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

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A B S T R A C T

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.

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NOMENCLATURE

h_{vv}	5.67+3.86Vw, convection heat transfer Coefficient
h_f	convection heat transfer coefficient between the fins and the air (W/m^2 K)
v_w	wind velocity (m/s)
T_a	ambient temperature (K)
T_{fe}	inlet air temperature of step in the collector (K)
T_n	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 (m^2)
F'	efficiency factor of solar collector
G	air mass flow rate (kg/h)
I_o	Global irradiance incident on solar air heater collector (W/m^2)
U_b	heat loss coefficient from the bottom of the duct to ambient air (W/m^2 K)
U_L	global heat loss coefficient (W/m^2 K)
h_{rc}	radiation heat transfer coefficient between the wall
h_1	convection heat transfer coefficient between the absorber plate and air (W/m^2 K)
v_f	average air velocity in the solar collector tunnel (m/s) caused by wind (W/m^2 K)
T_f	air stream temperature of step in the collector (K)
T_{fs}	outlet fluid temperature of step in collector (K)
A_d	cross surface area in channel duct of collector(m^2)
A_f	total surface area of fins (m^2)

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F_R	heat removal factor of solar collector
m	air mass flow rate (kg/s)
S	absorber solar energy (W/m^2)
U_t	heat loss coefficient from the absorber plate to ambient air (W/m^2 K)
Q_u	useful energy gain of solar air collector (W)
h_r	radiation heat transfer coefficient between the inner wall of the absorber plate and the bottom of the casing (W/m^2 K)
	Greek letters
e_{is}	thickness of the insulating (m)
δ	ratio t/l, dimensionless
d_x	step of section (m)
β	collectors tilt (degrees)
ρ_f	air density (kg/m^3)
η_f	fin efficiency
τ_v	transparent cover transmittance
ε_n	emissivity of black plate
ε_1	emissivity of the inner wall of the absorber plate
λ_{is}	thermal conductivity of insulator (W/m K)
α	aspect ratio s/h dimensionless
γ	ratio t/s, dimensionless
σ	constant of Stefan Boltzman
ν_s	dynamic viscosity of air (m^2/s)
μ_f	Kinematic viscosity of air
α_n	absorber plate absorption coefficient
ε_b	wood emissivity
ε_v	emissivity of glass
ϕ	quantity define by eq. (12)
λ_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 leads 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 fins 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.

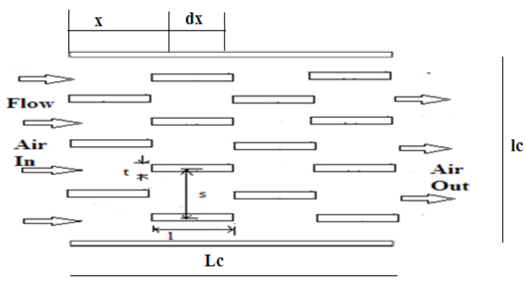

Figure 1. The top view of the absorber plate showing offset finned.

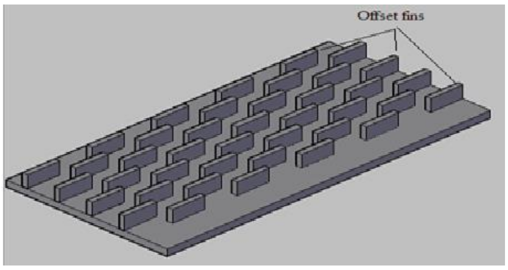

Figure 2. Schematic diagram of bottom view absorber plate attached with offset fins (Auto-CAD2007).

The one dimensional mathematical formulation in the flow direction in steady state condition for offset finned absorber solar air heater has been considered as shown in Figs.1 and 2.The following simplifying assumptions have been made:

- Thermal performance of collector is steady state.
- There is a negligible temperature drop through the glass cover, the absorber plate and the bottom plate.
- There is one-dimensional heat flow through the back insulation which is in the direction perpendicular to the air flow.

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

$$I_o(\tau_v a_n)=U_t.(T_n-T_a)+h_1.(T_n-T_f)+h_f\phi\,(T_n-T_f)+h_r(T_n-T_2). \quad (1)$$

Bottom plate

$$h_r.(T_n-T_2)=h_1.(T_n-T_f)+U_b.(T_2-T_a) \quad (2)$$

Air stream

$$\left(\frac{\dot{m}C_p.dT_f}{l_c d_x}\right)=h_1.(T_2-T_f)+h_1.(T_n-T_f)+h_f.\phi\,(T_n-T_f) \quad (3)$$

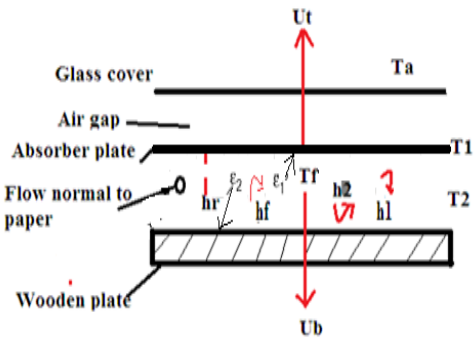

Figure 3. Heat transfer exchanges in the flat plate collector

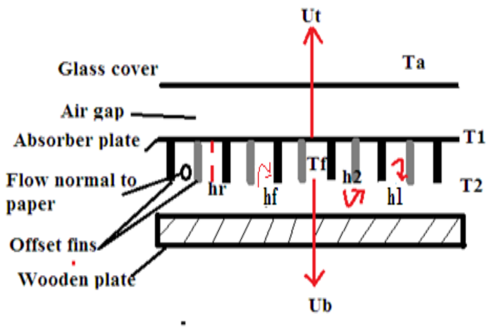

Figure 4. Heat transfer exchanges in offset fins collector

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:

$$j=0.6522R_e^{-0.5403}\alpha^{-0.1541}\delta^{0.1499}\gamma^{-0.0678}[1+5.269\times10^{-5}\alpha^{0.504}\delta^{0.456}\gamma^{-1.055}]^{0.1} \quad (4)$$

where

$$j=\frac{N_u}{R_e P_r^{\frac{1}{3}}}$$

Reynolds number is given by

$$R_e = v_f \left(\frac{d_e}{v_s}\right) \quad (5)$$

The average velocity is

$$v_f = \left(\frac{\dot{m}}{\rho_f A_d}\right) \quad (6)$$

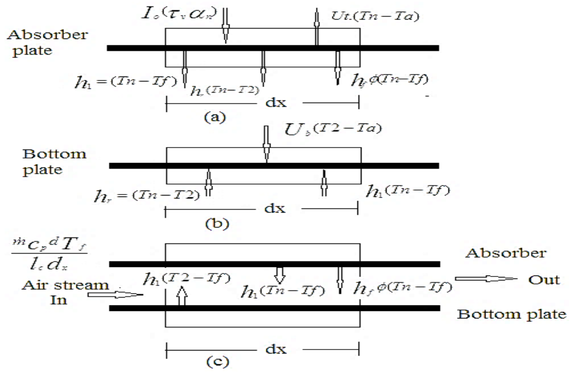

Figure 5. One dimensional steady state flow mathematical formulation (a) absorber plate (b) glass cover and (c) air stream in the offset finned solar air heater

The cross surface area 'A_d' in the air channel duct of the collector is defined as follows:

$$A_d = \left(\frac{lc.D - \left(lc.t(s+h+t)\right)}{s+t}\right) \quad (7)$$

The hydraulic diameter is given by the following definition [15]:

$$d_e = \left(\frac{4.s.h.l}{2.(s.l + h.l + t.h)}\right) \quad (8)$$

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]:

$$h_r = \frac{\sigma(T_n + T_2)(T_n^2 + T_2^2)}{\left(\frac{1}{\varepsilon_1}\right) + \left(\frac{1}{\varepsilon_2}\right) - 1} \quad (9)$$

With regard to the forced convection, the average heat transfer coefficient is given as:

$$h_1 = h_2 = h_f = \left(\frac{N_u \lambda_f}{d_e}\right) \quad (10)$$

The collector efficiency factor F' and collector overall loss coefficient U_L 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 $\emptyset$ takes into account the fins area ,which is supposed with the aluminum plate temperature as shown in Fig. 3.

$$F' = \frac{h_r h_2 \phi + h_r h_1 + U_b h_1 + h_1 h_2 \phi}{(U_t + h_1 + h_r)\left(U_b + h_2 \phi + h_r\right) - h_r^2} \quad (11)$$

$$U_L = \left(\frac{(U_b + U_t)\left(h_r h_1 + h_r h_2 \phi + h_1 h_2 \phi\right) + U_b U_t \left(h_1 + h_2 \phi\right)}{h_r h_2 \phi + h_r h_1 + U_b h_1 + h_1 h_2 \phi}\right) \quad (12)$$

The collector heat removal factor can be expressed as:

$$F_R = \left(\frac{\dot{m} c_p}{A_c U_L}\right)\left[1 - \exp\left(-\frac{A_c U_L F'}{\dot{m} c_p}\right)\right] \quad (13)$$

The dimensionless quantity ϕ is defined as:

$$1 + \left(\frac{A_f}{A_c}\right)\eta_f = \phi \quad (14)$$

In which the total surface area of fin is calculated as

$$A_f = \frac{4\left((l.h + t.h) \times L.lc\right)}{2.l.(s+t)} \quad (15)$$

Where the fin efficiency is

$$\eta_f = \frac{\tanh(MD)}{MD} \quad (16)$$

In which

$$M = \sqrt{\frac{2h}{k_f t}} \quad (17)$$

An empirical equation for the loss coefficient through the top of the collector U_t was developed by [15] following the basic procedure,

$$U_t = \frac{\frac{1}{N}}{\left(\frac{C}{T_n}\right)\left(\frac{T_n - T_a}{(N+f)^e} + \left(\frac{1}{h_{vv}}\right)\right)} + \frac{\sigma(T_n + T_a)(T_n^2 + T_a^2)}{\left(\varepsilon_n + 0.00591 N h_{vv}\right)^{-1} + \frac{2N + f - 1 + 0.133\varepsilon_n}{\varepsilon_v} - N} \quad (18)$$

where N=1, number of glass cover,
$f = 1 + 0.089 h_{vv} - 0.1166 h_{vv} \varepsilon_n (1 + 0.07866 N)$,
$C = 520(1 - 0.000051\beta^2)$,
For 0°< β <70°
$e = 0.430(1 - \frac{100}{T_n})$,
The loss coefficient through the bottom of the collector is

$$U_b = \left[\left(\frac{e_{is}}{\lambda_{is}}\right) + \left(\frac{1}{h_{vv}}\right)\right]^{-1} \quad (19)$$

Temperature distribution

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

$$\frac{T_{fs}-T_a-\left(\frac{S}{U_L}\right)}{T_{fe}-T_a-\left(\frac{S}{U_L}\right)}=exp\left(-\frac{A_cU_LF'}{\dot{m}c_p}\right) \qquad (20)$$

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

$$T_n\text{-}T_f=\left(\frac{S\left(U_b+h_2\phi+h_r\right)-(T_f-T_a)\left(U_bU_t+U_bh_r+U_th_2\phi+U_th_r\right)}{(U_t+h_1+h_r)\left(U_b+h_2\phi+h_r\right)-h_r^2}\right) \qquad (21)$$

And

$$T_2-T_f=\frac{h_rS-\left(T_f-T_a\right)(U_bU_t+U_bh_r+U_th_r+U_bh_1)}{(U_t+h_1+h_r)\left(U_b+h_2{}^{\phi}+h_r\right)-h_r^2} \qquad (22)$$

The mean air stream temperature by Klein [9]:

$$T_f=T_{fe}+\left(\frac{\frac{Q_u}{A_c}}{F_RU_L}\right)\left(1-\left(\frac{F_R}{F'}\right)\right) \qquad (23)$$

The absorber solar energy is defined by:

$$S=(\tau_v\alpha_n)I_o \qquad (24)$$

While introducing the collector overall loss coefficient between the absorber and the ambient air U_L, the useful energy gain provided by;

$$Q_u=A_cF_R\left(S-U_L\left(T_{fe}-T_a\right)\right) \qquad (25)$$

Thermal efficiency

The thermal efficiency can be expressed as:

$$\eta_{th}=\frac{\dot{m}C_p(T_{fs}-T_{fe})}{I_oA_c} \qquad (26)$$

.

Exergy analysis

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

$$E_{Xi}+E_{Xs}+E_{Xw}-E_{Xo}=IR \qquad (27)$$

E_{Xi} and E_{Xo} are the exergy associated with the mass flow of flowing air entering and leaving the control volume, respectively. E_{Xs}, E_{Xw} and IR 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_{Xi} and E_{Xo} are:

$$E_{Xi}=(h_i-h_a)-T_a(S_i-S_a) \qquad (28)$$
$$E_{Xo}=(h_o-h_a)+T_a(S_o-S_a) \qquad (29)$$

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

$$E_{Xs}=I_oA_c\psi=I_oA_c(1-\frac{4}{3}\left(\frac{T_a}{T_s}\right)+\frac{1}{3}\left(\frac{T_a}{T_s}\right)^4) \qquad (30)$$

where ψ 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:

$$E_{Xu.p}=E_{Xo}-E_{Xi}-E_{Xw}=\dot{m}C_p\left[\left(T_{fs}-T_{fe}\right)-T_a\ln\left(\frac{T_{fs}}{T_{fe}}\right)\right]-\frac{T_a}{T_{fe}}W_p \qquad (31)$$

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

$$W_p=\dot{m}\Delta P/(\rho\eta_{pm}) \qquad (32)$$

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

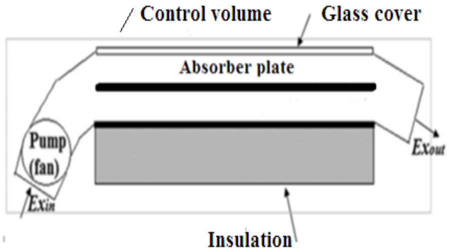

Figure 6. Considered control volume (CV) for the solar air heater [14]

$$f_f=9.6243R_e^{-0.7422}\alpha^{-0.1856}\delta^{0.3053}\gamma^{-0.2659}\times[1+7.669\times10^{-8}R_e^{4.429}\alpha^{0.920}\delta^{3.767}\gamma^{8.236}]^{0.1} \qquad (33)$$

Hence, Pressure drop

$$\Delta P=\frac{4f_f\rho L_cv_f^2}{2d_e}$$

η_{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]:

$$\eta_{II} = \frac{E_{Xu.p}}{I_o A_p \psi} = \frac{E_{Xu.p}}{I_O A_p (1 - \frac{4}{3}\left(\frac{T_a}{T_s}\right) + \frac{1}{3}(\frac{T_a}{T_s})^4)} \qquad (34)$$

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_c=1m, l=0.02m, t=0.003m,
$\tau_v \alpha_n$=0.85, e_{is}=0.04m, λ_{is}=0.033 W/mK,
$T_s = 5762K, T_a = 298K, \varepsilon_b = 0.93,$
$\varepsilon_v = 0.88, \varepsilon_n = 0.9, \mu_f = 18.97 \times \frac{10^{-6} m^2}{s}, C_p = 1.005$
$\frac{kJ}{kgK}, \lambda_f = 0.02826 \frac{W}{mK}, G = 50\ to\ \frac{300kg}{h},$
$D = 4cm, h = 3.8cm, I_o = 950W/m^2.$

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 checked 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 Io=750 and Io=950W/m^2. It is seen that use 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 apparently 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 Io=750 and Io=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 Io=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 Io=950W/m2; 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 Io=950W/m2.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 Io=750 and Io=950W/m^2. From the figure it is

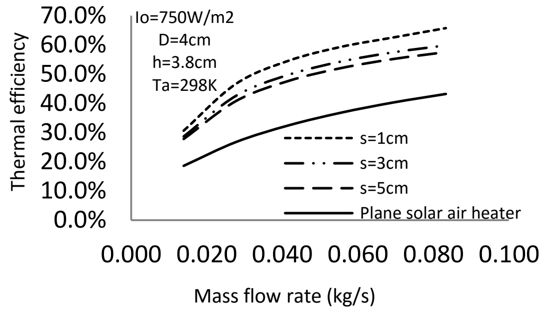

Figure7. Thermal efficiency vs. mass flow rate for various fin spacing s=1, 3 and 5cm

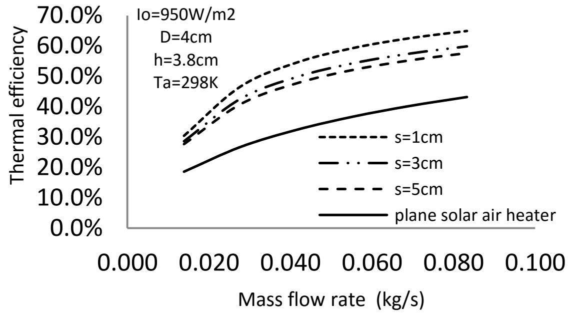

Figure 8.Thermal efficiency vs. mass flow rate for various fin spacing s=1, 3 and 5cm

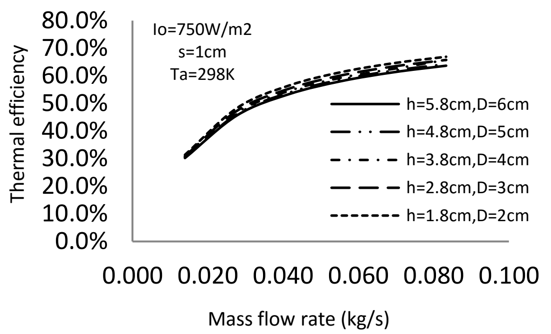

Figure 9. Thermal efficiency vs. mass flow rate

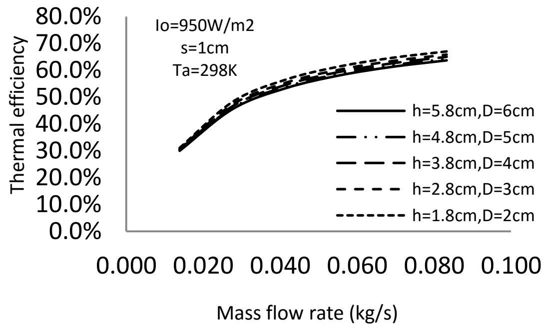

Figure 10.Thermal efficiency vs. mass flow rate

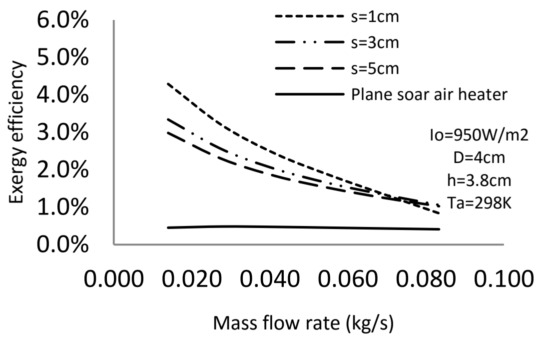

Figure 11. Exergy efficiency vs. mass flow rate

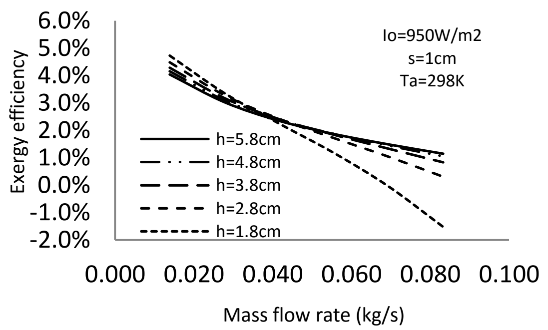

Figure 12. Thermal efficiency vs. mass flow rate

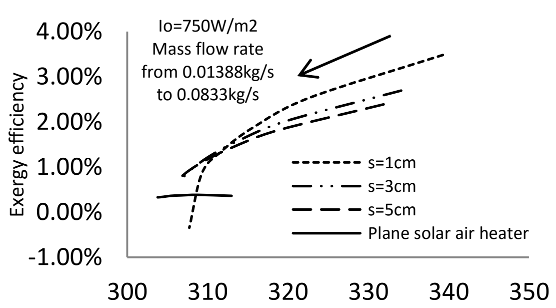

Figure 13. Exergy efficiency vs. outlet temperature at different mass flow rate for Io=750W/m^2

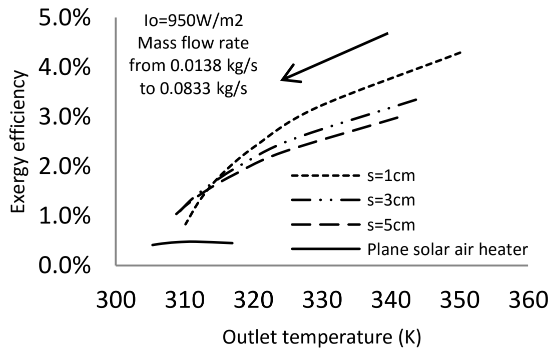

Figure 14. Exergy efficiency vs. outlet temperature at different mass flow rate for Io=950W/m^2

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.

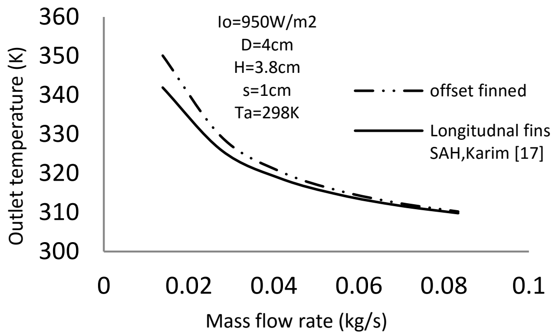

Figure 15. Outlet temperature vs. mass flow rate for fin height h=3.8cm.

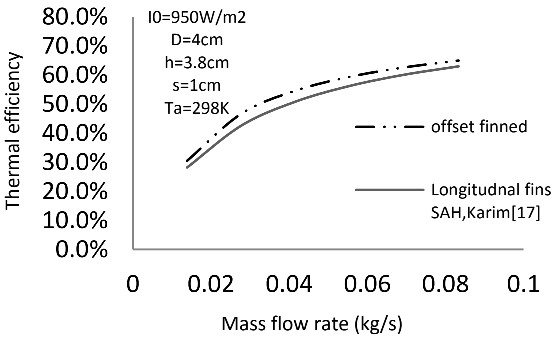

Figure 16.Thermal efficiency vs. mass flow rate for fin height h=3.8cm

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.

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Persian Abstract

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چکیده

این مقاله نتایج تحقیقات نظری عملکرد انرژی و اکسرژی گرمکن هوایی خورشیدی بالدار را نشان میدهد.مطالعه پارامتری اثرات تغییر پارامترهایی از قبیل خمیدگی بالها،فاصله بالها(۱-۵ سانتی متر)، ارتفاع بالها(۱,۸-۵,۸ سانتی متر) در نرخ دبی جرمی متفاوت (۰,۰۱۳۸۸-۰,۰۸۳۳ کیلوگرم بر ثانیه) در مورد بازده انرژی و اکسرژی انجام شده است. نتایج نشان می دهد که بالهای خمیده ی چسبیده زیر صفحه جاذب در دبی های جرمی پایین منجر به افزایش قابل توجه راندمان انرژی می شود.نتایج همچنین نشان می دهد که روند تغییرات راندمان انرژی و اکسرژی مشابه نیستند و بازده اکسرژی در ارزیابی عملکرد یک معیار مهم می باشد. کاهش ارتفاع بالها، کاهش فاصله بالها در دبی های جرمی پایین موثر هستند. اما در دبی های جرمی بالا روند معکوس مشاهده شده است. همچنین راندمان اکسرژی با شیب تند کاهش می یابد.راندمان جمع کننده خورشیدی با بالهای خمیده با جمع کننده صفحه تخت سنتی و جمع کننده بالدار مقایسه شده است.