International Journal of Mechanical, Industrial and Aerospace Sciences
ISSN: 2517-9950
Vol:7, No:6, 2013

Double Pass Solar Air Heater with Transvers Fins and without Absorber Plate

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***Abstract*—**The counter flow solar air heaters, with four transverse fins and wire mesh layers are constructed and investigated experimentally for thermal efficiency at a geographic location of Cyprus in the city of Famagusta. The absorber plate is replaced by sixteen steel wire mesh layers, 0.18 x 0.18cm in cross section opening and a 0.02cm in diameter. The wire mesh layers arranged in three groups, first and second include 6 layers, while the third include 4 layers. All layers fixed in the duct parallel to the glazing and each group separated from the others by wood frame thickness of 0.5cm to reduce the pressure drop. The transverse fins arranged in a way to force the air to flow through the bed like eight letter path with flow depth 3cm. The proposed design has increased the heat transfer rate, but on other hand causes a high pressure drop. The obtained results show that, for air mass flow rate range between 0.011-0.036kg/s, the thermal efficiency increases with increasing the air mass flow. The maximum efficiency obtained is 65.6% for the mass flow rate of 0.036kg/s. Moreover, the temperature difference between the outlet flow and the ambient temperature, ΔT, reduces as the air mass flow rate increase. The maximum difference between the outlet and ambient temperature obtained was 43°C for double pass for minimum mass flow rate of 0.011kg/s. Comparison with a conventional solar air heater collector shows a significantly development in the thermal efficiency.

***Keywords*—**Counter flow, solar air heater (SAH), Wire mesh, Fins, Thermal efficiency.

I. 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, grain, seeds, fruit, and vegetables. Also, solar air heaters are utilized as preheaters in industries and as auxiliary heaters in building to save energy during winter-times. There are many different parameters affecting the solar air heater efficiency, e.g. collector length, collector depth, type of absorber plate, glass cover plate wind speed, inlet air, etc. The absorber plate area and heat transfer coefficient are two important parameters affecting thermal efficiency of the collector. When increase absorber plate shape area, the heat transfer rate to the flowing air increases, but on the other hand the pressure drops will increase inside the collector and increase the pumping power required. Different modifications are suggested and applied on double pass solar air heater to increase the heat transfer coefficient between the absorber plate and the air stream; these modifications are improving the efficiency of SAHs. Alvarez, Cabeza, Muniz, and Varela investigated new design by using corrugated plate as absorber plat, the experimental and numerical analysis showed improve in performance of thermal efficiency by increasing the contact area with transport air. [1]. On the other hand various techniques used by Romdhane and Ben Salma, for improving the heat transfer coefficient between the absorber plate and working fluid such as fixation of small wings at absorber plate [2]. High thermal efficiencies were obtained by Labed and Moummi for double-pass SAHs with obstacles in the air channel fixed in the channel [3]. Esen and Ozgen modeled new SAH by using least square support vector machines and superior capability for generalization, and this capability is independent on the dimensionality of the input data [4]. Also Esen, and Ozgen worked for improving SAH efficiency by inserting an absorbing plate made of aluminum cans into the double pass channel in a flat-plate SAH [5]. Utilization of packed porous materials in the second air passage double pass counter flow has been proposed by researchers as one of the effective alternative to improve its thermal performance. Many investigators such as [6] investigated porous material as an attractive choice for improving the thermal performance of conventional for double pass solar air collector. Experimentally and theoretically thermal performance of a double-pass SAH with packed bad above heater absorber plate [7], and lower channel [8], [9] were investigated. The Prashant Dhiman [10] showed that the thermal efficiency of the counter flow packed bed solar air heater was 11-17% higher than thermal efficiency of parallel flow packed bed solar air heater. Although benefit of using porous medium is to increase the absorber surface area per unit volume ratio but on other hand resulting in significant frictional losses therefore more pumping power would be needed [11]. The purpose of this work is: First, investigate experimentally the thermal performance of double pass solar air heater using sixteen steel wire mesh layers as an absorber plate and longitude fins in lower channel. The aim of using the wire mesh as porous media is to increase the absorber surface area, high porosity to reduce the pressure drop in bed. Five transverse fins were insulated and fixed within the channel to give as letter 8 path of air, this lead to increase both of the airflow path length and air velocity at same mass flow rate with uniform air flow. Second, used 3cm collector high of solar air heater result in increased the average velocity, heat transfer coefficient between the absorber area and air flow, which best thermal

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efficiency will achieve. Third, reduced of inlet air area to increase air inlet velocity in order of increased the rate of heat transfer between the absorber wire mesh and air at the first partition of collector.

TABLE I
UNITS AND ABBREVIATION

Symbol	Quantity	SI Units
A_c	Area of the collector	(m^2)
C_p	C_p specific heat of the fluid	(kJ/kg.K)
h	Fluid deflection inside the incline manometer	(m)
I	Solar radiation	(W/m^2)
m	Air mass flow rate	(kg/s)
Q	Volume flow rate	(m^3/s)
T_{in}	Inlet temperature	(°C)
T_{out}	Outlet temperature	(°C)
T_{air}	film air temperature	(°C)
ΔT	Temperature difference (T_{out}- T_{in})	(°C)
ρ	Density of air	(kg/m^3)
η	efficiency of the solar collector	(-)
ΔP	Pressure difference, $\Delta P = \rho.g.h.\sin 15$	(N/m^2)
ω	Uncertainty for the mass flow rate	(-)
Φ	Porosity	(-)

II. EXPERIMENTAL SET-UP AND EQUIPMENTS

The double pass solar air heaters, with transverse fins and wire mish used as an absorber plate, is constructed and tested for thermal efficiency at a geographic location of Cyprus in the city of Famagusta. The length and the width of the collector are 1.47m and 1.0m respectively. The distance between the first glass and second glass 4mm in thickness, is 2cm. The distance between the second glass and the bottom of the collector is 3cm Fig. 1. 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 porosity of the bed was calculated to be 0.98. The wire mesh layers arranged in three groups, first and second include 6 layers, while the third include 4 layers, were fixed in the duct parallel to the glazing. The distance between each group is 0.5cm to reduce the pressure drop. The wire mesh layers were painted with black paint before installing it into the collector. Four Aluminum fins of 80cm, 45cm length by 2.7cm height and 0.3cm thickness painted with black color and positioned transversely along the bed such that four equally spaced sections were created. A black slot rubber band, 0.5cm in height and 0.3cm thickness, was used to prevent contact between fins and glass. In this way, air flows in a path same as letter 8 gaining warmth as it goes along the path channel also avoid air pass from the upper of the fins and stopped loss of heat from fins to environment if it attached the glass.

As the air passageway was increased, the velocity of it was also increased by way of the cross-sectional area of the air passage channel decreased for the same mass flow rat. It was effective on the efficiency and the outlet temperature. 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 [12].

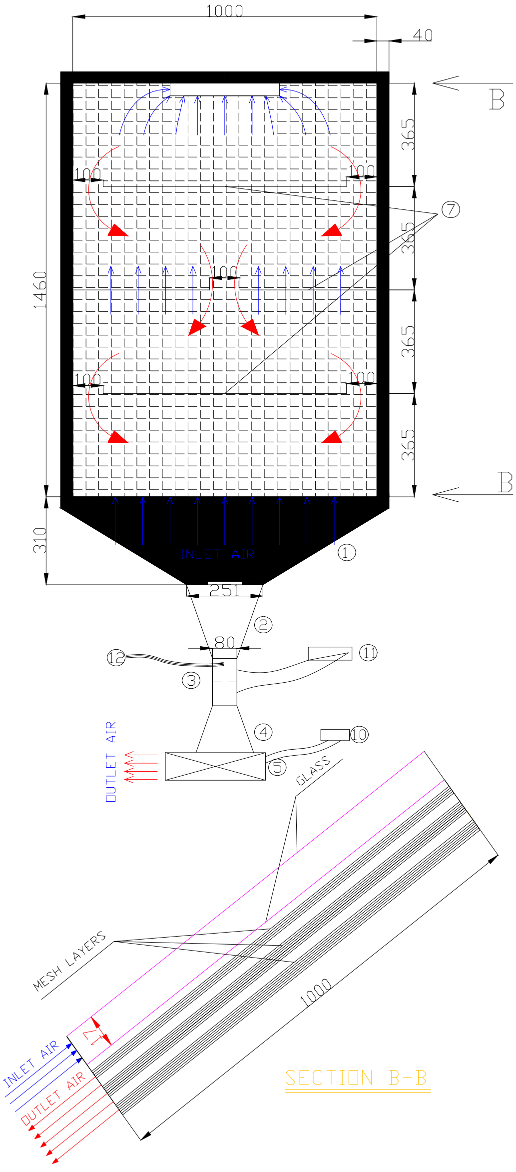

Fig. 1 (a) schematic assembly of the SAH system (b) section B- B
1- Converging section 2-Converging duct 3- Orifice meter
4- Diverging duct 5-Air blower 6-Bed thermocouples
7- Fins 8-Glass thermocouples 9- Glass
10- Speed controller 11-Incline manometer 12- Outlet air thermocouples

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 0.595cm diameter and are 2.5cm long. A radial 0.62kW 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 803kg/m3. The angle of the manometer was fixed at 15°. 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, Tin, 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, Tout, distributed in three groups. Each group contains 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°C accuracy. A calibration test showed that the accuracy of the thermocouples reading were within ±0.15°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 2800W/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° with respect to the horizontal to maximize the solar radiation incident on the glass covers. Air is circulated for 60min 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 60min time intervals. All tests began at 8:00 am and ended at 5:00 pm daily.

III. 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 [12], [14]:

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

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 (3) is a function of ΔT, m, and I, considering C_p and Ac 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, Tin, Tout, m, and η were found to be 21.8°C, 34.92°C, 56.7°C, 0.0246kg/s, 687.575W/m^2 and 48.62% respectively. The fractional uncertainty of the mass flow rate and the efficiency are found to be 0.0034 and 0.0091 respectively.

IV. Results and Discussion

This experimental work investigates the effect of partitioning single pass mesh wire packed bed SAH under Gazimagusa prevailing weather conditions during the summer months, 15.07.2011- 25.07.2011, with clear sky condition. Gazimagusa is a city in North Cyprus located on 35.125°N and 33.95°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 14.8m/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 3cm 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 0.036kg/s. The uncertainty of the mass flow rate is calculated to be 0.0084 [13].

The average hourly documented mean value of the wind speed and relative humidity ratio which were taken from the metrological office of Famagusta city was 20.27m/s and 59.08% respectively. Solar air heaters were performance in this study, and six air mass flow rates were investigated at the experiments from 0.012kg/s to 0.036kg/s. Usual hourly values of solar radiation between 8 am and 5 pm are shown in Fig. 2. The highest values of Solar radiation was 987W/m^2 at noon (at about 13:00 pm) and the minimum value was in the afternoon until sunsets as were expected. Calculating the mean solar intensity for each day, there was constancy in the solar radiation as all mean averages are within the little change and close range. The mean average solar intensity for all the days of the experiment was 689.85W/m^2.

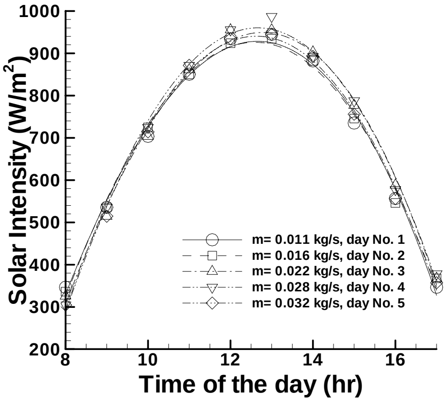

Fig. 2 Solar intensity versus different standard local time of days

Fig. 3 shows variations of the ambient temperatures were similar to inlet temperatures and between 29 to 40°C in morning, in general increasing from morning till evening with slightly reduced at 5:00 pm. In some days, the inlet temperature continues increasing from morning until 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. Fig. 4 shows results of experiments including the difference of ambient air temperatures and outlet temperature, with versus time of the day of different mass flow rates for double pass solar air heater. For all cases the ($\Delta T = T_{out} - T_{in}$) increases in the morning to a peak value at noon and starts to decrease in the afternoon, depended directly on solar radiation, as expected. For the formation of the solar air heater which studied in these experiments, ΔT be influenced by air mass flow rate, ΔT increased to maximum differences at 1:00 pm for the minimum mass flow rate 0.011kg/s, it was also found that the magnitude of the ΔT to reach 43°C at solar intensity 982W/m^2.

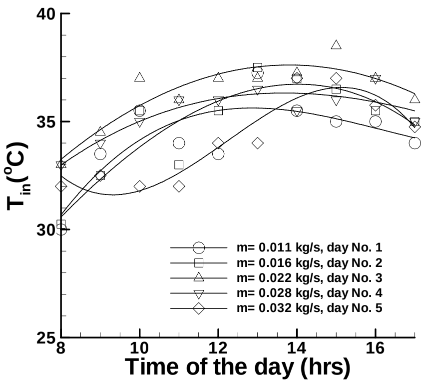

Fig. 3 Ambient temperatures versus different standard local time of days

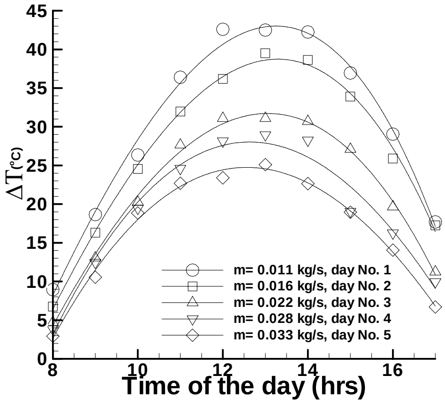

Fig. 4 Temperature difference versus standard local time of the day at different mass flow rates

The weather conditions such as solar intensity, humidity and wind speed, also it was affected on the outlet temperature and ΔT, so there were important when designing solar air heaters. From the collected works analysis, El-Sebaii [14] studied theoretically and experimentally a double pass solar air heater also double-glass with a packed bed beyond the heater absorber plate. They stated that the maximum ΔT was 38°C at solar intensity 853W/m^2 and mass flow rate of air was 0.0157kg/s. Yeh [15] investigated numerically for flat bed with internal fine attached by operating with external recycle air heater. They reported that the maximum ΔT was 20°C at solar intensity 830W/m^2 and mass flow rate of air was 0.01kg/s. Ozgen [16] experimentally investigated a flat plat solar air heater used aluminum cans as absorber into the double-pass channel. They reported that the maximum ΔT was 20°C at solar intensity 900W/m^2 and mass flow rate of air was 0.05kg/s. Figs. 5 and 6 presented 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. Where T_{bed} and T_g was the average bed and glass temperature respectively.

The maximum temperature difference of the bed was found to be 58.35°C for double flow for the air mass flow rate of 0.011kg/s at 1:00 pm. A little change between ΔTbed and ΔT double pass gives a good evident that there was a good heat transfer in the channel of the SAH from the bed to the air. On other side, the high temperature difference of the glass, ΔT_g, means that loss quantity of heat transfer from the glass to the surrounding Fig. 6.

These heat loses defiantly decreased the efficiency of the SAH. 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 1.0cm. 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 mass flow rate are presented in Fig. 7. The efficiencies were increased consequently with increased works reported by El. Sebaii [17].

Depending on the inlet and outlet air temperature, the efficiency for most of the experiments were found to be increased with the standard local time to peak value between 12:00 am and 1:00 pm, then began to decrease later on the afternoon, such that similar as previous study for Omojaro [18], but efficiency for minimum mass flow rate not decreases at afternoon, these observations were similar with El-khawajah [19] results, that mean the amount of absorber heat stored in channel of SAH and air carry more heat from the bed, this was due to the low mass flow rate with large number of wire mesh layers in the matrices such that increasing the contact surface area in the SAH. Parasad [20] discussed that the bed efficiency increased significantly when number of wire mesh layers were increased. Also, increasing the porosity of the bed leads to decrease in the pressure drop. However, their explanations were not comparable as in this study. This was due to study on flow depth of 3cm with 4 transvers fins. Also, found high pressure drop increased when increased mass flow rate to reach 314.3Pa with m= 0.036kg/s, the minimum pressure drop was 49Pa for double flow respectively for m= 0.01kg/s.

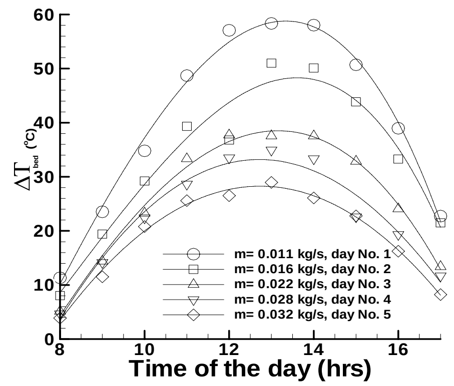

Fig. 5 Temperature bed difference versus standard local time of the day at different mass flow rates

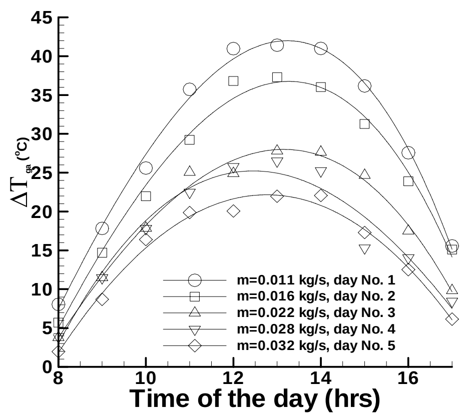

Fig. 6 Temperature glass difference versus standard local time of the day at different mass flow

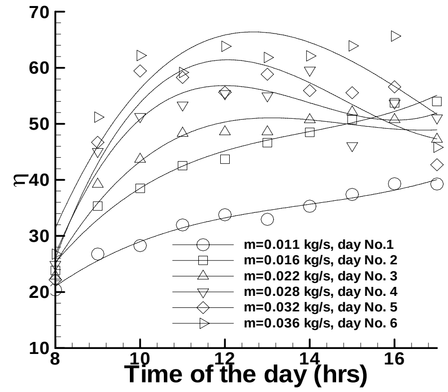

Fig. 7 Variation of collector efficiency at different mass flow rates

This agreement with Bashria [21] results that the pressure drop increased with the decreased the flow depth and this raised more when channel flow was longer. Also found that the pressure drop was purpose of mass flow rate, it was increased by increasing the mass flow rate. El-khawajah [19] reported in his design that the maximum pressure drop was 68.5Pa for maximum air flow rate (m = 0.042kg/s), 7cm flow depth with 4 transversal longitudinal fins and 12 wire mesh layers. Ben Slama [2] the pressure drop of his design for partially transversal baffles which are not touching the absorber was 130.8Pa for mass flow rate 0.031kg/s.

The comparison of the thermal performance of the absorber wire mesh matrix with fins SAH with other SAHs reported in the literature for double pass SAH are presented in Fig. 8. Omojaro [18] used seven wire mesh layers, 1.5m×1m, as an absorber plate with longitudinal fins and 7cm bed width. El-khawajah [19] used twelve wire mesh layers, 1.5m×1m, as an absorber plate with four transverse longitudinal fins, 7cm air flow depth. Sopian [22] presented double-pass solar collector with saturated porous media, efficiency between 32% and 67% for mass flow rate different from 0.03 kg/s to 0.07kg/s.

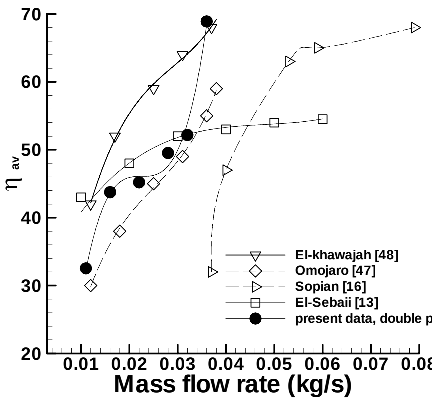

Fig. 8 Efficiency comparison between the double pass SAH with some double pass SAHs in literature

The upper channel is various from 3.5cm to 10.5cm and the lower channel is varied from 7cm to 14cm. El-Sebaii [17] shows experimentally double pass-copper finned plate solar air heater, bed dimensions $1\times1m^2$ area and 0.1m air flow depth.

V. Conclusion

An experimental simulation for four Aluminum fins and 16 wire mish as absorber in double pass air flow with air flow depth of 3cm presented the results of many parameters on system thermal performance and bed pressure drop. It was found that increasing the air mass flow rate through channel results a higher thermal efficiency, on the other hand it was increased a bed pressure drop. The low flow depth results in increasing the efficiency and temperature difference between the outlet air and the inlet air, but at the same time it increased the bed pressure drop, using many layers as porous media increased the system efficiency and an exit air temperature but also increased pressure drop of SAH. This increment of the pressure drop results increasing the pumping power of the collector.

Finally the thermal efficiency and temperature difference ΔT was associated with several of the reported. It was found that in the proposed simulation of the system thermal efficiency was higher related to the other simulations.

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