

# An Experimental Investigation of the Energy and Exergy Efficiency of Extended Surface Solar Air Heaters

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**Abstract**—In this study, the energy and exergy efficiencies of newly developed four different solar air collectors have been investigated experimentally. The developed air heaters were compared with conventional type heaters. Solar air heaters with dimensions of 900 mm x1900 mm x170 mm have been used in the experiments. The experiments were carried out at mass flow rates of 0.02 kg/s, 0.04 kg/s and 0.06 kg/s. Collector outlet temperature, absorber surface temperature, glass door temperature, ambient temperature, solar radiation value and wind speeds were measured. Energy and exergy efficiencies have been calculated according to the data acquired from the experiments. It has been determined as a result of the experiments that the energy efficiency of the conventional type heater at a mass flow rate of 0.06 kg/s is %42, whereas that of Type-III is % 77.1; and that the exergy efficiency of the conventional system is mass flow rate of 0.02 kg/s is %8.31 whereas that of Type III is %30.32. It has been observed that the energy and exergy efficiencies of Type-III are higher in comparison with other types.

**Keywords**— Solar Energy, Solar Air Heater, Solar air collector.

## I. INTRODUCTION

Solar air heaters (SAH) are generally used for the heating and drying processes of internal spaces. Many different types of solar air heaters have been developed. The studies carried out in this field are generally related with the increase of the thermodynamic efficiency of SAH. The heat transfer coefficient between the absorber surface and the used fluid (air) has to be increased in order to increase the efficiency of such type of systems. Various methods used to accomplish this: (i) increasing the absorbing surface area, (ii) increasing the turbulence of the fluid, (iii) ensuring that the fluid used (air) stays more in the collector [1].

Hans [2] et.al. have examined the heat transfer and friction factor of a channel type solar air heater in their studies. They have placed V-shaped fins to the absorbing surface in order to increase the surface area of the absorbing surface and to increase the turbulence of the fluid. They have examined the effects of the fin width, height and attack angle on the Nusselt Coefficient and friction factor. El-Sebaei [3] et.al. have examined the thermal performance of a flat and V-type corrugated double pass air heater theoretically and experimentally. El-Sebaei et.al. [4] have examined the effect of the selective surface on a flat type collector on the thermal performance theoretically and experimentally. They have used; CuO, CoO, Cr-Cr<sub>2</sub>O<sub>3</sub>, Ni-Sn and black dye as selective

surface. They have determined that Ni-Sn coating has a %31 higher performance in comparison with black painted surface. Karslı [5] has carried out the experimental energy and exergy analysis of four tubular and pool type collectors with 70° and 75° inclined fins. Moumami [6] et.al. have experimentally examined the effect of fins with a height of 20 mm and a gap between them of 25 mm on efficiency. They have also examined the effects of selective and non-selective surfaces on thermal performance. They have used four types of collectors in their study. They have observed that the lowest efficiency in terms of thermal performance was for the non-selective and non-finned surface. Karim and Hawlader [7,8] have carried out the thermal analysis of a corrugated collector to be used in drying processes placed as V shaped on an absorbing surface. They have observed that the V type corrugated collector has a %7-12 higher efficiency in comparison with the conventional type collector. They used five different types of finned collectors in their study. Another collector was the conventional type collector. Peng, et.al. [9] have experimentally examined the effects of 4 mm diameter metal pins with different length on absorbing surface. They have observed the energy efficiency as 75 % maximum. Koyuncu [10] has used a plate made of wood fiber (chipboard) as an absorber bottom plate. In the study carried out, the plate was painted black in Type 1. In Type 2, an aluminium sheet metal painted black was placed over the absorber plate, whereas in Type 3 zig zag shaped sheet plate was used. Air flow in Type 4 was passed through the flat absorber plate and wood fiber plate. In Type 5, air flow was passed through the zig-zag shaped absorber plate and wood fiber plate. Whereas in Type 6, a second plastic lid was used 30 mm below the glass lid. It has been observed that the highest efficiency was obtained in Type 2 with %45,12 whereas the lowest efficiency was observed in Type 6 with %36,94. Alvarez, et.al. [11] have painted waste beverage cans to black and placed them on the absorber plate of the collector. They have determined the maximum efficiency as %74. Chaube [12] has placed different ribs on the square absorber plate and examined the heat transfer and flow characteristics between 3000<Re<20000 using finite elements method (FEM).

In this study, the performances of 5 different SAHs were examined experimentally. The experiments were carried at 3 different air flow rates. Energy and exergy efficiencies were calculated according to the acquired results.

II. EXPERIMENTAL SETUP

The collectors used have the dimensions of 900 mm x 1900 mm x 170 mm. The lower and side sections have been insulated using 50 mm glass wool. The top section has been closed with a 4 mm thick glass in order to decrease losses. The interior section of the collector was dyed matte black. 5 different collector types were designed in order to determine the suitable collector type of these collectors, Type 1 is the conventional type collector. V type absorber surface and turbulence increasing elements were used in other types. Each V type element used has the dimensions of 80x200 mm in each side. 3 holes of 23 mm diameter were opened one on one side and two on the other side with Fig.1. 27 of these elements were used. These elements were placed perpendicular, ordered, cross and parallel to the flow direction as shown in Fig.2 and thus the experiments were carried out using these four different placement types. Fig.3 show the experimental setup.

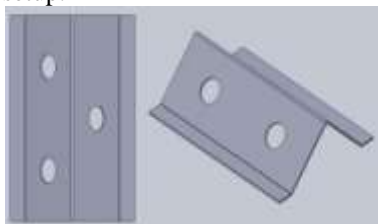


Fig.1 Schematic diagram of the V type Element

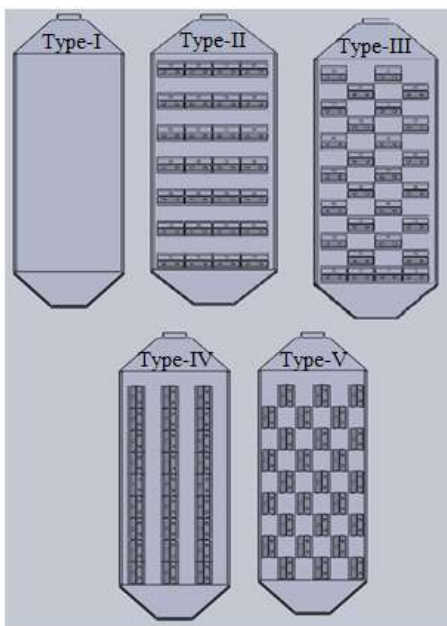


Fig.2 Schematic diagram of the tested solar air collectors



Fig.3 Experimental setup

A radial fan with 1800 rev/min and 0.75 kW power that has a maximum air flow rate of 1.8 kg/s was used in order to provide air flow in the collectors. The used fans were assembled to below the collectors.

Kipp&Zonen CMP3 brand solar meter was used to measure radiation in the experiments. The solar meter was placed to the top of the collector in all the experiments with the same angle with respect to the horizontal axis. Elimko E-680 channel selective thermometer was used to measure temperature. J type iron-Constantan thermocouple was used as an intermediate member. The air inlet, air outlet, environment, surface, glass lid and ambient temperatures were measured. CEM DT-8880 brand anemometer with telescopic probe was used to measure the air flow rate in the collector; whereas is Prova AVM-07 brand finned type anemometer was used to measure wind speed.

The experiments were carried out under similar climate conditions and between the hours of 09:00-17:00. Measurements were taken every 15 minutes.

III. THERMAL ANALYSIS

Solar energy collectors are accepted as heat exchangers that increase the temperature of the operating air. In these types of collectors, energy efficiency can be calculated as the ratio of the available heat to the solar radiation on the collector [1,13].

$$Q_c = \dot{m}C_p (T_{out} - T_{in}) \tag{1}$$

$$\eta_{en} = \frac{Q_c}{(IA_C)} \tag{2}$$

$$\eta_{en} = \frac{\dot{m}C_p (T_{out} - T_{in})}{(IA_C)} \tag{3}$$

In exergy calculations, air was assumed to be ideal gas and that the experiments were carried out under steady state, steady flow conditions. In addition, the kinetic and potential energy effects as well as nuclear and chemical reactions were neglected. The heat transfer direction to the system and work transfer from the system were accepted as positive. According to these assumptions, the energy and exergy equalities of the system are as below [1,13,14].

$$\sum \dot{E}_{in} = \sum \dot{E}_{out} \tag{4}$$

$$\sum \dot{E}x_{in} - \sum \dot{E}x_{out} = \sum \dot{E}x_{dst} \tag{5}$$

$$\dot{E}x_{heat} - \dot{E}x_{work} + \dot{E}x_{mass,in} - \dot{E}x_{mass,out} = \dot{E}x_{dst} \tag{6}$$

General exergy equality [13] can be expressed as below using Eq. (6) .

$$\sum \left( 1 - \frac{T_e}{T_s} \right) \dot{Q}_s - \dot{W} + \sum \dot{m}_{in} \psi_{in} - \sum \dot{m}_{out} \psi_{out} = \dot{E}x_{dst} \tag{7}$$

The input and output specific exergy [13,14] can be written as such;

$$\psi_{in} = h_{in} - h_e - T_e (S_{in} - S_e) \tag{8}$$

$$\psi_{out} = h_{out} - h_e - T_e (S_{out} - S_e) \tag{9}$$

If Equation (8) and (9) are placed inside equation (7);

$$\left( 1 - \frac{T_e}{T_s} \right) \dot{Q}_s - \dot{m} (h_{out} - h_{in} - T_e (S_{out} - S_{in})) = \dot{E}x_{dst} \tag{10}$$

is obtained.

The solar energy absorbed by the absorber surface is as follows [13]:

$$\dot{Q}_s = I \alpha A_c \tag{11}$$

The enthalpy and entropy changes of the air in the collector [13,14];

$$\Delta h = h_{out} - h_{in} = c_p (T_{f,out} - T_{f,in}) \tag{12}$$

$$\Delta s = s_{out} - s_{in} = c_p \ln \frac{T_{f,out}}{T_{f,in}} - R \ln \frac{P_{out}}{P_{in}} \tag{13}$$

When Equations (11),(12) and (13) are placed in equation (10)

$$\left( 1 - \frac{T_e}{T_s} \right) I \alpha A_c - \dot{m} c_p (T_{f,out} - T_{f,in}) + \dot{m} c_p T_e \ln \frac{T_{f,out}}{T_{f,in}} - \dot{m} R T_e \ln \frac{P_{out}}{P_{in}} = \dot{E}x_{dst} \tag{14}$$

is obtained.

Exergy loss (irreversibility) [13,14] can be calculated as below:

$$\dot{E}x_{dst} = T_e S_{gen} \tag{15}$$

Dimensionless exergy loss [13];

$$\dot{E}x_D = \frac{\dot{E}x_{dst}}{Q_c} \tag{16}$$

is thus obtained.

Whereas exergy efficiency [13] is expressed as;

$$\eta_{II} = 1 - \frac{T_e S_{gen}}{\left[ 1 - T_e / T_s \right] \dot{Q}_s} \tag{17}$$

IV. RESULTS AND DISCUSSIONS

Efficiencies of 5 different collectors have been examined in three different air flow rates in this study. The air flow rates are 0.02 kg/s , 0.04 kg/s and 0.06 kg/s. The experiments were carried out during 11.07.2012 and 22.07.2012 at the city of Elazığ in Turkey. Radiation values and wind speeds during these dates were approximately similar.

The average radiation values for the 12 days during which the experiments were carried out have been shown in Fig.4 Radiation was calculated as 633 W/m<sup>2</sup> at 09:00 on the morning of the first day of the experiments, as 975 W/m<sup>2</sup> at 13:00 and as 434 W/m<sup>2</sup> at 17:00. Whereas radiation was calculated as 610 W/m<sup>2</sup> at 09:00 in the morning on the 12th day of the experiments, as 960 W/m<sup>2</sup> at 13:00 and as 415 W/m<sup>2</sup> at 17:00.

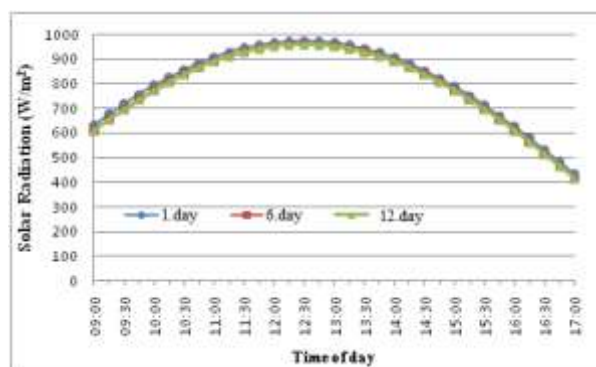


Fig. 4 Solar radiation on the collecting area

Available heat amounts at different flow rates were calculated for each collector using Equality (1). The change of available heat amount with time has been shown in Fig. 5 for each flow rate. The highest available heat was acquired in Type 3 at  $m=0.06$  kg/s. Available heat for this situation varies between 483 J/s and 1180 J/s and on average it is 990 J/s. Whereas the lowest available heat was observed in Type-I at a flow rate of  $m=0.02$  kg/s. Available heat varied between 160 J/s and 382 J/s for this and the average was determined as 311 J/s. Whereas Fig.6 shows the change of available heat in Type-III at different flow rates. As can be seen in the figure, available heat increases with increased flow rate.

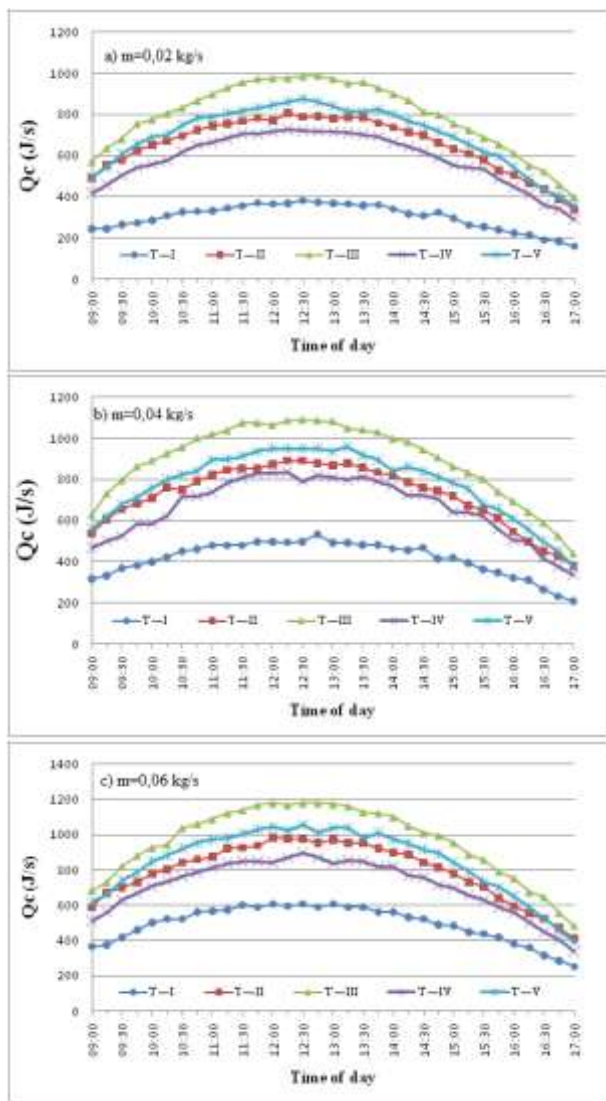


Fig.5. Variation of the useful heat rate versus time at different types of collector, for (a) 0.02 kg/s, (b) 0.04 kg/s, (c) 0.06 kg/s different mass flow rates.

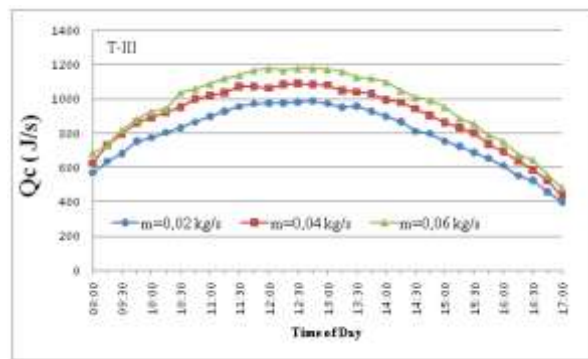
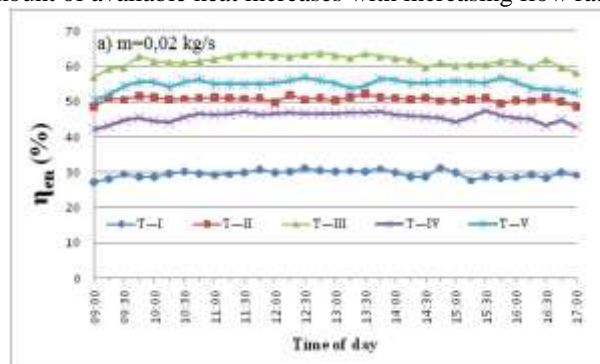


Fig. 6 Variation of the useful heat rate versus time at different mass flow rates, for Type-III

The reason for the higher available heat amount in collectors with increased turbulence effect using V type elements is that the absorbing surface area is greater in comparison with that of the conventional type collectors. This area is about  $1.71 \text{ m}^2$  in conventional type collectors, whereas in other collectors it is about  $2 \text{ m}^2$ . In addition, the fluid was forced to stay in the collector more in turbulence with the increased turbulence in the collector with V-type element placed inside. The reason why available heat increases with increasing flow rate is the fact that it has more contact with the fluid surface. In this manner, the absorber plate will be in contact with more flowing molecules and a greater heat transfer will occur. However, greater mass flow rate will result in the decrease in collector outlet temperature.

Energy efficiency for each collector was examined using Equality (3). According to the obtained results, the change of flow rate with energy efficiency has been shown in Fig.7 for each flow rate. The highest energy efficiency was calculated as %77.1 for Type-III at a flow rate of  $m=0.06$  kg /s. Whereas the lowest energy efficiency was calculated as %30.38 for Type-I at a flow rate of  $m=0.02$  kg/s. Fig.8 shows the change of efficiency for different flow rates for Type-III as a function of time. The increasing flow rate increases energy efficiency as well.

Lowest energy efficiency has been observed in the conventional type collector. The reason for this is that the lowest available heat is also observed in this collector type. Just as the case for available heat, the increase of flow rate also increases energy efficiency. The reason for this is that the amount of available heat increases with increasing flow rate.



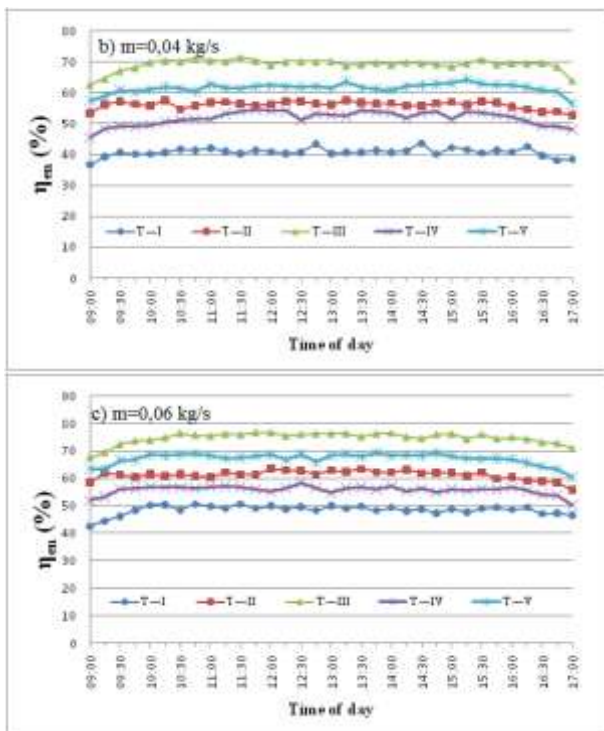


Fig.7. Variation of energy efficiency versus time at different types of collector, for (a) 0.02 kg/s, (b) 0.04 kg/s, (c) 0.06 kg/s different mass flow rates.

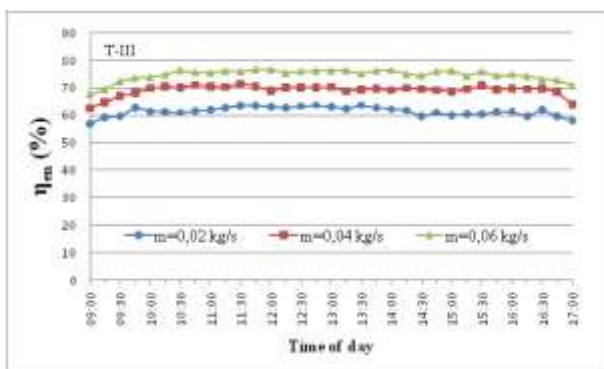


Fig.8 Variation of energy efficiency versus time at different mass flow rates, for Type-III

Exergy efficiency has been calculated using Equality (17). Solar air heater collectors are accepted as heat exchangers. That is why the  $T_s$  in the equation has been used as the temperature of the absorber plate. The change of exergy efficiency with time has been shown in Fig.9 according to obtained results. The highest exergy efficiency has been calculated as %30.32 for Type-III at a flow rate of  $m=0.02$  kg/s. Whereas the lowest exergy efficiency has been calculated as %4.83 for Type-I at a flow rate of  $m=0.06$  kg/s. Fig.10 shows the change of exergy efficiency with time for Type-III at different flow rates.

The collector with the lowest exergy efficiency is Type-I. Since the heat transfer amount is low in this collector, exergy efficiency was also low. Meanwhile, exergy efficiency is the indication of the energy quality obtained in a system. Hence,

the change of exergy efficiency obtained from a system is inversely proportional with the flow rate of the fluid.

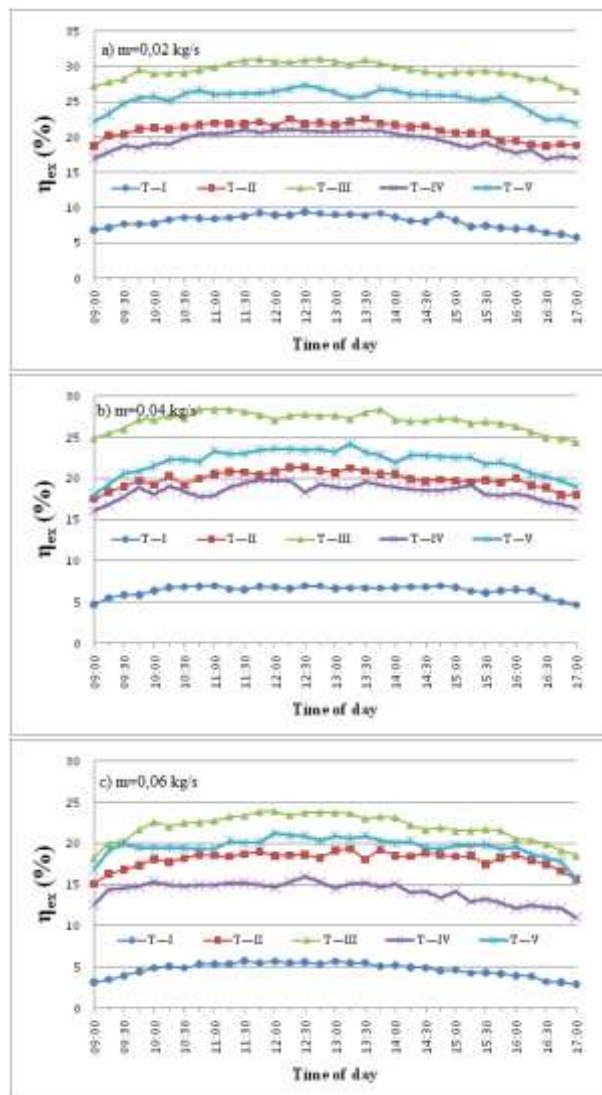


Fig.9 Variation of exergy efficiency versus time at different types of collector, for (a) 0.02 kg/s, (b) 0.04 kg/s, (c) 0.06 kg/s different mass flow rates.

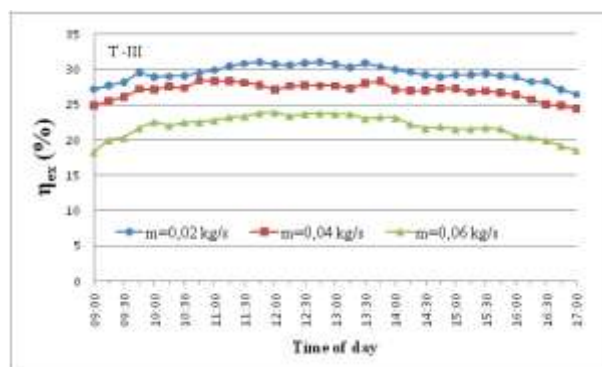


Fig.10 Variation of exergy efficiency versus time at different mass flow rates, for Type-III

Dimensionless exergy ratio was calculated using Equality (18). According to the obtained results, Fig.11 show the change of energy efficiency with time for each flow rate. Minimum dimensionless exergy ratio was calculated as 0.084476 for Type-III at a flow rate of  $m=0.06$  kg/s. Whereas the highest dimensionless exergy ratio was calculated as 0.292775 for Type-I at a flow rate of  $m=0.02$  kg/s. Fig.12 shows the change of dimensionless exergy ratio with time for Type-III at different flow rates.

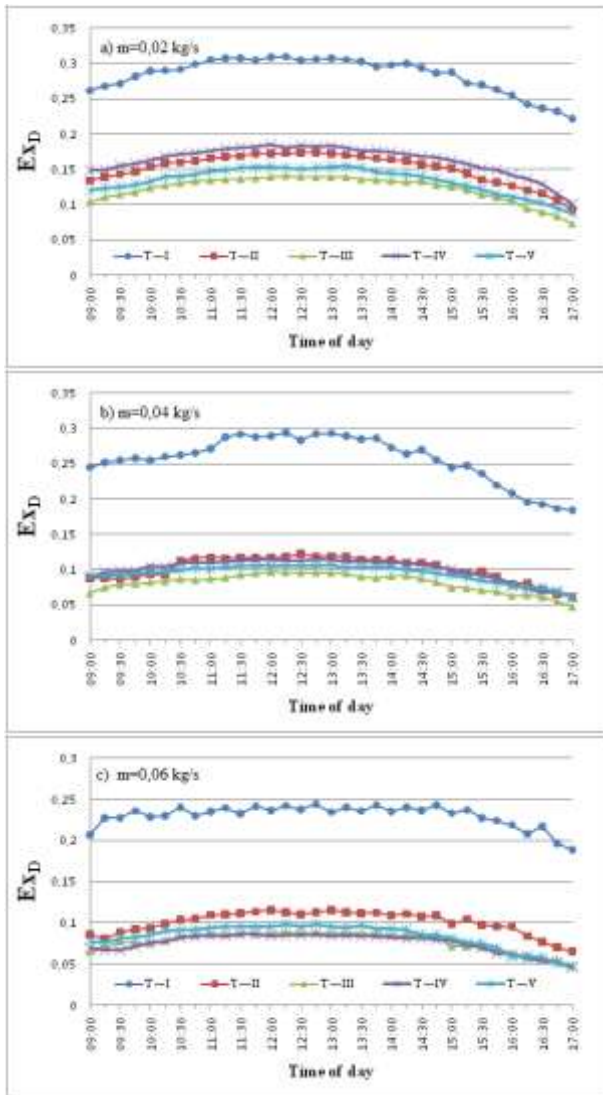


Fig.11. Variation of dimensionless exergy ratio versus time at different types of collector, for (a) 0.02 kg/s, (b) 0.04 kg/s, (b) 0.06 kg/s different mass flow rates.

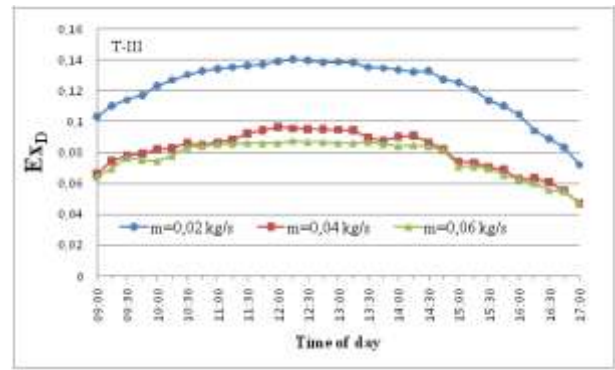


Fig.12 Variation of dimensionless exergy ratio versus time at different mass flow rates, for Type-III

## V. CONCLUSIONS

Energy and exergy efficiencies of 5 different solar air heaters have been examined experimentally in this study. The following results have been obtained as a result:

V type plates placed on the absorber surface have increased the heat transfer surface area. As a result, the available heat amount has been observed to be 51.5 % greater than the conventional type SAHs ( $m=0.06$  kg/s).

V-type surface expanding elements have increased turbulence. It has been observed that the placement has greater importance on turbulence. It has been observed that diagonal placement causes greater turbulence in contrast with straight placement.

It has been determined that energy and exergy efficiencies increase with increasing turbulence and surface area. It has been determined that energy efficiency increases with the increase of flow rate, whereas the exergy efficiency decreases.

It has been determined in this study that Type-III SAH has the highest energy and exergy efficiency.

## NOMENCLATURE

- A Surface are ( $m^2$ )
- $c_p$  Specific heat (J/kg K)
- $\dot{E}$  Energy rate (W)
- $\dot{E}_x$  Exergy rate (W)
- $\dot{E}_{x,D}$  Dimensionless exergy loss (dimensionless)
- $\dot{E}_{x,dst}$  Rate of irreversibility (W)
- h Enthalpy (J/kg)
- I Solar radiation ( $W/m^2$ )
- m Mass flow rate (kg/s)
- P Fluid pressure (Pa)
- R Universal gas constant (J/kg K)
- $Q_c$  Useful heat rate (W)
- $Q_s$  Incident energy in the collector area (W)
- s Entropy (J/kg K)
- $S_{gen}$  Entropy generation rate (W/kg K)
- T Temperature ( $^{\circ}C$ )
- $\dot{W}$  Work rate or power (W)
- $\alpha$  Absorptivity (dimensionless)

$\eta_{en}$	Energy efficiency (dimensionless)
$\eta_{ex}$	Exergy efficiency (dimensionless)
$\tau\alpha$	Effective transmission (dimensionless)
$\psi$	Specific exergy (J/kg)

*Subscripts*

a	Air
c	Collector
e	Environment
f	Fluid
in	Inlet
out	Outlet
gen	Generation

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