Energy and Exergy Analysis of Dual Channel Solar Air Collector with Perforating "V" Corrugated Absorber Plate

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Abstract
In This paper, an experimental study was carried out on a dual channel with perforating "V" corrugated absorber plate of solar air collector which the air flows both in upper channel and lower channel of the absorber plate for increasing heat transfer coefficient and improving thermal performance. The results of experimental procedures for dual channel with perforating "V" corrugated absorber plate of solar collector were compared with the flat plate dual channel of solar collector. Experimental calculations had been performed under Baghdad (33.34° North latitude, 44.4° East longitude) climatic conditions at different values of mass flow rates 0.021 kg/s, 0.027 kg/s and 0.032 kg/s. The results showed that the dual channel with perforating "V" corrugated absorber plate of solar collector is found to perform more efficiently than the flat plate dual channel of solar collector with increased 39% in thermal efficiency. Then, it showed that the efficiency increases with increasing mass flow rates. Also, it showed that the heat removal factor of the dual channel with perforating "V" corrugated absorber plate of solar collector is 36% more than the flat plate dual channel of solar collector. At the last part of the study, the exergy relations were derived for both collectors. The results of this part showed that the flat plate dual channel of solar collector is having largest irreversibility (exergy loss) and the dual channel with perforating "V" corrugated absorber plate of solar collector is having a greatest exergetic efficiency.
Keywords:Solar air collector; Dual channel; Double flow; Exergy analysis; perforating "V" corrugated; Heat removal factor

I. Introduction

The fast exhaustion of fossil fuel resources has required an insistent searching for renewable sources of energy to achieve the growing demand for energy for the future needs and subsequent generations. As can be seen, the reserves and resource base of fossil fuels are limited and exhaustible. Therefore, It is needful to take into consideration that renewable energy sources as a device of power generation. Solar energy is the largest popular source of non-conventional energy. It is better in quality, quantitatively plentiful, available everywhere, unfailing for most workable applications and has no polluting effect on the environment when transformed into useful forms. Iraq has an excellent geographic location [1], it is one of the relatively warm areas and the solar energy is available in large quantities throughout the seasons of the year.

Flat-plate collectors consider most common types of collectors for ease of manufacturing and low price compared to other collectors. It is used in applications that relatively require thermal energy at low temperatures as air heating process and drying the agricultural crops to minimize the use of electrical energy and thus, it relies partly on solar power as aide energy.

Recently, modern publications in the scope of solar thermal energy tend to adopt Exergy as a basis to conduct analytical studies. Exergy could represent the maximum possible useful work carrying out by the system of solar collector during the air heating process. An exergy is a useful tool for measuring the performance of energy transformation processes [2]. Consequently, exergy analysis applies to various thermal solar systems. In the last few years, a few studies about exergy analysis of solar air collector.

Ozturk and Demirel, 2004 [3], investigated heat transfer characteristics of a packed bed solar air collector. The packed bed of solar air collector was stuffed with gravel cylindrical rings which called Raschig rings for storing solar energy. The results showed that the packing improves the heat transfer and the energetic and exergetic efficiencies of the solar air collector with the packed bed are improved 33.78% and 2.16% respectively.

Ozturk, 2005 [4], investigated experimentally of solar air collectors with heat storage unit. A paraffin wax as a PCM was used in a solar collector for heating of the greenhouse and it was contacted each other in a parallel method. Energy and exergy analyses were applied in order to evaluate the system efficiency. The results showed that the energy efficiency of the greenhouse is improved about 40.4% and the exergy efficiency is improved about 4.2%.

Ucar and Inalli, 2006 [5], investigated the shape and arrangement absorber plate of the solar air collector. The surface of absorber plate arranged to increase heat transfer which in turn increased the efficiency of the collector. The five types of arrangement of absorber plate shape were used. The results showed that the energetic and exergetic efficiencies of the solar air collector with this modifications are improved approximately from 10% to 30% in comparison with the conventional solar collector.

Esen, 2008 [6], investigated experimentally solar air collector with several obstacles and without obstacles. The air was flowing upper and under absorber plate and the several types of obstacles were attached on top and bottom of absorber plate. The exergy analysis was presented and derived the exergy relations. The results showed that the energetic and exergetic efficiencies of the solar air collector with the obstacles are improved in comparison with the conventional solar collector as 45% and 38% respectively. Also, It showed that the conventional channel is having largest irreversibility.

Bahrehmanda and Ameri, 2015 [7], investigated theoretically and experimentally a flat plate with several glass cover. He has studied the effect of placing several glass covers on the surface of the solar collector and calculated the thermal loss coefficient from the top of the surface of the solar collector. The results showed that the thermal loss was reduced and the energetic and exergetic efficiencies were improved which mean the efficiency is 53.6% in comparison with single glass cover.

The work presented in this paper aims to analysis the performance of two kinds of collectors, that is a dual channel with perforating “V” corrugated absorber plate of solar air collector as well as the flat plate dual channel of solar collector. The results obtained from the collector of the perforating “V” corrugated are compared with the collector of flat plate dual channel. An experimental rig described in the next section, it is constructed and tested in Engineering College of Baghdad University, Baghdad, Iraq. The efficiency of the solar air collectors evaluated by using the test measurements. The different parameters such as the configuration of fluid flow in absorber plate and the mass flow rate of air in the duct, the effect of these parameters on the energetic and exergetic efficiencies were ensured In this study.

II. Experimental Apparatus

An image view of the experimental rig of solar air collectors system and the details described in it as shown in Fig.1-3. It is measured in Engineering College of Baghdad University, Baghdad, Iraq and situated at latitude 33.3° N, longitude 44.4° E and 41 m elevation above the sea level [1].

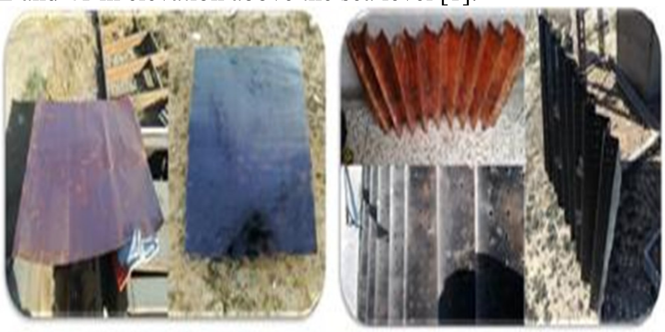


Fig 1. Photographic view for two models of absorber plates.

1	Anemometer	4	Data loggers	7	Air blower
2	Solar meter	5	Glass cover	8	Connection pipe
3	Thermocouples	6	Absorber plate	9	Stand with tilt 43˚

Fig.2. Photographic view of perforating “V” corrugated plate (right) and flat plate (left) of dual channel SAH.

In this paper, two kinds of flow channels of collectors were used. The plates of absorber were made of copper metal with a black matte coating. Size and thickness of the absorber plate for two collectors was 0.9 m, 0.7 m and 0.6 mm respectively, as shown in fig.1-3. A single of a normal window glass of thickness 6 mm was used as a transparent cover. The housing made of galvanized steel gauge 22 and it coated with a layer of a fiberglass (thickness 5 cm and density 48 kg/m3) was used for insulating the collector from sides and bottom to reduce heat losses. An air blower was used with a horsepower (0.5 hp) at a rate of 2800 rpm. The air flow rate can be changed by an external gate that consists of vane damper at the inlet of fan.

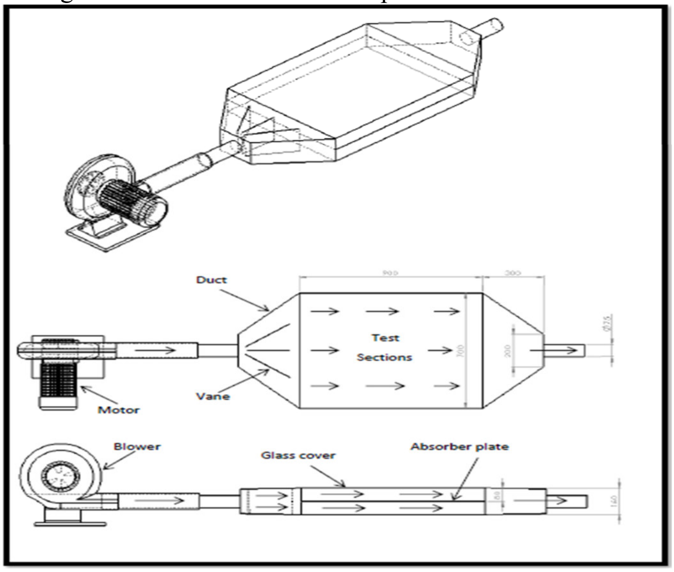


Fig. 3. Schematic diagram of dual channel solar collector.

III. Thermal analysis

The performance of solar air collectors (in fig.4) evolved by the energy balance. It is determining the distribution of solar energy incident relative to the energy used, useful about it, and various losses [8]. The incident solar energy is:

$$Q_i = I \,.\, A \qquad (1)$$

Useful heat gain by the solar collector is found as the following equation:

$$Q_u = F_R\, A[(I\tau\alpha) - U_L\,(T_i - T_a)] \qquad (2)$$

Then, the heat removal factor can be expressed of solar air collector as follows equations [8]:

$$F_R = \frac{mc_p(T_o - T_i)}{A[I(\tau\alpha) - U_L\,(T_i - T_a)]} \qquad (3)$$

A measure of performance solar air collector can be evolved by calculating the thermal efficiency:

$$\eta = \frac{Q_u}{AI} \qquad (4)$$

The collector overall heat transfer coefficient (U_L) is founded by relation as[8]:

$$U_L = U_t + U_b + U_e \qquad (5)$$

The coefficient of edge loss can be expressed as follows equation:

$$U_e = \frac{K_e}{X_e} * \frac{A_e}{A} \qquad (6)$$

The coefficient of top loss from the absorber plate to the environment is calculated as the following equation [8]:

$$U_t = \left[\frac{1}{h_{c,f\text{-}g} + h_{r,p\text{-}g}} + \frac{1}{h_{c,g\text{-}a} + h_{r,g\text{-}s}}\right]^{-1} \qquad (7)$$

The heat transfer radiation coefficient from the glass cover to the sky ($h_{r,g-s}$) is calculated as:

$$h_{r,g\text{-}s} = \varepsilon_g \sigma \left(T_g + T_s\right)\left(T_g^2 + T_s^2\right) \qquad (8)$$

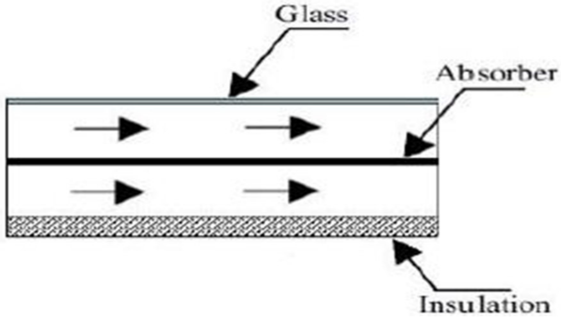


Fig. 4. Schematic of dual channel solar collector, a)Flat plate

The coefficient of heat transfer radiation from the absorber plate to the glass ($h_{r,p-g}$) is calculated as:

$$h_{r,p-g}=\frac{\sigma(T_p+T_g)(T_p^2+T_g^2)}{\frac{1}{\varepsilon_p}+\frac{1}{\varepsilon_g}-1} \quad (9)$$

The back loss coefficient U_b is found from [9];

$$U_b=[\frac{1}{h_{c,fl-b}+h_{r,p-b}}+\frac{X_b}{K_b}]^{-1} \quad (10)$$

The heat transfer radiation coefficient from the absorber plate to the bottom plate ($h_{r,p-b}$) as follows:

$$h_{r,p-b}=\frac{\sigma(T_p+T_b)(T_p^2+T_b^2)}{\frac{1}{\varepsilon_p}+\frac{1}{\varepsilon_b}-1} \quad (11)$$

The heat transfer convective coefficient (h) of free convection depends on wind speed (v) [10]:

$$h_w=5.7+3.8\ v \quad (12)$$

The heat transfer convective coefficient is found as:

$$h_{c,p-f}=\frac{NuK}{D_h} \quad (13)$$

The empirical correlation to define the Nusselt number for the turbulent flow (Re > 6000) at flows in rectangular duct exposed to a constant heat flux from top and insulation from bottom for flat plate as follows[11]:

$$Nu = 0.036\ Re^{0.8}\ Pr^{1/3}(D_h/L)^{0.055} \quad (14)$$

when the hydraulic diameter (D_h) is twice the spacing.

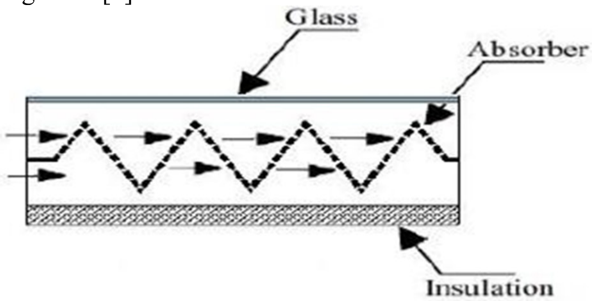
For perforating "V" corrugated plate:

$$Nu = 0.09\left(\frac{P_t}{d}\right)^{0.137} Re^{0.527}\ Pr^{-5.66} \quad (15)$$

P_t : pitch of holes, d : diameter of holes.

$$D_h = H_{i,min} + b \quad (16)$$

where b is the half height of the "v" groove and Hi,min is the space between the glass cover and the tip of "v" groove [8].



b)Perforating "V" corrugated plate

IV. EXERGY ANALYSIS OF CASE STUDY

This research includes evaluating the exergetic efficiency for conventional and dual channel solar air collectors and exergy loss (work loss) because of the internal irreversibility and work loss due to heat transfer to the environment in various thermal process across the solar collector [12].

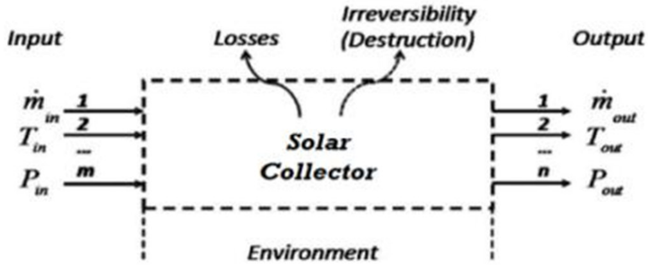


Fig. 5. Components of exergy balance

The exergy balance in fig. 5. that determines the distribution of thermal exergy as follows [6]:

$$\Sigma \dot{E}_{in} = \Sigma \dot{E}_{out} \quad (17)$$

$$\Sigma \dot{E}x_{in} - \Sigma \dot{E}x_{out} = \Sigma \dot{E}x_{dest} \quad (18a)$$

or

$$\dot{E}x_{heat} - \dot{E}x_{work} + \dot{E}x_{mass,in} - \dot{E}x_{mass,out} = \dot{E}x_{dest} \quad (18b)$$

Thence:

$$\sum \left(1-\frac{Te}{Ts}\right) Q_s - \dot{W} + \Sigma\, \dot{m}_{in}\, \psi_{in} - \Sigma\, \dot{m}_{out}\, \psi_{out} = \dot{E}x_{dest} \quad (19)$$

Where

$$\psi_{in} = (h_{in} - h_e) - T_e (s_{in} - s_e) \quad (20)$$

$$\psi_{out} = (h_{out} - h_e) - T_e (s_{out} - s_e) \quad (21)$$

The variance of the entropy and the enthalpy in the channels of air flows may be expressed as follows equations [5]:

$$\Delta h = h_{out} - h_{in} = c_p (T_{f,out} - T_{f,in}) \quad (22)$$

$$\Delta s = s_{out} - s_{in} = c_p \ln\frac{T_{f,out}}{T_{f,in}} - R \ln\frac{P_{out}}{P_{in}} . \quad (23)$$

The exergy destruction can be calculated according to the below relation:

$$\dot{E}x_{dest} = T_e\, S_{gen.} \quad (24)$$

The exergetic efficiency is founded as the following equation:

$$\eta_{II} = 1 - \frac{T_e\, S_{gen}}{[1-(T_e/T_s)]\, Q_s} \quad (25)$$

The cp of air can be found from the mean bulk temperature as the following relation:

$$\Delta T_m = (T_{in} + T_{out}) /2. \quad \mathbf{(26)}$$

V. RESULTS AND DISCUSSIONS

The outdoor test was conducted during the December (2016) in Baghdad. The collector was oriented south at 43° (For winter load, the tilt should be (latitude + 10) degrees and for year round use, the tilt equal latitude) [13]. The experiments were carried out from 8:30 AM to 16:00 PM for clear days. Different mass flow rates (0.021, 0.027, 0.032) kg/s are also investigated at the experiments.

Fig. 6 shows the variation of an inlet and outlet air temperature of the two models of solar air collectors with day hours on (20th Dec. 2016 and 12th Jan., 23rd Jan., 5th Feb. 2017). These figures show that the maximum of outlet air temperature for two collector models sequentially was (39.7°C) in model (III) and (31.2°C) in model (II) on (12th Jan. 2017) experimentally.

Fig. 7 shows the variation of thermal efficiency with mass flow rates (0.0217, 0.0271, 0.0325 kg/s). The results show the efficiency of the collector increases at the mass flow rates of the air inlet the solar collectors is increased and these results are scientifically agreed upon. The average thermal efficiency was (51.6 %) in model (II) on (20th Dec. 2016) and (72.2 %) in model (III) on (12th Jan. 2017) experimentally.

Fig. 8. Shows the variation of heat removal factor of the five models of solar air collectors with day hours. These figures show that the average of heat removal factor for two collector models sequentially was (0.68) in model (II) and (0.93) in model (III) on (12th Jan. 2017) experimentally.

The variation of the Nusselt number through collector for five models of solar collectors experimentally was shown in fig. 9. These curves represent the increase in the Nusselt number through collector with Reynolds number at clear climate, where the Nusselt number through collector increases steadily with Reynolds number and reaches its peak value (107.2) in model (III) at the highest Reynolds number (29208.4) due to high turbulence created through this model of the collector.

The change of the exergy destruction with day hours for two models of solar collectors experimentally was shown in fig. 10. during the winter season for days on (20th Dec. 2016 and 12th Jan., 23rd Jan., 5th Feb. 2017). These curves represent the increase in the exergy destruction with day hours at a constant flow rate and clear climate, where the exergy destruction increases steadily with time and reaches its peak value at the mid-day and then begins to decline gradually.The maximum of the exergy destruction at midday was (404.5 W) in model (II) and (328.7 W) in model (III) on (12th Jan. 2017) experimentally.

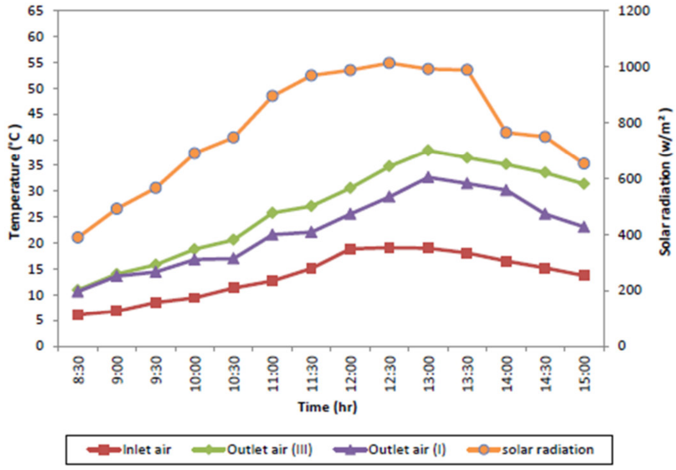


Fig. 6. Variation of inlet and outlet air temperature with time and solar radiation for model (III) and model (I).

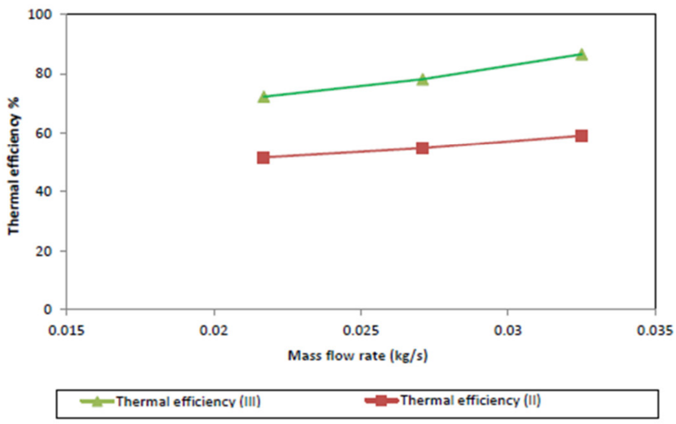


Fig. 7. Variation of thermal efficiency values with mass flow rates for two models of solar air collectors.

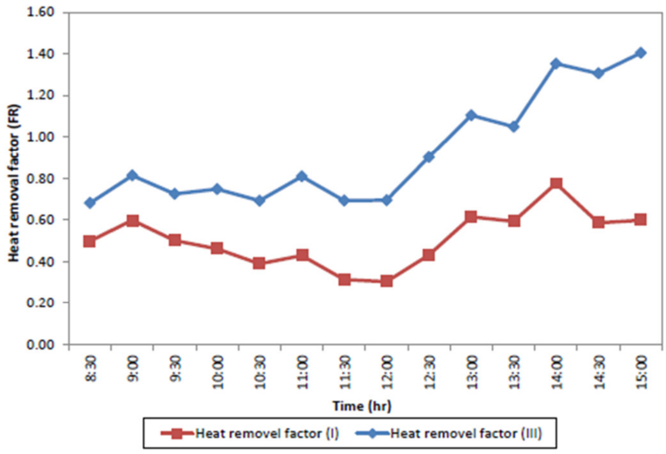


Fig. 8. Variation of heat removal factor values with time for model (III) and model (I).

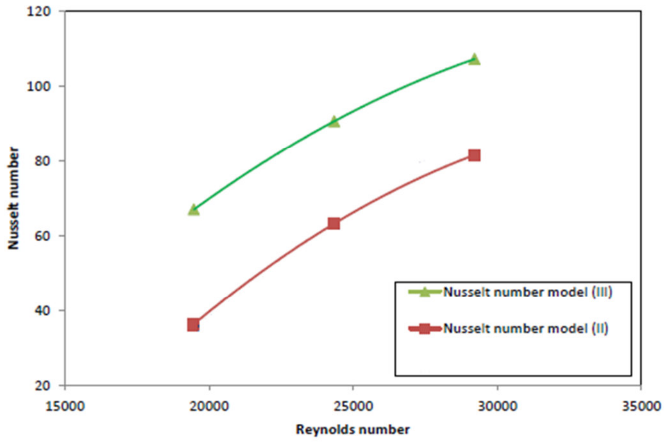


Fig. 9. Variation of Nusselts number values with Reynolds number for two models of solar air collectors.

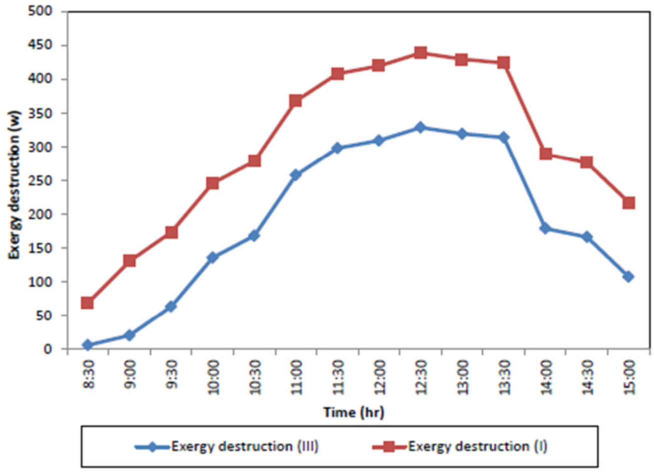


Fig. 10. Variation of exergy destruction values with time for model (III) and model (I)

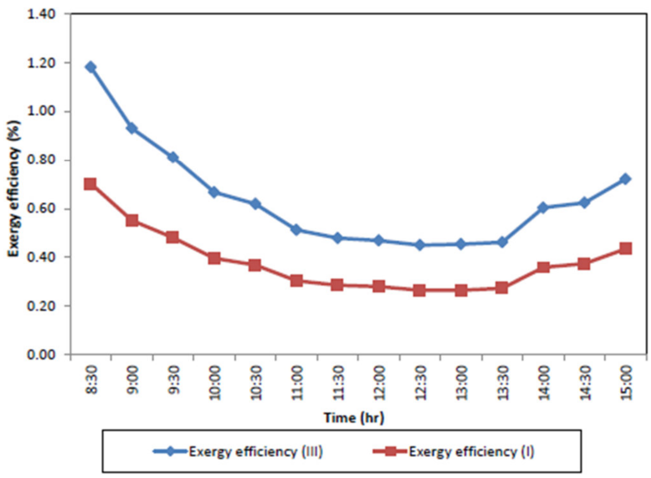


Fig. 11. Variation of exergy efficiency values with time for model (III) and model (I).

The variation of the exergy efficiency with day hours for two models of solar collectors experimentally was shown in fig. 11. during the winter season for days on (20th Dec. 2016 and 12th Jan., 23rd Jan., 5th Feb. 2017). These curves represent the increase in the exergy efficiency with day hours at a constant flow rate and clear climate. The maximum of the exergy efficiency was (64.1 %) in model (III) on (12th Jan. 2017) experimentally.

VI. CONCLUSIONS

The dual channel with perforating "V" corrugated absorber plate type of solar air collector used for increasing heat transfer coefficent and heat removal factor that is leading to improved thermal performance and exergetic efficiency. Based on results obtained from the experimental tests, The thermal efficiency of dual channel with perforating "V" corrugated absorber plate is 39% higher than that of the flat plate collector and it increases for higher mass flow rates. The heat removal factor of dual channel is 36% more than the flat plate collector. The flat plate collector is having largest irreversibility. Dual channel collector with perforating "V" corrugated absorber plate is having a greatest exergetic efficiency.

VII. NOMENCLATURE

Symbol	Definitions	Units
A	Collector area	(m2)
C_P	Specific heat	(kJ/kg K)
Dh	Hydraulic diameter	(m)
$\dot{E}$	Energy rate	(kW)
$\dot{E}x$	Exergy rate	(kW)
h	Coefficient of heat transfer	(W/m2.K)
I	Solar radiation	(W/m2)
k	Thermal conductivity	(W/m. K)
L	Length of collector	(m)
$\dot{m}$	Mass flow rate	(kg/s)
Nu	Nusselt number	
Pr	Prandtl number	
P	Pressure	(N/m²)
Q	Energy	(W)
S	Entropy	(kJ/kg K)
T	Temperature	(˚C)
U	Heat loss coefficient	(W/m2 ˚C)
V	Velocity	(m/s)
$\dot{W}$	Work rate or power	(kW)
X	Thickness	(m)

Subscripts

a	Ambient	m	Mean
b	Bottom	o,out	Outlet
c	Convection	p	Plate
e	Edge, Environment	r	Radiation
f	Fluid	s	Sky, Source
g	Glass	u	Useful
i,in	Inlet,Incident	w	Wind
des.	Destruction	gen.	Generation

Greek letters

α	Absorptance	η_{II}	Exergetic efficiency
ε_p	Emittance of surface	σ	Bultzmann constant
ε_g	Emittance of glass cover	τ	Glass transmittance
η	Thermal efficiency	ψ	Specific exergy

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