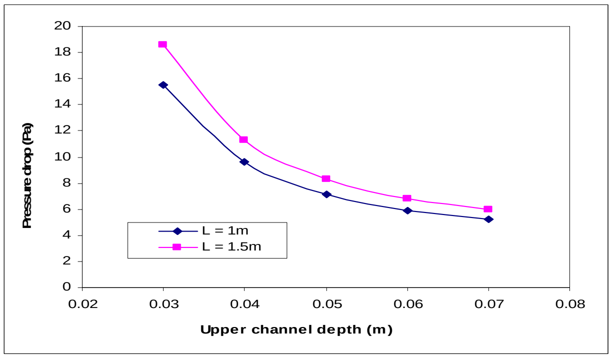
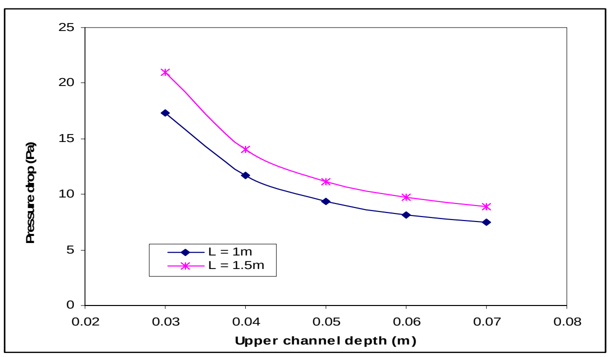


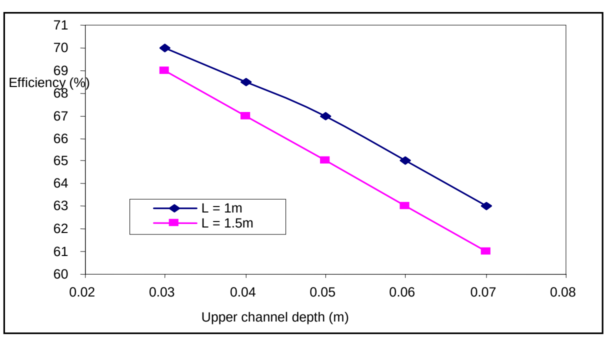
Figure 8: Outlet temperature drop variation with flow channel depth for single pass mode



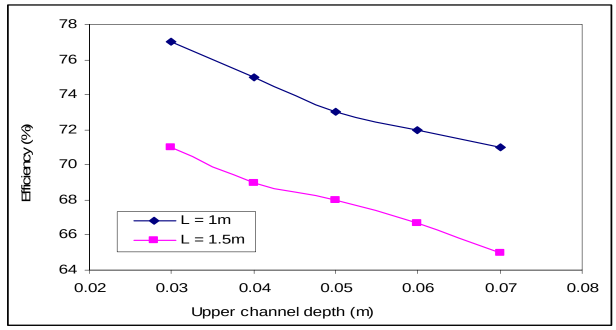
Double pass without using porous media



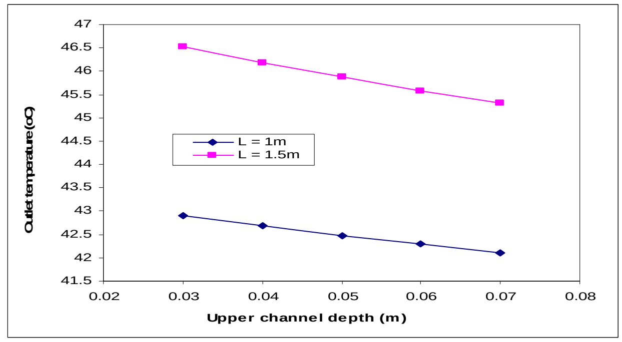
Double pass with using porous media
Figure 9: The variation of pressure drop with upper channel depth



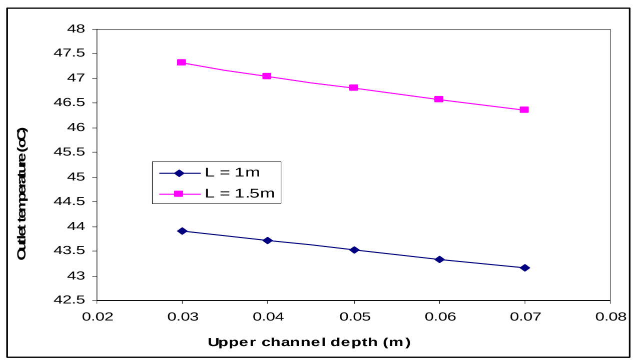
Double pass without using porous media



Double pass with using porous media
Figure 10: The variation of efficiency with upper channel depth



Double pass without using porous media



Double pass with using porous media
Figure 11: The variation of outlet temperature with upper channel depth

4. CONCLUSIONS

A mathematical simulation to predict the effect of different parameters on system thermal performance and pressure drop, for V-groove absorber 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. The channel length also has an effect on the thermal efficiency hence the system efficiency is more increased 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 made and the using of porous media increase the system efficiency and the outlet temperature. This increment will result in the increase of the pressure drop thus increasing the pumping power expanded in the collector. Finally the adoption of the aforementioned internet based mathematical simulation that have been developed to predict the thermal performance of solar air heaters, and to find the influence of different parameters is seems to be promising, hence it is capable to predict a reasonable results according to chosen parameters with design rules that incorporate the human expertise in the field. What's more the use of internet will help in sharing and distributing the knowledge.

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
P Pressure drop across the duct, Pa
Q_u Rate of useful energy gain, W
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

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