

ENGINEERING

PREDICTION STUDY ON SINGLE AND DOUBLE FLOW SOLAR AIR HEATER

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Abstract

This paper presents part of a developed internet-based system able to predict thermal performance for solar air heaters. The solution procedure is performed for a flat plate collector in single and double flow in order to achieve an efficient design suitable for collecting energy to be used in solar drying. The thermal performance was determined over a wide range of operating conditions, but in the same configuration for the two types of flow. The double flow is found to be more efficient than the single flow. The optimum operating conditions were determined. The use of the internet offers attractive features for interconnecting people throughout the world.

Keywords: Flat plate collector, single flow, double flow double duct, mathematical simulation

Introduction

The analysis of solar radiation in several main towns in Malaysia shows that solar radiation has the potential to be used for drying purposes and several other applications (Mohd et al., 1996). Typically, open air sun drying has been used to dry plants, seeds, fruits, meat, wood and other agricultural and forest products. But for large scale production, many limitations of open air drying have appeared (Sopian et al., 1999). Thus in order to reap the benefits from abundant and free solar energy, numerous attempts have been made in recent years to develop simple solar drying systems mainly for drying agricultural and forest products. Several studies to determine the thermal performance of solar air heaters have been conducted, theoretically and/or

experimentally, and different modifications have been suggested and applied to improve the heat transfer coefficient between the absorber plate and air (Ong, 1995a; Ong, 1995b; Metwally et al., 1997; Yeh et al., 2002; Froson and Nazha, 2003). But there is no work that has been carried out on using an expert system to support the designs of solar air heaters, where the expert system is an interactive computer-based decision tool that uses both facts and heuristics to solve difficult decision problems based on knowledge acquired from an expert. Thus our objective is to develop an internet-based expert system able to predict the thermal performance of different designs of solar air heaters (with a flat or V-groove absorber, a single, double or triple glass cover, in single

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pass or double pass double duct and with or without porous media in the lower duct). This paper presents part of the developed internet-based system; the study here is devoted to a flat plate collector in single or double flow pass, in order to achieve an efficient design suitable for collecting energy to be used in solar drying. The use of the internet offers attractive features that are useful at the development and delivery stages, as well as expanding and sharing knowledge from any location in the world. The developed system is fully implemented to run on theweb and provides an easy and attractive way to share knowledge.

Theoretical Analysis

In single pass flat plate collector energy is transferred from the plate to the ambient air at T_a through the back loss coefficient U_b , to the fluid at a temperature T_f through the convection heat transfer coefficient h_2 , and to the bottom of the cover through the radiation heat transfer coefficient h_r . Energy is transferred from fluid to the cover through the convection heat transfer coefficient h_1 . Finally energy is lost to the ambient air through the combined convection and radiation coefficient U_t . Schematic drawings for flat and thermal net work are shown in Figure 1. The steady-state energy equations yield the following equations (Bashria *et al.*, 2004a): Collector cover

$$h_{l}\left(T_{f}-T_{c}\right)+h_{r}\left(T_{p}-T_{c}\right)=U_{t}\left(T_{c}-T_{a}\right) \tag{1}$$

Flat plate absorber

$$U_b(T_p - T_a) + h_2(T_p - P_f) +$$

$$h_r(T_p - T_c) = I\tau\alpha$$
(2)

Fluid medium

$$h_2\left(T_p - T_f\right) = \left(\frac{mC_p}{W}\right) \left(\frac{dT_f}{dx}\right) + h_1\left(T_f - T_c\right)$$
(3)

In double Pass Double Duct Flat Plate Collector energy is transferred from the plate at T_P to the cover through the radiation heat transfer coefficient h_{r1}, and to the fluid in the upper duct flowing between the glass cover and the plate at a temperature T_{f1} through the convection heat transfer coefficient h₂, and to the fluid in the lower duct flowing between the absorber plate and the base plate at a temperature T_{f2} through the convection heat transfer coefficient h₃. Also energy is transferred to the base plate through the radiation heat transfer coefficient h_{r2}. Energy is transferred from the fluid flowing in the upper duct to the cover glass through the convection heat transfer coefficient h_1 and from the fluid in the lower duct at T_{f2} to the base plate through convection heat transfer coefficient h₄. Finally, energy is lost to the ambient air through the combined convection

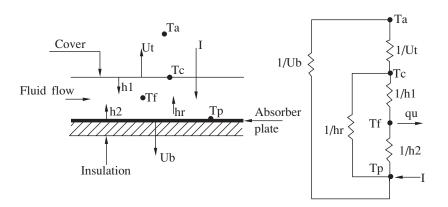


Figure 1. Schematic diagram of single pass solar collector

and radiation coefficient U, through the cover glass. Figure 2 illustrates the double pass double duct flat plate collector and the thermal net work. The steady-state energy balance on the cover, the plate and the fluid in the upper and lower ducts gives the following equations:

Collector cover

$$h_1(T_{f1} - T_c) + h_{r1}(T_p - T_c) = U_t(T_c - T_a)$$
 (4)

Fluid 1

$$h_2(T_p - T_{f1}) = \left(\frac{\dot{m}C_{p1}}{W}\right) \left(\frac{dT_{f1}}{dx}\right) + h_1(T_{f1} - T_c)(5)$$

Absorber Plate

$$I\tau\alpha = h_{r1} \Big(T_p - T_c \Big) + h_2 \Big(T_p - T_{f1} \Big) + h_3 \Big(T_p - T_{f2} \Big) + h_{r2} \Big(T_p - T_r \Big)$$
(6)

Fluid 2

$$h_3(T_p - T_{f2}) = \left(\frac{\dot{m}C_{p2}}{W}\right) \left(\frac{dT_{f2}}{dx}\right) +$$

$$h_4\Big(T_{f2} - T_r\Big) \tag{7}$$

Bottom Plate

$$h_4(T_{f2} - T_r) + h_{r2}(T_p - T_r) = U_b(T_r - T_a)$$
 (8)

Losses and Heat Transfer Coefficients

The overall heat transfer loss from the collector is the summation of three separate components, the top loss coefficient, the bottom loss coefficient, and the edge loss coefficient. The empirical relations for these coefficients are as follows (Duffie and Beckman, 1991; Al-Ajlan et al., 2003)

Fluid 1
$$h_2\left(T_p - T_{f1}\right) = \left(\frac{\dot{m}C_{p1}}{W}\right) \left(\frac{dT_{f1}}{dx}\right) + h_1\left(T_{f1} - T_c\right) (5) \qquad U_t = \left\{\frac{N}{\frac{C}{T_p}\left[\frac{T_p - T_a}{N + f}\right]^e} + \frac{1}{h_w}\right\}^{-1} + \tag{9}$$
Absorber Plate

(6)
$$\frac{\sigma\left(T_p + T_a\right)\left(T_p^2 + T_a^2\right)}{\left(\varepsilon_p + 0.00591Nh_w\right)^{-1} + \frac{2N + f - 1 + 0.13\varepsilon_p}{\varepsilon_a} - N}$$

$$U_b = \frac{k_b}{x_b} \tag{10}$$

(7)
$$U_e = \frac{U_{e'}A_e}{A} = \frac{k_e}{x_e} * \left[\frac{2*(L+W)*H}{L*W} \right]$$
 (11)

 $U_{e'}$ is the edge loss coefficient and A_e the area base on the perimeter of the collector

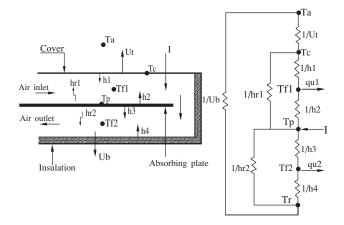


Figure 2. Schematic diagram of double pass solar collector

$$U = U_t + U_b + U_e \tag{12}$$

The radiation heat transfer coefficient between the absorber and the air heater surfaces may be estimated by assuming a mean radiant temperature equal to the mean fluid temperature as follows:

$$h_{r_1} = \frac{\sigma \overline{T}_{f1}^3}{\frac{1}{\varepsilon_p} + \frac{1}{\varepsilon_c} - 1}$$
 (13)

$$h_{r2} = \frac{\sigma \overline{T}_{f2}^3}{\frac{1}{\varepsilon_n} + \frac{1}{\varepsilon_r} - 1}$$
 (14)

The convective heat transfer coefficient for air flowing over the outside surface of the glass cover, will be calculated by the following equation (Mc Adams, 1985):

$$h_{w} = 5.7 + 3.8V \tag{15}$$

The convective heat transfer coefficients are calculated using the following relations:

$$h = \frac{Nuk}{D_b} \tag{16}$$

$$Nu = 0.0158 \left(Re \right)^{0.8} \tag{17}$$

$$Re = \frac{\dot{m}D_h}{A_f \mu} \tag{18}$$

$$D_h = 4 \frac{\text{wetted area}}{\text{perimeter}} \tag{19}$$

Collector Efficiency

The useful extracted energy of a collector is the difference between the absorbed energy and the energy losses, while the thermal efficiency of a collector is defined as the ratio of the useful energy to the total incident solar radiation. Mathematically, the useful energy and efficiency

of the collector are expressed as:

$$Q_{u} = A \left[\left(\tau \alpha \right)_{e} I - U \left(T_{Pm} - T_{a} \right) \right]$$
 (20)

Eqn. 19 is reformulated in terms of the inlet fluid temperature and the collector heat removal factor as shown below (Duffie and Beckman, 1991):

$$Q_{u} = A F_{R} \left[\left(\tau \alpha \right)_{e} I - U \left(T_{i} - T_{a} \right) \right]$$
 (21)

$$\eta = \frac{Q_u}{A I} = F_R \left(\tau \alpha \right) - F_R U \frac{\left(T_i - T_a \right)}{I}$$
 (22)

Methods

Dreamweaver combined with the Active Server Pages (ASP) as an extension environment, have been chosen as a development software tool to predict the thermal performance for solar air heaters. The programmatic code was written by VBScript as scripting language to execute commands on a computer; the databases were entered using the Microsoft Access as shown in Figure 3.

The required data are classified into three groups, the first group is named general input data which is concern about meteorological condition, the second group named collector characteristics contains all specific manufacture attributes related to the collector and the third group named energy characteristic contains measured data about inlet temperature and mass flow rate related to the transfer media inside the collector (Bashria *et al.*, 2004b). All the entered data related to the parameters which are needed for the prediction of the thermal performance of the system, the insertion menu is shown in Figure 4.

The major steps of the program for single flow and double flow pass in a flat plate collector are shown in flow chart Figure 5 and are discussed below.

- **Step 1:** The program will start with the word chosen which enables the user to choose one of the different solar air heater types
- **Step 2:** General input data will be required to be entered; these data contain meteo-

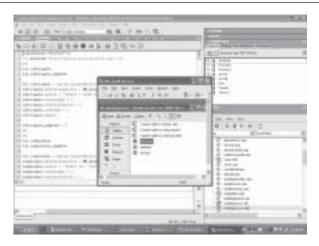


Figure 3. Dreamweaver combined with the Active Server Pages (ASP), VBScript and Microsoft Access

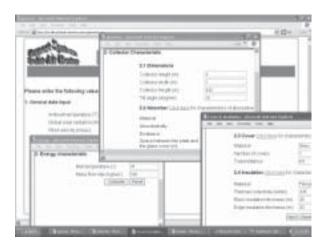


Figure 4. Insertion menu

rological and weather conditions, collector specification and heat transfer media parameters. These data will be inserted to the input data file as a dynamic data-base and each solar air heater will have a separate input data file. The dynamic database is enriched during each execution of the program but the information is lost when the execution is terminated.

Step 3: The static knowledge database which represents the used equations in certain formalism is created once, when the system is being developed.

- **Step 4:** Back and edge losses will be calculated according to the input data file and to the suitable equations in the static database.
- **Step 5:** Initial values for mean plate temperature and mean fluid temperature are assumed.
- Step 6: Using the above assumed values, top losses, radiative heat transfer coefficient between the cover and the plate, convective heat transfer coefficients, collector thermal efficiency, collector flow factor and the heat removal factor can be found.

- **Step 7:** According to these values useful energy will be calculated
- **Step 8:** The new temperature values of mean plate temperature and mean fluid temperature will be calculated
- Step 9: Step 5 to Step 8 will be repeated until $|(T_{pnew} T_p) \le 0.001| \text{ and }$ $|(T_{fnew} T_f) \le 0.001|$
- **Step 10:** Once the rule is verified, the outlet temperature can be calculated
- Step 11: Using useful heat gain, the thermal efficiency will then be calculated and this will be the final result for single the flow flat plate collector.

For the double flow double duct flat plate collector, the flow will enter to the lower duct; in this case the inlet air temperature in the lower duct will be the outlet air temperature from the upper duct. So the program will continue its execution according to the following steps:

- Step 12: Initial value for mean fluid temperature in the lower duct is assumed to be greater than the outlet temperature from the upper duct.
- Step 13: Read the air properties at assumed mean fluid temperature then calculate the radiative heat transfer coefficient between the plate and the bottom plate, Reynolds number and convective heat transfer coefficients in the lower duct.
- **Step 14:** According to the above, the second useful energy will be calculated
- **Step 15:** The new mean fluid temperature will be calculated
- **Step 16:** Step 12 to Step 15 will be repeated until $|(T_{f2new} T_{f2})| \le 0.001|$ **Step 17:** Once the rule is verified, the final
- **Step 17:** Once the rule is verified, the final outlet temperature can be calculated
- **Step 18:** Using total useful heat gain, the thermal efficiency will be calculated.

The above internet-based computer program is in the process of continual development, but it can be accessed through the website http://esrg.eng.upm.edu.my/BashriaSolarheaters/default.asp

Results and Discussion

Performance of the double flow solar air heater is studied and compared with the performance of the single pass. In the analysis the flow rate in each air heater is varied from 0.01 to 0.10 kg/sec. The standard design dimensions for a solar air heater system have been recommended by Beckman et al. (1977) as the length of duct from 1.25 to 2.50 m, the spacing between the absorber plate and the glass cover should be from 1 to 4 cm. According to these recommendations the air heater collector is 2.4 m in length and the spacing between the glass cover and the absorber is 0.035 m. The lower duct height is fixed to 0.075 m. The input data have been entered to give the same configuration for comparison. The detailed specifications are given below:

- Collector tilt angle (degree) 10
- Collector lengh (m)collector width (m)
- Plate typeAbsorber materialFlat plateBlack steel,
 - $\alpha = 0.95$ and
- Cover material $\epsilon = 0.89$ Ordinary
 - clear glass, $\tau = 0.85$ and $\epsilon = 0.88$
- Number of cover 1
- Insulation naterial Fiber glass, k = 0.045 W/m k
- Back insulation thickness (m) 0.05
- Edge insulation thickness (m) 0.03

The combined efficiency and outlet temperature variation with mass flow rate for a single and double flow solar air heater shown in Figure 6, have been predicted at solar radiation of 500 W/m², ambient temperature at 33°C, air velocity at 1.5 m/sec and pre-heated inlet temperature at 35°C.

The Figure shows that the efficiency of air collectors is dependent on the mass flow rate; it is increased by the increase of the mass flow rate. The double flow is 4 - 10% more efficient than the single flow; this is due to the increased heat removal factor in the double flow compared

to the single flow, while the outlet temperature decreased with the increased mass flow rate. From Figure 6 the optimum operating conditions with respect to efficiency and outlet temperature for both air collectors are determined; for the single flow at mass flow rate 0.055 kg/sec, the outlet temperature 49.8°C and the efficiency 50%, while for the double flow the optimum operating condition at 0.04 kg/sec where the outlet temperature is 55°C and the efficiency is 56%.

One of the unique works in collector testing is preparing the efficiency curves (Karim and Hawlader, 2004). ASHRAE (1977) recommends drawing the efficiency curve at two flow rates. In this study, a lot more efficiency curved was constructed at different flow rates for each collector. This was done to investigate the trend of the efficiency curves at higher flow rates, as drying applications may consider a wide range of flow rates and operating temperatures. The efficiency curves for both air heater types are drawn for different mass flow rates and straight lines resulted. Where the steady state thermal performance Eqn. 22 Indicates that

if the efficiency is plotted against
$$\frac{\left(T_i - T_a\right)}{I}$$

it will represent a straight line with the slope F_RU_L and y-intercept equals to $F_R(\tau\alpha)$, (Karim and Hawlader, 2004; Bashria *et al.*, 2004a).

Figure 7 shows the efficiency curves of the single flow at different flow rates. The efficiency equations with F_RU_L and $F_R(\tau\alpha)$ values are presented in Table 1. From the Figures, it is obvious that the slope of the efficiency curves decrease with an increase of flow rate which means at a higher flow rate the overall loss is lower. Figure 8 presents the efficiency curves of the double flow, and Table 2 shows its efficiency equations with F_RU_L and $F_R(\tau\alpha)$ values. From the figures it can be seen that the loss coefficient is higher in the single flow than the double flow mode.

Validation

The output from the developed program

has been run according to same configuration and parameters done in experimental work done by Karim and Hawlader (2004) for single pass flat plate collector and by Elradi (2001) for the double pass flat plate collector. The experimental performance curves and results from the experimental work reproduced from Karim and Hawlader (2004) and Elradi (2001) in order to compare them with predicted results obtained by the developed program.

For the single pass flat plate collector the same result obtained from the developed program has been obtained by the experimental work done by Karim and Hawlader (2004) as shown in Figure 9. From the Figures, it is obvious that comparison between the predicted output and the experimental results shows a great agreement with error difference lies between 1% to 5%.

Elradi (2001) has found the effect of mass flow rates on temperature rise of double pass flat solar collector at 540 W/m² solar radiation and 33°C ambient temperature. At the same conditions the effect of mass flow rates on temperature rise of double pass flat solar collector has been predicted by the developed program. The results is shown in Figure 10. By comparing the experimental and predicted results it is found that, the temperature rise is decreased with the increased of mass flow rate, and there is an error deficiency occurred lies from 0.6% to 2.2% between the experimental and predicted results.

Karim and Hawlader (2004) have also done an experimental work on the double pass flat plate collector; the input data for the developed program is entered as same as the experimental work. Figure 11 shows the variation of efficiency and outlet temperature with flow rate for double pass flat plate collector by predicted and experimental work (Karim and Hawlader, 2004). It is shown that the efficiency increased by increasing the mass flow rate while the outlet temperature decreased by increasing the mass flow rate.

The figures and analyses conducted show a good agreement between the predicted and experimental work with an error deficiency equal to 5% at maximum.

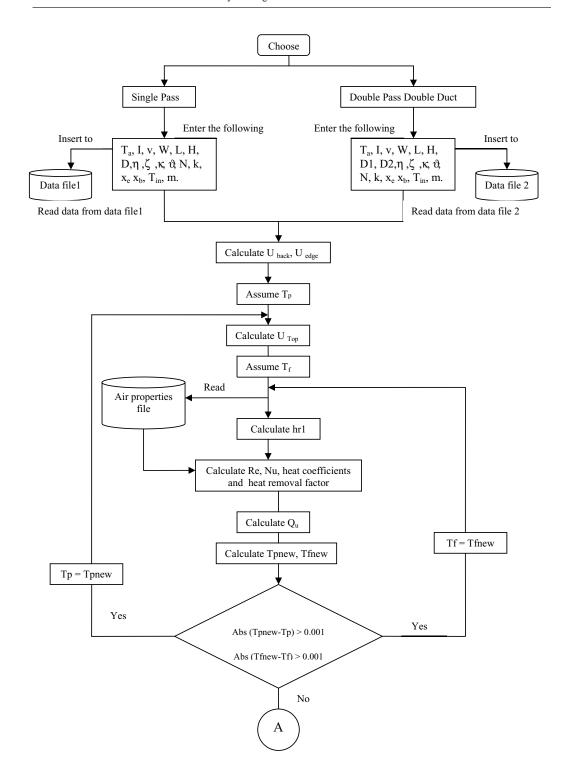


Figure 5. Flow chart of the computer program

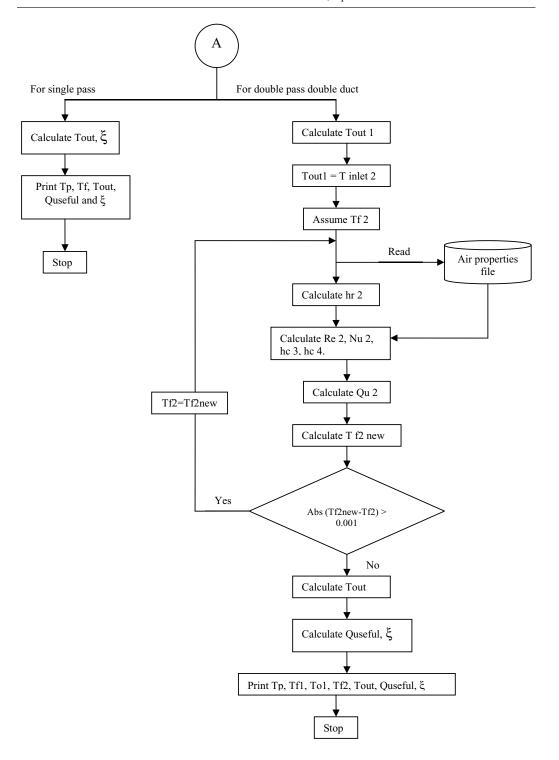


Figure 5. (Continued)

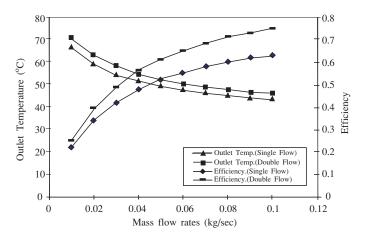


Figure 6. Combined efficiency and outlet temperature variation with mass flow rate for single and double flow

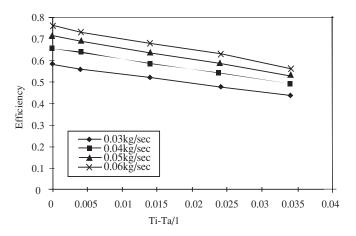


Figure 7. Efficiency curve for single flow

Table 1. Efficiency equations, $F_R(\tau\alpha)$ and F_RU_L for single flow

Flow rate (kg/sec)	$F_{R}(\tau\alpha)$	$\mathbf{F}_{\mathbf{R}}\mathbf{U}_{\mathbf{L}}$	Efficiency equation
0.03	0.4359	3.1542	Y = -3.1542 Z + 0.4359
0.04	0.4917	3.5345	Y = -3.5345 Z + 0.4917
0.05	0.5380	4.0771	Y = -4.0771 Z + 0.5380
0.06	0.5696	4.3154	Y = -4.3154 Z + 0.5696

 $Z = (T_i - T_a)/I$

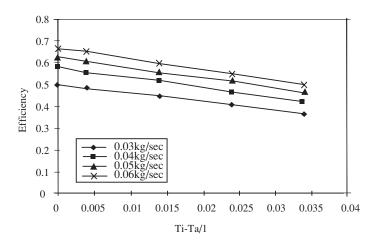


Figure 8. Efficiency curve for double flow

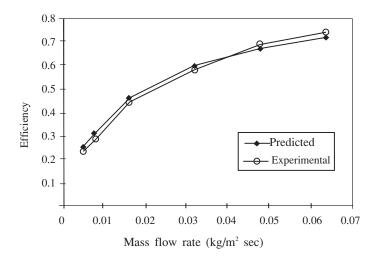


Figure 9. Efficiency variation with mass flow rate

Table 2. Efficiency equations, $F_{\mbox{\tiny R}}(\tau\alpha)$ and $F_{\mbox{\tiny R}}U_{\mbox{\tiny L}}$ for double flow

Flow rate (kg/sec)	$F_{\rm R}(aulpha)$	$\mathbf{F}_{\mathbf{R}}\mathbf{U}_{\mathbf{L}}$	Efficiency equation
0.03	0.5030	3.8844	Y = -3.8844 Z + 0.5030
0.04	0.5809	4.6653	Y = -4.6653 Z + 0.5809
0.05	0.6287	4.6501	Y = -4.6501 Z + 0.6287
0.06	0.67	5	Y = -5 Z + 0.67

 $Z = (T_i - T_a)/I$

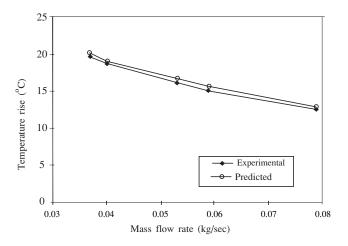


Figure 10. Effect of mass flow rate on temperature rise of double pass solar

Conclusion

A mathematical simulation study is conducted to perdict the thermal performance of flat plate solar collector in single and double flow pass. It is found that double pass operation increases the efficiency of the collector; it is become 4 - 10% more efficient than the single flow mode. The efficiency of both types of flow is a function in mass flow rate; hence the efficiency increases with the increases of flow rate.

The system is able to predict the collector performance for different types of flow according to chosen parameters with design rules that incorporate the human expertise in the field. This technique seems to be very promising and the internet based system can help in sharing and distributing the knowledge.

In the near future the study will investigate the effect of various design and operational parameters on the efficiency, the air temperature increment and the pressure drop in the system by using the developed program. Also the computer program shall encompass the V-groove absorber in single and double pass and the economic aspect for the different types for solar air heaters.

Notations

- A Area of collector that absorbs solar radiation, m²
- $C = 520*(1 0.000051 \beta^2)$
- C_p Specific heat of working fluid, J/kg K
- D_h Hydraulic diameter, m
- e = 0.43*(1 (100/Tpm))
- $f = (1 + 0.089 \, h_w 0.1166 \, h_w \, \epsilon_n) * (1 + 0.07866 \, N)$
- F_R Heat removal factor
- H Collector height, m
- h Fluid heat transfer coefficient, W/m² K
- h_r Radiation heat transfer coefficient, W/m² K
- h_w Wind convective heat transfer coefficient, W/m² K
- I Solar radiation, W/m²
- k Fluid thermal conductivity, W/m K
- k_b Back insulation conductivity, W/m K
- k_e Edge insulation conductivity, W/m K
- L Collector length, m
- m Collector flow rate, kg/sec
- N Number of glass cover
- Nu Nusselt number
- Q_{μ} Rate of useful energy gain, W
- Re Reynolds number
- T_a Ambient air temperature, K
- T_c Cover temperature, K
- T_f Mean fluid temperature, K

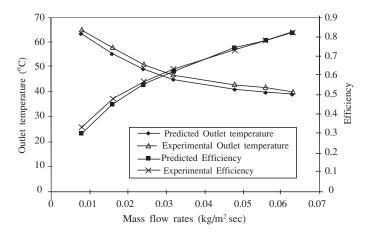


Figure 11. Variation of efficiency and outlet temperature with flow rate

- T_i Fluid inlet temperature, K
- Mean absorber plate surface temperature, K
- T_r Bottom plate temperature, K
- U Overall heat loss coefficient, W/m² K
- U_b Back loss coefficient, W/m² K
- U_e Edge loss coefficient, W/m² K
- U_t Top loss coefficient, W/m² K
- V Air velocity, m/s
- W Collector width, m
- x_b Back insulation thickness, m
- x_e Edge insulation thickness, m

Greek Symbols

- β Collector tilt angle, degree
- ε_p Emittance of absorber plate
- ε_c Emittance of glass cover
- σ Stephen-Boltzman constant
- au Solar transmittance of glazing
- α Solar absorption of collector plate
- μ Dynamic viscosity, Pa.sec

Subscripts

- 1 in upper duct
- 2 in lower duct
- a ambient
- b back
- c cover

- e edge
- f fluid
- h hydraulic
- p plate
- u useful

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