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Thermohydraulic Performance of Double Flow Solar Air Heater with Corrugated Absorber

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***Abstract*—**This paper deals with the analytical investigation of thermal and thermohydraulic performance of double flow solar air heaters with corrugated and flat plate absorber. A mathematical model of double flow solar air heater has been presented, and a computer program in C^{++} language is developed to estimate the outlet temperature of air for the evaluation of thermal and thermohydraulic efficiency by solving the governing equations numerically using relevant correlations for heat transfer coefficients. The results obtained from the mathematical model is compared with the available experimental results and it is found to be reasonably good. The results show that the double flow solar air heaters have higher efficiency than conventional solar air heater, although the double flow corrugated absorber is superior to that of flat plate double flow solar air heater. It is also observed that the thermal efficiency increases with increase in mass flow rate; however, thermohydraulic efficiency increases with increase in mass flow rate up to a certain limit, attains the maximum value, then thereafter decreases sharply.

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

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)
f	friction coefficient
h	convective 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	resistance factor
Re	Reynolds number
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)

Greek Symbols

α	absorptivity
ε	emissivity
φ	fraction of mass flow rate
ΔP	pressure drop (N/m^2)
η	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
$\boldsymbol{\theta}$	angle of v-groove absorbing plate (60˚)

Subscripts

a	ambient
ap	absorber plate
B	bottom
bp	bottom plate
c	convective
ch	channel
e	net
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
in	inlet
L	overall
m	mean
max	maximum
min	minimum
o	outlet
r	radiative
T	top
eff	thermohydraulic (effective)
u	useful
w	wind
1	duct above the absorber plate
2	duct under the absorber plate

I. INTRODUCTION

ENERGY plays an important role in world economic progress. The fossil fuels (coal, oil, natural gas) provide the major source of energy which depletes in coming years. Non-conventional renewable energy sources have an important role for industrialization and development. Among all renewable energy sources solar energy is clean, sustainable and abundant form of energy. It has great potential for moderate temperature applications such as for drying of

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agriculture products, preheaters in industries and space heating in building to save energy. Solar air heater is the most important type of solar collector because it is simple in design, has no moving parts and require little maintenance. It is a specific type of heat exchanger which transfers heat energy, obtained by absorbing insolation. Due to the poor thermal conductivity and small heat capacity of air, the convective heat transfer rate inside the air flow channel, where air is heated, is low. One of the effective ways to augment the convective heat transfer rate in channel flows is to increase the heat transfer surface area and turbulence. The heat transfer area and turbulence increase by using fins and baffles. An alternative way for increasing the heat transfer area is the double flow, in which air is flowing simultaneously and separately over and under the absorbing plate, instead of only single flow over or under the absorbing plate.

Several configurations of solar air heaters have been developed in literature for enhancing the performance. The double flow solar air heaters have been introduced for increasing the heat-transfer area, leading to improve thermal performance [1]. A parametric study of cross corrugated solar air collectors with different parameters such as mass flow rate, insolation and inlet temperature have been performed by Wenxian et al. [2] and Wenfeng et al. [3] and found that the cross corrugated collectors have superior thermal performance with respect to that of the flat plate. Paisarn [4] numerically studied heat transfer characteristics and performance of double pass flat plate solar air heater with and without porous media. Stasiek [5] carried out experimental studies on the heat transfer and fluid flow across corrugated-undulated heat exchanger surfaces. Chouksey and Sharma [6] theoretically investigated the thermal performance of packed bed solar air heater with wire screen matrices. Taymaz et al. [7] experimentally investigated the convective heat transfer characteristics in a periodic converging diverging heat exchanger channel. Fabbri [8] analysed the heat transfer in a channel having smooth and corrugated wall under laminar flow conditions and used finite element model to determine the velocity and temperature distributions. Naphon [9] investigated heat transfer characteristics and pressure drop in the channel with v-corrugated upper and lower plates. Piao et al. [10], [11] investigated experimentally natural, forced and mixed convective heat transfer in a cross-corrugated channel solar air heater. Noorshahi et al. [12] conducted a numerical study on the natural convection in a corrugated enclosure with mixed boundary conditions. Gao et al. [13], [15] numerically simulated the natural convection inside the channel formed by a flat cover and a wavelike absorbing plate.

This paper presents analytical investigations of thermal and thermohydraulic performance of double flow corrugated absorber solar air heater. Three types of solar air heaters have been considered; SAH-I (Fig. 1) has double flow corrugated absorber plate, SAH-II (Fig. 2) has double flow flat plate absorber, and SAH-III has conventional solar air heater. A mathematical model of double flow solar air heater has been presented for calculating the outlet air temperature, thermal and thermohydraulic efficiency.

II. Theoretical Analysis

The double flow corrugated absorber solar air heater is presented in Fig. 1. The formation of mathematical models is based on following assumptions: (i) air temperature variation is the function of the flow direction only, (ii) negligible temperature drop across the glass covers, absorbing and bottom plate, (iii) glass cover and flowing air do not absorb radiant energy, (iv) thermal losses through side and bottom insulation are negligible.

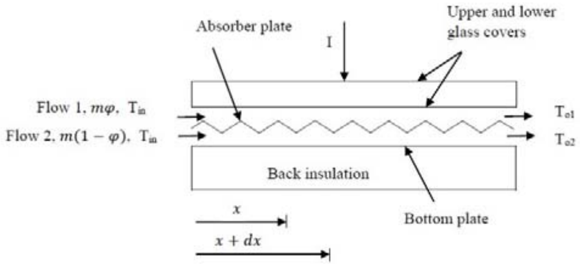

Fig. 1 The double flow corrugated absorber solar air heater

A. Energy Balance Equations

To formulate the energy balance equations, we considered a differential element of length dx at a distance x from the inlet. For the studied systems, the energy balance equations are written as:

- For the lower glass cover (cover 1),

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

- For the absorbing plate,

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

- For the bottom plate,

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

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

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

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

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

Solving (1) – (5) with the boundary condition:At, $\zeta = 0$, $T_{f1} = T_{f2} = T_{f,i}$

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

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

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

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

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

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

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

where $z = \frac{mc_p}{A_c}$ and Ys and Cs are the functions of M, and Ms are in the terms of the convective heat transfer coefficient, loss coefficients, and physical properties [165].

The total thermal energy gain is

$$Q_{uf} = Q_{uf1} + Q_{uf2} \quad (10)$$

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

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

The energy efficiency can be calculated by

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

B. Heat Transfer Coefficients

An empirical equation, derived by Klein [16], is used to calculate top loss coefficient, U_T, and is given by

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

The convective heat transfer coefficient from the outer glass cover (gc2) due to wind is calculated by the expression given by McAdams [17].

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

The heat transfer coefficient between two glass covers, inner glass cover (gc1) and outer glass cover (gc2) is expressed by Hottel and Woertz [18] empirical equation as

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

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

$$h_{c,ap-f1} = h_{c,f1-gc1} \quad (17)$$

and,

$$h_{c,ap-f2} = h_{c,f2-bp} \quad (18)$$

The radiation heat transfer coefficients between the absorbing plate and inner glass cover and between the absorbing plate and bottom plate may be expressed by assuming that the mean radiant temperature is equal to the mean fluid temperature as,

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

and

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

The radiation heat transfer coefficients between the two glass cover inner glass cover and outer glass cover and outer glass cover and air are respectively as,

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

and

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

For corrugated absorber, the developed area of the corrugated plate is greater than the flat plate by a factor of $1/sin(\theta/2)$ [19], thus the heat transfer coefficient between absorbing plate to fluid is

$$h_{c,ap-f1} = \frac{Nu_1 k_{f1}}{D_{h1}} \times \frac{1}{sin\left(\frac{\theta}{2}\right)} \tag{23}$$

$$h_{c,ap-f2} = \frac{Nu_2 k_{f2}}{D_{h2}} \times \frac{1}{sin\left(\frac{\theta}{2}\right)} \tag{24}$$

Karim et al. [19] modified the Hollands and Shewen [21] correlation of Nusselts number (Nu) as:
If $Re < 2800$

$$Nu = 2.821 + 0.126\, Re \frac{2b}{L} \tag{25}$$

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

$$Nu = 1.9 \times 10^{-6} Re^{1.79} + 225 \frac{2b}{L} \tag{26}$$

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

$$Nu = 0.0302 Re^{0.74} + 0.242 Re^{0.74} \frac{2b}{L} \tag{27}$$

For flat plate absorber, the convective heat transfer coefficient for air.

$$h_{c,ap-f1} = \frac{Nu_1 k_{f1}}{D_{h1}} \tag{28}$$

$$h_{c,ap-f2} = \frac{Nu_2 k_{f2}}{D_{h2}} \tag{29}$$

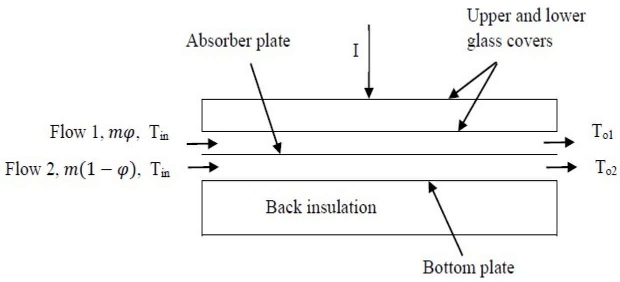

Fig. 2 The double flow flat plate absorber solar air heater

For laminar flow, the equation presented by Heaton et al. [21],

$$Nu_i = 4.4 + \frac{0.00398(0.7 Re_i D_{hi}/L)^{1.66}}{1 + 0.0114(0.7 Re_i D_{hi}/L)^{1.12}} \tag{30}$$

For turbulent flow the correlation derived from Kays [22], data with the modification of McAdams [17],

$$Nu_i = 0.0158 Re_i^{0.8}[1 + (D_{hi}/L)^{0.7}] \tag{31}$$

where $i = 1, 2$ (for flow 1, flow 2).

For the conventional solar air heater, the calculation procedure of collector efficiency is much simpler than for double flow solar air heater, and it is 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

$$T_{f1,m} = \left[\frac{Y_1 - \frac{M_5}{1-\varphi}}{\frac{M_4}{(1-\varphi)}}\right] C_1 \frac{Z}{Y_1} e^{(\frac{Y_1}{Z}-1)} + \left[\frac{Y_2 - \frac{M_5}{1-\varphi}}{\frac{M_4}{(1-\varphi)}}\right] C_2 \frac{Z}{Y_2} e^{(\frac{Y_2}{Z}-1)} - \frac{M_5}{M_4}\left(\frac{M_3M_4 - M_1M_6}{M_1M_5 - M_2M_4}\right) - \frac{M_6}{M_4} + T_a \tag{32}$$

$$T_{f2,m} = C_1 \frac{Z}{Y_1} e^{(\frac{Y_1}{Z}-1)} + C_2 \frac{Z}{Y_2} e^{(\frac{Y_2}{Z}-1)} + \frac{M_3M_4 - M_1M_6}{M_1M_5 - M_2M_4} + T_a \tag{33}$$

The mean absorbing plate temperature can be calculated by

$$T_{ap,m} = \frac{I\alpha_{ap}\tau_{gc}^2 + U_L T_a + h_{c,ap-f1}T_{f1,m} + h_{c,ap-f2}T_{f2,m}}{U_L + h_{c,ap-f1} + h_{c,ap-f2}} \tag{34}$$

The mean temperature of inner glass cover (gc1)

$$T_{gc1,m} = \frac{U_{gc1-a}T_a + h_{r,ap-gc1}T_{ap,m} + h_{c,f1-gc1}T_{f1,m}}{U_{gc1-a} + h_{r,ap-gc1} + h_{c,f1-gc1}} \tag{35}$$

The mean temperature of outer glass cover (gc2)

$$T_{gc2,m} = \frac{(h_{c,gc1-gc2} + h_{r,gc1-gc2})T_{gc1,m} + h_w T_a}{h_{c,gc1-gc2} + h_{r,gc1-gc2} + h_w} \tag{36}$$

D. Thermohydraulic Efficiency

The net energy gain, Q_e, of the collector can be expressed as the difference between the useful thermal energy gain, Q_{uf}, and the equivalent thermal energy required for producing the work energy necessary to overcome the pressure energy losses [23]. This net energy can be written as:

$$Q_e = Q_{uf} - P_m/C_f \tag{37}$$

where P_m is the work energy lost in friction in the heater channel, given by:

$$P_m = m\Delta P/\rho \tag{38}$$

C_f is the conversion factor to transform different efficiencies (thermal to mechanical) and is taken 0.2 [24].

The pressure drop ΔP is calculated from;

$$\Delta P = \Delta P_{ch} + \Delta P_{en} + \Delta P_{ex} \tag{39}$$

The pressure drop through the upper and lower channel ΔP_{ch} is calculated by the relation [25]-[27];

$$\Delta P_{ch} = 2\rho v_{ch}^2 f L/D_{hi} \tag{40}$$

Hydraulic diameter for flat plate (smooth channel) solar air heater;

$$D_{hi} = 2H_c W/(H_c + W) \tag{41}$$

Hydraulic diameter for v-corrugated absorber solar air heater;

$$D_{hi} = 2H_c W/(H_c + W) \times sin(\theta/2) \tag{42}$$

The friction factor is given by [25]-[28];
for turbulent flow

$$f = 0.059Re^{-0.2} \tag{43}$$

for laminar flow

$$f = 16/Re \tag{44}$$

The sum of the inlet and outlet pressure drop $(\Delta P_{en} + \Delta P_{ex})$ can be determined by Hegazy [29];

$$\Delta P_{en} + \Delta P_{ex} = (R_{en} + R_{ex})\frac{\rho v_p^2}{2} \tag{45}$$

where the sum of the entrance and exit resistance factor $(R_{en} + R_{ex})$ is taken as 1.5 [30].

The thermohydraulic efficiency of the solar air heater can be expressed as;

$$\eta_{eff} = \frac{Q_e}{A_c I} = \frac{Q_{uf} - (P_m/C_f)}{A_c I} \tag{46}$$

E. Calculation Method for Temperatures, Thermal Efficiency and Thermohydraulic Efficiency

The procedure for calculation of efficiency is to solve, first using guessed temperature to calculate the heat transfer coefficients, (14)–(31), and then, to estimate new temperatures by using (32)–(36). 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 values. Using (13), we calculate the thermal efficiency. The pressure drop has been calculated by using (39) including channel pressure drop, entrance and exit pressure drop, then we calculate the thermohydraulic efficiency by (46).

III. Results and Discussion

In this section, results of thermal and thermohydraulic performance of double flow corrugated plate and flat plate absorber solar air heaters are presented. The following values are used given in Table I for the systems and operating parameters.

Fig. 3 illustrates the effect of mass flow rate on air temperature rise at φ = 0.5; for double flow corrugated plate (SAH-I), double flow flat plate (SAH-II) and conventional (SAH-III) solar air heaters. This figure reveals that the air temperature rise decreases with increase in mass flow rate for all types of solar air heaters. It is seen that double flow solar air heaters have much higher air temperature rise to the conventional solar air heater at lower mass flow rate, and this is due to enhancement in heat transfer surface area. It is also observed that, at higher mass flow rate, air temperature rise of SAH-I and SAH-II is almost equal. The maximum value of air temperature rise is 45.3 °C for SAH-I, 38.0 °C for SAH-II, and 17.2 °C for SAH-III at m = 0.014 kg/s.

TABLE I
SYSTEMS AND OPERATING PARAMETERS USED WITH PRESENT INVESTIGATION

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

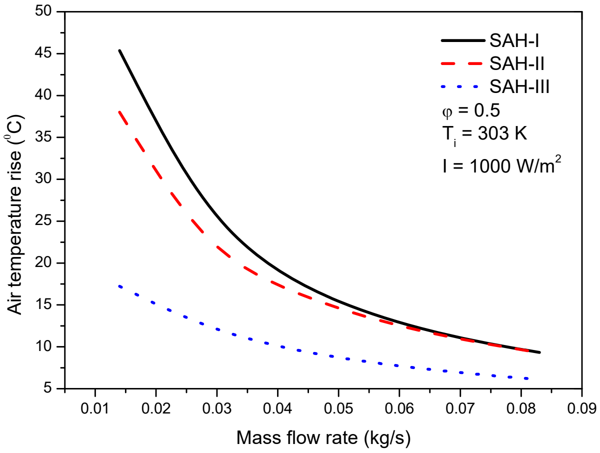

Fig. 3 Effect of mass flow rate on air temperature rise

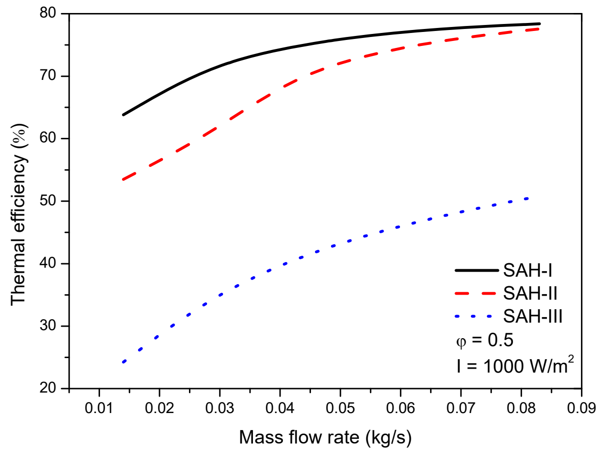

Fig. 4 Effect of mass flow rate on thermal efficiency

The variation of thermal efficiency as a function of mass flow for a set of parameters is shown in Fig. 4. It is revealed that the SAH-I and SAH-II collectors have much higher efficiency than the conventional (SAH-III) collector and SAH-I collector performs superior to the SAH-II collector. This is because of 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 corrugated plate collector to conventional collector is 163.32% at m = 0.014 kg/s and 54.27% at m = 0.083 kg/s.

The effect of mass flow rate on the channel pressure drop is shown in Fig. 5. It is clearly seen from this figure that the pressure drop increases with the increase in mass flow rate. This is because of fact that increase in mass flow rate increases the velocity of air leading to higher pressure drop. It is seen that the corrugated absorber collector (SAH-I) has higher pressure drop as compared to flat plate collector (SAH-II) because of increased friction, and SAH-II have higher pressure drop than SAH-III because increased surface area causes increased resistance during air flow.

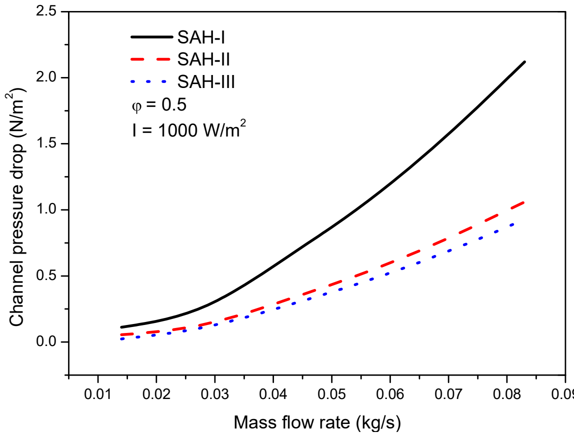

Fig. 5 Effect of mass flow rate on channel pressure drop

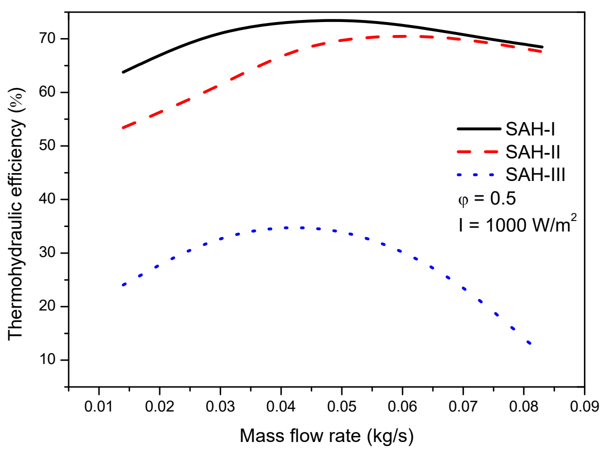

Fig. 6 Effect of mass flow rate on thermohydraulic efficiency

Fig. 6 shows the effect of mass flow rate on thermohydraulic efficiency for double flow corrugated, flat plate, and conventional solar air heaters at I = 1000 W/m^2. It is revealed that the thermohydraulic efficiency increases with the increase in mass flow rate upto a critical value of flow rate at which it attains a maximum value and there after decreases sharply. Double flow collectors have much higher thermohydraulic efficiency than conventional collector. Results indicate that the maximum thermohydraulic efficiency of SAH-II collector reaches maximum value at m = 0.0575 kg/s, whereas for SAH-I collector pick values of thermohydraulic efficiency shifted towards lower mass flow rates of 0.0475 kg/s. This type of trends is observed due to increase in pressure drop of flowing air in case of corrugated absorber/channels.

The present mathematical models of solar air heaters have been compared with the experimental values obtained from El Sebaii et al. [31]. Fig. 7 shows the comparison of analytical with experimental values of thermal efficiency of El Sebaii et al. [31] of double flow flat plate and corrugated absorber solar air heater. The maximum deviation in efficiency for flat plate and corrugated collectors are found to be ±5.47 % and 4.46%, respectively. This shows good resemblance of analytical and experimental values, which makes the validation of calculated numerical data with present mathematical modelling.

IV. Validation of Mathematical Model

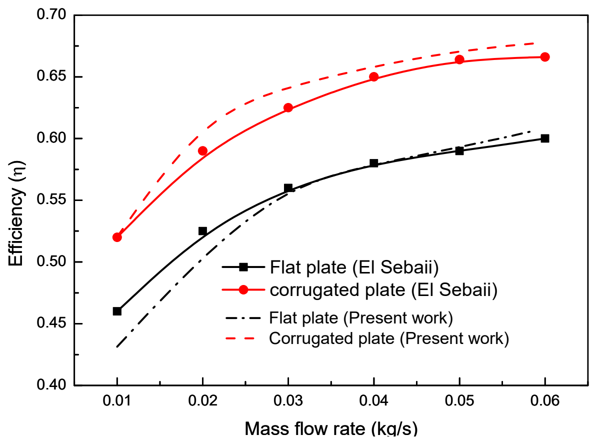

Fig. 7 Comparison of analytical efficiency data with available experimental data of El Sebaii [31]

V. Conclusions

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

i. The mathematical model for double flow solar air heater has been developed to study the effect of mass flow rate, fraction of mass flow rate and angle of v-corrugated absorber on the thermal and thermohydraulic performance of collector.
ii. A computer program in C++ language has been developed to solve the mathematical model and we obtained the results of air temperature rise, thermal efficiency, pressure drop and thermohydraulic efficiency to analyse the effect of system and operating parameters.
iii. It is observed that thermal efficiency increases with the increase in mass flow rate but thermohydraulic efficiency increases upto a certain limit of mass flow rate and there after it decreases.
iv. Double flow collectors perform better than conventional

collector. However, corrugated absorber double flow collector has the best thermal and thermohydraulic performance.

v. The analytical and experimental values of thermal efficiency have been found to compare reasonably well in the range of parameters investigated. The percentage deviation in efficiency of double flow flat plate and corrugated absorber solar air heater is found to be in the range of ± 5.47 % and 4.46%, respectively.

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