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SINGLE PASS SOLAR AIR HEATER WITH TRANSVERSE FINS AND WITHOUT ABSORBER PLATE

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ABSTRACT:

The single pass solar air heater, with transverse fins and wire mesh used as an absorber plate, is constructed and tested for thermal efficiency at a geographic location of Cyprus in the city of Famagusta. The absorber plate was replaced by sixteen steel wire mesh layers, 0.18 x 0.18cm in cross section opening and a 0.02cm in diameter. The fins were painted with black color and positioned transversely along the bed such that four equally spaced sections were created. The transversely fins are arranged in a way to force the air to flow through the bed like eight letter path. The obtained results show that for air mass flow rate rang between 0.011-0.032 kg/s, the thermal efficiency increases with increasing the air mass flow. The maximum efficiency obtained is 58% for the mass flow rate of 0.037 kg/s. Moreover, the temperature difference between the outlet flow and the ambient, ΔT, reduces as the air mass flow rate increases. The maximum difference between the outlet and ambient temperature obtained was 40.6 ^{o}C for mass flow rate of 0.011kg/s. Comparison with a conventional single pass collector shows a substantially enhancement in the thermal efficiency.

INTRODUCTION:

Heating air with solar energy is much cleaner than heating with fossil fuel, the delivered heat from air solar device can be used for drying agricultural products such as crop, grains, seeds, fruits, and vegetables. Solar air heaters are also used as pre heaters in industries and as auxiliary heaters in building to save energy during winter times [1].

Conventional solar air heaters mainly consist of panels, insulated hot air ducts and air blowers in active systems. The panel consists of an absorber plate and a transparent cover. There are many different parameters affecting on the solar air heater efficiency, e.g. collector length, collector depth, type of absorber plate, glass cover plate, wind speed, etc. [2].

The absorber plate area and heat transfer coefficient between the air and the absorber plate are two important parameters affecting the efficiency of the collector. When increased these parameters will increase the collector efficiency. On the other hand it will increase the pressure drops inside the solar air heater and increases the pumping power required [2]. The convective heat transfer rate between air flow and absorber plate could be augmented by increasing the absorber surface area and by increasing turbulence inside the bed [3]. The use of V- groove absorber was found to be more efficient by 12% than the flat plate collector of similar design [4] [5]. Using corrugated plates is a suitable method to increase the thermal performance and provides higher compactness [6]. A mathematical model that allows the determination of the thermal performances of the single-pass solar air collector with offset rectangular plate fin absorber plate is developed by [7]. Experimental and theoretical investigations are carried out on sheet metal absorber in the form of a chevron pattern [8]. The effect of using absorber plates coated with various selective coating materials on the heater performance was also investigated by [9]. Comparisons were made among four types of solar air collectors by Suleyman [10]. It has been found that the performance of a solar air heating system can be improved by operating several sub collectors in series in place of a single large collector with the same total area [11] [12]. The study concludes that the performance of solar air heater with single plastic glazing and flat plate absorber were approximately 9% more than the collector with double plastic glazing. Another study was carried out to determine the comparative performance of one-pass for fixed and free fins,

NOMENCLATURE:

A_c	(m2)	Area of the collector
C_p	(kJ/kg.K)	Cp specific heat of the fluid
h	(m)	Fluid deflection inside the incline manometer
I	(W/m2)	Solar radiation
m	(kg/s)	Air mass flow rate
Q	(m^3/s)	Volume flow rate
T_{in}	(^{o}C)	Inlet temperature

T_{out}	(°C)	Outlet temperature
T_{air}		film air temperature
ΔT	(°C)	Temperature difference (T_{out}- Tin)
ρ	(kg/m3)	Density of air
η	(-)	efficiency of the solar collector
ΔP	(N/m2)	Pressure difference, $\Delta P = \rho gh \sin 15$
ω	(-)	Uncertainty for the mass flow rate
Φ	(-)	Porosity

fixed fins collector is more effective than free fin collector [13]. The great improvement in performance is obtained with porous flat plate collectors as compared with analogous nonporous types. The study concluded that the presence of porous media in the channel increases the outlet temperature and as a result the thermal efficiency of the systems increases [14]. The thermal performances of single solar air heaters with steel wire mesh layers are used instead of a flat absorber plate are investigated experimentally [15]. Yousef and Adam [16] with using of porous media found, decreasing the flow depth result in increasing the collector thermal efficiency due to the increasing in out let temperature. On other hand, the pressure drop of their collector is increased too.

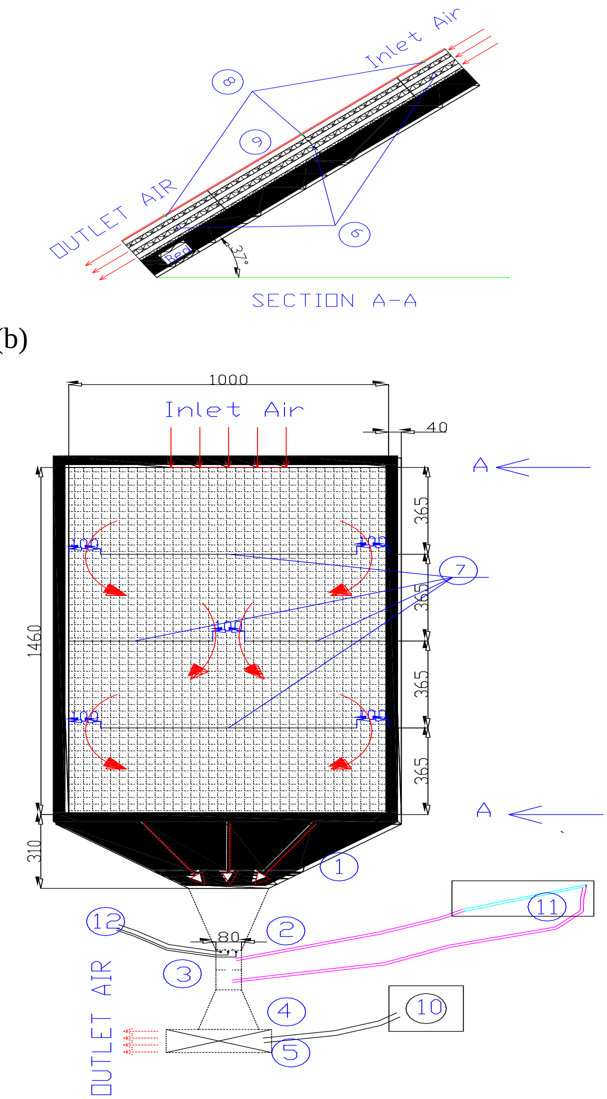
Different modifications are suggested and applied to increase the heat transfer coefficient between the absorber plate and the air stream, these modification include using finned absorber plates and porous material like wire mesh screen [17]. Aldabbagha [18] obtained results showed that the thermal efficiency of collector has been increased with increasing the number of fins with wire mesh as absorber.

Studies on single pass solar air heater with 3cm height experimentally are rather scarce. In this work, the single pass solar air heaters, with transverse fins and wire mesh used as an absorber plate was investigated experimentally. In order to increase the area of the collector and reduce the pressure drop, sixteen steel wire mesh layers arranged in three groups with three transverse fins were used.

EXPERIMENTAL SET-UP AND EQUIPMENTS:

The schematic diagram of an experimental set-up is shown in Fig 1. The set up was designed and constructed in order to obtain data for the investigation. The set-up consists of a wooden collector of 1.47m long and 1m wide and 3cm channel height. The frame of the collector was made of 2cm thick plywood painted with black and externally insulated with 2cm thick Styrofoam. Normal window glass of 0.4cm thick was used as glazing. The distance between the glass and the bottom of the collector was fixed to be 3cm. In order to increase the area and the air path length of the collector, four Aluminum fins, two of them 80cm long and others 45cm, with 2.7cm in height and 0.3cm in thickness were positioned transversely in the channel.

The fins were painted with black color to increase the absorptivity and reduce the reflectivity of the fins. The transverse fins were positioned along the bed such that four equally spaced sections were created. The transverse fins are arranged in a way to force the air to flow through the bed like eight letter path. A black slot rubber band, 0.5cm in height and 0.3cm thickness, was used to separate the fins from the glass.



1- Converging section.
2- Converging duct.
3- Orifice meter.
4- Diverging duct.
5- Air blower.
6- Bed thermocouples.
7- Fins.
8- Glass thermocouples.
9- Glass.
10- Speed controller.
11- Incline manometer.
12- Outlet air thermocouples.

Figure 1 (a) schematic assembly of the SAH system (b) section A- A

The aim of using the black slot rubber is to prevent the air to pass from the upper of the fins and heat transfer from fins to the glass.

Three wire mesh matrices was packed in the bed. The first and second matrix consist of 6 layers fixed at bottom and mid of the bed, where as the third matrix consists of four layers. The distance between the matrices was fixed to be 0.5cm. Arranging the wire mesh matrices in this way in the collector will reduce the pressure drop buildup as a result of using the porous media. The wire screen matrices are placed between the fins and were painted with black before installing (Fig. 1b). These wires mesh replace the absorber plate in the traditional solar air collectors; hence, the design is cheaper compared to the solar air heater having absorber plates because the wire mesh is really cheap and it is always available in the market. In operation, hot air flowing through the four equal sections. The lower section channel will be forced hot air to pass through the converging section and then into the orifice meter. The orifice meter is insulated and fixed between blower and bed by two galvanized ducts. A calibrated orifice meter was installed inside the pipe for measuring the volume flow rate of the air. The orifice meter was designed according to Holman [19]. Two flow straighteners are installed inside the pipe before and after the orifice meter to obtain a uniform flow through the orifice meter. Each straightener is consisted of plastic straw tubes having 0.595 cm diameter and are 2.5 cm long. A radial 0.62 kW fan (Type OBR 200 M-2K) was connected to the discharge of the solar air heater. The pressure difference through the orifice was measured by using an inclined tube manometer filled with alcohol having a density of 803 kg/m^3. The angle of the manometer was fixed at 15^o. Different mass flow rates were obtained by using a speed controller which is connected to the radial fan in order to control the fan speed. The inlet temperature, T_{in}, was measured by using two mercury thermometers fixed underneath the solar collector to measure the ambient air temperature. Nine thermocouples, T type, were used to measure the temperature of bed, T_{bed}, glass, T_g, and outlet air, T_{out}, distributed in three groups. Each group contain three thermocouples. The first three thermocouples were fixed inside the wire mesh to measure the temperature of bed. The first thermocouple was fixed inside the wire mesh and positioned at a mid upper section, i.e near the air entrance. The second thermocouple was fixed at a distance 75cm from the top of the SAH. Whereas the third one was fixed at mid fourth section near the outlet. The other three thermocouples were fixed on the glass inside a bed and above the bed thermocouples. The last three thermocouples were fixed inside the pipe before the orifice meter to measure the outlet air temperature of the working fluid from the bed. All temperatures reading were recorded by Digital Thermometer (OMEGASAYS) ±0.5 ^{o}C accuracy. A calibration test showed that the accuracy of the thermocouples reading were within ±0.15 ^{o}C. The solar intensity on an inclined surface was measured using an Eppley Radiometer Pyranometer (PSP) coupled to an instantaneous solar radiation meter model HHM1A digital, Omega 0.25% basic dc accuracy and a resolution of ±0.5% from 0 to 2800 W/m^2. The Pyranometer was fixed beside the glass cover of the collector. The solar heater was oriented facing south and tilted to an angle of 37^o with respect to the horizontal to maximize the solar radiation incident on the glass covers [20]. Air is circulated for 60 min prior to the period in which data is taken. The measured variables, ambient temperatures, outlet air temperatures of the collector, wind speed, and relative humidity ratio were recorded. The inclined tube manometer reading and the solar radiation were also recorded at 60 min time intervals. All tests began at 8:00 am and ended at 5:00 pm daily.

THERMAL ANALYSIS AND UNCERTAINTY:

Errors associated with the experimental measurements are presented in the previous section. Thermal efficiency, uncertainty due to the air mass flow rate and the thermal efficiency are presented here. The equation for mass flow rate (m) is

$$m = \rho . Q \tag{1}$$

where, ρ is the density of air and Q is the volume flow rate which depends on the pressure difference at the orifice which is measured from the inclined manometer.

The fractional uncertainty, ω_m /m, for the mass flow rate is [19, 21]

$$\frac{\omega_m}{m} = \left[\left(\frac{\omega_{T_{air}}}{T_{air}} \right)^2 + \left(\frac{\omega_P}{P} \right)^2 \right]^{1/2} \tag{2}$$

The efficiency of the solar collector, η, is defined as the ratio of energy gain to solar radiation incident on the collector plane,

$$\eta = \frac{m C_p \, (T_{out} - T_{in})}{I \; A_c} \tag{3}$$

The uncertainty for efficiency from Eq. (3) is a function of ΔT, m, and I, considering C_p and A_c as constants.

$$\frac{\omega_\eta}{\eta} = \left[\left(\frac{\omega_m}{m} \right)^2 + \left(\frac{\omega_{\Delta T}}{\Delta T} \right)^2 + \left(\frac{\omega_I}{I} \right)^2 \right]^{1/2} \tag{4}$$

Performance investigations for different mass flow rates were carried out; the average values of each variable were calculated daily. Then, the mean values of each variable for all the days were obtained and used to calculate the fractional uncertainty. The mean average values for ΔT, T_{in}, T_{out}, T_{air} , m, I , and η were found to be 21.69^{o}C, 33.72 ^{o}C, 55.41 ^{o}C, 44.57 ^{o}C, 0.0218 kg/s, 699.3 W/m^2 and 42.57% respectively. The fractional uncertainty of the mass flow rate and the efficiency are found to be 0.0077 and 0.0092 respectively.

RESULTS AND DISCUSION:

This experimental work investigates the effect of partitioning single pass mesh wire packed bed SAH under Gazimagusa prevailing weather conditions during the summer months, 3.07 2011 - 11.07.2011, with clear sky condition. Gazimagusa is a city in North Cyprus located on 35.125^o N and 33.95^o E longitude. Generally, Gazimagusa sky was clear and the average hourly recorded mean value of the wind speed and relative humidity ratio which were taken from the metrological office of Gazimagusa city was 14.8 m/s and 60.78% respectively. The performance of the proposed single pass solar air heater was done by used fins and 16 steel wire mesh layers as absorber with 3 cm high of bed was studied and compared with the performance of a conventional solar air heater. The mass flow rate of the air was varied from 0.011 to 0.032 kg/s. The uncertainty of the mass flow rate is calculated to be 0.0077 [21].

Figure 2 shows the hourly variation of solar intensity versus local time from 8:00 am to 5:00 pm, the experiment was done. The solar radiation was increasing from morning to a peak value at 1:00 pm and then, decreasing in the afternoon until sunsets.

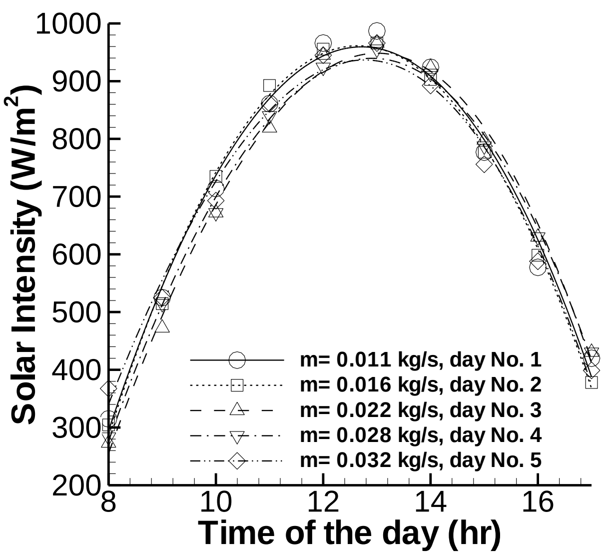


Figure 2 Solar intensity versus different standard local time of days for single pass solar air heater

The highest daily solar radiation obtained was 982 W/m^2 at noon and the average values of the solar radiation were 699.3 W/m^2. Calculation of all means averages solar intensity for each day was within the same and close range. Figure 3 shows the variation of air inlet temperature with time of the day for all the days of the experiment. The input temperature varies between 29 ^{o}C at morning to 40 ^{o}C. The inlet temperature in general increasing from morning till evening with slightly reduced at 5:00 pm. In some of the day the inlet temperature found to continue increasing from morning till evening as a result of low wind speed. Wind speed had an effective impact on humidity ratios and inlet temperature, which caused fluctuation during some of the days from the morning to evening.

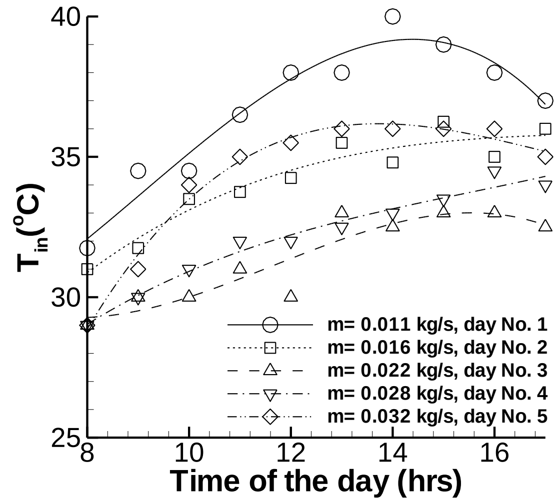


Figure 1 Ambient temperatures versus different standard local time of days

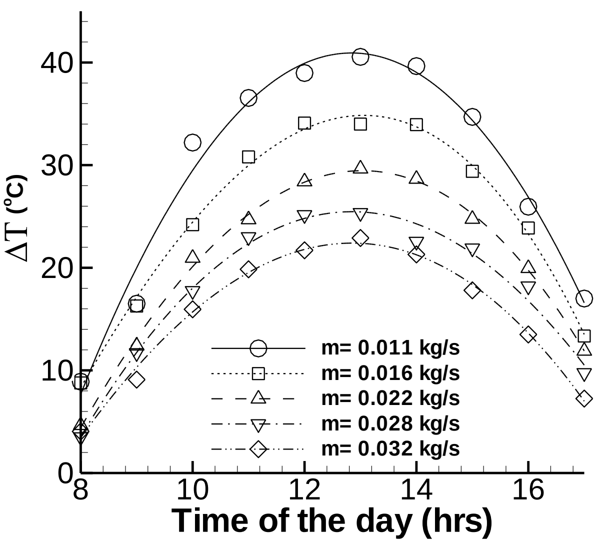


Figure 4 Temperature difference versus standard local time of the day at different mass flow rates

The hourly temperature differences ($\Delta T = T_{out} - T_{in}$) for different mass flow rates and with four partition solar air heater is shown in Fig. 4. As expected the temperature differences increased to a peak value at noon, 1:00 pm, and decreased in the afternoon until sunset the same as the solar intensity behaviors (Fig. 2). In general, ΔT was found to reduce with increasing air mass flow rate. The maximum ΔT obtained in this work is 40 ^{o}C at 1:00 pm with minimum mass flow rate, 0.011 kg/s. Sawi [8] with single pass solar air heater, reported that maximum value of ΔT was 40^{o}C when chevron pattern copper was used as absorber plate where the air mass flow rate was 0.0048 kg/s. The peak value of ΔT for six transfer fins with porous media wire mesh reached to 62.1 ^oC for m= 0.0121 kg/s and solar radiation was 1126 W/ m^2 [18]. The temperature difference between the outlet and inlet was 27 ^oC when used ten wire mesh layers for mass flow rate 0.012 kg/s and 1050 w/m^2 solar intensity.

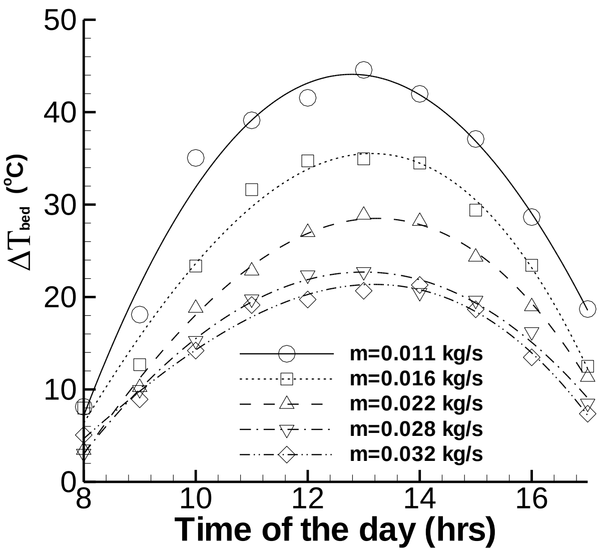


Figure 5 Bed temperature difference versus standard local time of the day at different mass flow rates

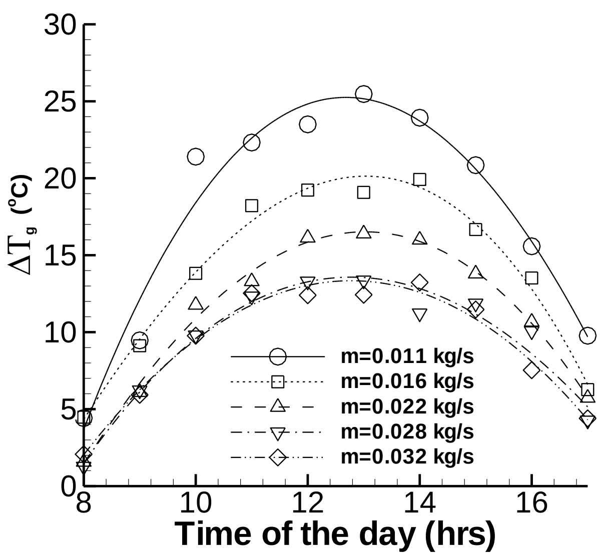


Figure 6 Glass temperature difference versus standard local time of the day at different mass flow rates

The bed temperatures difference, $\Delta T_{bed} = T_{bed} - T_{in}$, and glass temperatures difference, $\Delta T_g = T_g - T_{in}$ versus standard local time of the day for all the days in the experiment was carried out are presented in figure 5 and figure 6. Where T_{bed} and T_g is the average bed and glass temperature respectively.

The maximum temperature difference of the bed is found to be 44.6 ^oC for the air mass flow rate of 0.011 kg/s at 1:00 pm. The small difference between ΔT_{bed} and ΔT give a good evident that there is a good heat transfer in the channel of the SAH from the bed to the air. In other side, the high temperature difference of the glass, ΔT_g, means that there is a beg amount of heat transfer from the glass to the surrounding (Fig. 6) which regard as a heat losses. This heat loses defiantly will reduce the efficiency of the SAH.

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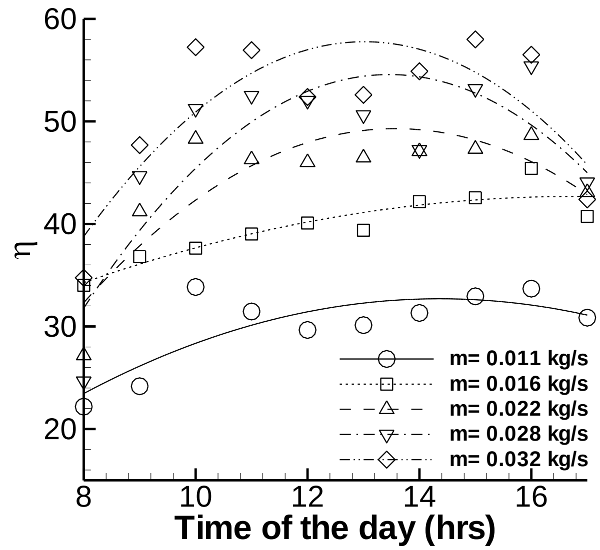

Figure 7 Variation of collector efficiency at different mass flow rates

The heat loses from the glass cover can be reduced by increasing the distance, 0.5cm, between the upper matrix layer of the wire mesh and the glass cover to 1.0 cm. This can be done by distributing and adding the four wire mesh layer of the upper matrix layer to the first and second matrix layer. More tests will be needed for the new distribution of the wire mesh to see their effect on the pressure drop and efficiency.

Efficiency versus time at various air rates are shown in figure 7. The efficiencies increase to a maximum value after 12:00 am and then start to decrease later in the afternoon. The efficiency was found to increase with increasing air mass flow rate.

The maximum efficiency obtained in this work 58% at 1:00 pm for m = 0.032 kg/s. The comparison of the thermal performance of the absorber wire mesh matrix with fins SAH with the other SAHs reported in the literature is presented in Fig. 8. It is clear that the proposed wire mesh with fins SAH has higher average efficiency at high mass flow rates. With the difference in structure of SAHs used by [4] and [17] these obtained results are very close to the present results. Whereas the work of Karem and Hawlader [4] used of V-corrugated as absorber plate, 1.8m × 0.7m, made of steel material and 10 cm height of collector, glass cover of 5mm in thickness. Omojaro and Aldabbagh [17] used seven wire mesh layers, 1.5m × 1m, as an absorber plate with longitudinal fins and 7cm height of bed. The higher present result is due to the small height channel collector used in this study. The effect of changes in channel depth on the thermal efficiency with and without the porous media was studied [17, 23, 24]. They recommended building collector with channel depth of 7cm. Generally, increasing the space between the glass cover and the bottom of the collector, absorber, reduces the average flow velocity and decreases the heat transfer coefficient.

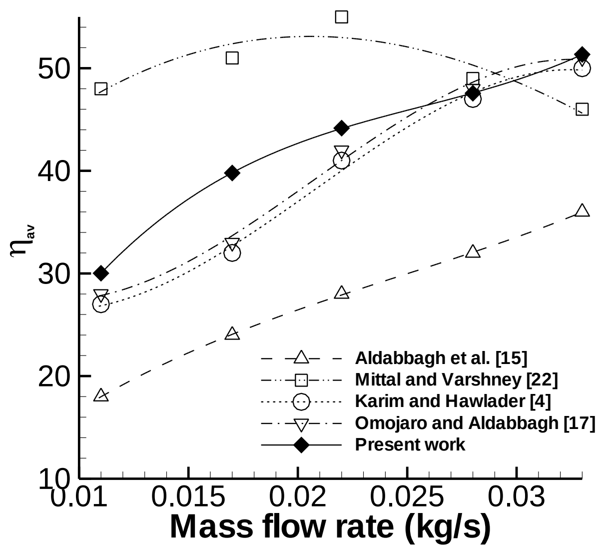

Figure 8 Comparison between the thermal efficiency with published data

CONCLUSION:

This study presents the design for a single pass solar air heater using matrix of wire meshes and fins instead of absorber plate, there is a significant increase in the thermal efficiency of the air heater. The experimental results show that the thermal efficiency increases with increasing air mass flow rate, between 0.011 kg/s and 0.032 kg/s. The temperature difference between the outlet air flow and the ambient was decreased with increasing air mass flow rate.

In addition, comparison of the results of a packed bed collector with those of a conventional collector show a substantial enhancement in the thermal efficiency as a result of using wire mesh screen layers as a packing material.

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