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SIMULATION OF A V–CORRUGATED SOLAR AIR HEATER

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Abstract

The performance characteristics of a V–corrugated plate type solar air heater was investigated analytically using computer simulation. A Visual Basic computer program was built to predict the hourly instantaneous collection efficiency, useful energy gained and the outlet air temperature of the solar air heater. The effects of various design and operating conditions, such as mass flux, wind speed, absorber plate emissivity, and collector tilt angle and inlet air temperature, on the predicted values were examined. Results of the simulation of the solar air heater indicated that increasing mass flux of the air through the collector increases the instantaneous efficiency and the useful energy gained but reduces the air outlet temperature rapidly. An air outlet temperature of about 69°C was obtained in winter at midday at a mass flux of .01 kg/s.m2 with an average collection efficiency of 34%. Also, it was found that absorber plate emissivity, and (inlet–ambient) temperature difference affect the solar air heater performance in varying degrees. An optimum tilt angle in Jalu for the solar air heater during the period from May to August and from November To February was found to be equal to 5°, 50 from the horizon, respectively. Also, changing the tilt angle by ±10° around its optimum value reduces the useful energy gained by only 2% during the same period in summer and winter. Experimental data of previous work were compared with the simulation results and found in very good agreement.

Keywords: Solar Energy, Air Heater, Computer Simulation

1. Introduction

Of the various applications of solar air heaters, is the heating of buildings and drying of agricultural products are considered the most successful. Solar air heaters may be classified into two main categories. The first type has a non–porous absorber in which the air stream does not flow through the absorber plate. Air may flow above and/or behind the absorber plate. The second type has a porous absorber where air may flow through the absorber matrix. Hollands [1] suggested that for the first type of solar air heaters, the flow passage should be below the absorber plate to minimize heat losses.

Many attempts had been made for the development of mathematical models to study the performance of these systems with respect to different climatic changes and various performance characteristics. Dhiman and Tiwari [2] presented an analytical model to study the performance of a two channel flat plate air heater.Diab et al.[3] made a detailed comparison study between the predictions of the one–node approximation and the six–nodes approximation for three different flow arrangement solar air heaters over a range of flow rates. Biondi et al. [4] analyzed the technical performance of seven solar air heaters having simple flat plate absorbers.

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The major limiting factor in solar air heater performance is the resistance to heat transfer between the air being heated and the collector absorbing plate. One of the important ways to augment convective heat transfer rate in channel flows is to alter the flow geometry by using fins or corrugated surfaces. Indrajit et al.[5] experimentally investigated the performance of two non–porous absorber solar air heaters, with and without fins. Their resultes indicated that the finned air heater was more efficient than the air heater without fins only at smaller flow rates.

Theoretical and experimental studies on the corrugated sheet type solar air heaters were first conducted by Buelow and Boyd [6]. The idea of a V–groove in the absorber plate for increasing the solar absorptance and turbulence in the air was given by Hollands [7]. An angle of 55–60° had been recomended by Sayigh [8] for such corrugations. He indicated that the heat transfer area is nearly doubled with such angles and losses due to outgoing long wave radiation are kept to a minimum. Hollands and Shewen [9] studied the optimization of flow passage geometry of solar air heaters. They concluded that the overall heat transfer coefficient between the V–corrugated absorber and the flowing air was from 47% to 300% higher than that for a flat plate absorber depending on whether the flow was laminar or turbulent, and on whether the V–corrugated plate was thermally bonded to the back plate. Joudi and Mohammed [10] investigated experimentally the performance of a V–corrugated plate type solar air heater. They provide systematic information on this type of solar air heaters under local weather conditions of Baghdad, Iraq.

The present work is a computer simulation study of the effect of various collector operating parameters on the air outlet temperature, useful energy gained and the instantaneous collection efficiency of the solar air heater for the city of Jalu, Libya. The validity of the computer simulation predictions was examined by comparing the simulated results with the experimental results of Ref.[10] using the same weather conditions and geographic data.

2. Mathematical Modeling

The solar air heater simulated in the present work is the same as that examined experimentally by Ref. [10]. It was a V–corrugated absorber plate type solar air heater with an angle of V–opening of 60 °. It uses single glass cover with air flowing through triangular passages made by the corrugated absorber plate and a flat rear plate. The design parameters of the solar air heater are given in Table (1), and Fig. (1) shows a schematic diagram of the solar air heater.

Table (1) : Design parameters of the simulated solar air heater : from Ref. [10]

Absorber plate area	**=2.7 m^2**
Length of collector	**= 2.8 m**
Width of collector	**= 1.2 m**

Thickness of collector	= 0.25 m
No. of triangular air ducts	= 13
Length of corrugation sides	= 0.075 m
Angle of corrugation	= 60°
Mean absorber - cover spacing	= 0.07 m
Number of glass cover	= 1
Glass cover thickness	= 0.004 m
Glass cover emissivity	= 0.88
Glass extinction coefficient	= 20 m^{-1}
Glass index of refraction	= 1.526
Absorber plate emissivity	= 0.95
Rear plate emissivity	= 0.9
Thickness of back insulation	= 0.13 m
Thickness of side insulation	= 0.055 m
Wooden frame plate thickness	= 0.02 m
Glass thermal conductivity	= 1 W/m.K
Insulation thermal conductivity	= 0.038 W/m.K
Wooden frame thermal conductivity	= 0.059 W/m.K

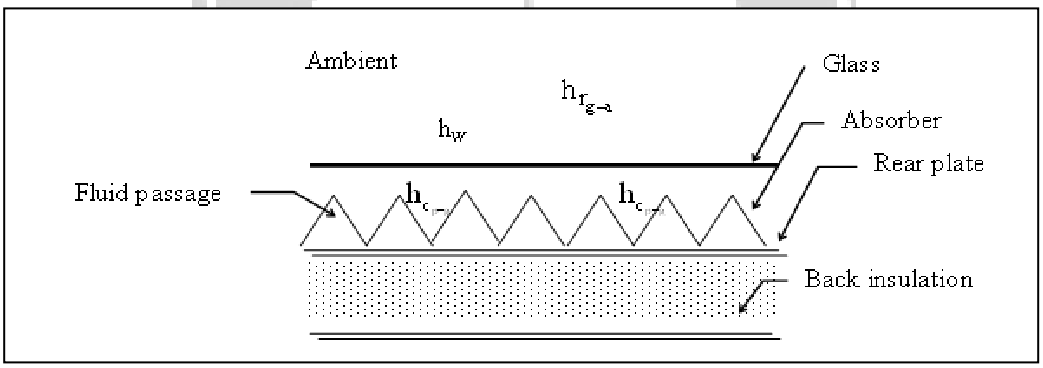

Fig. (1) : Schematic diagram of absorber plate - glass cover - ambient subsystem

In the modeling of the solar air heater, several approximations and assumptions were incorporated in order to simplify the analysis, without obscuring the basic physical situation, while retaining an acceptable accuracy of the simulation results. These important assumptions are as follows:

1– The solar air heater is simulated during the period from 8 a.m. to 5 p.m. with a time interval of one hour.

2– The thermal capacitance effects of the air heater components are negligible.

3– The solar air heater is simulated under steady state conditions.

4– The temperature of the cover and the absorber plate are spatially uniform.

5– For thermally developing flow in a heater channel, suitable averaged heat transfer coefficients available in the literature can be used.

6– Heat flow is one dimensional through the glass cover as well as through the sides and back insulation.

7– Effects due to air leakage are negligible.

With the previous simplifying assumptions, the useful energy gain from a solar air heater can be calculated from the Hottel – Whillier – Bliss equation [11].

$$q_u = F_R \left[S - U_L \left(T_{in} - T_a\right)\right] \quad (1)$$

Where

$$F_R = \frac{G\,Cp}{U_L}\left[1 - \exp\left(\frac{-F'\,U_L}{G\;Cp}\right)\right] \quad (2)$$

The heat removal factor is equivalent to a conventional heat exchanger effectiveness. It depends on the collector design and fluid flow rate.

Solar air heater performance is usually expressed by the collection efficiency defined by the equation:

$$\eta_c = \frac{q_u}{I_{gb}} \quad (3)$$

From equations 1 and 2, it is clear that the analytical value of the useful energy gain depends on the determination of the overall heat transfer coefficient UL, absorbed solar energy by the absorber plate S, and the collector efficiency factor F′. Other quantities may then be easily determined.

2.1 Overall Heat Loss Factor

In the case of flat plate solar collectors, the major heat loss is from the top through the front glass cover . It occurs in the form of convection and thermal radiation from the glass cover to the environment. In the present work the top heat loss factor was calculated by using the analytical technique. This technique is based on the heat transfer analysis of the absorber – cover–ambient subsystem as shown in Fig. (1). Using the concept of thermal resistance, the top heat loss factor for a single glass cover solar air heater is given as:

$$U_t = \frac{1}{R_1 + R_2} \tag{4}$$

Where R1 and R2 are the resistances to heat transfer from absorber to glass cover and from glass cover to ambient air, respectively. These resistances can be written as:

$$\left.\begin{aligned} R_1 &= \frac{1}{h_{c_{p\text{-}g}} + h_{r_{p\text{-}g}}} \\ R_2 &= \frac{1}{h_w + h_{r_{g\text{-}a}}} \end{aligned}\right] \tag{5}$$

Where the radiative heat transfer coefficient between the absorber and the glass cover and between the glass cover and the ambient were calculated using the conventional heat transfer equations, as given in Ref. [11], with the sky temperature being related to the ambient temperature by an equation suggested by Swinbank [12]:

$$T_s = 0.0552\, T_a^{1.5} \tag{6}$$

The natural convective heat transfer coefficient between the V– corrugated absorber plate and the flat glass cover was obtained from an equation suggested by Meyer et al. [13] in the form:

$$\overline{Nu}_L = c\, Gr_L^n \tag{7}$$

Where c and n are constants depending on the collector tilt angle and the Vee – aspect ratio which is the ratio of mean plate spacing to Vee – height. The values of c and n for the simulated solar air heater are given in Table (2).

Table (2) - Constants for Equation 7, Ref. [13]

tilt angle	c	n
0	0.06	0.41
10	0.065	0.4
20	0.07	0.39
30	0.075	0.38
40	0.08	0.367

The wind heat transfer coefficient was calculated by the dimensional equation [11].

hw = 5.7 + 3.8 V (8)

The estimation of the top heat loss factor by using heat transfer analysis required an iterative procedure since the glass cover temperature is unknown. In the present work, the glass cover temperature was first approximated using an equation suggested by Samdarshi and Mullick [14] as:

$$T_g = T_p - \frac{T_p - T_a}{1 + f} \tag{9}$$

where the factor f is given by the empirical equation:

$$f = \sqrt{5 / h_w}$$

After the evaluation of the top heat loss factor using the above equations, the mean glass cover temperature may then be determined analytically from.

$$T_g = T_p - \frac{U_t\left(T_p - T_a\right)}{\left(1 / R_1\right)} \tag{10}$$

Finally, the overall heat loss factor is the sum of the top, bottom and edge heat loss factors:

$$U_L = U_t + U_{bot} + U_{ed} \tag{11}$$

where the last two terms in Eq. (11) were calculated from familier equations given in Ref. [11].

2.2 Absorbed Solar Energy

The performance of a solar air heater depends to a great extent on the amount of solar radiation absorbed by its absorber plate. The incident radiation has three different spatial distributions: beam radiation, diffuse sky radiation, and ground reflected radiation. In the current simulation each component was treated separately.

On an hourly basis the absorbed solar radiation is calculated from the equation [11]:

$$S = D_f \,.\, S_f \left[I_b \,(\tau\alpha)_{e,b} + I_d (\tau\alpha)_{e,d} + I_r (\tau\alpha)_{e,r} \right] \tag{12}$$

Where $(\tau\alpha)_{e,b}$, $(\tau\alpha)_{e,d}$ and $(\tau\alpha)_{e,r}$ are the effective transmittance– absorptance product for beam, diffuse and reflected solar radiation, respectively. The dust factor which accounts for the reduction in absorbed energy due to dust on the glass cover was assumed equal to 0.99, while the shade factor which accounts for the reduction in absorbed energy when some of the air heater structure intercepts solar radiation was assumed 0.99 during midday and 0.98 before and afternoon [10].

The three components of solar radiation received by a south–facing surface with any angle of inclination were calculated from [15]:

$$I_b = I_{DN} \cos \theta_i \ ,$$

$$I_d = I_{DN}\, C \left(\frac{1+\cos\beta}{2} \right) \qquad (13)$$

$$I_r = \rho_g\, H_{gb} \left(\frac{1-\cos\beta}{2} \right)$$

Where ρ_g = ground reflectivity.

$H_{gb} = I_{DN}(\sin \alpha s + C)$

I_{DN} = the direct normal irradiance calculated hourly by the established equation :

$$I_{DN} = A \exp\left(- \frac{P}{P_o} \frac{B}{\cos\theta_z} \right) \qquad (14)$$

The ratio p/po is the pressure at the location altitude concerned relative to the standard atmospheric pressure. It was taken as unity for the present work because the altitude of Jalu is minimal.

Solar irradiance calculations are usually based on values published by ASHRAE [16] for the apparent extraterrestrial solar intensity A, the atmospheric extinction coefficient B, and the diffuse radiation factor C. The values of these three variables are listed for the 21st. day of each month. Table (3) shows these values as reported by [15] which are universally used in hourly, daily, and monthly averages of solar irradiance components determination. In the present work the values of A, B, and C are calculated as a function of day number using equations derived by Joudi [17]. These equations are:

$$A = 1158\left[1 + 0.066 \cos\left(\frac{360N}{370}\right)\right]$$

$$B = 0.175\left[1 - 0.2\cos(0.93\,N)\right] - 0.0045\left[1 - \cos(1.86\,N)\right] \qquad (15)$$

$$C = 0.0965\left[1 - 0.42\cos\left(\frac{360\,N}{370}\right)\right] - 0.0075\left[1 - \cos(1.95\,N)\right]$$

Where N is the day number in the year.

The effective transmittance–absorptance products for the beam, diffuse and reflected components of solar radiation were calculated by using the procedure outlined in Ref.[11].

2.3 Collector Efficiency Factor

For a collector with a triangularly corrugated absorber and a flat rear plate attached to the absorber, which is the case at hand, Duffie and Beckman [11] reported the following equation for the collector efficiency factor:

$$F' = \left[1 + \frac{U_L}{\dfrac{h_c}{\sin\dfrac{\gamma}{2}} + \dfrac{1}{\dfrac{1}{h_c} + \dfrac{1}{h_r}}} \right]^{-1} \qquad (16)$$

where h_c is the convective heat transfer coefficient between the air stream and the absorber plate, and hr is the radiative heat transfer coefficient between the absorber and the rear plate given by:

$$h_r = \frac{4\sigma\, T_p^3}{\dfrac{1}{\varepsilon_p} + \dfrac{1}{\varepsilon_{rp}} - 1} \qquad (17)$$

In equation 16, the heat transfer coefficient between the air stream and the absorber plate and between the air stream and the rear plate are assumed equal. Also, the heat transfer coefficient between the air and the absorber plate is divided by sine half the angle of corrugation to account for the corrugated absorber. Furthermore, the temperature difference between the absorber and the rear plate has a negligible effect on F′ within the range of operation temperature employed and may be neglected.

The convective heat transfer coefficient in the equation for the collector efficiency factor was determined from the correlation that was given by Anderson [18] for fully developed turbulent flow between parallel plates, with the upper being heated and the lower plate insulated, in the following form:

$$NU_{Dh} = 0.0158 \quad Re_{Dh}^{0.8}\, Pr^{1/3} \tag{18}$$

The air properties are evaluated at the mean fluid temperature through the collector which was shown by Klein et al. [19] to be:

$$T_{f,m} = T_{in} + \frac{q_u}{U_L F_R}\left(1 - \frac{F_R}{F'}\right) \tag{19}$$

Finally, the mean absorber plate temperature and the heated fluid outlet temperature can be evaluated from:

$$T_{p,m} = T_{in} + \frac{q_u}{U_L F_R}\left(1 - F_R\right) \tag{20}$$

$$T_{F,out} = \left(T_a + \frac{S}{U_L}\right)\left(1 - \exp\left(-F'.U_L \,/\, G\, C_P\right)\right) + T_{F,in}.\ \exp\left(-F'\, U_L \,/\, G\, C_P\right) \tag{21}$$

The evaluation of the useful energy gain requires the knowledge of the overall heat loss factor and the internal fluid heat transfer coefficient. These factors are themselves functions of the mean plate and mean fluid temperatures. However, the mean plate temperature as well as the mean fluid temperature can be determined only by knowing the values of the useful energy gain and the overall heat loss factor. Therefore, an iterative procedure must be used to evaluate the solar air heater performance.

3. Solution Procedure

Upon entering the initial design parameters of the air heater system, the main program calls a subprogram in which the three components of solar radiation received by the collector glazing are determined for each hour of the simulation period. The main program then calls another subprogram and the computation of the solar air heater performance is started by calling the meteorological data and the collector design data. Then, the total amount of solar radiation absorbed by the collector absorber is computed for each hour. This is accomplished by computing the hourly effective transmittance–absorptance product for each component of solar radiation and multiplying these values by the corresponding radiation components.

Modeling of the solar air heater was accomplished by an iterative procedure. Starting with an initial absorber plate temperature, the properties of air are calculated at the mean ambient – absorber temperature. These properties are then used to compute the convective heat transfer coefficient between the absorber and the flowing air from equation (18). Based on the initial absorber plate temperature, the top, bottom and edge heat loss factors from the collector to the surroundings are calculated. The collector efficiency factor, heat removal factor, useful energy gained and outlet air temperature are then calculated using equations (16), (2), (1) and (21), respectively. The new mean absorber plate temperature and the new mean fluid temperature are then obtained from the useful energy gained from equations (20) and (19). The new mean absorber plate temperature is then compared with the initial reference value for a given accuracy. If the required accuracy is not obtained, the new mean absorber plate temperature is taken as a new reference value, and the process is repeated with air properties evaluated at the new mean fluid temperature. The convergence criteria for the difference in mean absorber plate temperature calculated from equation (20) was set equal to 0.00001% .

The calculation of the top heat loss factor using the heat transfer analysis technique requires itself an iterative procedure since the glass cover temperature is unknown. For this technique, the glass cover temperature is first approximated from equation (9). The air properties are then calculated at the mean absorber–cover temperature. After the evaluation of the top heat loss factor, the new glass cover temperature is obtained from equation (10). The new cover temperature is then compared with the approximated reference value for a given accuracy. Also, if the required accuracy is not obtained, the new glass cover temperature is taken as a new reference value and the process is repeated. The convergence criteria for the difference in cover temperature calculated from equation (10) was set equal to 0.00001% . The above process is repeated for each absorber plate temperature and the whole process for modeling the solar collector is repeated for each hour during the simulation period.

4. Results & Discussion

The results of the simulation of the solar air heater include absorbed solar radiation by the absorber plate, useful energy gained by the working fluid, mean absorber plate temperature, mean working fluid temperature through the collector, outlet working fluid temperature and collector instantaneous efficiency. These parameters were predicted on hourly basis during the simulated period because the simulation of the collector for a day, a month or a year can not give the true picture of it’s performance. This may be attributed to the fact that the factors controlling the performance such as solar radiation intensity and ambient conditions are variable by themselves and change from hour to hour. Therefore, the suitable way of expressing the performance of the collector would be in terms of instantaneous values of the factors influencing it. Also, when hourly meteorological data is used, the collector can be represented as a zero capacitance model [19]. This implies that the effects of thermal capacitance of the solar air heater components on it’s performance can be neglected and the air heater may be considered to be in equilibrium with it’s environment at any instant of time. In addition, the assumption of uniform glass cover and absorber plate temperatures which was made in

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the analysis of the solar air heater in section 2 provides that the air heater can be represented as a single node in the flow direction [3].

The solar air heater parameters that were simulated in the present work are given in Table (4). The range of each of these parameters is the most practical design and operating condition for the case at hand. Baseline values represent fixed values of these parameters when the effect of the range of a certain other variable is studied. For example, all values are those of baseline except wind velocity which ranges from 0 to 10 m/s when the effect of wind velocity on the solar air heater performance is studied. In addition, the day numbers selected for each month during the summer season were based on a recommendation by Klein [20] but of course with the local weather conditions. Typical results for December and June will be presented here for brevity.

Table (4) - Collector operating parameters

Variables	Range	Baseline Value
Mass flow rate [kg / s . m^2]	0.01-0.06	0.03
Absorber plate emissivity	0.1-0.95	0.95
(Inlet-Ambient) temperature difference, °C	0-20	0
Wind velocity [m / s]	0-10	2
Collector tilt angle [degree]	0-40	50
Day numbers	198 and 355	355

The performance of a flat solar collector is highly influenced by it's orientation and tilt angle. Both these two parameters influence the amount of solar radiation reaching the collector absorber surface and thermal losses from the collector. Maximum solar irradiance is obtained when the altitude angle between the solar rays and the collector surface becomes normal or near normal. The value of the tilt angle for maximum solar radiation intensity depends on the latitude of the location concerned and also the time of year. The simulated solar air heater was assumed to be oriented facing south which is the usual orientation of fixed–position flat plate solar collectors in the northern hemisphere. The optimum tilt angle for the city of Jalu and for day numbers 198 and 355 was calculated using the computer simulation. The optimization was based on maximizing the total amount of useful energy gained during the simulated period. Fig. (2) and Fig.(3) shows the variation in the total daily absorbed energy during the simulated period with the collector tilt angle for the above days. It can be observed that the optimum tilt angle for maximum absorbed energy is approximately 5° and 50° for June and December, respectively. It was also found that using the computer simulation, if the collector tilt angle is fixed at 5° during the period of May to August for the city of Jalu, the total energy gained during this period was reduced by a very small amount as compared with the total energy gained for the

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same period by using monthly optimum tilt angles. The difference between these two cases was found to be less than 0.25% .

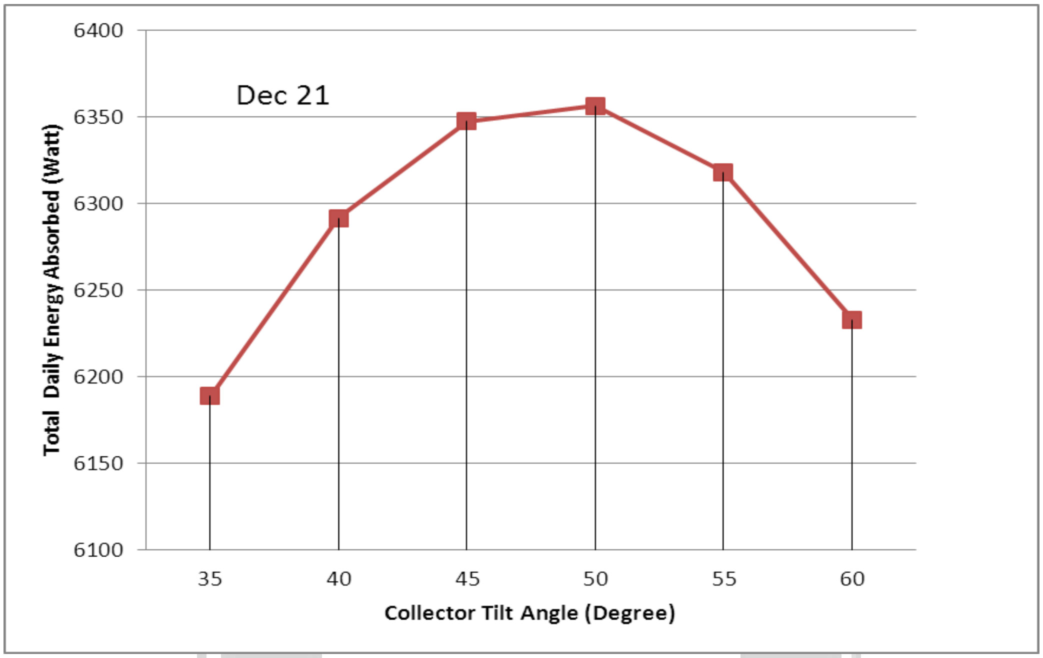


Fig. (2) : Variation of total daily energy absorbed with collector tilt angle for December, 21

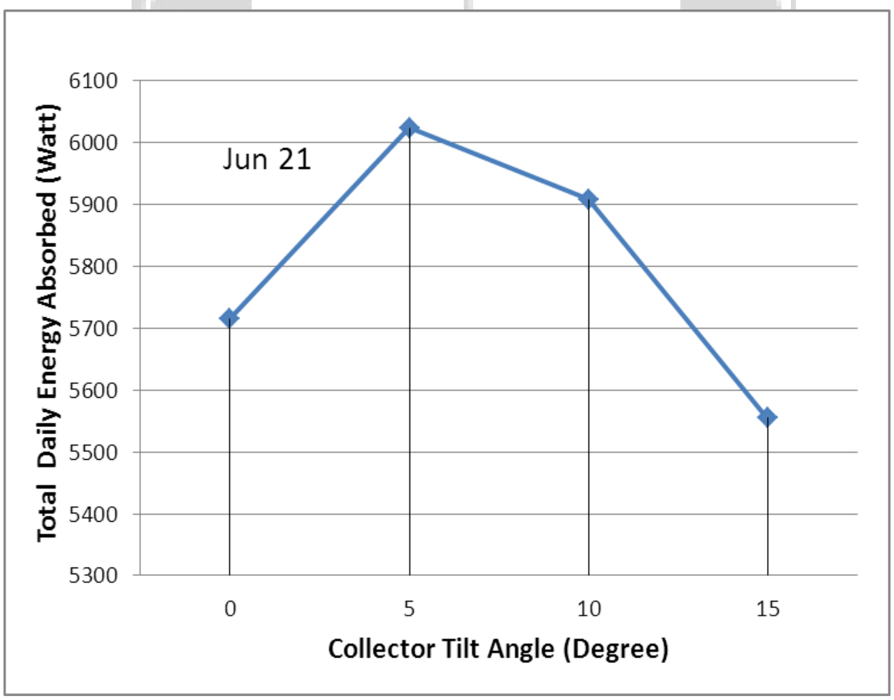


Fig. (3) : Variation of total daily energy absorbed with collector tilt angle for June, 21

Fig. (4) Shows the variation of solar radiation intensity and absorbed solar radiation during the simulated period. The amount of solar energy absorbed by a solar collector depends essentially on the optical properties of it's glass cover and absorber plate, while the amount of useful energy gained depends mainly on the thermal characteristics of the collector.

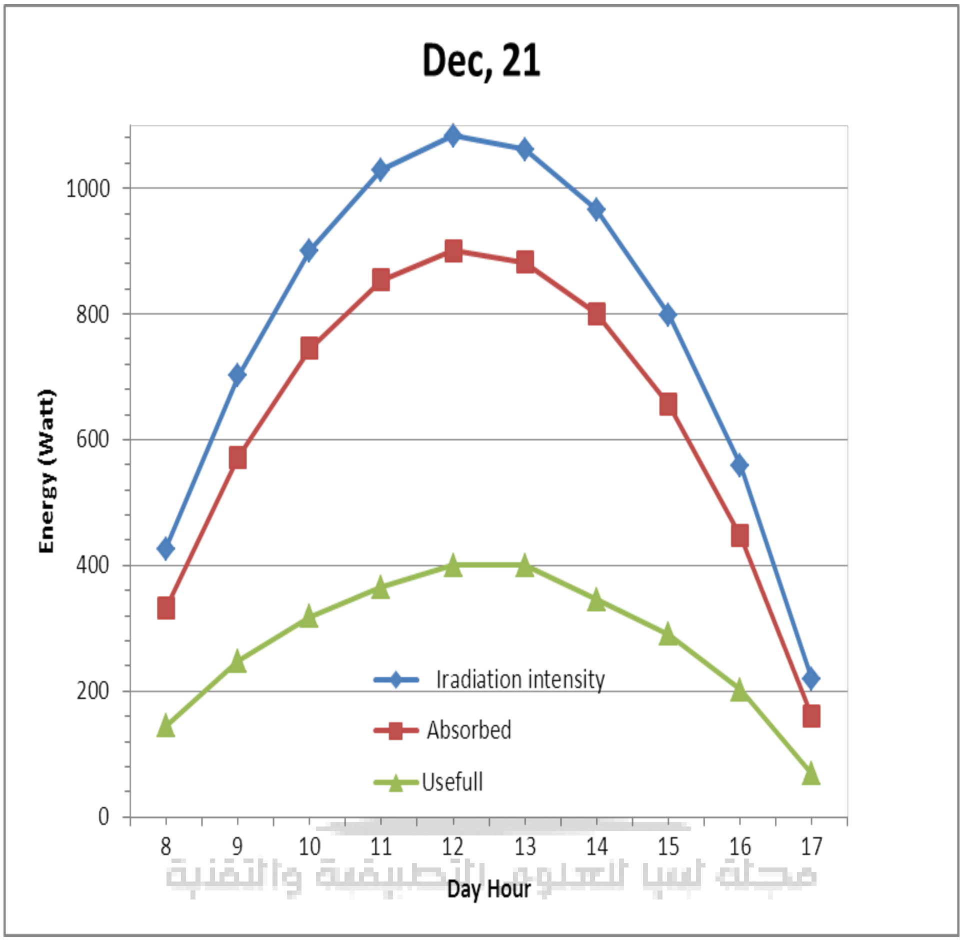


Fig. (4) : Variation of the solar irradiance, absorbed energy and useful energy gained during typical winter day hours.

Fig. (5) Shows the variation of the instantaneous collection efficiency during the simulated period. From this figure it is clear that the efficiency takes higher values at higher mass flow rates because a lower flow rate increases the average operating temperature of the collector, thereby decreasing the collection efficiency. Also, the variation in values of the collection efficiency from 9 a.m. to 3 p.m. is very small. This is mainly due to the combined effect of the fraction of absorbed solar radiation and the thermal losses from the collector during the day. However, there is a distinct drop in the value of the collection efficiency at 8 a.m. and 4 p.m. due to the high angle of incidence of solar rays which causes a reduction in the transmissivity and an increase in the reflectivity of the rays by the collector glazing.

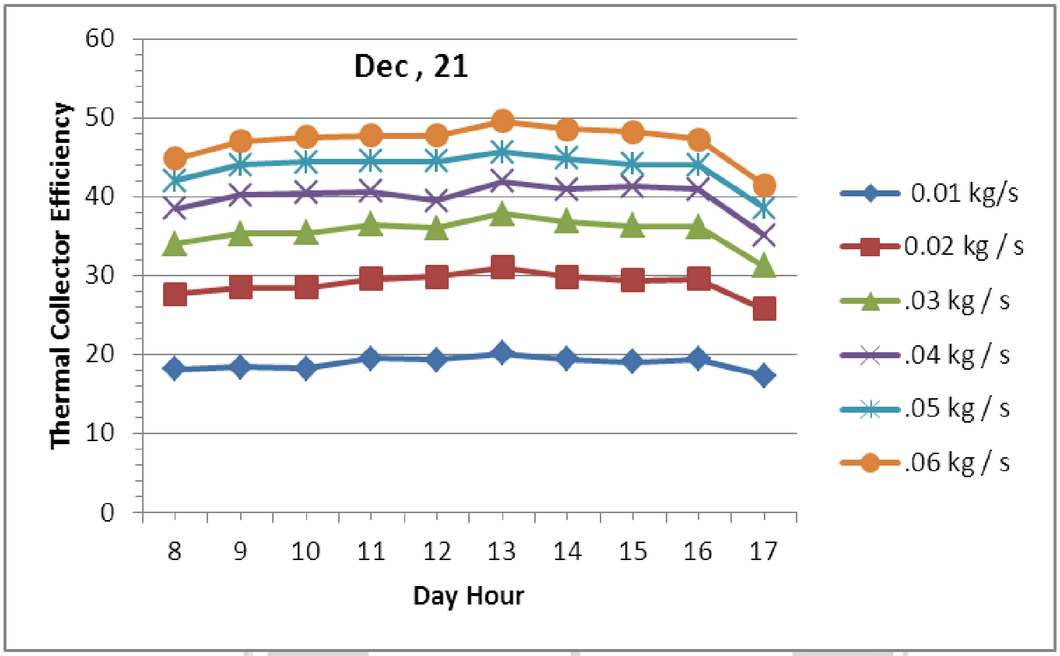


Fig. (5) : Variation of the thermal collector Efficiency with air mass flow rate during typical winter day hours.

The effect of air mass flow rate on the outlet air temperature from the collector is shown in Fig. (6). These temperatures are inversely variant with mass flow rate. The above results are in agreement with all the results cited in the literature. A maximum of about 69°C was obtained for the outlet air temperature during midday at a flow rate of 0.01 kg / s.m^2 with a collection efficiency of about 34%.

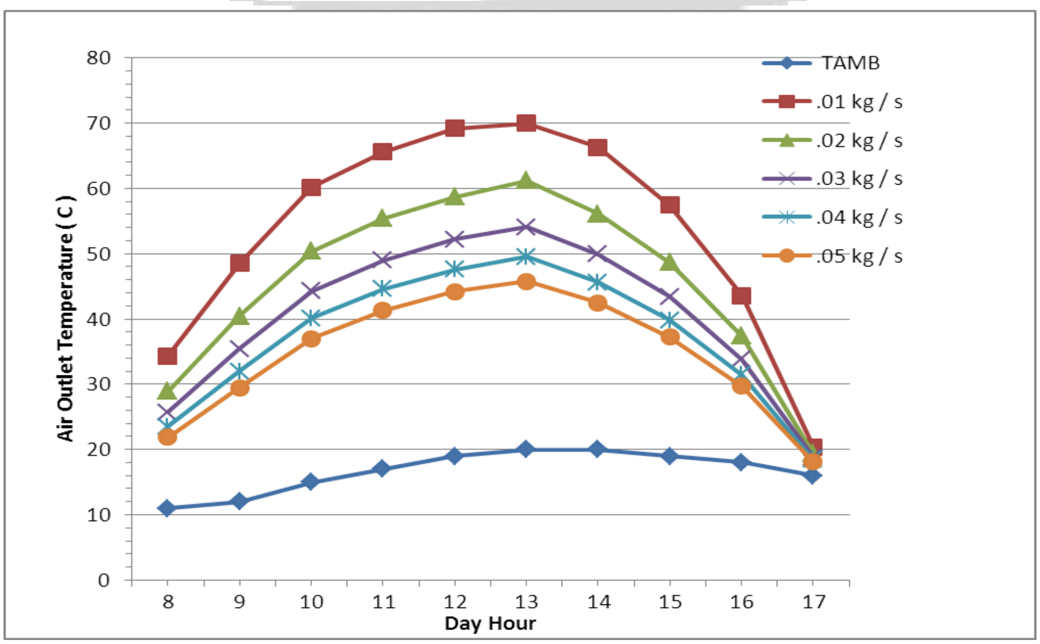


Fig. (6) : Variation of the air outlet temperature with air mass flow rate during a typical winter day typical winter day hours

The effect of using selective absorber surfaces on the useful energy gained is shown in Fig. (7). Decreasing the long wave emissivity of the absorber surface results in an increase in the useful energy gained and thus the collector efficiency, which is due to a reduction in the radiative heat loss from the absorber plate. It was observed that, by using the computer simulation, if the emissivity of the corrugated absorber plate is reduced from 0.95 to 0.1 through the use of selective absorber coatings, the collection efficiency improved by approximately 13% , 11% and 9 % for mass flow rates of 0.01, 0.02 and 0.03 kg/s.m^2, respectively. However, it should be pointed out that these selective coatings increase the cost of the solar air heater and often deteriorate with time.

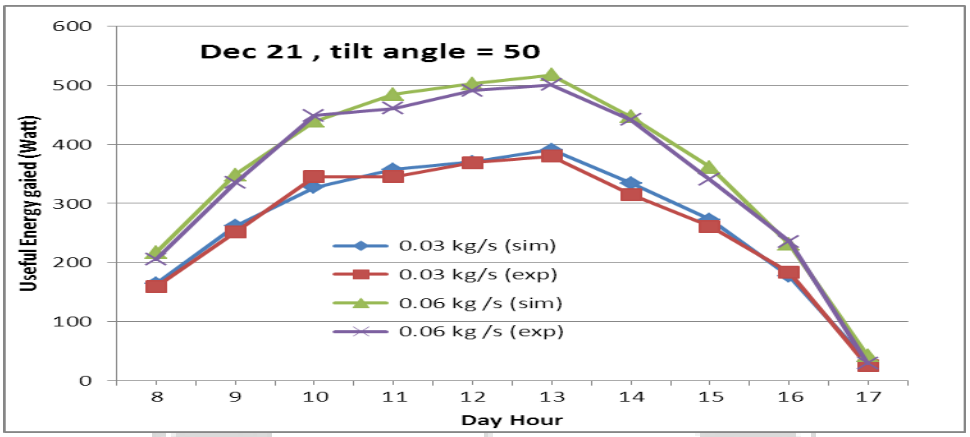
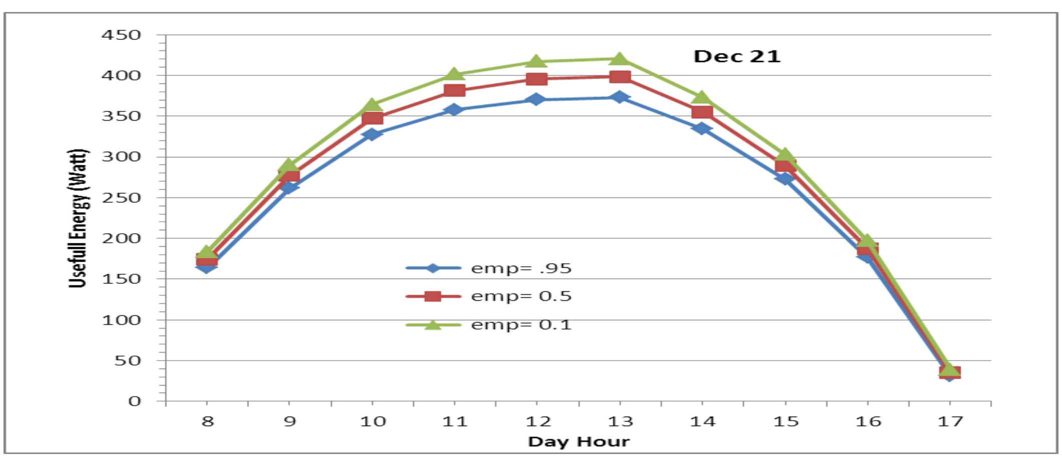


Fig. (7) : Variation of the Useful Energy Gained with collector plate emissivity during a typical winter day typical winter day hour

In order to validate the results obtained in the present simulation of the solar air heater, the computer model predictions were compared with the results obtained from a series of experiments conducted by Ref. [10]. For comparison purposes the same operating condition of this reference were, of course, introduced into the present computer model.



Fig(8): comparison between the experimental and the computer simulation values of the useful energy gained during the day for the city of Benghazi

Fig. (8) Shows a comparison between the experimental and the computer simulation values of the useful energy gained during the day. It can be observed from Fig. (8) that the useful energy gained increases with increasing mass flow rate of air for all hours during the simulated period for both experimental and computer results. In this figure the computer results were based on values of solar radiation calculated from the clear sky model presented in section 2. The maximum difference between experimental and analytical values of the useful energy gained was within 7.5%.

5. Conclusions

- Increasing the mass flow rate of air through the solar air heater increases the instantaneous collection efficiency and the useful energy gained but reduces the outlet air temperature correspondingly.
- Air outlet temperatures of 69°C were obtained in winter at midday for a mass flow rate of 0.01 kg/s.m^2 with a collection efficiency of 34 %.
- Based on maximizing the total useful energy gained during the simulated period, the optimum tilt angles for the period from May to August and from October to February in Jalu were found to be 5° and 50° from the horizontal, respectively. Changing this angle by ±10° around it's optimum value reduces the useful energy gained by only 2%.
- Experimental data of previous research were compared with simulation results and found in very good agreement.

Nomenclature

	Symbols	Units
A	Apparent extraterrestrial solar intensity	[W / m^2]
B	Atmospheric extinction coefficient	[(air mass) $^{-1}$]
C	Diffuse radiation factor	-
Cp	Specific heat of air	[J / kg . K]
D_f	Collector dust factor	-
D_h	Hydraulic diameter of air passage	[m]
F′	Collector efficiency factor	-
F_R	Collector heat removal factor	-
g	Acceleration due to gravity (9.81)	[m / s^2]
G	Air flow rate per unit of collector area (mass flux)	[kg / s m^2]
Gr_L	$\frac{g\beta_T\left(T_p - T_g\right) L^3}{\nu^2}$ = Grashof number based on L	-
h_c	Convective heat transfer coefficient	[W / m^2 . K]
h_r	Radiative heat transfer coefficient	[W / m^2 . K]

h_w	Wind heat transfer coefficient	[W / m^2 . K]
H	Solar irradiance on a horizontal surface	[W / m^2]
I	Solar irradiance on a tilted surface	[W / m^2]
I_{DN}	Direct normal irradiance	[W / m^2]
K	Thermal conductivity	[W / m . K]
L	Mean absorber plate - glass cover spacing	[m]
Nu_L	$\frac{h_c L}{K}$ = Nusselt number based on L.	-
Nu_{Dh}	$\frac{h_c D_h}{K}$ = Nusselt number based on D_h	-
p	pressure	[Pa]
Pr	$\frac{\mu Cp}{K}$ = Prandtle number	
q_u	Useful energy gained per unit collector area	[W / m^2]
Re_{D_h}	$\frac{\rho_a u D_h}{\mu}$ = Reynolds number based on D_h	-
R	Thermal resistance	[m^2 . K / W]
S	Absorbed solar radiation	[W / m^2]
S_f	Collector shade factor	-
T	Temperature	[K]
u	Velocity of air in the collector	[m / s]
U	Heat loss factor	[W/ m^2 .K]
V	Wind speed	[m / s]

Greek Symbols

	Symbols	Units
α	Absorptance	-
α_s	Sun's altitude angle	[degree]
β	Collector tilt angle with horizon	[degree]
β_T	Coefficient of volumetric expansion	[K^{-1}]
γ	Angle of corrugation	[degree]
ε	Emissivity	-
η	Efficiency	-
θ_i	Solar radiation incident angle	[degree]
θ_z	Solar zenith angle	[degree]
μ	Dynamic viscosity of air	[Pa . s]
ν	Kinematic viscosity of air	[m^2 / s]
ρ	Reflectivity	
ρa	Density of air	[kg / m^3]
σ	Stefan - Boltzmann constant (5.669 * 10-8)	[W / m^2 . K^4]

($\tau\alpha$)	Transmittance - absorbtance product	-

Subscripts

a	Ambient .
b	Beam .
bot	Bottom of collector .
c	Collector .
d	Diffuse.
e	Effective.
ed	Edge of collector.
f	Fluid .
g	Glass.
gb	Global .
in	Inlet to collector .
ins	Insulation.
L	Overall .
m	Mean .
out	Outlet from collector .
p	Absorber plate .
r	Reflected .
rp	Rear plate of collector .
s	Sky
t	Top .

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