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International Communications in Heat and Mass Transfer 33 (2006) 1281 – 1290
International Communications in HEAT and MASS TRANSFER
www.elsevier.com/locate/ichmt

Thermal and exergy analysis of solar air collectors with passive augmentation techniques ☆

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Available online 28 September 2006

Abstract

One of the main disadvantages of solar air collectors in practical applications is their relatively low efficiency. In this experimental investigation, the shape and arrangement of absorber surfaces of the collectors were reorganised to provide better heat transfer surfaces suitable for the passive heat transfer augmentation techniques. The performance of such solar air collectors with staggered absorber sheets and attached fins on absorber surface were tested. The exergy relations are delivered for different solar air collectors. It is seen that the largest irreversibility is occurring at the conventional solar collector in which collector efficiency is smallest.

Keywords: Solar air collectors; Thermal efficiency; Exergy loss

1. Introduction

The solar air collector has an important place among solar thermal systems because it is widely used in many commercial applications such as the supply of hot air to school buildings and agricultural and industrial drying, etc [1]. Its advantages are low cost, no freezing and no high pressure problems. However, the efficiency of solar air collectors is low because of the low Prandtl number of air. There have recently been numerous studies to increase performance of the solar air heater. The effect of collector aspect ratio on the collector efficiency of upward-type flat-plate solar air heaters has been investigated both theoretically and experimentally by Yeh and Lin [2]. Depending on the collector aspect ratio, up to 10% improvement has been observed at the solar air heaters efficiency. In another study, Yeh and Ting [3] experimentally investigated the performance of solar air heaters with the iron fillings placed in the gap between the absorbing plate and glass cover packed part and found considerable improvement in the performance of these types of solar air heaters. However, they ended up that the procedure would increase the pressure drop of air across the heater, and so may lead to higher power consumption. High performance improvement in a solar air heater was also seen in the case of the fins provided with attached baffles in another study [4]. It was observed that the collector efficiency was increased by the density of baffles due to an extended heat-transfer surface. Hamdan and Jurban [5] developed a mathematical model for analysis of the thermal performance of three types of solar air collectors (bare, covered and finned-plate) using

☆ Communicated by W.J. Minkowycz.

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0735-1933/$ - see front matter
doi:10.1016/j.icheatmasstransfer.2006.08.006

a general energy-balance technique. The average solar insolation over the period of this study was 623 W/m^2 at 26 °C in Jordan. A counter-current solar air heater was proposed for better efficiency by Mohamad [6]. The main idea of his study was to preheat the air stream before passing it over absorber plate. It was found that the gain in the thermal efficiency was in the order of 18% to 20% compared to conventional solar air heaters. Garg et al. [7] developed a steady-state model for the performance of rectangular fin air collectors in which the effects of the rectangular fins, air-flow rates, length of fin, depth of the air duct, etc. on the performance of the air heater were studied. The efficiency of the finned air heater was found to be in the range from 45% to 61% for various duct depths and fin lengths. Choudhury and Garg [8] have conducted a comparative theoretical parametric analysis of solar air heaters with and without packing of different materials, shapes and sizes in the flow passage above the back plate. Parker et al. [9] conducted thermal analyses for three types of solar air collectors with air flow over, under and on both sides of the absorber.

In the most recent works, the performance improvement of solar air collectors was mostly studied using extended heat transfer surfaces such as fin, barrier, package filled with iron, etc. in the air flow channels. However, most of the proposed methods have considerably increased the pumping power. Therefore, the passive heat transfer augmentation techniques with less pumping power have recently gained much more importance. Although the methods for heat transfer augmentation by creating the turbulent flow over the plates are well known without originality, their applications

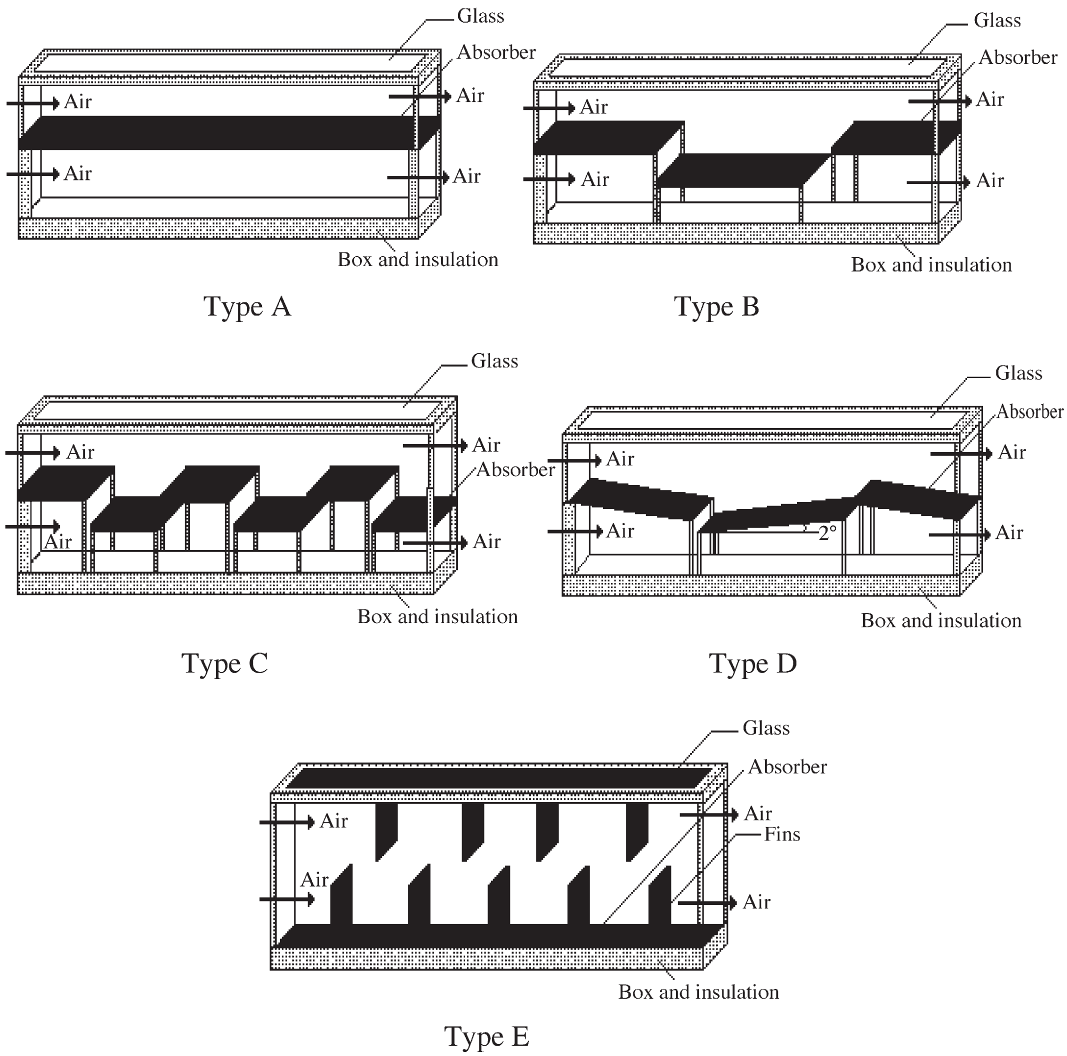

Fig. 1. Cross-sectional views of the tested solar air collectors.

and combinations with various arrangements which increase the collector efficiency, such as that by placing the overlapped glass plates in the flow channel suggested by Selçuk [10], should be taken into consideration as a new progress. The staggered absorber bare surfaces have been tested in the solar air collectors with some differences such as geometrical sizes and arrangements from the former studies [2–11].

In this study, the solar collectors with different shape and arrangement of absorber surfaces have tested and thermal performance of these collectors have been compared with conventional solar air collector. In addition, irreversibility of each of solar air heaters has been calculated.

2. Theory

The efficiency of solar collectors is the ratio of useful energy obtained in collector to solar radiation incoming to collector. It can be formulated as the following [12]:

$$\eta = Q_u / A_c H \tag{1}$$

The useful energy transferred to fluid is:

$$Q_u = A_c F_R \lfloor (\tau\alpha) H - U_L (T_f - T_a) \rfloor \tag{2}$$

The collector heat removal factor (F_R) is the ratio of useful heat obtained in collector to total heat collected by collector when the absorber surface temperature is equal to fluid entire temperature on every point of the collector surface.

$$F_R = \frac{mc_p}{A_c U_L}\left[1 - e^{-\left(A_c U_L F'\right)/\left(mC_p\right)}\right] \tag{3}$$

The collector overall heat loss coefficient (U_L) is the sum of top, bottom and edge heat loss coefficients. It means:

$$U_L = U_t + U_b + U_e \tag{4}$$

The top heat loss coefficient (U_t):

$$U_t = \left\{ \frac{N}{\frac{C}{T_o}\left[\frac{T_p - T_a}{N + f}\right]^e} + \frac{1}{h_w} \right\} + \frac{\sigma\left(T_p + T_a\right)\left(T_p^2 + T_a^2\right)}{\frac{1}{(\varepsilon_p + 0.00591N \ h_w)} + \frac{2N + f - 1 + 0.133\varepsilon_p}{\varepsilon_g} - N} \tag{5}$$

where

$$C = 520(1 - 0.000051\beta^2) \quad (0° < \beta < 70°) \tag{6}$$

$$f = \left(1 + 0.089h_w - 0.1166h_w\varepsilon_p\right)(1 + 0.07866N) \tag{7}$$

$$h_w = 5.7 + 3.8V_R \tag{8}$$

$$e = 0.43 \ (1 - 100/T_o) \tag{9}$$

The bottom heat loss coefficient (U_b):

$$U_b = k/L \tag{10}$$

The edge heat loss coefficient (U_e):

$$U_e = (UA)_e / A_c \tag{11}$$

where:

$$(UA)_{\mathrm{e}} = \frac{h_{\mathrm{s}}}{L_{\mathrm{s}}} PL_{\mathrm{c}} \tag{12}$$

The following equation is acceptable for the term of collector efficiency factor (F') mentioned in Eq. (3):

$$F' = \frac{h_{\mathrm{r}}h_1 + h_2U_t + h_2h_{\mathrm{r}} + h_1h_2}{(U_{\mathrm{t}} + h_{\mathrm{r}} + h_1)(U_{\mathrm{b}} + h_2 + h_{\mathrm{r}}) - h_{\mathrm{r}}^2} \tag{13}$$

where h_{r} is:

$$h_{\mathrm{r}} = \frac{\sigma\left(T_1^2 + T_2^2\right)(T_1 + T_2)}{\frac{1}{\varepsilon_{\mathrm{g}}} + \frac{1}{\varepsilon_{\mathrm{p}}} - 1} \tag{14}$$

The Nusselt number is needed for calculation of h_1 and h_2 heat transfer coefficients in the air channel of solar collector. The Nusselt numbers can be used as the following equation for various Reynolds numbers:

$$100 < Re < 2100 \quad Nu = 0.344\ Re^{0.35} \tag{15}$$

$$2100 < Re < 2850 \quad Nu = 16810^{-09}\ Re^{2.25} \tag{16}$$

$$2850 < Re < 5650 \quad Nu = 2.5510^{-03}\ Re^{1.04} \tag{17}$$

$$5650 < Re < 100000 \quad Nu = 19.810^{-03}\ Re^{0.8} \tag{18}$$

3. Experimental procedure

3.1. Features of the solar collectors

The solar air collectors used in this study are shown schematically in Fig. 1. The type A is conventional solar air collector in which air flows over both sides of the absorber. This type is taken as the reference collector in the comparison of experimental results. In the type B, the total absorber surface consists of three staggered sheets placed into the air flow

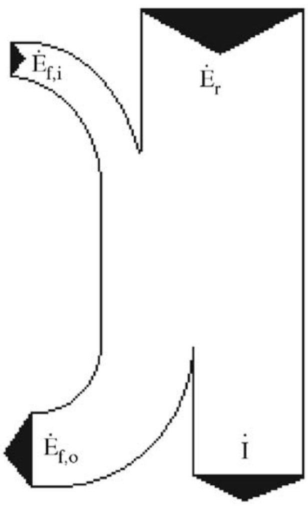

Fig. 2. Exergy flows in a solar collector.

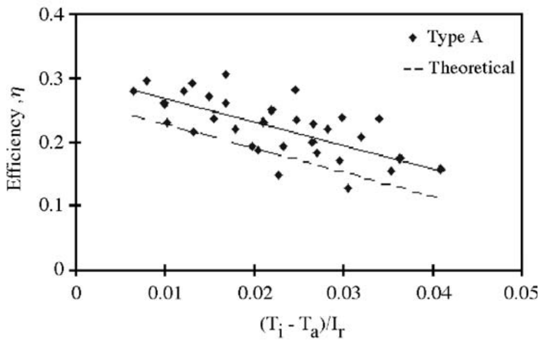

Fig. 3. The comparison of experimental and theoretical performance curves for the conventional solar air heater (Type A).

channel. The type C has the same form as the type B except for having six sheets. The type D has three absorber sheets, but its characteristics are different from the type B. In this type collector, the staggered sheets are placed into the air channel at 2° to the flow axis. The type E has been manufactured by attaching fins on absorber surface into the air flow channel. Each collector case is 80 cm wide and 125 cm in length. These wooden collector cases are covered with 4 mm thick glass. All parts of the solar air collectors are thermally well insulated except for the glass cover.

3.2. Experimental set-up

Collector positioned towards the south at an angle of 23.7° which was optimum for summer in Elazığ–Turkey (36–42° N). The collectors were tested to the ASHRAE Standard 93–77 [13]. The incident solar radiation to the collectors was measured by a Kipp–Zonen pyranometers. Two collectors were connected to the suction entrance of a radial fan having 2850 rpm and 2.5 kW. The adjustable volumetric air flow was measured by using a rotameter. To measure the inlet and outlet temperatures, mercury thermometers (0–100 °C) were employed. The ambient temperature was also recorded by a mercury thermometer. An inclined manometer connected to inlet and

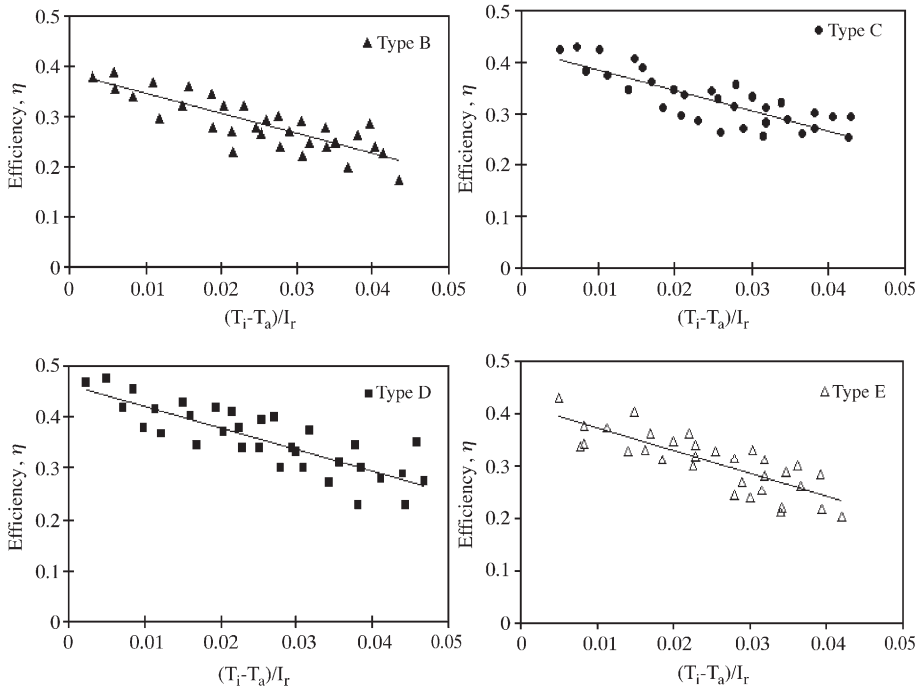

Fig. 4. The performance curves for solar air heaters of type B, type C, type D and type E.

outlet was used to find out the pressure drop of air through the solar collectors. All measurement devices have been calibrated.

3.3. Experimental data analysis

Experimental measurements were done in the clear days of July and August of 1998. The air temperatures at the inlet and outlet of the collector, the ambient temperature, the solar insolation, the wind velocity and the volume flow rate of air were measured for each set of experiments during the steady-state period. All physical properties of air were selected according to the following bulk mean temperature:

$$\Delta T_{\mathrm{m}} = (T_{\mathrm{i}} + T_{\mathrm{o}})/2 \tag{19}$$

4. Exergy analysis

Exergy analysis is a useful method to establish strategies for the design and operation of many industrial processes where the optimal use of energy is considered an important issue [14]. Fig. 2 schematically shows the exergy flows in the solar collector. If the effects due to the kinetic and potential energy changes are neglected, the general exergy balance can be expressed in rate form as given [15]

$$E_{\mathrm{i}} - E_{\mathrm{o}} = E_{\mathrm{dest}} \tag{20}$$

or

$$E_{\mathrm{s}} + E_{\mathrm{f,i}} + W = E_{\mathrm{f,o}} + I \tag{21}$$

Using Eq. (22), the rate form of the general exergy balance can also be written as

$$\Sigma\left(1 - \frac{T_{\mathrm{a}}}{T_{\mathrm{s}}}\right)Q_{\mathrm{s}} - W + \Sigma m_{\mathrm{i}}\Psi_{\mathrm{i}} - \Sigma m_{\mathrm{o}}\Psi_{\mathrm{o}} = I \tag{22}$$

Where

$$\Psi_{\mathrm{in}} = (h_{\mathrm{i}} - h_{\mathrm{a}}) - T_{\mathrm{a}}(S_{\mathrm{i}} - S_{\mathrm{a}}) \tag{23}$$

$$\Psi_{\mathrm{out}} = (h_{\mathrm{o}} - h_{\mathrm{a}}) - T_{\mathrm{a}}(S_{\mathrm{o}} - S_{\mathrm{a}}) \tag{24}$$

If Eqs. (23) and (24) are substituted in Eq. (22) and it is arranged

$$\left(1 - \frac{T_{\mathrm{a}}}{T_{\mathrm{s}}}\right)Q_{\mathrm{s}} - m[(h_{\mathrm{o}} - h_{\mathrm{i}}) - T_{\mathrm{a}}(S_{\mathrm{o}} - S_{\mathrm{i}})] = I \tag{25}$$

where Q_{s} is solar energy absorbed by the collector absorber surface and it is evaluated by the expression [16]:

$$Q_{\mathrm{s}} = H(\tau\alpha)A_{\mathrm{c}} \tag{26}$$

The enthalpy and entropy change of the air in the collector is expressed by

$$\Delta h = h_{\mathrm{o}} - h_{\mathrm{i}} = c_p\left(T_{\mathrm{f,o}} - T_{\mathrm{f,i}}\right) \tag{27}$$

$$\Delta S = S_{\mathrm{o}} - S_{\mathrm{i}} = c_p \ln\frac{T_{\mathrm{f,o}}}{T_{\mathrm{f,i}}} - R\ln\frac{P_{\mathrm{o}}}{P_{\mathrm{i}}} \tag{28}$$

Substituting Eqs. (26), (27) and (28) in Eq. (25), it may be rewritten as

$$\left(1-\frac{T_\mathrm{a}}{T_\mathrm{s}}\right)H(\tau\alpha)A_\mathrm{c}-mc_\mathrm{p}\left(T_\mathrm{f,o}-T_\mathrm{f,i}\right)+mc_\mathrm{p}T_\mathrm{a}\ln\frac{T_\mathrm{f,o}}{T_\mathrm{f,i}}-mRT_\mathrm{a}\ln\frac{P_\mathrm{o}}{P_\mathrm{i}}=I \tag{29}$$

The irreversibility I can be directly evaluated from the following equation

$$I=E_\mathrm{dest}=T_\mathrm{a}S_\mathrm{gen} \tag{30}$$

The second law efficiency is calculated as

$$\eta_\mathrm{II}=1-\frac{T_\mathrm{a}S_\mathrm{gen}}{[1-(T_\mathrm{a}/T_\mathrm{s})]Q_\mathrm{s}} \tag{31}$$

5. Results and discussion

The performance curves of five types solar air heaters tested in this study are shown in Figs. 3, 4. The experimental and theoretical efficiencies for conventional solar air heater (type A) can be seen in Fig. 3. It is seen that the theoretical predictions agree reasonably well with experimental results. The differences between theoretical and experimental efficiencies were negligible. The agreement

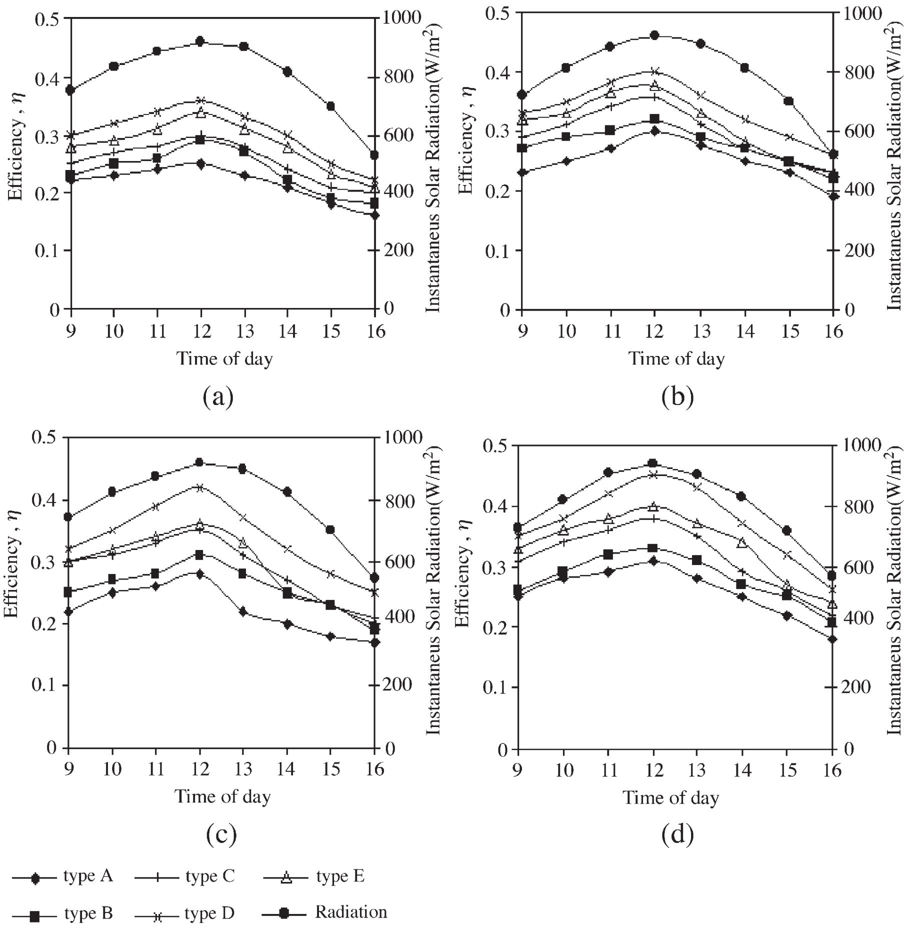

Fig. 5. The efficiency of the collectors at various air rates; a) m=0.008 kg/s, b) m=0.013 kg/s, c) m=0.02 kg/s, d) m=0.026 kg/s.

Table 1
Exergy analysis of tested solar collectors

Solar collectors	Exergy input, E_{in} (kW)	Exergy output, E_{out} (kW)	Irreversibility, I (kW)	Exergy loss (%)	Second law efficiency, η_{II} (%)
Type A	0.365	0.129	0.236	64.38	35.61
Type B	0.334	0.137	0.197	58.98	41.01
Type C	0.320	0.148	0.172	53.75	46.25
Type D	0.312	0.175	0.137	43.91	56.09
Type E	0.318	0.157	0.161	50.62	49.37

between the results obtained with the comparison-proof experiments and the previous studies shows that the possible entrance effects were absent.

The performance curves for solar air collector of type B, type C, type D and type E are given in Fig. 4. It is found that the collector performance of type B is approximately 10% higher than that of type A. The comparison between type A and type C showed that the efficiency of type C was approximately 15% higher than that of type A (the conventional solar air heater). The collector efficiency of type D with each staggered absorber sheet was found as nearly 30% higher than the efficiency of the conventional air heater (type A). If the collector performance of type E and type A are compared, efficiency of type E was approximately 25% higher than that of type A.

By least-squares fitting of the experimental data, the efficiencies of the types A, B, C, D and E can be summarized as

$$\eta_{\text{type A}} = 0.31–3.72(T_i–T_a)/H \tag{32}$$

$$\eta_{\text{type B}} = 0.38–3.92(T_i–T_a)/H \tag{33}$$

$$\eta_{\text{type C}} = 0.43–3.94(T_i–T_a)/H \tag{34}$$

$$\eta_{\text{type D}} = 0.46–4.21(T_i–T_a)/H \tag{35}$$

$$\eta_{\text{type E}} = 0.44–4.0(T_i–T_a)/H \tag{36}$$

$$\eta_{\text{theoretical}} = 0.26–3.78(T_i–T_a)/H \tag{37}$$

Fig. 5 shows the efficiency of the collectors at four various mass flow rates of air and daily incident solar radiation levels. It is found that the collector performance improves with increasing mass flow rate of air. Furthermore, increasing the incident radiation, gives an enhancement to collector efficiency. It seems that the collector efficiency is maximum at midday at which the incident radiation is maximum. The collector efficiency of type D reaches 50% on midday at 0.026 kg/s mass flow rate.

Irreversibility of each of solar air heaters has been calculated and the results have been shown in Table 1. The exergy analysis was carried out on August of 1998 in Elazığ at 12:00 p.m. for mass flow rate 0.026 kg/s. It can be seen that the type A is the collector in which the exergy loss is the highest (64.38%). In the type D with each staggered absorber sheet, exergy loss is the lowest (43.91%).

6. Conclusions

By using the present passive techniques such as dividing three or six sheets to absorber surface, attaching fins on absorber surface and giving an oblique angle (2°) to the three sheets absorber surface, the efficiency of solar collector has been increased approximately 10% to 30% in comparison with the conventional solar collector. The largest irreversibility occurs at the conventional solar air heater, since, in conventional collector (type A) only a little part of solar energy absorbed by the collector can be used.

Nomenclature

A_c	Collector surface area (m^2)
c_p	Specific heat of air (J/kg K)
F_R	Collector heat removal factor
F'	Collector efficiency factor
H	Total solar radiation incident on collector (W/m^2)

I	Irreversibility, (J/kg)
k	Thermal conductivity of insulator (W/m K)
L	Thickness of the insulator (m)
L_c	Collector length (m)
m	Mass flow rate (kg/h)
h	Enthalpy (J/kg)
h_1	Heat convection coefficient between the glass cover and air (W/m^2 K)
h_2	Heat convection coefficient between the absorber plate and air (W/m^2 K)
h_r	Radiation coefficient between the air-duct surfaces (W/m^2 K)
h_s	Heat convection coefficient between the insulation and ambient (W/m^2 K)
h_w	Heat convection coefficient for air flowing over the outside surface of the glass cover (W/m^2 K)
N	Number of glass cover
Q_u	Collector useful energy (W)
S	Entropy (J/kg K)
T	Temperature (K)
U_b	The bottom heat loss coefficient (W/m^2 K)
U_e	The edge heat loss coefficient (W/m^2 K)
U_L	The collector overall heat loss coefficient (W/m^2 K)
U_t	The top heat loss coefficient (W/m^2 K)
V_R	Wind velocity (m/s)
W	Work (W)
ΔT_m	Bulk mean air temperature (K)
η	Collector efficiency
ε_g	Emissivity of the glass cover
ε_p	Emissivity of the absorbing plate
β	Collectors tilt (degree)
ρ	Density of air (kg/m^3)
$(\tau\alpha)$	Effective transmission

Subscripts

a	Ambient
f	Fluid
i	Inlet
o	Outlet
p	Absorbing plate
r	Radiation
s	Sun

Acknowledgements

The authors gratefully acknowledge the financial support from the Scientific Research Projects Administration Unit of Firat University for this study.

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