International Journal of Electrical, Electronic and Communication Sciences
ISSN: 2517-9438
Vol:11, No:12, 2017

Exergy Based Performance Analysis of Double Flow Solar Air Heater with Corrugated Absorber

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Abstract**—**This paper presents the performance, based on exergy analysis of double flow solar air heaters with corrugated and flat plate absorber. A mathematical model of double flow solar air heater based on energy balance equations has been presented and the results obtained have been compared with that of a conventional flat-plate solar air heater. The double flow corrugated absorber solar air heater performs thermally better than the flat plate double flow and conventional flat-plate solar air heater under same operating conditions. However, the corrugated absorber leads to higher pressure drop thereby increasing pumping power. The results revealed that the energy and exergy efficiencies of double flow corrugated absorber solar air heater is much higher than conventional solar air heater with the concept involving of increase in heat transfer surface area and turbulence in air flow. The results indicate that the energy efficiency increases, however, exergy efficiency decreases with increase in mass flow rate.

Keywords**—**Corrugated absorber, double flow, exergy efficiency, solar air heater.

NOMENCLATURE

A_c	area of collector (m^2)
b	half height of v-groove (m)
C_p	specific heat of air at constant pressure (J/kg K)
D_h	hydraulic diameter (m)
Ex	exergy (W)
f	friction coefficient
h	specific enthalpy (J/kg)
h_c	convective heat transfer coefficient (W/m^2 K)
h_r	radiative heat transfer coefficient (W/m^2 K)
H_c	average height of air flow channel (m)
H_{gc}	height of glass cover (m)
I	insolation (W/m^2)
k	thermal conductivity (W/m K)
L	collector length (m)
l	thickness (m)
m	mass flow rate (kg/s)
Nu	Nusselt number
Q	energy gain by air (W)
R	universal gas constant (J/kgK)
Rf	resistance factor
Re	Reynolds number
S	specific entropy (J/kg/K)
T	temperature (K)
U	loss coefficient (W/m^2 K)
V	velocity of wind (m/s)
v	velocity of air (m/s)
W	collector width (m)
$\dot{W}$	work rate or power (W)

Greek Symbols

α	absorptivity
ε	emissivity
φ	fraction of mass flow rate
ΔP	pressure drop (N/m^2)
η	efficiency
η_{exe}	exergy efficiency
μ	viscosity of air (Ns/m^2)
ρ	density of air (kg/m^3)
σ	Stefan-Boltzmann constant (W/m^2 K^4)
τ	transmissivity
Ψ	specific exergy (J/kg)
$\boldsymbol{\theta}$	angle of v-groove absorbing plate (60˚)

Subscripts

a	ambient
ap	absorber plate
B	bottom
bp	bottom plate
ch	channel
$dest$	destruction
en	entrance
ex	exit
f	total flow
$f1$	flow above the absorber plate
$f2$	flow under the absorber plate
gc	glass cover
$gc1$	lower glass cover
$gc2$	upper glass cover
ht	heat
in	inlet
L	overall
m	mean
max	maximum
min	minimum
o	outlet
T	top
u	useful
w	work

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I. INTRODUCTION

NOWADAYS, solar thermal systems are usually recognized as one of the most effective methods to exploit solar energy. Among various thermal systems, solar air heaters are more popular for the use of air heating by using solar energy because of its simplicity in design and economic. It is best for low to moderate air temperature applications such as space heating, agriculture products drying, timber seasoning and many more. Generally the efficiency of solar air heater is low due to low thermal capacity of air and low heat transfer coefficient between absorber to flowing air [1]. To improve the efficiency, many researchers developed various design such as honeycomb collector, packing of air duct, extended surface absorber, artificial roughness on absorber. The use of corrugated absorber in place of flat plate absorber increases the heat transfer surface area as well as turbulence in the flowing fluid which increase the heat transfer to flowing air [2].

The first law of thermodynamics analysis is based on energy balance method. However, second law of thermodynamics involves the reversibility or irreversibility process which is very important for exergy analysis of an energy system. The exergy analysis is an effective method for designing and performance evaluation of energy systems [3].

A mathematical model has been developed by Abhishek et al. [4] to investigate the energy and exergy performance of wavy fin solar air heater for the evaluation of the effect of various complex geometries of wavy fin. Sahu and Prasad [5] analytically investigated the exergetic performance of arc shaped wire rib roughened absorber solar air heater and found that the 56% enhancement in exergetic efficiency to the smooth plate absorber solar air heater for the relative roughness height (e/d) of 0.0422. The exergetic analysis and optimization for different parameters of double pass corrugated absorber solar air heater have been studied by Mahdi et al. [6]. The effect of depth, length, fin shape and Reynolds number have been studied by Bahrehmand et al. [7] for the energy and exergy analysis of single and two glass cover solar air collector. The study showed that the system with fin and thin metal sheet are more efficient than other systems. The energy and exergy analysis of five solar air heaters with different configuration have been experimentally investigated by Huseyin [8] and reported that heat transfer coefficient and pressure drop increases with shape of absorber surface. Lalji et al. [9] carried out exergy analysis of a packed bed solar air heater and correlations have been developed for friction factor and heat transfer coefficient. Gupta and Kaushik [10] studied the energy, effective and exergy performance of different types of roughened absorber solar air heater. Hikmet [11] experimentally investigated the energy and exergy performance of a novel design flat plate air collector with and without obstacles. Author found that the double flow collector having obstacles gives better performance than without obstacles. Akpinar and Kocyigit [3] performed experimental analysis for a flat plate solar air heater having several obstacles and no obstacles, for the mass flow rate of 0.0074 and 0.0052 kg/s. The first law and second law of efficiencies were obtained 20% to 82% and 8.32% to 44.00% respectively. Salwa et al. [12] proposed a packed bed solar air heater with PCM spherical capsules as packing material. They obtained experimentally energy efficiency 32% to 45% and exergy efficiency 13% to 25% using first and second laws of thermodynamics.

This paper presents mathematical models to investigate energy and exergy performance of double flow flat plate and corrugated absorber solar air heaters. Three types of solar air heaters have been considered; AH-1 has double flow corrugated absorber plate, AH-2 has double flow flat plate absorber, and AH-3 has conventional solar air heater.

II. Theoretical Analysis

Fig. 1 shows the schematic diagram of double flow solar air heater with two glass cover. The double flow solar air heaters with corrugated absorber (AH-1) and flat plate absorber (AH-2) are shown in Figs. 1 (a) and (b) respectively. The present work is based on theoretical analysis and the energy balance equations are made under the following assumptions:

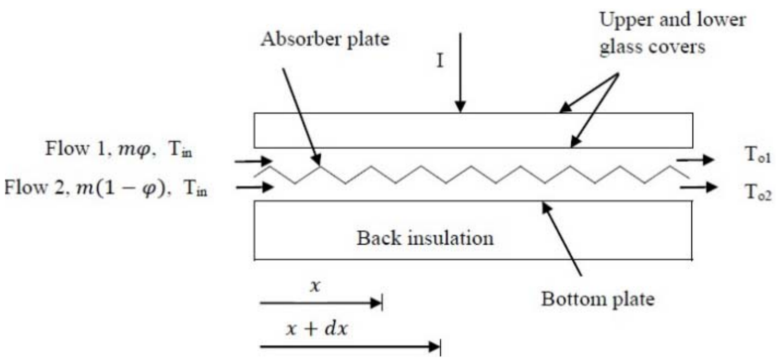
- Air temperature variation is the functions of the flow directions only.
- Temperature drop across the glass cover, absorber plate and bottom plate is negligible.
- The systems operate under quasi steady state.
- Glass covers and flowing air do not absorb radiant energy.

A. Energy Balance Equations

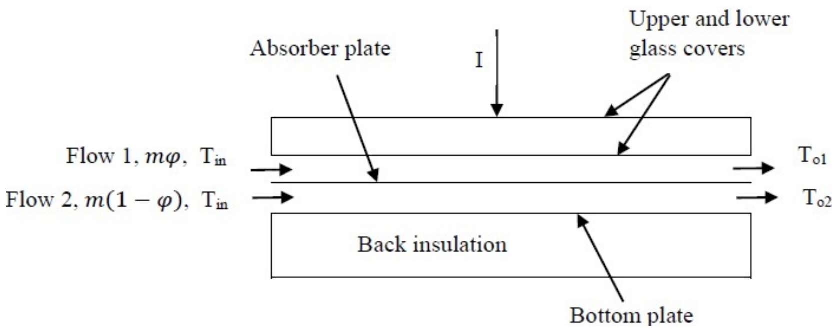
The energy balance equations for the element length of dx can be written as:

- For the lower glass cover (cover 1),

$$h_{r,ap-gc1}(T_{ap}-T_{gc1})Wdx + h_{c,f1-gc1}(T_{f1}-T_{gc1})Wdx = U_{gc1-a}(T_{gc1}-T_a)Wdx \quad (1)$$



(a) AH-1



(b) AH-2

Fig. 1 The schematic diagram of double flow solar air heater with corrugated absorber and flat plate absorber

- For the absorbing plate,

$$I\alpha_{ap}\tau_{gc}^{2}Wdx = U_T(T_{ap}-T_a)Wdx + U_B(T_{ap}-T_a)Wdx + h_{c,ap-f1}(T_{ap}-T_{f1})Wdx + h_{c,ap-f2}(T_{ap}-T_{f2})Wdx \tag{2}$$

- For the bottom plate,

$$h_{r,ap-bp}(T_{ap}-T_{bp})Wdx + h_{c,f2-bp}(T_{f2}-T_{bp})Wdx = U_{bp-a}(T_{bp}-T_a)Wdx \tag{3}$$

- For flow 1 (air flowing over the absorbing plate),

$$h_{c,ap-f1}(T_{ap}-T_{f1})Wdx = m\varphi c_p dT_{f1} + h_{c,f1-gc1}(T_{f1}-T_{gc1})Wdx \tag{4}$$

- For flow 2 (air flowing under the absorbing plate),

$$h_{c,ap-f2}(T_{ap}-T_{f2})Wdx = m(1-\varphi)c_p dT_{f2} + h_{c,f2-bp}(T_{f2}-T_{bp})Wdx \tag{5}$$

Solving (1)-(5) with the boundary condition: at, $\zeta = \frac{x}{L} = 0$, $T_{f1} = T_{f2} = T_{f,in}$.

We obtain the temperature distributions of flow 1 and flow 2 as

$$T_{f1} = \left[\frac{Y_1-\frac{M_5}{1-\varphi}}{\frac{M_4}{(1-\varphi)}}\right]C_1 e^{\frac{Y_1}{Z}\zeta} + \left[\frac{Y_2-\frac{M_5}{1-\varphi}}{\frac{M_4}{(1-\varphi)}}\right]C_2 e^{\frac{Y_2}{Z}\zeta} - \frac{M_5}{M_4}\left(\frac{M_3M_4-M_1M_6}{M_1M_5-M_2M_4}\right) - \frac{M_6}{M_4} + T_a \tag{6}$$

$$T_{f2} = C_1 e^{\frac{Y_1}{Z}\zeta} + C_2 e^{\frac{Y_2}{Z}\zeta} + \frac{M_3M_4-M_1M_6}{M_1M_5-M_2M_4} + T_a \tag{7}$$

The outlet temperature of flow 1, can be obtained from (6), for, $\zeta = 1$, $T_{f1} = T_{f1,o}$

$$T_{f1,o} = \left[\frac{Y_1-\frac{M_5}{1-\varphi}}{\frac{M_4}{(1-\varphi)}}\right]C_1 e^{\frac{Y_1}{Z}} + \left[\frac{Y_2-\frac{M_5}{1-\varphi}}{\frac{M_4}{(1-\varphi)}}\right]C_2 e^{\frac{Y_2}{Z}} - \frac{M_5}{M_4}\left(\frac{M_3M_4-M_1M_6}{M_1M_5-M_2M_4}\right) - \frac{M_6}{M_4} + T_a \tag{8}$$

The outlet temperature of flow 2, can be obtained from (7), for, $\zeta = 1$, $T_{f2} = T_{f2,o}$

$$T_{f2,o} = C_1 e^{\frac{Y_1}{Z}} + C_2 e^{\frac{Y_2}{Z}} + \frac{M_3M_4-M_1M_6}{M_1M_5-M_2M_4} + T_a \tag{9}$$

where $z = \frac{mc_p}{A_c}$ and Y's and C's are in the terms of the convective heat transfer coefficient, loss coefficients and physical properties [13]. The total thermal energy gain is

$$Q_{uf} = Q_{uf1} + Q_{uf2} \tag{10}$$

where,

$$Q_{uf1} = m\varphi c_p(T_{f1,o} - T_{f,in}) \tag{11}$$

and

$$Q_{uf2} = m(1-\varphi)c_p(T_{f2,o} - T_{f,in}) \tag{12}$$

The energy efficiency can be calculated by

$$\eta = \frac{mc_p}{IA_c}(T_{f,o} - T_{f,in}) = \frac{Z}{I}\Delta T \tag{13}$$

B. Heat Transfer Coefficients

An empirical equation, derived by Klein [14], is used to calculate top loss coefficient, U_T, and is given by

$$U_T = \left[\frac{2(T_{ap,m}/520)}{\left\{\frac{(T_{ap,m}-T_a)}{2+(1+0.089h_w-0.1166h_w\varepsilon_{ap})(1+0.07866\times 2)}\right\}^{0.43(1-100/T_{ap,m})}} + \frac{1}{h_w}\right]^{-1} + \frac{\sigma(T_{ap,m}+T_a)(T_{ap,m}^2+T_a^2)}{\left[(\varepsilon_{ap}+2\times 0.00591h_w)^{-1}+\{2\times 2+(1+0.089h_w-0.1166h_w\varepsilon_{ap})(1+0.07866\times 2)-1+0.133\varepsilon_{ap}\}/\varepsilon_{gc}-2\right]} \tag{14}$$

The convective heat transfer coefficient from the outer glass cover (gc2) due to wind is calculated by the expression given by McAdams [15].

$$h_{c,gc2-a} = 5.7 + 3.8V \tag{15}$$

The heat transfer coefficient between two glass covers, inner glass cover (gc1) and outer glass cover (gc2) is given by Hottel and Woertz [16] as

$$h_{c,gc1-gc2} = 1.25\left(T_{gc1,m} - T_{gc2,m}\right)^{0.25} \tag{16}$$

The convective heat transfer coefficient between the absorber plate and flow 1 is assumed to be equal to the convective heat transfer coefficient between flow 1 and glass cover 1, and the convective heat transfer coefficient between the absorber plate and flow 2 is assumed to be equal to the convective heat transfer coefficient between flow 2 and bottom plate.

$$h_{c,ap-f1} = h_{c,f1-gc1} \tag{17}$$

and,

$$h_{c,ap-f2} = h_{c,f2-bp} \tag{18}$$

The radiative heat transfer coefficients between the absorber plate to inner glass cover and bottom plate are expressed as,

$$h_{r,ap-gc1} \approx \frac{4\sigma T_{f1,m}^3}{\frac{1}{\varepsilon_{ap}}+\frac{1}{\varepsilon_{gc1}}-1} \tag{19}$$

and

$$h_{r,ap-bp} \approx \frac{4\sigma T_{f2,m}^3}{\frac{1}{\varepsilon_{ap}}+\frac{1}{\varepsilon_{bp}}-1} \tag{20}$$

The radiative heat transfer coefficients between both glass covers and upper glass cover to air are respectively,

$$h_{r,gc1-gc2} = \frac{\sigma(T_{gc1,m}^2+T_{gc2,m}^2)(T_{gc1,m}+T_{gc2,m})}{\frac{1}{\varepsilon_{gc1}}+\frac{1}{\varepsilon_{gc2}}-1} \tag{21}$$

and

$$h_{r,gc2-a} = \varepsilon_{gc2}\sigma(T_{gc2,m}^2 + T_a^2)(T_{gc2,m} + T_a) \tag{22}$$

For AH-1 solar air heater (i.e. corrugated absorber), the absorber plate surface area is increased by a factor of $1/sin(\theta/2)$ compared to the AH-2 solar air heater (i.e. flat plate absorber) [17]; hence, the heat transfer coefficient between absorber plate to flowing air is

$$h_{c,ap-f} = \frac{Nuk_f}{D_h} \times \frac{1}{sin\left(\frac{\theta}{2}\right)} \tag{23}$$

Karim et al. [17] modified the Hollands and Shewen [18] correlation of Nusselts number (Nu) as:

$$\text{If } Re < 2800 \; Nu = 2.821 + 0.126\, Re\frac{2b}{L} \tag{24}$$

$$\text{If } 2800 \le Re \le 10^4 \; Nu = 1.9 \times 10^{-6}Re^{1.79} + 225\frac{2b}{L} \tag{25}$$

$$\text{If } 10^4 \le Re \le 10^5 \; Nu = 0.0302Re^{0.74} + 0.242Re^{0.74}\frac{2b}{L} \tag{26}$$

For AH-2 solar air heater (i.e. flat plate absorber), the convective heat transfer coefficient between absorber plate and flowing air is

$$h_{c,ap-f} = \frac{Nuk_f}{D_h} \tag{27}$$

For laminar flow [19],

$$Nu = 4.4 + \frac{0.00398(0.7ReD_h/L)^{1.66}}{1+0.0114(0.7ReD_h/L)^{1.12}} \tag{28}$$

For turbulent flow [20],

$$Nu = 0.0158Re^{0.8}[1 + (D_h/L)^{0.7}] \tag{29}$$

For the conventional solar air heater, the calculation procedure of collector efficiency is much simpler than for double flow solar air heater, and not described here.

C. Mean Temperature

The mean air temperatures in the ducts can be found by integrating (6) and (7) from $\zeta = 0$ to $\zeta = 1$, and the expressions are

$$T_{f1,m} = \left[\frac{Y_1-\frac{M_5}{1-\varphi}}{\frac{M_4}{(1-\varphi)}}\right]C_1\frac{Z}{Y_1}e^{(\frac{Y_1}{Z}-1)} + \left[\frac{Y_2-\frac{M_5}{1-\varphi}}{\frac{M_4}{(1-\varphi)}}\right]C_2\frac{Z}{Y_2}e^{(\frac{Y_2}{Z}-1)} - \frac{M_5}{M_4}\left(\frac{M_3M_4-M_1M_6}{M_1M_5-M_2M_4}\right) - \frac{M_6}{M_4} + T_a \tag{30}$$

$$T_{f2,m} = C_1\frac{Z}{Y_1}e^{(\frac{Y_1}{Z}-1)} + C_2\frac{Z}{Y_2}e^{(\frac{Y_2}{Z}-1)} + \frac{M_3M_4-M_1M_6}{M_1M_5-M_2M_4} + T_a \tag{31}$$

The mean absorbing plate temperature can be calculated by

$$T_{ap,m} = \frac{I\alpha_{ap}\tau_{gc}^2+U_LT_a+h_{c,ap-f1}T_{f1,m}+h_{c,ap-f2}T_{f2,m}}{U_L+h_{c,ap-f1}+h_{c,ap-f2}} \tag{32}$$

The mean temperature of glass cover gc1 is calculated by

$$T_{gc1,m} = \frac{U_{gc1-a}T_a+h_{r,ap-gc1}T_{ap,m}+h_{c,f1-gc1}T_{f1,m}}{U_{gc1-a}+h_{r,ap-gc1}+h_{c,f1-gc1}} \tag{33}$$

The mean temperature of glass cover gc2 is calculated by

$$T_{gc2,m} = \frac{(h_{c,gc1-gc2}+h_{r,gc1-gc2})T_{gc1,m}+h_wT_a}{h_{c,gc1-gc2}+h_{r,gc1-gc2}+h_w} \tag{34}$$

D. Pressure Drop

The pressure drop ΔP is calculated from

$$\Delta P = \Delta P_{ch} + \Delta P_{en} + \Delta P_{ex} \tag{35}$$

The pressure drop through the upper and lower channel ΔP_{ch} is calculated by [21];

$$\Delta P_{ch} = 2\rho v_{ch}^2 f L / D_{hi} \tag{36}$$

The sum of the inlet and outlet pressure drop ($\Delta P_{en} + \Delta P_{ex}$) can be determined by Hegazy [22];

$$\Delta P_{en} + \Delta P_{ex} = (Rf_{en} + Rf_{ex}) \frac{\rho v_p^2}{2} \tag{37}$$

where the sum of the entrance and exit resistance factor ($Rf_{en} + Rf_{ex}$) is taken 1.5 [23].

E. Exergy Analysis

The general exergy balance equation can be expressed in rate form as [3]

$$\sum Ex_{dest} = \sum Ex_{in} - \sum Ex_o \tag{38}$$

Equation (38) can be written as

$$Ex_{dest} = Ex_{ht} - Ex_w + Ex_{in} - Ex_o \tag{39}$$

The rate form of the general exergy balance equation can also be written as:

$$Ex_{dest} = \sum \left(1 - \frac{T_e}{T_s}\right) Q_s - \dot{W} + \sum m_{in} \Psi_{in} - \sum m_o \Psi_o \tag{40}$$

where,

$$\Psi_{in} = (h_{in} - h_e) - T_e (S_{in} - S_e) \tag{41}$$

and

$$\Psi_o = (h_o - h_e) - T_e (S_o - S_e) \tag{42}$$

On substitution of (41) and (42) in (40):

$$Ex_{dest} = \left(1 - \frac{T_e}{T_s}\right) Q_s - m[(h_o - h_{in}) - T_e (S_o - S_{in})] \tag{43}$$

where,

$$Q_s = I \alpha_{ap} \tau_{gc}^2 A_c \tag{44}$$

The change in enthalpy and entropy is given as:

$$\Delta h = h_o - h_{in} = C_p (T_{f,o} - T_{f,in}) \tag{45}$$

$$\Delta S = S_o - S_{in} = C_p ln \frac{T_{f,o}}{T_{f,in}} - R ln \frac{P_o}{P_{in}} \tag{46}$$

By substituting (44)-(46) into (43), we have

$$Ex_{dest} = \left(1 - \frac{T_e}{T_s}\right) I \alpha_{ap} \tau_{gc}^2 A_c - m C_p (T_{f,o} - T_{f,in}) + m C_p T_e ln \frac{T_{f,o}}{T_{f,in}} - m\, R T_e ln \frac{P_o}{P_{in}} \tag{47}$$

The exergy efficiency (η_{exe}) can be calculated as,

$$\eta_{exe} = \frac{Ex_o}{Ex_{in}} = \frac{m(\Delta h - T_e \Delta S)}{\left(1 - \frac{T_e}{T_s}\right) Q_s} \tag{48}$$

F. Calculation Procedure for Energy Efficiency and Exergy Efficiency

For the numerical calculation of energy and exergy efficiencies a program in C++ language have been developed considering the following system and operating parameters:

$$L = 1.25\, m,\ W = 0.80\, m,\ H_g = 0.025\, m,\ H_{min} = 0.025\, m,\ H_{max} = 0.0375 m,\ \tau_g = 0.875,\ \alpha_{ap} = 0.96,\ \varepsilon_g = 0.94,\ \varepsilon_{ap} = 0.80,\ \varepsilon_{bp} = 0.94,\ U_B \approx 0,\ T_a = 30°\text{C} = 303K,\ V = 1\, m/s,\ b = 0.00625\, m,\ I = 1000\, W/m^2,\ m = 0.014 - 0.083\, kg/s,\ \varphi = 0.5.$$

The procedure for calculation of thermal and exergy efficiencies are, first using guess temperature the heat transfer coefficients has been calculated from (14)-(29) then new temperatures are obtained by using (30)-(34). If the calculated values of temperature are different from the assumed values continued calculation by iteration method, these new temperatures will be used as the guessed temperatures for next iteration, and the process will be repeated until all the newest temperatures obtained are their respective previous valuesand their respective previous values were less than 0.001. Using (13) energy efficiency is calculated. Further pressure drop has been estimated using (35). The change in enthalpy and entropy has been calculated by using (45) and (46) respectively, then exergy efficiency has been calculated by using (48).

III. Results and Discussion

In this section, results of energy and exergy performance of double flow corrugated plate and flat plate absorber solar air heaters are presented.

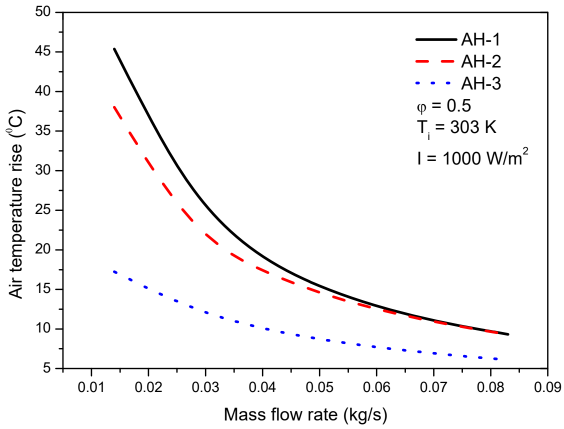


Fig. 2 Air temperature rise vs. mass flow rate

Fig. 2 shows the variation of air temperature rise for double flow corrugated plate (AH-1), flat plate (AH-2) and conventional (AH-3) solar air heaters with mass flow rate for φ = 0.5 (i.e. at equal mass flow rate through both channels). It can be seen from the figure that the air temperature rise decreases with increase in mass flow rate. The air temperature rise through double flow solar air heater has higher value than the conventional solar air heater at lower mass flow rate, because of increase in heat transfer surface area. The surface area becomes double in double flow arrangement with respect to single flow. It is also observed that at higher mass flow rate air temperature rise of AH-1 and AH-2 is almost equal. The maximum value of air temperature rise is 45.3 ˚C for AH-1 collector at m = 0.014 kg/s.

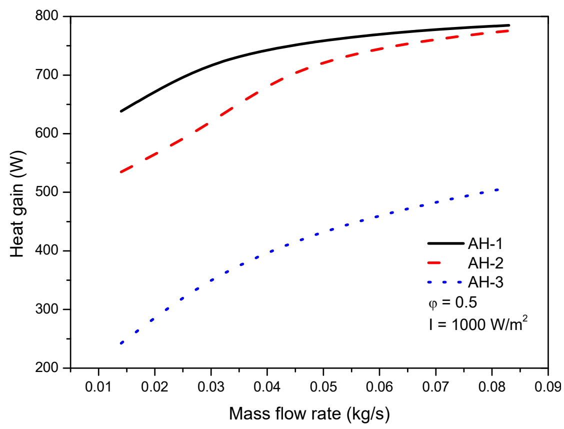


Fig. 3 Heat gain vs. mass flow rate

Fig. 3 shows the plot of heat gain as a function of mass flow rate. Heat gain continuously increases with increase in mass flow rate for all collectors; this is due to increase in heat transfer rate. The double flow corrugated absorber solar air heater has the highest value of heat gain throughout the mass flow rate investigated. The minimum and maximum heat gain by AH-1 collector is 638.24 W and 784.75 W for the mass flow rate of 0.014 kg/s and 0.083 kg/s respectively.

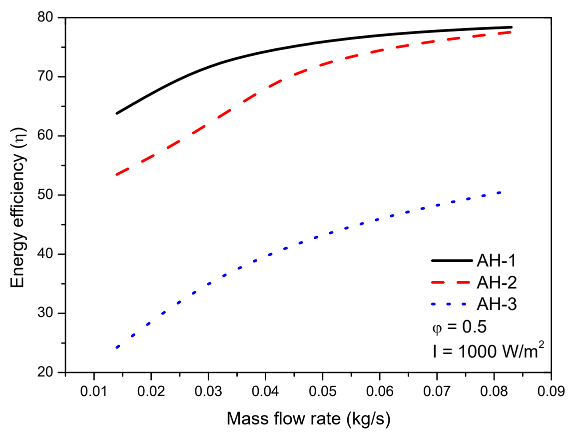


Fig. 4 Energy efficiency vs. mass flow rate

Fig. 4 illustrates the variation of energy efficiency with mass flow rate. It is observed from the figure that the double flow solar air heaters (AH-1 and AH-2) have much higher efficiency than the conventional (AH-3) solar air heater. It is also observed that in double flow collectors, corrugated absorber have higher efficiency than flat plate absorber. The reason behind this is increase in heat transfer surface area and turbulence in air flow which leads to increase convective heat transfer rate. The percentage enhancement in energy efficiency of AH-1 collector is found to be 19.34% over AH-2 collector and 1.6 times over AH-3 collector at m = 0.014 kg/s.

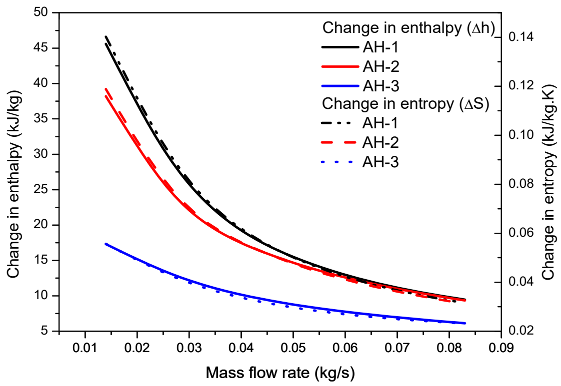


Fig. 5 Change in entropy and enthalpy vs. mass flow rate

The change in enthalpy (Δh) and entropy (ΔS) with respect to mass flow rate is shown in Fig. 5. The figure reveals that, as mass flow rate increases, Δh and ΔS decrease due to irreversibility. The AH-1 collector has highest value of Δh and ΔS for entire mass flow rate. The calculation of Δh is based on specific heat capacity and temperature of air at inlet and outlet, whereas ΔS is calculated based on the temperature and pressure drop. The maximum values of Δh and ΔS for AH-1 is 45.58 kJ/kg and 0.14 kJ/kg.K, for AH-2 is 38.17 kJ/kg and 0.11 kJ/kg.K, for AH-3 is 17.31 kJ/kg and 0.05 kJ/kg.K respectively for the mass flow rate of 0.014 kg/s.

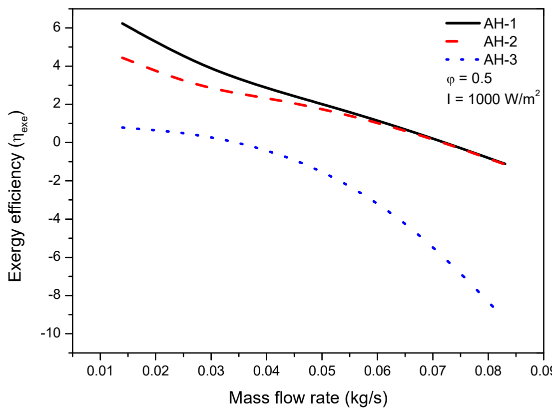


Fig. 6 Exergy efficiency vs. mass flow rate

Fig. 6 presents the effect of mass flow rate on exergy efficiency for double flow corrugated, flat plate and conventional solar air heaters at I = 1000 W/m^2. The exergy efficiency decreases with increase in mass flow rate and it is negative at higher mass flow rate (i.e. m > 0.035 kg/s for AH-3 and m > 0.072 kg/s for Ah-1 and AH-2). It can be observed that double flow corrugated and flat plate leads to exergy efficiency increase compared to conventional solar air heater. It is also observed from the figure that corrugated plate collector is more efficient at lower mass flow rate and enhancement in exergy efficiency decreases with increase in mass flow rate, due to decrease in outlet temperature of air at higher mass flow rate.

IV. Conclusions

On the basis of above results the following conclusions are drawn:

i. The mathematical model has been developed for double flow solar air heaters having corrugated and flat plat absorber, to study the effect of mass flow rate on energy and exergy performance of collector.
ii. A computer program in C^{++} language has been developed to solve the mathematical model and the results are obtained for air temperature rise, heat gain, energy efficiency, pressure drop, change in enthalpy, change in entropy and exergy efficiency to analyze the effect of system and operating parameters.
iii. It is observed that heat gain and energy efficiency increases, however, air temperature rise, change in enthalpy, change in entropy and exergy efficiency decrease with increase in mass flow rate.
iv. Double flow solar air heaters perform better than conventional solar air heaters. However, double flow corrugated absorber solar air heater has the best energy and exergy performance.
v. The exergy efficiency decreases with increase in mass flow rate, due to decrease in outlet temperature of air at higher mass flow rate. At higher mass flow rate i.e. $m > 0.035$ kg/s for conventional solar air heater and $m > 0.072$ kg/s for double flow corrugated and flat plate solar air heaters, the exergy efficiency become negative.

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