Investigation of an Offset Finned Solar Air Heater Based on Energy and Exergy Performance

S. Rai¹, P. Chand², S. P. Sharma²

¹Research scholar, Mechanical Engineering Department, NIT Jamshedpur, Jharkhand, India
²Mechanical Engineering Department, NIT Jamshedpur, Jharkhand, India

This paper represented theoretically investigation of energy and exergy performance of an offset finned solar air heater. Parametric study was done to investigate the effect of variation of offset fin parameters i.e. fins spacing (1 to 5cm) and fins height (1.8 to 5.8cm) at different mass flow rates (0.01388 to 0.0833kg/s) on the energy and exergy efficiency. The results indicated that attaching offset finned below the absorber plate at low mass flow rates can lead to noticeable enhancement of exergy efficiency. The results revealed that the trend of variation of the energy and exergy efficiencies are not the same and the exergy efficiency is the chief criterion for performance evaluation. Decreasing the fins height, reducing the fins spacing are effective at low mass flow rates, but at high mass flow rates the inverse trend is observable, such that exergy efficiency reduces sharply. The efficiencies of offset finned solar collector were compared with conventional flat-plate collectors and longitudinal fins collector.

Received 24 February 2016
Accepted in revised form 27 May 2016

NOMENCLATURE

- \( h_{\text{conv}} = 5.67+3.86v_w \), convection heat transfer coefficient
- \( h_f \) convection heat transfer coefficient between the fins and the air (W/m² K)
- \( v_w \) wind velocity (m/s)
- \( T_a \) ambient temperature (K)
- \( T_{fe} \) inlet air temperature of step in the collector (K)
- \( T_{fa} \) mean absorber plate temperature (K)
- \( C_p \) specific heat of air (J/kg K)
- \( C_f \) Conversion factor (0.18)
- \( M \) quantities define by eq. (14) (m⁻¹)
- \( L_c \) length of the flat plate collector (m)
- \( l_c \) width of the flat plate collector (m)
- \( t \) fin thickness (m)
- \( l \) length of fin (m)
- \( h \) height of the offset fin (m)
- \( D \) height of air tunnel in solar collector (m)
- \( s \) lateral fin spacing (m)
- \( d_e \) hydraulic diameter (m)

\( A_c \) dx.lc, collector surface area of the step dx (m²)
\( F' \) efficiency factor of solar collector
\( G \) air mass flow rate (kg/h)
\( I_o \) Global irradiance incident on solar air heater collector (W/m²)
\( U_b \) heat loss coefficient from the bottom of the duct to ambient air (W/m² K)
\( U_L \) global heat loss coefficient (W/m² K)
\( h_{rc} \) radiation heat transfer coefficient between the wall
\( h_f \) convection heat transfer coefficient between the absorber plate and air (W/m² K)
\( v_f \) average air velocity in the solar collector tunnel (m/s) caused by wind (W/m² K)
\( T_{fs} \) outlet fluid temperature of step in collector (K)
\( A_d \) cross surface area in channel duct of collector(m²)
\( A_f \) total surface area of fins (m²)

* Corresponding author: Shalini Rai
E-mail: rai.shalin@gmail.com
heat removal factor of solar collector

\( F_R \)

air mass flow rate (kg/s)

\( m \)

absorber solar energy (W/m²)

\( S \)

heat loss coefficient from the absorber plate to ambient air (W/m² K)

\( U_l \)

useful energy gain of solar air collector (W)

\( Q_u \)

radiation heat transfer coefficient between the inner wall of the absorber plate and the bottom of the casing (W/m² K)

\( h_r \)

Greek letters

\( \epsilon_{is} \)

thickness of the insulating (m)

\( \delta \)

ratio t/L, dimensionless

\( d_s \)

step of section (m)

\( \beta \)

collectors tilt (degrees)

\( \rho_t \)

air density (kg/m³)

\( \eta_f \)

fin efficiency

\( \tau_v \)

transparent cover transmittance

\( \varepsilon_n \)

emissivity of black plate

\( \varepsilon_1 \)

emissivity of the inner wall of the absorber plate

\( \lambda_{is} \)

thermal conductivity of insulator (W/m K)

\( \alpha \)

aspect ratio t/h dimensionless

\( \gamma \)

ratio t/s, dimensionless

\( \sigma \)

constant of Stefan Boltzman

\( \nu \)

dynamic viscosity of air (m²/s)

\( \mu \)

Kinematic viscosity of air

\( \alpha_p \)

absorber plate absorption coefficient

\( \varepsilon_p \)

wood emissivity

\( \varepsilon_r \)

emissivity of glass

\( \phi \)

quantity define by eq. (12)

\( \lambda_f \)

thermal conductivity of air (W/m K)


dimensionless number

\( N_u \)

Nusselt number

\( P_r \)

Prandtl number

\( R_e \)

Reynolds number

\( j \)

colburn factor

\( T_2 \)

mean temperature of the channel back in collector(K)

\( T_s \)

sun temperature(K)

\( k_f \)

thermal conductivity of fin (W/m K)

INTRODUCTION

Solar thermal systems are used to utilize solar energy. Among different types, solar air heaters are widely employed due to simplicity in design, maintenance as well as low cost of materials required for construction. Nevertheless, in spite of these multiple benefits of solar air heaters, their fundamental deficiency is the low rate of heat transfer between absorber plate and flowing air due to unfavorable thermo-physical properties of air. Thus, researchers have focused their studies toward diverse performance improvement strategies. It has been shown that attaching fins and baffles on the absorber plate are an effective strategy used to enhance the heat transfer rate of solar air heater by extending the heat-transfer area and creating more turbulence [1, 2]. In addition to use fins and baffles, many different studies have been reported in terms of using absorber plate with several kinds of artificial roughness [9,12], using packed bed materials [3, 4], and utilizing corrugated surfaces such as V-corrugated and cross corrugated surfaces [5, 6]. External and internal recycling of the flowing air in different types surfaces [7, 8]. Nonetheless, in contrast with desirable influence of all of these improvement strategies on the performance of solar air heaters which will eventually lead to higher efficiency, an adverse effect exist in which the higher pump work is required due to increased friction losses.

In order to balance the quality of energy gain and friction losses, the exergy analysis is more appropriate in comparison to the energy analysis. This analysis can be considered as a fruitful manner to supplement, not to replace, the energy analysis [9]. Exergy (or availability) is the maximum work potential that can be obtained from a form of energy. Due to this fact that exergy analysis deals with irreversibility minimization or maximum exergy delivery, it usually provides more helpful results as well as more realistic sight of process. The exergy analysis has proved to be a powerful tool in the design, optimization, and performance evaluation of energy systems [10]. Hence, many attentions have been drawn by researchers to investigate the exergy based performance of solar air heaters. Gupta and Kaushik [11] established the optimal performance parameters for the maximum exergy delivery in a flat-plate solar air heater. They investigated the effects of aspect ratio, mass flow rate per unit area and the channel depth on the energy and exergy output rates of the solar air heater. They found that based on energy-based evaluation, the energy output rate increases with mass flow rate and aspect ratio and decreases with the channel depth and the inlet temperature of air. On the other hand, in terms of exergy based evaluation criterion, authors concluded that the energy output rate is not a monotonically increasing function of mass flow rate and aspect ratio, and a decreasing function of channel depth and inlet temperature of air. Singh et al. [12] analytically studied the exergetic performance of a solar air heater having discrete V-down rib roughness and compared the obtained results with a conventional flat-plate solar air heater. They investigated the effects of Reynolds number and rib roughness parameters on exergetic efficiency and suggested the use of the discrete V-down rib roughened solar air heater for the Reynolds numbers less than 18,000. Alta et al. [13] presented an experimental study based on energy and exergy analysis 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. So far, no efforts have been conducted in order to investigate the exergy based
performance of a solar air heater with offset finned attached below the absorber plate. In this study, a theoretical model has been presented to evaluate the exergetic performance of solar air heater with offset finned attached below the absorber plate. The offset fins parameters, i.e. fins spacing, and fin height as parametric variables as well as mass flow rate, solar radiation intensity have been varied to study their effect on the exergetic performance.

Theoretical analysis
The air heater considered consists of a flat glass cover and a flat absorber plate with offset finned attached below the absorber plate. The schematic diagram of top view of offset finned solar air heater illustrated in Fig. 1. Combination of parallel flat absorber plate with offset fins and bottom plate makes a channel, where the air is flowing through the channel, is heated by the absorbed solar radiation.

The energy balance equations for offset finned solar air heater absorber plate, glass cover and air stream shown in Figs. 3 to 5 and can be written as follow:

**Absorber plate**

\[ I_o \left( \tau v_a n \right) = U_t \left( T_n - T_a \right) + h_f \left( T_n - T_f \right) + h_r \left( T_n - T_2 \right). \]  

**Bottom plate**

\[ h_r \left( T_n - T_2 \right) = h_1 \left( T_n - T_f \right) + U_b \left( T_2 - T_a \right) \]

**Air stream**

\[ \left( \frac{m_c \cdot d \cdot T_f}{l_{cel}} \right) = h_1 \left( T_2 - T_f \right) + h_t \left( T_n - T_f \right) + h_f \left( T_n - T_f \right) \]

For calculating the Nusselt number, the correlation of the colburn factor \( j \) is recommended [14], and is used for plate fin heat exchangers and represents the data continuously in the laminar, transition, and turbulent flow regions:

\[ j = 0.6522 \cdot R_e^{0.5403} \cdot \delta^{0.1544} \cdot \theta^{0.1499} \cdot 0.0678 \cdot \left[ 1 + 5.269 \times 10^{-5} \cdot \theta^{0.504} \cdot \phi^{0.456} \cdot \psi^{1.055} \right]^{0.1} \]

where

\[ j = \frac{N_u}{\frac{N_t}{P_r}} \]

Reynolds number is given by
The average velocity is
\[ v_t = \frac{m}{\rho_t A_d} \]  

The cross surface 'A_d' in the air channel duct of the collector is defined as follows:
\[ A_d = \left( \frac{l_c D - (l_c t(s + h + t))}{s + t} \right) \]  

The hydraulic diameter is given by the following definition [15]:
\[ d_e = \left( \frac{4.9 s h l}{2 (s l + h l + t h)} \right) \]  

The radiation heat transfer coefficient between the inner wall of the absorber plate and the aluminum plate, where the temperature \( T_n \) and \( T_2 \) are expressed in Kelvin, is written as [15]:
\[ h_r = \frac{\sigma(T_n + T_2)(T_n^2 + T_2^2)}{\left( \frac{1}{T_2} + \frac{1}{T_n} \right) - 1} \]  

With regard to the forced convection, the average heat transfer coefficient is given as:
\[ h_1 = h_2 = h_t = \left( \frac{N_u A_t}{d_e} \right) \]  

The collector efficiency factor \( F' \) and collector overall loss coefficient \( U_t \) for the offset rectangular fin absorber plate shown in Fig. 2 are obtained from energy balances on the absorber plate, bottom plate and the air stream like in [15] where the quantity \( \phi \) takes into account the fins area ,which is supposed with the aluminum plate temperature as shown in Fig. 3.

\[ P' = \frac{h_t h_2 \phi + h_t h_1 + U_b h_1 + h_t h_2 \phi}{(U_t + h_t + h_r)(U_b + h_2 \phi + h_r) - h_r^2} \]  

\[ U_t = \left( \frac{u_b + u_l}{h_2 \phi + h_1 + u_l h_1 + h_2 \phi} + u_l \right)^{1/2} \]  

The collector heat removal factor can be expressed as:
\[ F_R = \left[ \left( \frac{\dot{m} c_p}{A_c U_t} \right) \right] \left[ 1 - \exp \left( \frac{A_t U_t F}{\dot{m} c_p} \right) \right] \]  

The dimensionless quantity \( \phi \) is defined as:
\[ 1 + \left( \frac{A_f}{A_c} \right) \eta_f = \phi \]  

In which the total surface area of fin is calculated as
\[ A_f = \frac{\pi J (1 + \rho \eta) x L c}{2 (s + t)} \]  

Where the fin efficiency is
\[ \eta_f = \frac{\text{tanh}(MD)}{MD} \]  

In which
\[ M = \sqrt{\frac{2h}{k_f t}} \]  

An empirical equation for the loss coefficient through the top of the collector \( U_t \) was developed by [15] following the basic procedure,
\[ U_t = \frac{1}{N} \frac{J}{(\frac{1}{h_{1f}} + \frac{1}{h_{2f}} + \frac{1}{h_{top}})} + \frac{\sigma(T_n + T_2)(T_n^2 + T_2^2)}{\left( \frac{1}{T_2} + \frac{1}{T_n} \right) - 1} \]  

where \( N=1 \), number of glass cover,
\( f = 1 + 0.0899 \rho_{np} - 0.1166 \rho_{np} \rho_n (1 + 0.07866 N) \),
\( C = 520(1 - 0.000051^2) \),
For \( 0^\circ \leq \beta < 70^\circ \)
\( e = 0.430(1 - \frac{100}{T_n}) \).

The loss coefficient through the bottom of the collector is
\[ U_b = \left[ \left( \frac{\rho_{np}}{\rho_{np}} \right) + \left( \frac{1}{h_{ppb}} \right) \right]^{-1} \]
**Temperature distribution**

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

\[
\frac{T_{fs} - T_a}{T_{fe} - T_a} = \exp \left( - \frac{A_d U_L F_1}{m c_p} \right)
\]

(20)

The mean temperature of the absorber plate and the back plate are obtained by solving the energy balance equation on these plates for:

\[
T_s T_i = \left( \frac{S}{U_i + h_2 \phi + h_i} \right) - (T_f - T_o) \left( \frac{U_b U_L + U_h h_r + U_b h_a}{U_b + h_2 \phi + h_r} \right) - h_r^2
\]

(21)

And

\[
T_2 - T_f = \frac{S \left( T_f - T_a \right)}{U_b U_L + U_h h_r + U_b h_a + U_b h_1}
\]

(22)

The mean air stream temperature by Klein [9]:

\[
T_f = T_{fe} + \left( \frac{q_u}{A_c F_R} \right) \left( 1 - \left( \frac{F_R}{F_1} \right) \right)
\]

(23)

The absorber solar energy is defined by:

\[
S = (\tau_s \alpha_n) I_o
\]

(24)

While introducing the collector overall loss coefficient between the absorber and the ambient air \( U_L \), the useful energy gain provided by:

\[
Q_u = A_c F_R \left( S - U_L \left( T_{fe} - T_a \right) \right)
\]

(25)

**Thermal efficiency**

The thermal efficiency can be expressed as:

\[
\eta_{th} = \frac{\dot{m} C_p(T_{fs} - T_{fe})}{I_o A_c}
\]

(26)

**Exergy analysis**

By considering the control volume, shown in Fig. 4, the general exergy balance for the solar air heater is [16]:

\[
E_{X_i} + E_{Xs} + E_{Xw} - E_{X_o} = IR
\]

(27)

\( E_{X_i} \) and \( E_{Xo} \) are the exergy associated with the mass flow of flowing air entering and leaving the control volume, respectively. \( E_{Xs} \) and \( E_{Xw} \) are the exergy of solar radiation falling on glass cover, exergy of the input work required to pump the air through the solar air heater and the irreversibility of the air heating process. The irreversibility or exergy loss occurs due to the temperature difference between absorber plate surface and sun, heat losses to the ambient and pressure drop in channel.

\( E_{X_i} \) and \( E_{Xo} \) are:

\[
E_{X_i} = (h_i - h_a) - T_a(S_i - S_a)
\]

(28)

\[
E_{X_o} = (h_o - h_a) + T_a(S_o - S_a)
\]

(29)

The \( E_{Xs} \) is calculated as [12]:

\[
E_{Xs} = \dot{I}_o A_c \psi = \dot{I}_o A_c \left( 1 - \frac{4}{3} \frac{T_a}{T_f} + \frac{1}{3} \left( \frac{T_a}{T_f} \right)^4 \right)
\]

(30)

where \( \psi \) is the exergy efficiency of radiation. Considering the pressure drop or pump work and assuming air as an incompressible fluid or perfect gas, the useful exergy gain, \( E_{Xu,p} \) is:

\[
E_{Xu,p} = E_{Xo} - E_{X_i} - E_{Xw} = m C_p \left( T_{fe} - T_a \right) - T_a \ln \left( \frac{T_{fe}}{T_{fe}} \right) - \frac{T_a}{T_{fe}} W_p
\]

(31)

The term \( \left( \frac{T_a}{T_{fe}} \right) W_p \) is the exergy destruction due to pressure drop. The required pump work is:

\[
W_p = \frac{\dot{m} \Delta P}{(\rho \eta_{pm})}
\]

(32)

The following correlation is developed for calculating the pressure drop [15] Friction factor:

\[
f_f = 9.6243 R_e^{-0.7422} \alpha_{-0.1856} \delta_{0.3053} \gamma_{-0.2659} \times \left[ 1 + 7.669 \times 10^{-8} R_e^{4.429} \alpha_{0.920} \delta_{3.767} \gamma_{8.236} \right]^{0.1}
\]

(33)

Hence, Pressure drop

\[
\Delta P = \frac{4 f_f \rho U_e u_e^2}{2 d_e}
\]

\( \eta_{pm} \) is the pump motor efficiency and is taken to be 0.85 [16]. The exergy efficiency, called second law efficiency, of the solar air heater is calculated by dividing the useful exergy gain to the exergy solar radiation as [12]:

![Figure 6. Considered control volume (CV) for the solar air heater [14](Image)](image-url)
\[
\eta_{\mu} = \frac{E_{xu,p}}{I_o A_p \psi} = \frac{E_{xu,p}}{I_o A_p \left(1 - \frac{4}{3} \frac{T_2}{T_0} + \frac{1}{3} \frac{T_0^3}{T_2^2}\right)}
\] (34)

**Calculation procedure**

For performing the calculations of this study a proper code in MATLAB 7.8.0 R2009a was developed considering the following configuration, system properties and operating conditions:

- \( L_c = 1.5m, l_o = 1m, t_o = 0.02m, t_i = 0.003m \),
- \( \tau_s \alpha_o = 0.85, e_u = 0.04m, \lambda_u = 0.033 W/mK \),
- \( T_s = 5762K, T_o = 298K, \epsilon_b = 0.93, \)
- \( \varepsilon_v = 0.88, \varepsilon_n = 0.9, \mu_f = 18.97 \times \frac{10^{-6}m^2}{s}, C_p \)
- \( k = 1.005 \)
- \( k/g, \lambda_f = 0.02826 W/mK, G = 50 to 300kg \)
- \( D = 4cm, h = 3.8cm, I_o = 950W/m^2 \).

The procedures followed for determination of exergetic performance briefly are:

1. Initial temperature values of \( T_n, T_f, T_2 \) were assumed.
2. Using initial value of \( T_f \) was used to predict the physical properties of air. Then using all of these initial temperatures, the heat transfer coefficients were found using Eqs. (9)-(10).
3. \( T_f, T_n \)and \( T_2 \) were obtained using Eqs. (21), (22) and (23) and the obtained temperature values were compared with the initial assumptions. If difference between obtained values and initial guesses were less than 0.001, assumed values were considered correct; otherwise, the procedure was repeated until the values reached to a convergence.
4. Upon completion of step (3), the values of mean temperatures were obtained and the exergy efficiency was calculated via Eq. (34).

**RESULTS AND DISCUSSION**

In the following section, results of exergetic performance evaluation of an offset finned solar air heater are presented. A large number of graphs can be drawn from the results of study but because of the space limitations, only typical results are shown.

Figs. 7 and 8 show the variation of thermal efficiency with mass flow rate of air for different fin spacing and insolation for \( I_o = 750 \) and \( I_o = 950W/m^2 \). It is seen that using of attaching offset finned below the absorber plate lead to higher thermal efficiency as compared to a conventional (plane) solar air heater. It has been found that the thermal efficiency monotonically increases with an increase in mass flow rate because of an increase in the convective heat transfer coefficient. The lowest fin spacing (1cm) maintains the highest efficiency value throughout the range of mass flow rate investigated. Furthermore, a slightly fall is observed in the rate of increase of efficiency as mass flow rate increases apparently due to relatively lower percentage increase in surface conductance as mass flow rate increases as also due to relatively higher heat flow rate. Also increase in fin spacing decreases the thermal efficiency of solar air heater. This is due to facts that increase in fin spacing resulted in decrease in heat transfer surface area and hence heat transfer rate.

Figs. 9 and 10 show variation of thermal efficiency as a function of mass flow rate for different fin height and insolation for \( I_o = 750 \) and \( I_o = 950W/m^2 \). From the figure it is seen that thermal efficiency increases with increase in mass flow rate. It is also observed that increase in height of offset fin slightly decreases the thermal efficiency of solar air heater. This is because of increase in offset fin height increase the heat transfer surface area; however it decreases the convective heat transfer coefficient.

Fig. 11 shows variation of exergy efficiency with mass flow rate for different fin spacing and for insolation \( I_o = 950W/m^2 \). It is clearly seen that attaching offset fins below the absorber plate, lead to exergy efficiency increase compared to a plane solar air heater. From the figure it is also observed that decrease in fin spacing gives higher exergy efficiency at lower mass flow rate for \( I_o = 950W/m^2 \); the improvement in exergy efficiency is due to enhanced heat transfer area and also creation of more turbulence which results in higher heat energy gain. Also result reveals that at higher mass flow rate exergy efficiency decreases rapidly with increase in fin spacing. This is due to increased exergy destruction due to higher pressure drop in the channel.

The variation of exergy efficiency with mass flow rate for different fin height is plotted in Fig. 12 for \( I_o = 950W/m^2 \). It is observed that the exergy efficiency decreases with increase in mass flow rate. From the figure it is clearly seen that there is drastic fall in exergy efficiency as mass flow rate increases for lower fin height of 1.8cm, whereas the exergy efficiency slightly decreases with increase in mass flow rate for higher fin height of 5.8cm. This trend continue for other heights of fin also. Further, it is noticed that lower fin height (1.8cm) maintains the higher exergy efficiency at lower mass flow rate, whereas for higher fin height (5.8cm) the reverse trend is observed.

Figs. 13 and 14 show effect of offset fin spacing on the exergy efficiency and outlet temperature at different mass flow rate along with conventional solar air heater for \( I_o = 750 \) and \( I_o = 950W/m^2 \). From the figure it is
Figure 7. Thermal efficiency vs. mass flow rate for various fin spacing s=1, 3 and 5 cm

Figure 8. Thermal efficiency vs. mass flow rate for various fin spacing s=1, 3 and 5 cm

Figure 9. Thermal efficiency vs. mass flow rate

Figure 10. Thermal efficiency vs. mass flow rate

Figure 11. Exergy efficiency vs. mass flow rate

Figure 12. Thermal efficiency vs. mass flow rate

Figure 13. Exergy efficiency vs. outlet temperature at different mass flow rate for Io=750W/m²

Figure 14. Exergy efficiency vs. outlet temperature at different mass flow rate for Io=950W/m²

clearly seen that increase in mass flow rate decreases the exergy efficiency for all values of fin spacing. The lower fin spacing (1cm) at lower mass flow rate of 0.01388 kg/s maintains the higher exergy efficiency and outlet
temperature of flowing fluid. It is found that at lower mass flow rate of 0.01388 kg/s the variation of exergy for different fin spacing is higher and at higher mass flow rate of 0.0833 kg/s, the trend of variation is reversed. Further it is seen that the outlet temperature is almost same for all fins spacing at higher mass flow rate of air.

Result validation

Due to lack of research related to performance assessment of offset finned solar air heaters with single glass cover, the verification was performed for the solar air heater used in the study of Karim et al. [16]. Figs. 15 and 16 show the comparison of thermal performance of offset finned absorber solar air heater with longitudinal fins [16]. It has been seen from the figures, that there is considerable enhancement in thermal performance by use of attaching offset finned below the absorber plate.

**Figure 15.** Outlet temperature vs. mass flow rate for fin height h=3.8cm.

**Figure 16.** Thermal efficiency vs. mass flow rate for fin height h=3.8cm

**CONCLUSION**

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, and extend the heat-transfer area and create more turbulence. In this study, performance evaluation of an offset finned solar air heater on energy and exergy efficiencies has been carried out theoretically. The most important findings of this study are summarized in the following:

- On the basis of the energy and exergy analysis, with attaching offset finned below the absorber plate and reducing fin spacing the thermal efficiency continuously enhances with the mass flow rate increase; whereas exergy efficiency vice versa.
- Comparison between the energy and exergy based analyses reveals that the exergy efficiency is a more accurate criterion for performance evaluation.
- According to the parametric study performed, the exergetic performance is very sensitive to the variation of fins spacing.
- Attachment of the offset finned below the absorber plate at lower mass flow rates enhances the exergy efficiency remarkably, however in higher mass flow rates; it does not result in significant improvement.

**REFERENCES**

چکیده
این مقاله نتایج تحقیقات نظری عملکرد انرژی و اکسپریزی درونی از باد با نشان میدهد، مطالعه پارامترات تغییر پارامترهایی از قبیل خمیدگی بالا و فاصله بالا، ارتفاع بالا و فاصله بالا در نظر می‌گیرند. در مورد برخی از پارامترات به این نتیجه می‌رسند که بالا و فاصله بالا باعث افزایش عملکرد می‌شوند. بنابراین نتایج نشان می‌دهد که بالا و فاصله بالا باعث افزایش عملکرد می‌شوند.

 DOI: 10.5829/idosi.ijee.2016.07.03.01