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Experimental and Theoretical Study for Performance Enhancement of Air Solar Collectors by Using Different Absorbers

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Abstract

An experimental and theoretical study has been done to investigate the thermal performance of different types of air solar collectors, In this work air solar collector with a dimensions of (120 cm x90 cm x12 cm) , was tested under climate condition of Baghdad city with a (43° tilt angel) by using the absorber plate (1.45 mm thickness, 115 cm height x 84 cm width), which was manufactured from iron painted with a black matt.

The experimental test deals with five types of absorber:-

Conventional smooth flat plate absorber , Finned absorber , Corrugated absorber plate, Iron wire mesh on absorber And matrix of porous media on absorber .

The hourly and average efficiency of the collectors were investigated for three values of mass flow rates (0.016 kg/s to 0.027 kg/s) for each type of collector and then the porosity for the last collector type was tested by changing the porosity of porous media.

A typical air solar collector has been studied Theoretically to build a standard software for testing any type of air solar collectors with local weather data .

From the experimental study it can be seen by using some obstacle material to the air flow (fins, corrugated absorber plate, iron wire mesh porous media on the absorber) could be enhanced the efficiencies not less than 4 % for finned type and 8 % for corrugated and 25 % for mesh and 30 % for porous media comparing with flat plate (smooth) collector .

Theoretically, the results showed that the collector with high convention heat transfer coefficient porous media has high hourly efficiency about (η = 56 %) and iron wire mesh on absorber (η = 52 %), on the other side the minimum performance occurred in the flat plate absorber (η = 28 %).

Comparison of results reveals that the theoretical predictions agree reasonably well with experimental results. And the difference between the theoretical and experimental efficiency in general was between (1─ 15 %).

Keywords: *air solar, collector performance, pours media.*

1. Introduction

Solar air heaters are very simple and cheap, therefore is most widely used as a collection devices of solar energy. The thermal efficiency of solar air heater has been found to be very poor because of their inherently low heat transfer capability between the absorber plate and the air flowing through the duct. Several designs of solar air heaters have been developed over the years in order to improve their performance. Providing artificial roughness on the absorber plate is an effective technique to enhance the heat transfer coefficient.[1]

Amer et al. (2010) , [2] induced a study for the effect of absorber surface shape of solar air collector (Flat V-Corrugated , Sinusoidal Wave-Corrugated,Rectangular-Corrugated) (1×1×0.1m) The results show that the best heat transfer coefficient enhancement is (63%) for (V Corrugated plate) compared with the flat plate.

and the maximum drag friction coefficient is occurred at (V-Corrugated) at (Re=8000).

Bhandari et. al. (2012), [3], made a comparative study of performance of different types of flat plate solar air heaters. The results showed that the analysis of various types of solar air heaters for the same mass flow rate in a double pass finned solar air heater has the highest efficiency. And, it was concluded that the values of mean absorber plate temperature (T_{pm}) and fluid outlet temperature decrease with the increase in mass flow rate.

Abdullah et. al. (2013), [4], showed in this work numerical solutions of the heat balance equations for single glazed flat plate collectors using computer program. The study shows that the empirical relations of (U_T) predict the values close to numerical solutions only for certain assumed conditions and cause large errors in the calculation of top heat loss factor and useful energy for other range of variables.

Chabane et. al. (2013), [5], Reported that heat transfer of a solar air heater duct can be increased by providing artificial roughness on the heated wall (i.e. the absorber plate). The results show that the maximum efficiency values obtained for the 0.016 kg/s with and without using fins were 51.50 % and 43.94% respectively.

Peng et. al. (2010), [6], theoretically and experimentally analyzed the performance of a solar air heater with fins attached to the absorber plate. The aim is to enhance the thermal performance of the solar air heater with a cheap and accessible collector design that has a long durability. For this purpose, an experimental setup was installed . In this study. 25 pin-finned collectors and a flat plate clear collector were discussed. It was concluded that with 19 m^3/h air flow rate, the pin-fin arrangement in a solar air collector could increase the heat transfer coefficient up to three times when compared with flat-plate collector.

Ahmed M. et. al. [7], (2007) studied the effect of ambient and inlet temperature to the solar air heater on the performance of solar air heater equipped with porous material. The analysis and design of the solar air heater to be operated in cold climatic regions. It was shown that a collector thermal and exergetic efficiency of about 85 % and 39 %, respectively, could be obtained for ambient temperature of 20º C , while the thermal efficiency will be around 55 % and the exergetic efficiency of 40 % for ambient temperature of -10º C.

The aim of present study is to review the design and to analyze the heat transfer of air solar collector. Experimental-theoretical study presents a comparison of a solar collector with smooth absorber plate , corrugated absorber plate and with using fins attached on the absorber plate and with using wire mesh & porous media to enhance the heat transfer coefficient.

2. Experimental Setup

The solar air collector facing south was mounted in Baghdad (33.3 Latitude) as in the Fig. 1 , with tilt angle 43 degree from horizontal ,The length of the collector equals to 1200 mm, width is 900 mm and the depth is 120 mm. (Fig. 2) .

An insulation layer (fiber glass) of 50 mm thickness is mounted on the back side of the collector, whereas sides are insulated with 30 mm of fiber glass. and outlet is placed as a one hole round duct 100 mm dia. in upper part in which the blower is directly connected.

2.1. Housing and Absorber

The collector housing is the container that provides structural integrity for the collector assembly.

Absorber plate painted with black matt (Flat Black paint, (α = 0.92 to 0.98,) [8] .It's been used single-glazed 4 mm thickness, (120 cm length X 89 cm width).

The insulation has been used is fiberglass with conductivity (ki= 0.037 W/m.C°[9]). Back-insulation thickness is 50mm, and Edge insulation thickness is 30mm.

Centrifugal fan blower, radial type, has been used , A nominal flow rate of fan 0.0255 m3/s are connected to the collector. This fan it has special vanes to regulate the mass flow rate.

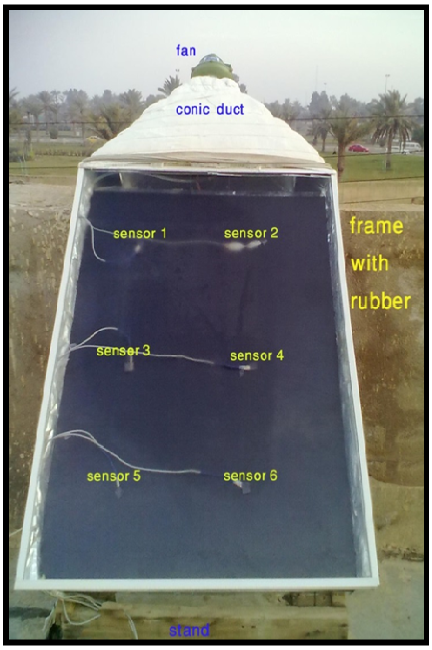

Fig. 1. The collector air solar.

2.2 Measurement Apparatus

- solar radiation meter.
- Dual function anemometer & intelligent thermometer.
- Temperature data logger(LabJack with Thermistors).
- Thermistors silicon type (LM35) .

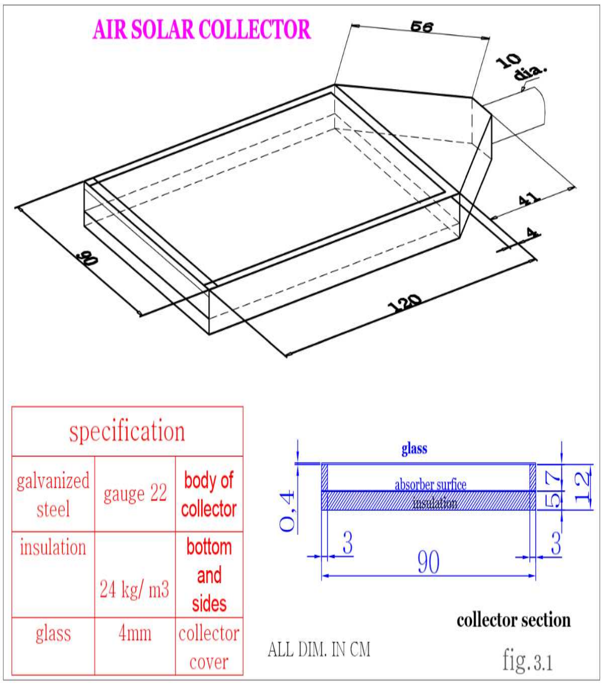

Fig. 2. Schematic air solar collector.

3. Procedures of Experiment

Below mentioned the measurements procedures to have to be performed to collect data of testing experiment :

1. Temperature measurements
2. Air flow measurements
3. Solar radiation measurements

Solar radiation and temperatures have been recorded from sunrise to sun set every 5 minutes for five types of collector absorber surfaces, as in Fig. 3, which they are:

1. Smooth flat surface
2. Fined surface
3. Corrugated surface
4. Wire mesh on surface
5. Porous media on surface

and using three different mass flow rates on every type of collector by changing fan inlet suction vanes to achieve the specific mass flow rate as shown in the following Table 1 .

Table 1,
Readings of five types of collector surfaces.

Type	date	Flow (kg/s)	matrix weight
flat	27/1/2014	0.027	
	13/2/2014	0.022	
	12/2/2014	0.019	
Corrugated	30/1/2014	0.027	
	22/2/2014	0.022	
	21/2/2014	0.019	
Fined	31/1/2014	0.027	
	10/2/2014	0.022	
	11/2/2014	0.019	
Wire mesh	25/1/2014	0.022	(830 g)
	19/2/2014	0.019	
	20/2/2014	0.017	
Porous media	18/2/2014	0.022	(585 g)
	14/2/2014	0.027	(885 g)
	17/2/2014	0.016	(885 g)

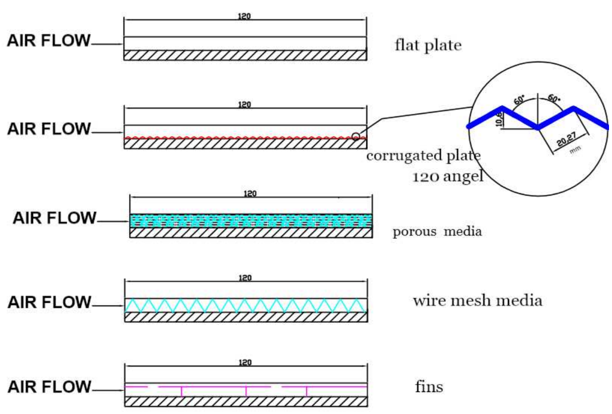

Fig. 3. schematic views of the five types of plate surfaces.

4. Experimental calculations

4.1. Air mass flow rate (m_a)

$$m_a = \rho_a \cdot V_a \cdot A_{duct}$$

4.2. Useful energy (Q_u)

4.3. $$Q_u = m_a \cdot C_p \cdot (T_{out} - T_{in})$$

collector efficiency (η)

$$\eta = \frac{Q_u}{A_c \cdot G_T}$$

Hourly solar radiation, useful energy

$$\eta = \frac{\int Q_U\, dt}{A_C \int G_T dt}$$

5. Theoretical Analysis

The present work involves a comparative study of performance analysis of different types of flat plate air solar collectors. A Fortran program code is built to carry out the whole analysis. The effect of mass flow rates, inlet temperature and intensity of solar radiation on the performance of solar air heaters are also investigated in the present study**,** see Appendix .

6. Flow Chart of Computer Program

Many relations have been obtained in the previous section to analyze the operation of air solar collector. Fortran computer program has been classified into three parts which are the main program, radiation subroutine and collector subroutine to obtain the theoretical performance of air solar collector , Fig. 4**,** shows Flow chart of the mathematical model.

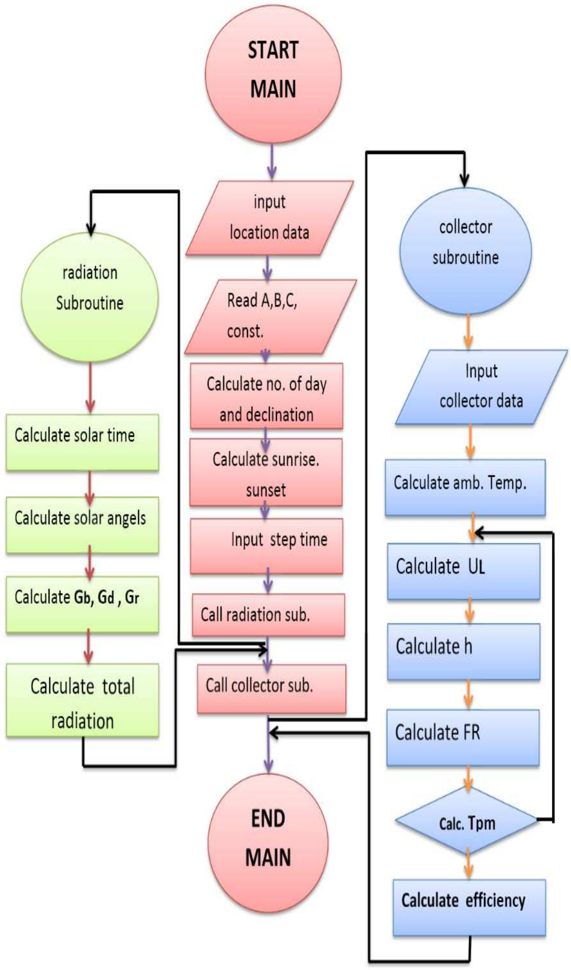

Fig. 4. flow chart of computer program.

7. Results and Discussion

7.1. Flat Smooth Absorber Plate

The Fig. 5, shows the relation of (Tai ,Tao, Tpm) inlet air ,outlet air and plate temperatures respectively with time, It can be seen that the maximum value reached at midday (Tai =30 C° ,Tao =50 C°, Tpm =80 C°) that because being the incident radiation maximum at the same time.

The Hourly solar radiation value I, and Hourly useful energy value Qu, along the day for flow rate 0.019 kg/s shown in Fig. 6 , both starting with the minimum value (I= 1.7 MJ/m2 & Qu= 0.2MJ/m2) and reaching maximum value (I= 4.2 MJ/m2 & Qu= 1.3 MJ/m2) at midday.

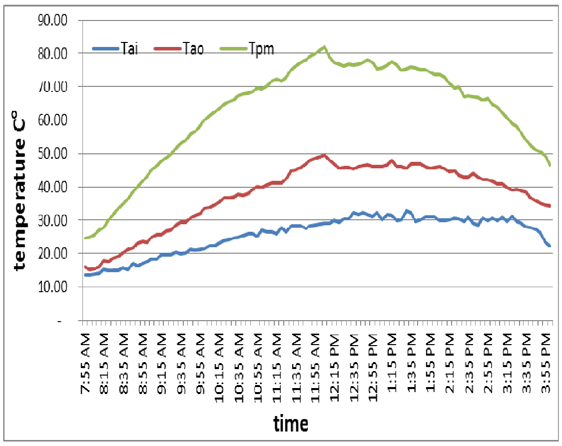

Fig. 5. The inlet air ,outlet air and plate temperatures with time of day at 0.022 kg/s flow rate.

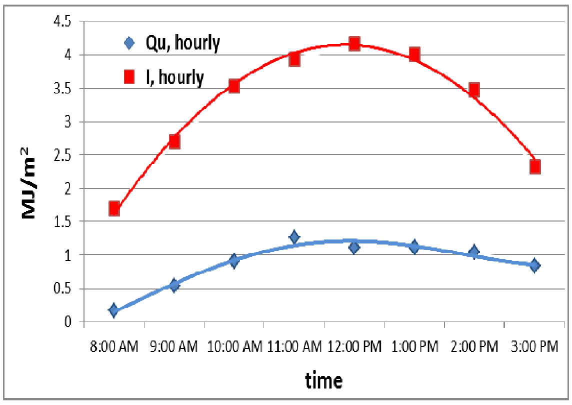

Fig. 6. Hourly useful energy and hourly solar radiation with time of day at 0.019 kg/s flow rate.

7.2. The Efficiency Curves of Five Types

The performance curves of experimental study for five types of air solar collectors tested in this study which are flat plate, finned, corrugated, mesh and porous types are shown in Fig. 6 and their values given Table 2.

Table 2,
Efficiency of five types

Absorber type	Average Efficiency
flat	26%
finned	30%
corrugated	34%
mesh	51%
porous	56%

From Fig. 7 can be seen that the collector Efficiency of finned plate is higher than that of flat plat, due to the fact that The Fins increase the surface area and heat transfer coefficient, and extension of the air flow path by creating new airflow passages between the vanes. It was also found that the curves tends to decrease slightly this attributed to the combined effects of the solar radiation and the top losses.

And from the comparison between flat plate and corrugated plate showed that the efficiency of corrugated was higher than that of flat plate, because the air flowing across a corrugated absorber plate creates turbulence along the plate, which increases the convective heat transfer coefficient , and also it can be seen that the Efficiency has a tendency to decrease , same reason of finned plate case .

It is further observed that the collector efficiency of mesh type was higher than the efficiency of the conventional air heater flat plate, this is a reasonable observation for the enhancement of the heat transfer. It is also observed that the efficiency curves are almost constant which is attributed to the fact that, solar radiations are progressively absorbed and stored by the layers of wire mesh and the remaining radiations are absorbed by the absorber plate.

If the collector performance of porous and flat plate type are compared, it will be found that the efficiency of porous is clearly higher than that of flat plate ,the reason that not only the absorber plate but the matrix material also works as the absorber of solar radiation, Therefore, the heat energy due to absorption of solar radiation is distributed throughout the packed material and the absorber plate and dissipated now more effectively to the flowing fluid due to very large surface area being in contact with the flowing fluid in a packed collector as compared to a plane collector where in only the absorber plate surface is in contact with the flowing fluid.

Another reason that the temperature difference between the porous material and air is less as compared to the temperature difference between the absorber plate and air in case of collector without matrix as well as for the conventional collectors because the matrix is shading the solar radiation.

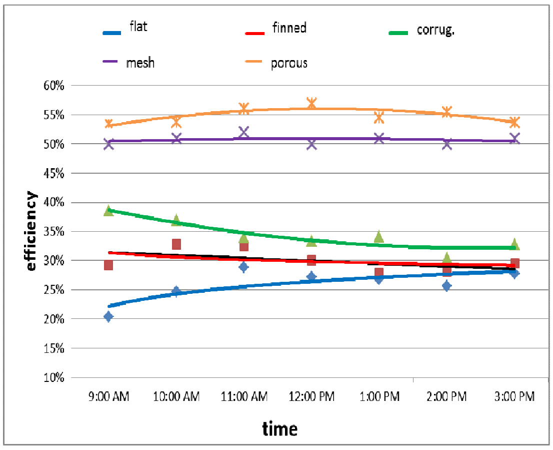

Fig. 7. The performance curves (efficiency) of five types air solar collector with time of day for 0.022 kg/s mass flow rate.

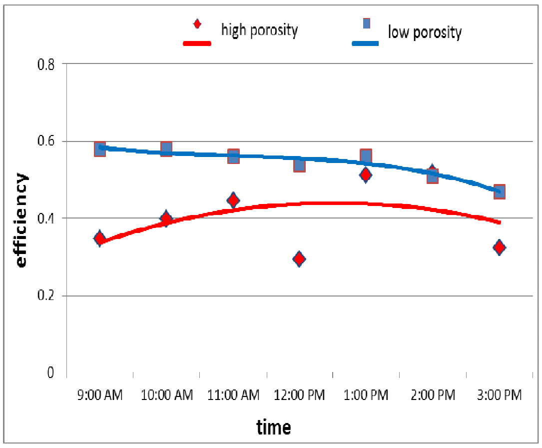

Fig. 8. Efficiency with time of day for different values of porosity for porous media absorber.

7.3. Effect of Porous Material

It is reveal that the thermal efficiency of packed collector (porous +wire mesh) increases as porosity decreases. This may be due to the decrease in porosity increases the effective heat transfer area. Therefore, the heat transfer coefficient between the matrix and flowing air increases resulting in higher thermal efficiency. Also, decrease in porosity reduces the flow channeling, i.e. increase in turbulent of the air flowing through the packed passage resulting in improved thermal performance. It is important to explore the influence of porosity on the thermal performance of air solar collector as shown in Fig. 8.

There is no doubt that the working with the optimum porosity is also required

.

7.4. Theoretical Study

The performance curves of five types of air solar collectors models tested in this study. the theoretical efficiencies can be seen in Fig. 9.

It's clear that the efficiencies were slightly degreased at midday. This is a reasonable observation due to the fact that the increase of solar intensities will increase the mean plate temperature as shown in Fig. 10, and then increase the losses. This has a significant influence on performance efficiency of the air solar collector .

A comparison can be made between the results of Fig. 7 & Fig. 9, to verify the degree of theoretical predictions agreement with the experimental results , As it can be seen that the difference between the theoretical and experimental efficiency was between (1— 15 %) , so its acceptable result with a good agreement.

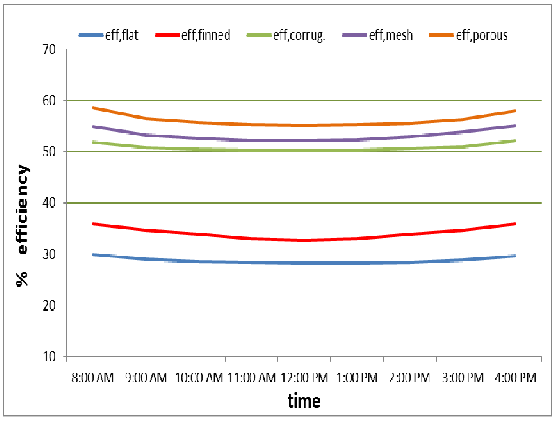

Fig. 9. The performance curves (efficiency) of five types air solar collector with time of day for 0.022 kg/s mass flow rate.

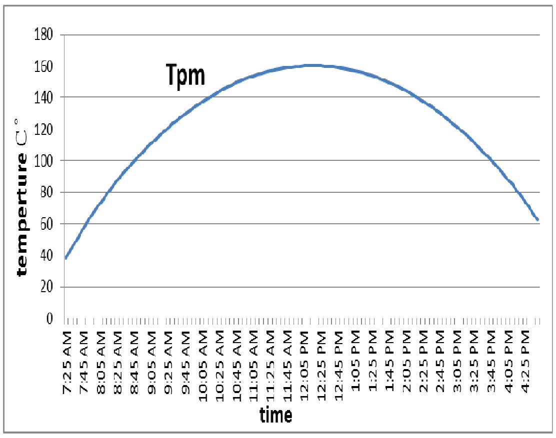

Fig. 10. The mean plate temperature with time of day for rate for Flat plate absorber.

8. Conclusions

The conclusions can be drawn from the experimental and analytical studies as the following :

- The hourly solar radiation, useful energy exhibit parallel changes with the incident radiation .i.e. they depend directly on the amount of the solar radiation for any air flow rate .
- The efficiency of the collector improves with increasing mass flow rates for every types of absorbers due to an enhanced heat transfer to the air flow.
- The heat transfer and thermal performance of solar air heater with artificial roughness(finned ,corrugated) gave higher values than that of non-artificial roughness solar collector ,Hence artificial roughness, are recommended.
- Solar air collector with(porous, mesh) material in the air pass gives maximum thermal efficiency among the solar collector types .
- It is reveals that the theoretical predictions agree reasonably well with experimental results. The differences between theoretical and experimental efficiencies were a non-significant.
- A standard Fortran software has been performed to test any type of air solar collector in local weather data before starting to manufacturing or using .

Appendix:

Solar radiation calculation , [10]

a. Latitude angle (ϕ)

It is the angular location north or south of the equator, for Baghdad city equal 33.3°.

b. Declination angle (δ)

$$\delta = 23.45 \times \sin\left[\frac{360}{365}(284 + n)\right]$$

c. Surface azimuth angle (γ)

The deviation of the projection on a horizontal plane of the normal to the surface from the local meridian ,zero due south.

d. Surface tilt angle (β)

The angle between the collector plane and the horizontal ($0^o \geq \beta \geq 180^o$).

e. Hour angle (ω)

The angular displacement of the sun east or west of the local meridian due to rotation of the earth on its axis at 15o per hour, morning negative and afternoon positive.

f. sunset hour angle(ωs) and No. of daylight hours

The sunrise hour angle is the negative of the sunset hour angle, and determined using Equation:

$$\omega_s = \cos^{-1}.(-\tan\phi . \tan\delta)$$

It also follows that the number of daylight hours is given by :

$$\text{No. of daylight} = \frac{2}{15}\cos^{-1}.(-\tan\phi . \tan\delta)$$

g. Solar Time

$$\text{Solar time} - \text{standard time} = 4(L_{st} - L_{loc}) + E$$

Where

$$E = 229.2(0.000075 + 0.001868 \cos B' - 0.032077 \sin B' - 0.014615 \cos 2B' - 0.04089 \sin 2B')$$

Where : $B' = (n - 1)\frac{360}{365}$

h. Solar altitude angle (α_s)

$$\sin\alpha_s = \cos\delta . \cos\phi . \cos\omega + \sin\phi . \sin\delta$$

i. Incidence angle (θ)

$$\cos\theta = \cos\theta z \cos\beta + \sin\theta z \sin\beta \cos(\gamma s - \gamma)$$

j. Zenith angle (θ_z) ,

$$\cos\theta_z = \cos\delta . \cos\phi . \cos\omega + \sin\phi . \sin\delta$$

k. Solar azimuth angle (γ_s)

$$\cos\gamma s = (\sin\alpha_s . \sin\phi - \sin\delta)/\cos\phi . \cos\alpha_s$$

Total Solar Irradiation,[1]

$$G_T = G_b + G_d + G_r$$

Beam radiation calculation

$$G_b = IDN.\cos\theta$$

$$IDN = A . \exp\left(\frac{-B}{\sin\alpha_s}\right)$$

Diffuse radiation calculation

$$G_d = C \ IDN \ Fss$$

$$Fss = (1 + \cos\beta)/2$$

Ground reflected radiation calculation

$G_r = (G_b + G_d)\ \rho g.\ Fsg$

$Fsg = (1 - \cos\beta)/2$

Collector overall heat losses

$$U_L = U_T + U_B + U_S$$

$$U_T = \left[\frac{N}{\left(\frac{C}{T_p}\right)\left[\frac{T_p - T_\infty}{(N+f)}\right]^e} + \frac{1}{h_\infty}\right]^{-1} + \frac{\sigma.(T_p + T_\infty).\ (T_p^2 + T_\infty^2)}{[\varepsilon_p + 0.00591.N.h_\infty]^{-1} + \left[\frac{(2N + f - 1 - 0.133\varepsilon_p)}{\varepsilon_g}\right] - N}$$

$$U_B = \frac{K_i}{X_i}$$

$$U_s = \frac{(L+W).H.K_i}{L \times W \times X_{si}}$$

$$C = 520.(1 - 0.000051.\beta^2)$$

$$f = (1 + 0.089.h_\infty - 0.1166.h_\infty.\varepsilon_p).(1 + 0.07866\ N)$$

$$e = 0.43.(1 - \frac{100}{T_p})$$

$$h_\infty = 2.8 + 3.0V \ldots$$

$$Q_u = A_p.[G_T.(\tau\alpha) - U_L(T_p - T_\infty)]$$

Where:

collector efficiency factor, (F'),[11]

$$F' = \left[\frac{1}{1 + \frac{U_L}{h}}\right]$$

$$h_a = \frac{N_{u_a}.k_a}{D_h}$$

$$R_{e_a} = \frac{m_a D_h}{\mu_a.Af}$$

Dh: hydraulic diameter $= \frac{2W\times t}{(W+t)}$

Calculation heat transfer coefficient

for flat plate, [10]

$$N_{u_a} = 0.0158\ R_{e_a}^{0.8}.$$

For fined plate ,[12].

$$N_{u_a} = 0.0333\ R_{e_a}^{0.8}.\ Pr^{(1/4)}$$

For corrugated plate , [13]

$$h = 303.2\ m_a^{0.632} / ARc^{0.986}$$

$$ARc = \frac{\text{duct corrugated duct height (H)}}{\text{corrugation cycle length (Lc)}}$$

For porous and mesh.[14]

$$h = 0.45\left(\frac{ka}{D_h}\right).\ R_{e_a}^{0.62}.p^{-4.56}.\left(\frac{pt}{dw}\right)^{-0.21}.Pr^{1/3}$$

$$p = \frac{Vt - Vs}{Vt}$$

Heat removal factor ,FR,[10].

$$FR = \frac{m_a.C_{p_a}}{U_L.A_a}\left(1 - \exp\left[-\frac{F'.U_L.A_a}{m_a.C_{p_a}}\right]\right)$$

The useful energy,[10]

fully fresh, outside air)

$$Q_u = FR.A_p.G_T(\tau\alpha)$$

Mean plate temperature calculation, [10]

$$T_{pm} = T_{ai} + \frac{Q_u}{A_p.FR.U_L}(1 - FR)$$

collector efficiency

$$\eta_a = \frac{Q_u}{A_p.I_T}$$

Nomenclature

English

A apparent solar radiation at air mass = 0
Ap apparent collector area
Arc aspect ratio for solar collector channel , height to width
Af Air flow section area
B The atmospheric extinction coefficient.
P Porosity of matrix media
B′ Parameter of the time equation
pt pitch of wire mesh or (pours media)
C Dimensionless value represent to the average ratio of diffuse to normal beam radiation.
D_w wire diameter
Cp_a Specific heat capacity
Dh Hydraulic diameter
E The equation of time
F` collector efficiency factor
FR heat removal factor
Fsg angle factor between the surface and the earth
Fss angle factor between the surface and the sky
GT Total incident solar radiation
Gb Beam solar radiation
Gd Diffuse solar radiation
Gr Ground reflected solar radiation

H Collector thickness
h heat transfer coefficient
h_∞ Convective heat transfer coefficient for air wind flowing over the surfaces of the collector
h_1 Convection heat transfer coefficient between glass cover and air flow in the channel
h_2 Convection heat transfer coefficient between absorber plate and air flow in the channel
hr Radiation heat transfer coefficient between absorber plate and glass cover
I Hourly solar radiation
IDN the direct normal solar radiation
ItH total horizontal irradiation
IT total Hourly irradiation
K Thermal conductivity
Ki Back insulation thermal conductivity
Ks Side insulation thermal conductivity
L Collector length
Lst the standard meridian for the local time zone
Lloc the longitude of the given location
m_a Mass flow rate
N Number of covers
n Number of day in year
Nu_a Nusselt's number for air
NTU rate number of transfer units
Pr Prandtl number
Qu Useful energy
Re_a Reynolds number for air
S solar radiation absorbed by a collector
T Temperature
t Air channel thickness
Tc cover temperature
Tg Glass cover temperature
T_∞ Ambient air temperature
Tai Air flow inlet temperature
Tao Air flow outlet temperature
Tp plate temperature
Tpm Average plate temperature
U_T Top loss coefficient
U_B Back loss coefficient
U_f Heat removal loss coefficient of fluid
U_L thermal resistance between absorber plate and ambient
$U_\circ$ thermal resistance between fluid and ambient
U_S Side loss coefficient
V Velocity of ambient wind
Va Average air velocity
Vt total volume
V_S solid volume
W Collector width
Xi Back insulation thickness
Xsi Side insulation thickness

Greek symbol

α Absorptivity
αs Solar altitude angle
β Surface tilt angle
γ Surface azimuth angle
γ_s Solar azimuth angle
δ Declination angle
εp Absorber plate emissivity
εg Glass cover emissivity
τg Glass cover transmission
η Collector efficiency
θ Incidence angle
θz Solar zenith angle
μa Viscosity
ρ reflectance
ρa Air Density
ρg Ground reflectivity
σ Stefan-Boltzman constant
ϕ Latitude angle
τα Transmisitance absorptiance
ω Hour angle
ω_s Sunset hour angle

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دراسة عملية و نظرية لتحسين أداء المجمعات الشمسية الهوائية باستخدام سطوح امتصاص مختلفة

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الخلاصة

تم القيام بدراسة عملية ونظرية لبحث الأداء الحراري لأنواع مختلفة من المجمعات الشمسية الهوائية، في هذا العمل ابعاد المجمع الشمسي الهوائي (120 cm x90 cm x12 cm) تم اختباره تحت الظروف المناخية لمدينة بغداد بزاوية ميل للمجمع، (43°) باستخدام لوح امتصاص (1.45 mm thickness) ،(115 cm height × 84 cm width) الذي تم تصنيعه من الحديد مطلي بطلاء اسود مطفي.

تم التعامل في الاختبار العملي مع خمسة أنواع من سطوح الامتصاص : -

السطح المستوي الاملس التقليدي (flat plat)، زعانف على سطح الامتصاص (fins)، سطح الامتصاص المتعرج (corrugated)، مشبك معدني من الحديد (wire mesh) ، وحشوه وسيط مسامي على لوح الامتصاص (porous).

ان الكفاءة الساعية ومعد ل الكفاءة تم تحقيقها على ثلاث قيم من معدل التدفق (0.016 kg/s to 0.027 kg/s) لكل نوع من المجمعات الشمسية وأيضا تم اختبار المسامية للنوع الاخير عن طريق تغيير قيمه المسامية لحشوه الوسيط المسامي.

تم دراسة نظريا أنموذج للمجمع الشمسي الهوائي وذلك لأنشاء برنامج قياسي لاختبار اداء أي نوع من المجمعات الشمسية على بياناتنا الجوية المحلية.

من الدراسة العملية يمكن ان نلاحظ بواسطة استعمال بعض الاعاقات لمجرى الهواء (fins, corrugated absorber plate, Iron wire mesh, pours media) بانه قد امكن من تحسين الكفاءات ليس باقل من (4 % for finned type) و(8 % for corrugated) و(25 % for wire mesh) و(30 % for porous media) وذلك بالمقارنة مع المجمع الشمسي المستوي التقليدي .

نظريا ، أظهرت النتائج بان المجمع الشمسي ذو المعامل العالي لانتقال الحرارة هو الاعلى كفاءه وهو (pours media η = 56 %) ومن ثم يأتي (Iron wire mesh η= 52%) في حين ان الاداء الاقل يحصل عند (flat plate η = 28 %.).

مقارنة النتائج تظهر أن التوقعات النظرية تتفق بشكل منطقي مع النتائج التجريبية . وكانت عموماً الفروقات بين الكفاءة النظرية والعملية ما بين (1─ 15 %) .