Thermal and Thermohydraulic Performance Analysis on Offset Finned Absorber Solar Air Heater

Shalini Rai*, Prabha Chand*, S.P. Sharma*
*Mechanical Engineering Department, NIT Jamshedpur, Jharkhand (India)
**Corresponding Author; Shalini Rai; NIT Jamshedpur, Tel:+9896338256, rai.shalin@gmail.com

Abstract-- In this paper, the thermal and thermohydraulic performance analysis on offset finned absorber solar air heater has been investigated theoretically. The thermal efficiency indicated the portion to which the solar energy added by insolation is converted to network output. Whereas the thermohydraulic (effective) efficiency invokes to how useful energy is to accomplish a purpose. A parametric study was done to investigate the effect of variation of system and operating parameters i.e. fin spacing, fin height, air mass flow rate and insolation 950 W/m² on the thermal and effective efficiency. The analytical approach gives adequate foresight of the performance of offset finned solar air heater and it can be helpful in designing such types of solar air heater.

Index Term-- plane solar air heater, offset fins, thermal efficiency, effective efficiency.

1. INTRODUCTION

With new advancements in scientific investigations, solar energy could be more economical in future with low cost and high efficiency. It is most important in the midst of renewable energy resources due to its quantifiable side. The effortless and the most efficient way to utilize solar energy are to convert it into thermal energy for heating applications. The flat plate solar air heater has a captivate radiative heat exchanger, convert solar radiant energy into heat, which is pass on through convection from the absorber plate to the air. Solar air heater can be used adequately for different applications, i.e. crops drying and therapeutic plants to repress the loss of medicinal effect from direct radiation, dehydration of food, heating of space and thermal storage. By virtue of lower thermo physical properties of air, extended surfaces, i.e. packed beds, artificial roughened absorber and fins and baffles are used to improve the heat transfer rate without increasing the solar air heater size (Duffie and Beckman, 1980; Kriers and Krieth, 1978). An experimental analysis was done by Youcef, A.S., (2005) to investigate the thermal performance of an offset rectangular plate fins attached below the absorber plate in staggered pattern and oriented parallel to the fluid flow. Results show that by high heat transfer surface area per unit volume are obtained high thermal performances with low pressure loss. Ming, Y., et al (2014) designed and optimized a solar air heater with offset strip fins by numerical modeling. The work was useful for developing energy efficient and cost effective solar air heaters. Another investigator, Youcef, A.S., Desmons, J.Y (2006) developed a mathematical model to evaluated the thermal performances of the single pass solar air collector with offset rectangular plate fin absorber plate. Result indicates that by high surface heat transfer area per unit volume, generate the low pressure loss and Hachemi, A (1999) studied the performance of offset rectangular plate fins and developed the new technique to enhance the heat transfer more with fully developed turbulent flow. From the result the high thermal performances were obtained with low flow friction and in consequence low electrical power consumption by the fan in comparison to the flat plate collector. Junqi, D., et al (2007) studied sixteen different types offset strip fins to evaluate the air-side heat transfer and pressure drop characteristics. And flat tube heat exchangers were performed experimentally. The heat transfer coefficients and pressure drop data with different fin space, fin height, and fin length were reported in terms of frontal air velocity. The theoretical investigation of the thermal and thermohydraulic performance of wavy finned absorber solar heater by Priyam, A. and Chand, P. (2016). They evaluated the effect of mass flow rate and fin spacing on the performance of solar air heater. An experimental study based on energy and exergy analysis has done by Alta, D., et al. (2010) in order to determine the performance of three different types of flat-plate solar air heaters. Their results showed that the energy and exergy efficiencies of air heater with fins and double glass cover are higher. An experimental evaluation of evaluated the energy and exergy efficiency by Esen, H (2008) on four types of flat-plate solar air heaters with and without obstacles on the absorber plate. From the analysis, found that all of three solar air heaters with obstacle on the absorber plate show better performance in comparison with the one without obstacle. Moreover, results indicated that the highest irreversibility happen for the solar air heater without obstacle, having the smallest value of efficiency. Recently Sahu, M.K., and Prasad, R.K. (2016) investigated the exergetic performance evaluation of solar air heater with arc-shaped wire rib roughened absorber plate. Results showed that the exergy analysis is one of the important methods to evaluate the performance of solar air heater.

It is widely spread that the collector geometric size controls the air velocity, also the effect of forced convection. A simple technique for flow control of air and the limitation of forced convection incorporate by adjusting the channel duct of flow system with offset finned attached below the absorber plate.

In the present paper, an analytical investigation on elemental volume of offset finned attached below the absorber solar air heater is being reported. Effect of system and operating parameters such as fin spacing, fin height and air mass flow
rate on thermal and thermohydraulic performance has been also reported and results are compared with conventional flat plate solar air heater.

2. THEORETICAL ANALYSIS

A solar air heater having offset finned below the absorber plate as shown in Figs.1-2 has been considered. The channel of width ‘\( W_{sc} \)’, duct height ‘\( D_h \)’, and length of the collector absorber plate ‘\( L_{sc} \)’, having one glass cover is uniformly heated from top by solar radiations transmitted through the glass cover. The height, length, thickness and spacing of the offset fin are ‘\( h_f \)’, ‘\( l_f \)’, ‘\( t_f \)’, and ‘\( s_f \)’ respectively.

![Fig. 1. Solar air heater absorber plate with parameters of offset fin](image)

![Fig. 2. Schematic diagram of bottom view absorber plate attached with offset fin.](image)

The general heat balance equations are written on the basis of following assumptions Duffie and Beckman, (1980):

2. There is no absorption of solar energy by glass cover.
3. One dimensional heat flow through glass cover and back insulation.
4. The covers are opaque to infrared radiation.
5. Temperature drop through glass cover, absorber plate and bottom plate are negligible small.
6. There is no heat generation.
7. Aspect ratio (\( W_{sc}/D_h \)) is very large.

2.1. General heat balance equations

Consider elemental view of width ‘\( W_{sc} \)’ and thickness ‘\( dx \)’ at a distance x from inlet as shown in Figs 3-4, then the energy balance equations for the absorber plate, the bottom plate and the air flowing in between can be written respectively as:

\[
I_0(\alpha_{t},\alpha_{0}) = U_I(\theta_{in} - \theta_{a}) + h_{aa}(\theta_n - \theta_t) + h_{fa}\phi_{fin}(\theta_n - \theta_t) + h_{ra}(\theta_n - \theta_2) \tag{1}
\]

\[
h_{ra}(\theta_n - \theta_2) = h_{aa}(\theta_n - \theta_t) + U_{bt}(\theta_2 - \theta_a) \tag{2}
\]

\[
C_p \frac{dT_{fin}}{W_{sc}dx} = h_{aa}(\theta_2 - \theta_t) + h_{aa}(\theta_n - \theta_t) + h_{fa}\phi_{fin}(\theta_n - \theta_t) \tag{3}
\]

The dimensionless quantity \( \phi_{fin} \) is defined as:

\[
\phi_{fin} = 1 + \left( \frac{A_t}{A_{sc}} \right) \eta_{fin} \tag{4}
\]

Total surface area of fin \( A_t \) and fin efficiency \( \eta_{fin} \) is calculated as:

...
where

\[ \eta_{fn} = \frac{\tanh(MD_n)}{MD_n} \]  

and

\[ M = \frac{2h_f}{k_{fin}t_f} \]

### 2.2. Heat transfer coefficient

The coefficient of an equivalent radiative heat transfer ‘\( h_{ra} \)’ can be expressed as Duffie and Beckman (1980):

\[ h_{ra} = \frac{\sigma_{sb}(\theta_n + \theta_g)(\theta_n^2 + \theta_g^2)}{\left(\frac{1}{\theta_n^2} + \frac{1}{\theta_g^2}\right) - 1} \]

(7)

where the temperature \( \theta_n \) and \( \theta_g \) are in Kelvin.

The average heat transfer coefficient is given as:

\[ h_{aa} = h_{ba} = h_{fa} = \left(\frac{\eta_{fin}k_{air}}{d_c}\right) \]

(8)

The Nusselt number, \( N_{fin} \) and the colburn factor \( j_f \) was recommended by Manglik, R.M., and Bergles, A.E (1995) was used to calculate heat transfer coefficient for fluid flowing through the channel duct in laminar, transition, and turbulent regions:

\[ j_f = 0.6522R^0.5403\theta^0.1541b0.1499e^{-0.0678}[1 + 5.269 \times 10^{-5}a0.5041b0.456c^{-1.055}]^{0.1} \]

(9)

where

\[ j_f = \frac{N_{un}}{R_{en}P_{in}} \]

Reynolds number can be evaluated as

\[ R_{en} = \nu_{aa} \frac{d_c}{\nu_a} \]

(10)

The average velocity is expressed as

\[ v_{aa} = \left(\frac{n}{\rho_aA_{cs}}\right) \]

(11)

Air channel duct cross surface area ‘\( A_{cs} \)’ of the collector is calculated as

\[ A_{cs} = \left(W_{sc}D_b - (W_{sc}t_f(s_f + t_f))\right) \]

(12)

The equivalent hydraulic diameter of duct channel is calculated as Kays and London (1958):

\[ d_e = \frac{4s_fh_f}{\left(3.6s_f + \pi h_f + t_f + h_f\right)} \]

(13)

The collector efficiency factor \( F'_{c} \) can be expressed as:

\[ F'_{c} = \frac{h_{ra}h_{ba}\Phi_{fin} + h_{ra}h_{aa} + h_{ba}h_{ba}\Phi_{fin} + h_{aa}h_{ba}\Phi_{fin}}{(U_{1f} + h_{aa} + h_{ra})(U_{bt} + h_{ba}\Phi_{fin} + h_{ra}) - h_{ra}^2} \]

The overall loss coefficient \( U_{OL} \) can be expressed by following empirical correlation Youcef, A.S. (2005):

\[ U_{OL} = \left(\frac{(U_{bt} + U_{1f})(h_{ra}h_{aa} + h_{ba}h_{ba}\Phi_{fin} + h_{ba}h_{ba}\Phi_{fin} + U_{bt}U_{1f}(h_{aa} + h_{ba}\Phi_{fin}))}{h_{ra}h_{ba}\Phi_{fin} + h_{ra}h_{aa} + U_{bt}h_{ba} + h_{ba}h_{ba}\Phi_{fin}}\right) \]

(15)

The collector heat removal factor can be expressed as:

\[ F_{HR} = \left(\frac{n_cP_{in}}{A_{sc}U_{OL}}\right) \left[1 - \exp\left(-\frac{A_{sc}U_{OL}F'_{c}}{n_cP_{in}}\right)\right] \]

(16)

The coefficient of top loss through the collector \( U_{tl} \) was used by Priyam, A. and Chand, P. (2016) given as:

\[ U_{tl} = \left(\frac{C}{2n} \left(\frac{\beta_n - \beta_g}{n + f}\right) + \left(\frac{1}{n_c}\right) + \frac{\sigma_{sb}(\theta_n + \theta_g)(\theta_n^2 + \theta_g^2)}{e_b + 0.0059n\theta_{cv}^{-1} + 2n^2 + 2 + 14.1333\theta_b} - n\right) \]

(17)

where

\( n = 1, \) number of glass cover,

\( f = 1 + 0.089h_{cv}^{-1} - 0.1166h_{cv}^{-1}(1 + 0.07866n),\)

\( C = 520(1 - 0.000051\beta_t^2).\)

For \( 0^\circ < \beta_t < 70^\circ \)

\( y = 0.430(1 - \frac{100}{\theta_n}).\)

The bottom loss coefficient through the collector is given as:

\[ U_{bt} = \left(\frac{1}{k_{bt}} + \left(\frac{1}{n_{cv}}\right)\right)^{-1} \]

(18)

### 2.3. Temperature distribution equation

The outlet air temperature collector can be obtained from an energy balance equation:

\[ \theta_{o_t} - \theta_{a_t} = \frac{\theta}{\rho_S - \theta_{a_t}} \]

(19)

The mean temperature of the absorber plate and the back plate are obtained by solving the energy balance equation given as:

\[ \theta_n - \theta_t = \frac{(\frac{U_{bt}U_{1f}(h_{aa} + h_{ba}\Phi_{fin} + h_{ra}) - (h_{ra} - \theta_t)(U_{bt}U_{1f} + h_{aa} + h_{ba}\Phi_{fin} + h_{ra})}{(U_{1f} + U_{bt} + h_{aa} + h_{ba}\Phi_{fin} + h_{ra})} - h_{ra}^2\]

(20)

\[ \theta_{s} - \theta_{t} = \frac{h_{ra}S - (h_{ra} - \theta_t)(U_{bt}U_{1f} + h_{aa} + h_{ba}\Phi_{fin} + h_{ra})}{(U_{1f} + U_{bt} + h_{aa} + h_{ba}\Phi_{fin} + h_{ra})} - h_{ra} \]

(21)

The mean air stream temperature can be expressed as

\[ \theta_t = \theta_{a_t} + \left(\frac{Q_{ug}}{F_{fin}U_{OL}}\right) \left[1 - \left(\frac{F_{fin}U_{OL}}{n_cP_{in}}\right)\right] \]

(22)

The absorbed solar energy is defined by:
The useful energy gain is expressed as:

\[ Q_{ug} = A_{sc} F_{HR} (S - U_{OL} (\theta_H - \theta_D)) \]  

(24)

2.4. Thermal efficiency

The thermal efficiency can be expressed as:

\[ \eta_{th} = \frac{Q_{ug} (\text{useful gain of energy carried away by air})}{I_0 A_{sc}}. \]

\[ \eta_{th} = \frac{n c_p (\theta_{fo} - \theta_{fl})}{I_0 A_{sc}} \]  

(25)

2.5. Thermohydraulic performance

Thermohydraulic performance of offset finned solar air heater can be optimized by considering the conversion factor which has been responsible for actual energy gain from conversion of primary energy to mechanical energy. Due to large amount of the energy lost in conversion. In order to investigate the thermohydraulic performance of the collector, the following expression for effective efficiency has been used in the present analysis.

\[ \eta_{eff} = \frac{Q_{ug} - P_{mech}}{I_0 A_{c}} \]  

(26)

where \( C_f \), conversion factor, the value of \( C_f \), as recommended by Cortes, A., and Piacentini, R (1990) is 0.18.

The mechanical power, \( P_{mech} \) required to blow the air through the duct is given by

\[ P_{mech} = \frac{\dot{m} \Delta p_d}{\rho_a} \]  

(27)

The pressure drop, \( \Delta p_d \) across the duct of solar air heater of length, \( L_{sc} \), can be determined from the following expression:

\[ \Delta p_d = \frac{f_{p} L_{sc} \rho_a v^2}{2d_e} \]  

(28)

The friction factor, \( f_p \), can be calculated using the correlation reported by Manglik, R.M., and Bergles, A.E (1995) as:

\[ f_p = 9.6243 \times 10^{-0.7422} a^{-1.856} b^{0.3053} c^{-0.2659} (1 + 7.669 \times 10^{-0.7422} a^{-1.856} b^{0.3053} c^{-0.2659}) \]

The effective efficiency has been evaluated using Eq. (26) for various values of system geometrical and operating parameters.

3. RESULTS AND DISCUSSION

A proper code in MATLAB 7.8.0 R2009a was developed by considering the following system, system properties and operating conditions as listed in Table I for an analytical investigation on thermal and thermohydraulic performance of offset finned absorber solar air heater.

<table>
<thead>
<tr>
<th>Input data</th>
<th>Numerical value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length of collector, ( L_{sc} )</td>
<td>1.5 m</td>
</tr>
<tr>
<td>Width of collector, ( W_{sc} )</td>
<td>1 m</td>
</tr>
<tr>
<td>Channel duct height, ( D_h )</td>
<td>2 cm to 6 cm</td>
</tr>
<tr>
<td>Offset fin height, ( h_f )</td>
<td>1.8 cm to 5.8 cm</td>
</tr>
<tr>
<td>Offset fin spacing, ( s_f )</td>
<td>1 cm to 5 cm</td>
</tr>
<tr>
<td>Offset fin length, ( l_f )</td>
<td>2 cm</td>
</tr>
<tr>
<td>Offset fin thickness, ( t_f )</td>
<td>3 mm</td>
</tr>
<tr>
<td>Mass flow rate in kg/s, ( \dot{m} )</td>
<td>0.0139 kg/s to 0.083 kg/s</td>
</tr>
<tr>
<td>Mass flow rate in kg/h, ( G_a )</td>
<td>50 kg/h to 300 kg/h</td>
</tr>
<tr>
<td>Product of transmissivity and absorptivity, ( \tau_c \alpha_a )</td>
<td>0.85</td>
</tr>
<tr>
<td>Solar insolation, ( I_o )</td>
<td>950 W/m²</td>
</tr>
<tr>
<td>Insulation thickness, ( t_{is} )</td>
<td>4 cm</td>
</tr>
<tr>
<td>Insulation thermal conductivity, ( k_{is} )</td>
<td>0.033 W/m-K</td>
</tr>
<tr>
<td>Ambient temperature, ( \theta_a )</td>
<td>298 K</td>
</tr>
<tr>
<td>Specific heat of air, ( c_p )</td>
<td>1.005 kJ/kg-K</td>
</tr>
<tr>
<td>Thermal conductivity of air, ( k_{air} )</td>
<td>0.02826 W/m-K</td>
</tr>
<tr>
<td>Kinematic viscosity of air, ( \mu_k )</td>
<td>18.97×10^{-6} m²/s</td>
</tr>
<tr>
<td>Emissivity of wood, ( \varepsilon_w )</td>
<td>0.93</td>
</tr>
<tr>
<td>Emissivity of glass, ( \varepsilon_g )</td>
<td>0.88</td>
</tr>
<tr>
<td>Emissivity of back plate, ( \varepsilon_w )</td>
<td>0.9</td>
</tr>
</tbody>
</table>

Fig. 5 shows the collector efficiency factor as a function of mass flow rate of air for different value of fin spacing and for insolation \( I_o=950W/m² \). It is seen from the figure that use of attaching offset finned below the absorber plate, a substantial enhancement of collector efficiency factor as compared to plane solar air heater is achieved. It has been found that the collector efficiency factor moderately increases with increase in mass flow rate and decrease in fin spacing.
The enhancement in the collector efficiency factor decreases with increase in mass flow rate as seen from the figure. This is due to the fact that the heat transfer coefficient increases as the fluid velocity increases with increase in mass flow rate and decrease in fin spacing, although the mean plate temperature decreases.

Fig. 6 shows the variation of collector efficiency factor as a function of mass flow rate for different fin height. From the figures it is clearly seen that collector efficiency factor increases with increase in mass flow rate for all values of fin height. It is also observed that increase in height of offset fin decreases the collector efficiency factor for all values of mass flow rate. This is because of increasing in offset fin height increases the heat transfer surface area; however it decreases the convective heat transfer coefficient.

Fig. 7 shows the effect of fin spacing of offset finned below the absorber plate solar air heater on the heat removal factor at various mass flow rate of air. From the plots it is seen that heat removal factor increases with increase in mass flow rate and decrease in fin spacing. The lower value of fin spacing shows the drastic increase in collector heat removal factor. This is probably because of increase in surface conductance of fins to flowing air.

Fig. 5. Collector efficiency factor vs. mass flow rate for different fin spacing.

Fig. 6. Collector efficiency factor vs. mass flow rate for different fin height.

Fig. 7. Heat removal factor vs. mass flow rate for different fin spacing.
Fig. 8 shows the variation of heat removal factor as a function of mass flow rate and at different fin height. Results show that heat removal factor randomly increases with increase in mass flow rate. It is also observed that increase in height of offset fin decreases the heat removal factor for different mass flow rate. This is due to facts that increase in fins height (i.e. duct height) decreases the surface conductance of the fins.

The variation of outlet fluid temperature and thermal efficiency with mass flow rate for different fin spacing along with plane solar air heater is plotted in Fig. 9. From the figures it is found that the trend of variation of outlet fluid temperature and thermal efficiency is reversed with mass flow rate. Outlet temperature of fluid decreases whereas thermal efficiency increases with increase in mass flow rate. Accordingly with increase in mass flow rate, the percentage of enhancement of outlet fluid temperature decreases. It is also seen the outlet fluid temperature and thermal efficiency increases for lower value ($s_f=1\text{cm}$) of fin spacing. This is because the heat flow rate increases at lower fin spacing due to increase in heat transfer rate.

Fig. 10 illustrates the variation of the outlet fluid temperature and thermal efficiency with mass flow rate for different fins spacing. It has been seen that the trend of variation of outlet temperature and thermal efficiency with respect to mass flow rate is reverse. It is also found that outlet temperature and thermal efficiency decreases for higher value of fin height ($h_f=5.8\text{cm}$) at different mass flow rate. This effect can be attributed to the facts that increase in fin height decreases the surface conductance of fins.
The variation of the calculated values of the pressure drop vs. mass flow rate for different fin spacing of offset finned absorber solar air heater along with plane solar air heater are presented in Fig. 11 for \( Io = 950 \text{W/m}^2 \). It is seen that pressure drop increases with increase in mass flow rate for all value of fin spacing. Also, the pressure drop increases drastically with decrease in fin spacing and increase in mass flow rate. This is because of fact that decrease in fin height/duct height increases the velocity of flowing air and subsequently increases the pressure drops.

Effect of fin height on pressure drop for different mass flow rate along with plane solar air heater has been represented in Fig. 12. From the figure it is clearly seen that the pressure drop increases randomly with increase in mass flow rate and decrease in fin height/duct height increases the velocity of air and subsequently increases the pressure drop.

Fig. 13 shows the variation of friction factor as a function of Reynolds number for different fin spacing along with plane solar air heater for insolation \( Io = 950 \text{W/m}^2 \). From the figure it can be seen that the trend of variation of friction factor and Reynolds number with respect to mass flow rate, reversed. It is found that the Reynolds number decreases and friction factor increases with decrease in mass flow rate for different value of fin spacing, whereas increases in mass flow rate the variations are vice-versa. It is also observed that for lower fin spacing and higher mass flow rate the percentage enhancement of Reynolds number are higher as compared to lower mass flow rate. Smaller fin spacing reduces the cross section area of flowing fluid, which increase the friction between the air and surface area of offset fins.
Fig. 14 shows the effect of mass flow rate on thermal and effective (thermohydraulic) efficiency of offset fins absorber along with plane solar air heater for different value of fin spacing and insolation $I_o=950\text{W/m}^2$. It is seen from figures that as the air mass flow rate increases, the thermal efficiency increases continuously due to increase in convective heat transfer coefficient. However, thermohydraulic efficiency increases up to a threshold value of mass flow rate, attains a maximum, and then decreases sharply, there exists an optimum value of thermohydraulic efficiency for a given fins spacing. It is found that at lower fins spacing ($s_f = 1\text{cm}$) the thermohydraulic efficiency is almost equal to thermal efficiency in between mass flow rate of 0.020 to 0.030 kg/s and thereafter start decreasing gradually up to 0.062 kg/s and then decreases sharply at higher mass flow rate as compared to thermal efficiency.

The effect can be attributed to the fact that Reynolds number is strong parameter that affects the pumping power and thermal energy gain, thereby affecting the effective efficiency. However for other fin spacing there is slight fall in effective efficiency is observed in the range of mass flow rate investigated.

Figs. 15 shows the effect of mass flow rate on thermal and thermohydraulic efficiency for different fin height and insolation $I_o=950\text{W/m}^2$. It is found that thermal efficiency increases continuously with increase in mass flow rate, whereas effective efficiency increases up to 0.030 kg/s and then decreases for higher mass flow rate. It is also seen that for lower fin height the thermal efficiency increases with increase in mass flow rate, and thermohydraulic efficiency decreases almighty beyond mass flow rate of 0.030 kg/s onwards. This is because increase in friction factor and pressure drop for lower fin height.

3.1. Validation of the results

The convective heat transfer coefficient between absorber plate to air flowing through the duct of offset finned absorber solar air heater have been calculated and compared with the experimental values of Junqi, D. et al (2007) for same values of system and operating parameters as shown in Fig. 16. The percentage deviation between the analytical and experimental values of convective heat transfer coefficient have been found to be in the range of $+9.36$ to $-5.26$. This establishes the validity of proposed analytical model for the investigation of thermal and thermohydraulic performance of offset finned solar air heater. Further, the results of the thermal efficiency of offset finned absorber solar air heater is compared with the experimental values of thermal efficiency of longitudinal fins absorber solar air heater used by Paisarn, N. (2004) and Karim, M.A., et al (2006) plotted in Fig. 17. The percentage enhancement in thermal efficiency of offset finned absorber solar air heater with respect to longitudinal fins solar air heater Paisarn, N. (2004) at mass flow rate of 0.02 kg/s and 0.08 kg/s is found to be 16.2 and 6.9 respectively, whereas these values are 13.7 and 4.96 respectively as compared to Karim, M.A., et al (2006).
4. CONCLUSIONS
In the present work, the effect of system parameters fin spacing and fin height and operating parameters air mass flow and insolation 950 W/m² on the thermal and thermohydraulic performance of offset finned absorber with plane solar air heater has been analytically. Attaching offset finned below the absorber plate is one of the effective techniques used to enhance the rate of heat transfer in solar air heaters, due to increase in thermal conductance of the absorber. On the basis of results and discussion the following conclusions can be drawn:

1. The predicated mathematical formulation gives adequate foresight of the performance of offset and plane solar air heater.

2. On the basis of parametric study on offset finned absorber plate the values of collector efficiency factor, heat removal factor, air outlet temperature, thermal efficiency, friction factor, effective efficiency and pressure drop has been found through the analytical approach.

3. Numerically calculated result of offset finned solar air heater was compared with plane solar air heater.

4. Results show that at lower fin spacing and fin height for lower air mass flow rate gives better thermal and thermohydraulic performance of offset finned absorber in comparison to plane solar air heater under same condition.

REFERENCES