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Analytical Investigations on the Thermal and Thermohydraulic Performance of Double Flow Double Exposer Solar Air Heater

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Abstract: This study presents an analytical investigation on thermal and thermohydraulic performance of double flow, double and single exposer solar air heaters with corrugated and flat plate absorber. The performance of collectors are analyzed with different parameters such as mass flow rate and insolation. In order to assess the accuracy of the present mathematical model, the results of single exposer collectors are compared with experimental results and found reasonably well. The results show that the performance of double exposer collectors have significantly higher than single exposer collectors. However, corrugated absorber performance is more superior to the flat plate absorber.

Key words: Solar air heater, corrugated absorber, double flow, double exposer, thermal efficiency, thermohydraulic efficiency

INTRODUCTION

Solar air heater has an important place in the utilization of solar energy. The great advantage of this collector as it absorbs both direct and diffuse solar radiation and found the application for low to medium grade thermal energy collection. The various applications of solar air heater are space heating, drying of agricultural products and medicinal plant, timber seasoning and power generation. The efficiency of solar air heater is low due to poor thermo physical properties of air, the heat transfer rate between absorber to flowing air is low which have attracted the attention of large number of researchers to improve its performance. Literature of various research papers indicate that the performance of solar air heater can be improve by using fins on absorber plate, roughened absorber, packing of flow duct, corrugated absorber, double flow and double pass (Rai *et al*., 2017; Gupta and Kaushik, 2009; Chouksey and Sharma, 2016; Eswaramoorthy, 2015; Karim *et al*., 2014; Assari *et al*., 2014; El-Sebaii *et al*., 2011).

The single exposer solar collector receive relatively low solar radiation and have an average efficiency whereas, using reflecting surfaces (reflectors), through which incoming solar radiation reflected towards to the collector surface increases solar flux which leads to increase the efficiency. Armenta *et al*. (2011) optimized the orientation of reflectors using simulation. They indicated that it is very difficult to find the exact position and orientation of reflectors for a particular location. Kostic and Pavlovic (2012) presented the influence of the position of flat plate reflectors on the thermal efficiency of solar thermal collector. Tanaka (2015) theoretically analyzed the gap between a solar collector and flat plate bottom reflector and proposed a graphical model to calculate the amount of solar radiation reflected by the reflector which is absorbed by the collector. Nikolic and Lukic (2015) theoretically and experimentally investigated the thermal performance of double exposer solar collector with flat plate reflector and found that double exposer flat plate solar collector have better performance than conventional solar collector. The thermal power of these collectors obtained in the range of 41.79-66.44%.

In the present work an analytical study has been carried out to investigate the thermal and thermohydraulic performance of double flow, double and single exposer solar air heaters (Fig. 1). Four different types of solar air heaters are considered, Double Exposer Corrugated Plate (DECP), Double Exposer Flat Plate (DEFP), Single Exposer Corrugated Plate (SECP) and Single Exposer Flat Plate (SEFP). The characteristic of these solar air heaters are each have double glass cover to reduce the heat loss to the environment, the average flow channel depth is 0.025 m, the angle of v-corrugated absorber is 60° (El-Sebaii *et al*., 2011) and the fraction of mass flow rate (r) is 0.5, i.e., mass flow rate of air through the each channel is same because of at r = 0.5, double flow solar air heater have maximum efficiency (Yeh *et al*., 1999).

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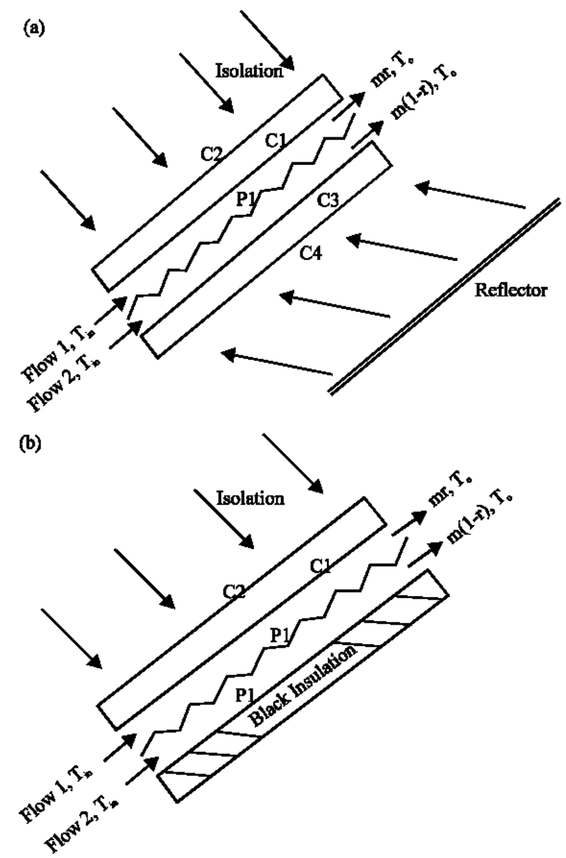

Fig. 1: Schematic diagram of double flow: a) Double exposer and b) Single exposer solar air heaters

MATERIALS AND METHODS

Theoretical analysis: The schematic diagram of double flow double exposer corrugated absorber solar air heater is shown in Fig. 1a. The energy balance equations have been formulated under the following assumptions: the temperatures of air varies only in the flow directions, there is no absorption of radiant energy by the glass covers and air streams, temperature gradient does not exist along the thickness of glass covers and absorbing plate.

Energy balance equation

Glass Cover (C2):

$$\left(h_{c,C1C2}+h_{r,C1C2}\right)\left(T_{C2}-T_{C2}\right)= \left(h_{W}+h_{c,C1C2}\right)\left(T_{C2}-T_{a}\right) \quad (1)$$

Glass Cover (C1):

$$h_{c,F1C1}\left(T_{F1}-T_{C1}\right)+h_{r,P1C1}\left(T_{P1}-T_{C1}\right)= \left(h_{C,C1C2}+h_{r,C1C2}\right)\left(T_{C1}-T_{C2}\right) \quad (2)$$

Flow 1 (F1):

$$h_{c,P1F1}\left(T_{P1}-T_{F1}\right)=mrC_{p}\left(T_{F10}-T_{Fin}\right)+ h_{c,F1C1}\left(T_{F1}-T_{C1}\right) \quad (3)$$

Absorber Plate (P1):

$$1\alpha_{P1}\tau_{C}^{2}+\rho 1\alpha_{P1}\tau_{C}^{2}=h_{C1,P1F1}\left(T_{P1}-T_{F1}\right)+ h_{r,P1C1}\left(T_{P1}-T_{C1}\right)+h_{C,P1F2}\left(T_{P1}-T_{F2}\right)+ h_{r,P1C3}\left(T_{P1}-T_{C3}\right) \quad (4)$$

Flow 2 (F2):

$$h_{c,P1F2}\left(T_{P1}-T_{F2}\right)=m\left(1\text{-}r\right)C_{p}\left(T_{F10}-T_{Fin}\right)+ h_{C,F2C3}\left(T_{F2}-T_{C3}\right) \quad (5)$$

Glass Cover (C3):

$$h_{c,F2C3}\left(T_{F2}-T_{C3}\right)+h_{r,P1C3}\left(T_{P1}-T_{C3}\right)= h_{c,C3C4}+h_{r,C3C4}\left(T_{C3}-T_{C4}\right) \quad (6)$$

Glass Cover (C4):

$$\left(h_{c,C3C4}+h_{r,C3C4}\right)\left(T_{C3}-T_{C4}\right)= \left(h_{W}+h_{c,C4S}\right)\left(T_{C4}-T_{a}\right) \quad (7)$$

Assuming:

$$T_{F1}=\left(T_{F10}+T_{Fin}\right)/2$$
$$T_{F2}=\left(T_{F20}+T_{Fin}\right)/2$$

And:

$$h_{c,P1F1}=h_{c,F1C1}$$
$$h_{c,P1F2}=h_{c,F2C3}$$

It is found from Eq. 1-7, respectively as:

$$T_{C2}=\frac{\left(h_{c,C1C2}\right)T_{C1}+\left(h_{W}+h_{c,C2S}\right)T_{a}}{h_{c,C1C2}+h_{r,C1C2}+h_{c,C2S}} \quad (8)$$

$$T_{C1}=\frac{\left(h_{c,C1C2}+h_{r,C1C2}\right)T_{C2}+h_{c,F1C1}T_{F1}+h_{r,P1C1}T_{P1}}{h_{c,C1C2}+h_{r,C1C2}+h_{c,F1C1}+h_{r,P1C1}} \quad (9)$$

$$T_{F1}=\frac{h_{c,P1F1}\left(T_{P1}+T_{C1}\right)+2mrc_{p}T_{Fin}}{2h_{c,P1F1}+2mrC_{p}} \quad (10)$$

$$T_{P1}=\frac{\left(1+\rho\right)1\alpha_{P1}\tau_{C}^{2}+h_{c,P1F1}T_{F1}+h_{r,P1C1}T_{C1}+h_{c,P1F2}T_{F2}+h_{r,P1C3}T_{C3}}{h_{c,P1F1}+h_{r,P1C1}+h_{c,P1F2}+h_{r,P1C3}} \quad (11)$$

$$T_{F2} = \frac{h_{c,P1F2}\left(T_{P1}+T_{C3}\right)+2m(1\text{-}r)C_pT_{Fin}}{2h_{c,P1F2}+2m(1\text{-}r)C_p} \quad (12)$$

$$T_{C3} = \frac{h_{c,F2C3}T_{F2}+h_{r,P1C3}T_{P1}+\left(h_{C,C3C4}+h_{C,C3C4}\right)T_{C4}}{h_{c,F2C3}+h_{r,P1C3}+h_{c,C3C4}+h_{r,C3C4}} \quad (13)$$

$$T_{C4} = \frac{\left(h_{c,C3C4}+h_{r,C3C4}\right)T_{C3}+\left(h_W+h_{C,C4S}\right)T_a}{h_{c,C3C4}+h_{r,C3C4}+h_W+h_{C,C4S}} \quad (14)$$

The useful heat gain:

$$Q_u = mC_p\left[rT_{F1}+(1\text{-}r)T_{F2}\text{-}T_{Fin}\right] \quad (15)$$

The collector efficiency is:

$$\eta = \frac{Q_u}{(1+\rho)1A_C} \quad (16)$$

For the single exposer collector energy balance equation are not described here because it is similar to the double exposer collector only the difference is in place glass covers C3 and C4 insulation is mounted.

Heat transfer coefficients: For corrugated absorber the heat transfer coefficient between absorbing plate to fluid is enhance by a factor of $1/\sin(\theta/2)$ to the flat plate absorber (Karim *et al*., 2014):

$$h_{c,P1F} = \frac{Nu\times k_F}{D_h}\times\frac{1}{\sin\left(\frac{0}{2}\right)} \quad (17)$$

The correlation of Nusselts number (Nu) by Karim *et al*. (2014) can be expressed as: if Re<2800:

$$Nu = 2.821+0.126Re\frac{2b}{L} \quad (18)$$

If $2800\leq Re\leq 10^4$:

$$Nu = 1.9\times10^{-6}\,Re^{1.79}+225\frac{2b}{L} \quad (19)$$

If $10^4\leq Re\leq 10^5$:

$$Nu = 0.0302Re^{0.74}+0.242Re^{0.74}\frac{2b}{L} \quad (20)$$

For flat plate absorber convective heat transfer coefficient as:

$$h_{C,P1F} = \frac{Nu\times k_F}{D_h} \quad (21)$$

Equation 22 and 23 are for laminar and turbulent flow, respectively (Yeh *et al*., 1999):

$$Nu = 4.4+\frac{0.00398\left(0.7ReD_{h/L}\right)^{1.66}}{1+0.0114\left(0.7ReD_{h/L}\right)^{1.12}} \quad (22)$$

$$Nu = 0.0158Re^{0.8}\left[1+\left(D_{h/L}\right)^{0.7}\right] \quad (23)$$

Thermohydraulic efficiency: The net energy obtained by the collector:

$$Q_e = Q_u\text{-}P_m/C_f \quad (24)$$

Where:

$P_m = m\Delta P/\rho_a$ = The work energy lost in friction in the heater channel

C_f = The conversion factor and taken as 0.2 (Bahrehmand *et al*., 2015)

The pressure drop is calculated from the following expression:

$$\Delta P = \Delta P_{ch}+\Delta P_{en}+\Delta P_{ex} \quad (25)$$

The pressure drop through channel is (El-Sebaii *et al*., 2011):

$$\Delta P_{ch} = 2\rho v_{ch}^2 fL/D_h \quad (26)$$

The sum of entrance and exit pressure drop can be determined by Hegazy (2000):

$$\Delta P_{en}+\Delta P_{ex} = \left(R_{en}+R_{ex}\right)\frac{\rho_a v_p^2}{2} \quad (27)$$

The sum of the resistance factor ($R_{en}+R_{ex}$) is taken 1.5 (Griggs and Khodabakhsh-Sharifabad, 1992). The thermohydraulic efficiency of the solar air heater can be expressed as:

$$\eta_{eff} = \frac{Q_e}{(1+\rho)1A_c} = \frac{Q_u\text{-}\left(P_m/C_f\right)}{(1+\rho)1A_c} \quad (28)$$

Models validation: The numerically calculated thermal efficiency of SECP and SEFP are compared with the experimental values obtained from El-Sebaii *et al*. (2011). Figure 2 shows the comparison of theoretical and experimental values of thermal efficiency. The maximum deviation in thermal efficiency for SECP is found 2.70% and for SEFP is 2.76%.

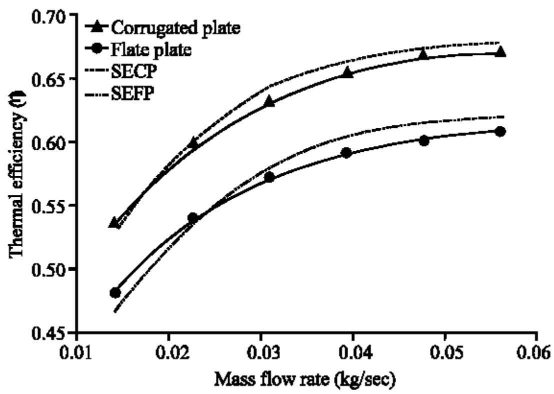

Fig. 2: Comparison of analytical results with available experimental results (El-Sebaii *et al*., 2011)

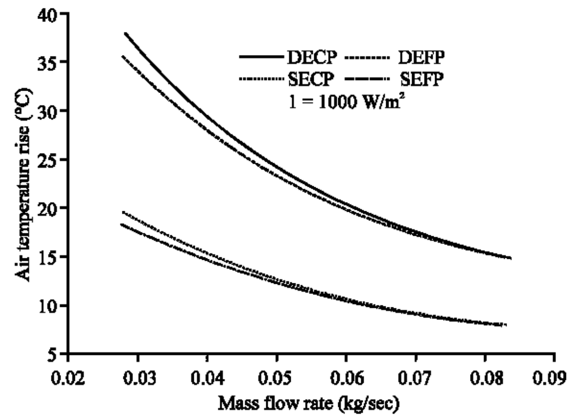

Fig. 3: Air temperature rise as a function of mass flow rate for different types of solar air heaters

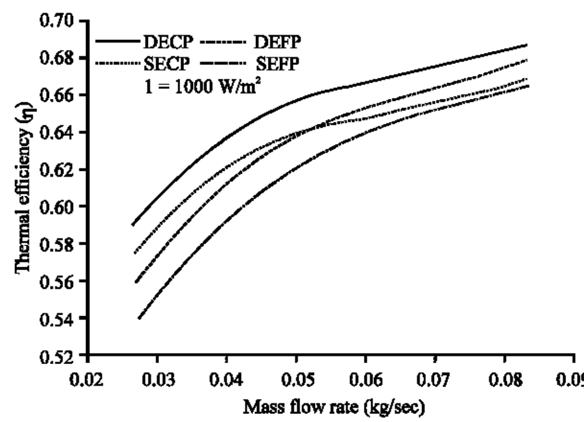

Fig. 4: Thermal efficiency as a function of mass flow rate for different types of solar air heaters

RESULTS AND DISCUSSION

This study presents the various results of the effect of mass flow rate and insolation on thermal and thermohydraulic performance of four types of double flow solar air heaters vig. Double Exposer Corrugated Plate (DECP), Double Exposer Flat Plate (DEFP), Single Exposer Corrugated Plate (SECP) and Single Exposer Flat Plate (SEFP). The following system, operating and metrological parameters are used for the numerical calculations:

- $L = 1.25$ m
- $W = 0.80$ m
- $H_C = 0.025$ m
- $H_F = 0.025$ m
- $\tau_C = 0.875$
- $\alpha_{P1} = 0.96$
- $\varepsilon_C = 0.94$
- $\varepsilon_{P1} = 0.80$
- $\varepsilon_{P2} = 0.94$
- $\rho = 0.9$
- $k_i = 0.025$ W/mK
- $t_i = 0.05$ m
- Ta = 30
- °C = 303 K
- V = 1 m/sec
- b = 0.00625 m
- I = 600-1000 W/m^2
- m = 0.0277-0.0833 kg/sec
- $\gamma = 0.5$

Figure 3 shows the air temperature rise as a function of mass flow rate for different types of double flow solar air heaters for insolation of 1000 W/m^2. Air temperature rise decreases with increase in mass flow rate for all types of solar air heaters. It is seen from figure that the double exposer collector have higher gain in air temperature rise with respect to single exposer collector due to energy gain from both (top and bottom) sides of the collector. It can also seen that at lower mass flow rate air temperature rise is higher because of high energy gain. The maximum value of air temperature rise are 40.12, 37.7, 20.46 and 19.05°C at mass flow rate of 0.0277 kg/sec for DECP, DEFP, SECP and SEFP, respectively.

Figure 4 presents the variation of thermal efficiency with mass flow rate for different types of solar air heaters for I = 1000 W/m^2. It is seen from the plot, the thermal efficiency increases with increase in mass flow rate because of increase in convective heat transfer coefficient. It is observed from the figure that double exposer collector have higher thermal efficiency than single exposer collector having same absorber plate. It is also observed that the double or single exposer solar air heater with corrugated absorber have higher efficiency

Table 1: Enhancement in air temperature rise of double flow, double and single exposer solar air heaters

	DECP		DEFP		SECP		SEFP
Mass flow rate (kg/sec)	ΔT (°C)	E_T (%)	ΔT (°C)	E_T (%)	ΔT (°C)	E_T (%)	ΔT (°C)
0.0277	40.12	110.62	37.71	97.92	20.46	7.38	19.05
0.0416	29.25	104.60	28.16	96.97	15.05	5.28	14.29
0.0555	22.51	99.59	21.93	94.49	11.44	1.46	11.28
0.0694	18.32	96.76	18.03	93.62	9.42	1.19	9.31
0.0833	15.56	96.30	15.40	94.27	7.96	0.40	7.93

Table 2: Thermal efficiency (%)

Factors	η	E_η	η	E_η	η	E_η	η
0.0277	58.79	10.85	55.25	4.17	56.96	7.39	53.04
0.0416	64.37	7.68	61.97	3.67	62.94	5.27	59.78
0.0555	66.09	5.05	64.40	2.36	63.84	1.46	62.91
0.0694	67.26	3.56	66.19	1.90	65.73	1.19	64.95
0.0833	68.58	3.31	67.47	1.64	66.64	0.40	66.38

Table 3: Thermohydraulic efficiency (%)

Factors	η_{eff}	$E_{\eta eff}$	η_{eff}	$E_{\eta eff}$	η_{eff}	$E_{\eta eff}$	η_{eff}
0.0277	58.59	12.18	55.05	5.40	56.57	8.31	52.23
0.0416	63.68	9.68	61.29	5.55	61.62	6.13	58.06
0.0555	64.45	7.73	62.77	4.93	60.72	1.50	59.82
0.0694	64.16	8.59	63.11	6.81	59.83	1.26	59.08
0.0833	63.22	12.37	62.55	11.18	56.46	0.36	56.26

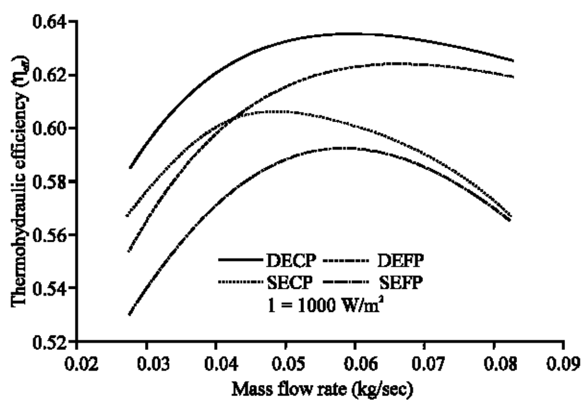

Fig. 5: Thermohydraulic efficiency as a function of mass flow rate for different types of solar air heaters

than the flat plate absorber solar air heater due to increase in thermal conductance. The percentage enhancement in thermal efficiency of corrugated absorber plate is higher at lower mass flow rate and lower at higher mass flow rate. The maximum and minimum enhancement in thermal efficiency of DECP with respect to the SEFP is 10.85% and 3.31% at mass flow rate of 0.0277 and 0.0833 kg/sec, respectively.

The effect of mass flow rate on thermohydraulic efficiency is shown in Fig. 5 for double and single exposer collectors for I = 1000 W/m^2. It is clearly seen from the figure that thermohydraulic efficiency increases with increase in mass flow rate upto a certain value of flow rate at which it attains a maximum value and there after decreases sharply for all types of solar air heaters. Results also indicated that the thermohydraulic efficiency of single and double exposer collectors having flat plate absorber reaches maximum value at m = 0.0555 and 0.0694 kg/sec, respectively whereas, corrugated absorber collector's pick value of thermohydraulic efficiency shifted towards lower mass flow rates. This type of trend is observed due to increase in pressure drop of flowing air in corrugated absorber/channel.

Table 1-3 present the results and percentage enhancement of air temperature rise, thermal efficiency and thermohydraulic efficiency of double flow DECP, DEFP and SECP with respect to SEFP solar air heater for the range of mass flow rate of 0.0277-0.0833 kg/sec for I = 1000 W/m^2. Results clearly indicate that there is considerable enhancement in ΔT, η and η_{eff} of double exposer to the single exposer collector. However, double expose solar air heater with corrugated absorber plate have the maximum enhancement for the entire range of mass flow rate investigated. It has been also observed from the table that the rate of enhancement in air temperature rise and thermal efficiency is more at lower mass flow rate than the higher mass flow rate. The reason behind this is at lower mass flow rate the rise in air temperature is high compared to the higher mass flow rate. The maximum enhancement in ΔT, η and η_{eff} are 110.62, 10.85 and 12.37 %, respectively for DECP solar air heater with respect to SEFP solar air heater.

Figure 6 shows the variation of air temperature rise as a function of insolation for DECP solar air heater for

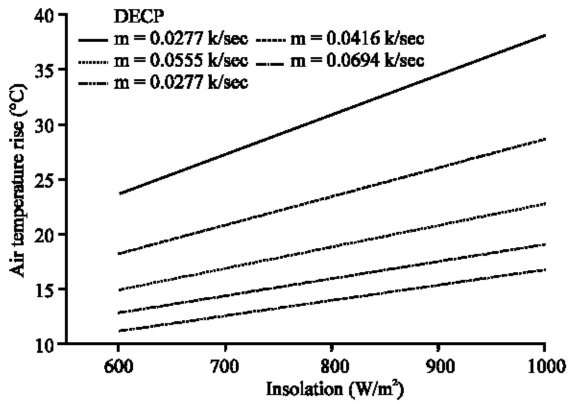

Fig. 6: Air temperature rise as a function of isolation

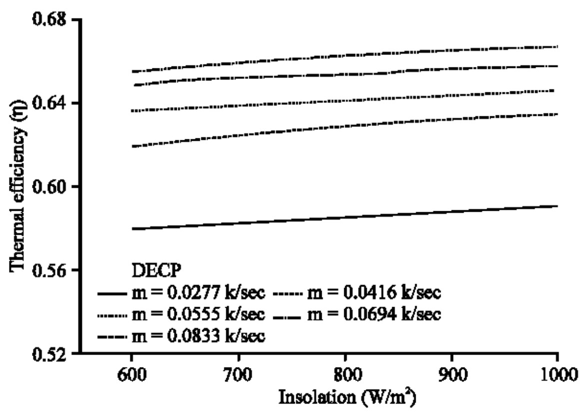

Fig. 7: Thermal efficiency as a function of isolation

different mass flow rates. It is found that air temperature rise increases linearly with increase in insolation for all mass flow rates, due to energy gain increases with increase in insolation. It is also found that air temperature rise increases with decrease in mass flow rate. The maximum and minimum value of air temperature rise are 40.12 and 9.12°C for the insolation of 1000 and 600 W/m^2, respectively.

The effect of insolation on thermal efficiency of DECP solar air heater for different mass flow rate is shown in Fig. 7. Figure reveals that the thermal efficiency increases with increase in insolation for all mass flow rates but the rate of increase is very low as insolation has no effect on convective heat transfer between absorber to flowing air. Figure 7 also reveals that the thermal efficiency increases, however, enhancement rate of thermal efficiency decreases with increase in mass flow rate because of lower percentage increase in surface conductance.

Figure 8 illustrate the variation of thermohydraulic efficiency with insolation for DECP solar air heater.

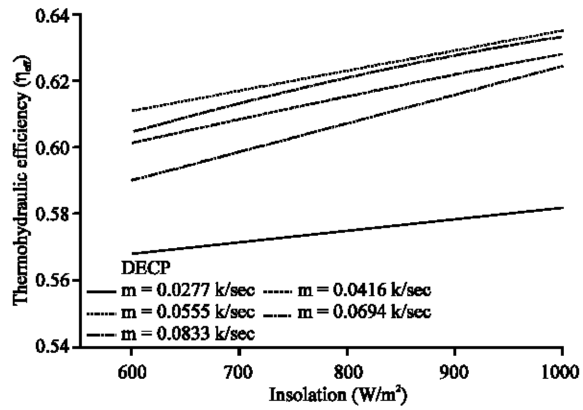

Fig. 8: Thermohydraulic efficiency as a function of insolation

Figure 8 shows the low increasing rate of thermohydraulic efficiency as insolation increases. Carefully observation of figure shows that the thermohydraulic efficiency increases as mass flow rate increases and maximum value obtained for the mass flow rate of 0.0555 kg/sec, further increase in mass flow rate decreases its values. This is because of increase in mass flow rate of air leads to increase in pressure drop and consequently decreases the value of thermohydraulic efficiency.

CONCLUSION

On the basis of results obtained from the analytical models the following conclusions are drawn: The thermal efficiency increases with increase in mass flow rate whereas thermohydraulic efficiency increases upto a certain limit of mass flow rate and there after decreases for all type of collectors. The considerable enhancement in thermal and thermohydraulic efficiencies have been achieved in double exposer solar air heaters however, using corrugated absorber is more superior to the flat plate absorber. Increasing the solar radiation intensity leads to increase the air temperature rise, thermal and thermohydraulic efficiencies.

NOMENCLATURE

A_c = Area of collector (m^2)
b = Half height of corrugated plate (m)
C_p = Specific heat of air at constant pressure (J/kg K)
D_h = Hydraulic diameter (m)
E = Enhancement
H = Height (m)
h = Heat transfer coefficient (W/m^2 K)
k = Thermal conductivity (W/m K)
T = Temperature (K)
m = Mass flow rate (kg/sec)
r = Fraction of mass flow rate
I = Insolation (W/m^2)

Lcollector length (m):
N_u = Nusselt number
Q = Energy gain (W)
R = Resistance factor
Re = Reynolds number
V = Velocity of wind (m/sec)
v = Velocity of air (m/sec)
W = Collector width (m)

Greek symbols:
α = Absorptivity
ε = Emissivity
η = Efficiency
τ = Transmissivity
ρ = Coefficient of reflection
ρ_a = Density of air (kg/m^3)
ΔP = Pressure drop (N/m^2)
θ = Angle of corrugated absorber (60°)

Subscripts:
c = Convective
e = Net
r = Radiative
C = Cover

C1, C2, C3, C4 glass cover:
ch = Channel
eff = Thermohydraulic
en = Entrance
ex = Exit
F = Flow

F1, F2 flow above and under the absorber:
I = Insulation

P1, P2 absorber and bottom plate:
W = Wind
a = Ambient
in = Inlet
o = Outlet
p = Pipe
t = Thickness
u = Usefu

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