Analytical and Experimental Studies on the Thermal Efficiency of the Double-Pass Solar Air Collector with Finned Absorber

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Abstract: Problem statement: The design of suitable air collectors is one of the most important factors controlling the economics of the solar drying. Air type collectors have two inherent disadvantages: Low thermal capacity of air and low absorber to air heat transfer coefficient. Different modifications are suggested and applied to improve the heat transfer coefficient between the absorber plate and air. These modifications include the use of extended heat transfer area, such as finned absorber. Approach: The efficiency of the solar collector has been examined by changing the solar radiation and the mass flow rate. An analytical and experimental study to investigate the effect of mass flow rate and solar radiation on thermal efficiency were conducted. The theoretical solution procedure of the energy equations uses a matrix inversion method and making some algebraic rearrangements. Results: The average error on calculating thermal efficiency was about 7%. The optimum efficiency, about 70% lies between the mass flow rates 0.07-0.08 kg sec\(^{-1}\). The thermal efficiencies increase with flow rate and it increase about 30% at mass flow rate of 0.04-0.08 kg sec\(^{-1}\). Conclusion: The efficiency is increased proportional to mass flow rate and solar radiation and .the efficiency of the collector is strongly dependent on the flow rate.

Key words: Double-pass solar air collector, finned absorber, thermal efficiency

INTRODUCTION

Depleting of fossil and gas reserves, combined with the growing concerns of global warming, has necessitated an urgent search for alternative energy sources to cater to the present day demands. An alternative energy resource such as solar energy is becoming increasingly attractive. Solar energy is a permanent and environmentally friendly source of renewable energy. The use of non-renewable fuels, such as fossil fuel has many side effects. Their combustion products produce pollution, acid rain and global warning.

Solar drying system is one of the most attractive and promising applications of solar energy systems in tropical and subtropical countries. The technical development of solar drying systems can proceed in two directions. Firstly, simple, low power, short life and comparatively low efficiency-drying system. Secondly, high efficiency, high power, long life expensive drying system (Fudholi et al., 2010).
capacity of air and low absorber to air heat transfer coefficient. Different modifications are suggested and applied to improve the heat transfer coefficient between the absorber plate and air (Supranto et al., 2009). These modifications include the used an extended heat transfer area, such as absorber with fins attached, V-corrugated collector and collector with porous media.

Sopian et al. (2009) studied on the thermal efficiency with and without porous media of the double-pass solar air collector for various operation conditions. They concluded that typical thermal efficiency of the double-pass solar air collector with porous media is about 60-70%. Pradhapraj et al. (2010) reviewed on porous and non porous flat plate air collector with mirror enclosure. They discussed the performances of porous and non-porous absorber plates, the possible methods of finding out air leakages and the methodology adopted for the performance and efficiency calculations.

Various designs of solar collectors have been the subject of many theoretical and experimental investigations. Helal et al. (2010) studied energetic performances of an integrated collector storage solar water heater. The systems shows little cost, simplicity and simpler to be installed on the building roof. Prasad et al. (2010) studied experiment analysis of flat plate collector and comparison of performance with tracking collector. Dammak et al. (2010) optimized hybrid of flat plate collector with a bubble pump for absorption-diffusion cooling systems. Reda (2010) studied the stability of luminescent solar collector prepared by sol-gel spin coating method using Ponceau 2R.

In the present study, the main concern is to study theoretically and experimentally on the thermal efficiency of the double-pass solar air collector with finned absorber.

**MATERIALS AND METHODS**

Figure 1 shows the cross section of the double-pass collector with the finned absorber. The collector consists of the glass cover, the insulated container and the black painted aluminum absorber. The size of the collector is 1.2 m wide and 2.4 m long. In this type of collector, the air initially enters through the first channel formed by the glass covering the absorber plate and then through the second channel formed by the back plate and the finned absorber. The size of the fins is 6 cm wide and 20 cm long. The fins have area of 1.512 m².

Figure 2-3 show the schematic of the indoor testing facility and the experimental setup of the double-pass solar collector. The simulator uses 45 halogen lamps, each with rated power of 500 W.
The maximum average radiation of 788 W/m² can be reached. Dimmers are used to control the amount of radiation that the test collector received. A Data acquisition recorder is used to record the required parameters such as the temperatures (inlet, outlet, absorber, glass cover and ambient) and intensity of the solar simulator. A type-T thermocouple is used in this experiment.

A pyranometer is used to measure the solar intensities. A vane anemometer probe is used to measure linear velocity of air flow. The lighting control of the simulator has been adjusted to obtain the required radiation levels. The solar collector has been operated at varying air mass flow rate and radiation conditions.

**Theoretical analysis:** Figure 4 shows the various heat transfer coefficients of double-pass solar collectors with finned absorbers. Figure 5 shows energy balance for each element of the fin with a height (dz) shown in Fig. 5 can be expressed as:

\[ Q_z = Q_{z+dz} + h_{sf}(2dz)(T_{in} - T_{i1}) \]  

Where:

\[ Q_{i1} = 2\dot{m}C(T_{i1} - T_{i1})/wL \]  
\[ Q_{z} = 2\dot{m}C(T_{i2} - T_{i2})/wL \]  

By making an energy balance for a differential element of a fin with a height (dz) shown in Fig. 5 can be expressed as:

\[ Q_z = Q_{z+dz} + h_{sf}(2dz)(T_{in} - T_{i1}) \]  

Where:

\[ Q_s = -kA_{sf}\frac{dT_{in}}{dz} \]  
\[ Q_{z+dz} = -kA_{sf}\left(\frac{dT_{in}}{dz}\right)_{z+dz} = -kA_{sf}\left(\frac{dT_{in}}{dz} + \frac{d^2T_{in}}{dz^2}dz\right) \]  

Substituting Eq. 9 and 10 in Eq. 8, we get:

\[ \frac{d^2T_{in}}{dz^2} - \frac{2h_{sf}l}{k_{sf}A_{sf}}(T_{i1} - T_{i1}) = 0 \]  

For simplicity, let:
Then Eq. 11 becomes:

\[
\frac{d^2\theta}{dz^2} - M^2 \theta = 0
\]  

(14)

It is a linear homogeneous, second order differential equation. The general solution for Eq. 14 is:

\[
\theta = \lambda_1 \cosh M(H-z) + \lambda_2 \sinh M(H-z)
\]  

(15)

where, \(\lambda_1\) and \(\lambda_2\) are constants and depend on the boundary conditions:

for \(z=0\), \(T_p-T_{f1}=0\)  

(16)

for \(z=H\), \(\frac{d\theta}{dz} = 0\)  

(17)

from boundary conditions, Eq. 15 can be written as:

\[
\theta = \frac{\theta_0}{\cosh MH} \cosh M(H-z)
\]  

(18)

The fin heat transfer rate from the fin base:

\[
Q_{as} = Q_{z=0} = -kA_s \left( \frac{dT_{as}}{dz} \right)_{z=0}
\]  

(19)

\[
= (2k_pA_{d}l_{h0})^{1/2} \left( T_p - T_{f1} \right) \tanh MH
\]

(19)

Theoretical solution procedure: The theoretical model assumes that for a short collector, the temperatures of the wall surrounding the airflow are uniform and temperatures of the airflow vary linearly along the collector. For the short collectors, the mean air temperature is then equal to the arithmetic mean (Choundhury et al., 1995).

Where:

\[
T_{i1} = \frac{T_{i1o} + T_i}{2}
\]  

(20)

\[
T_{i2} = \frac{T_{i2o} + T_{i2b}}{2}
\]  

(21)

In general, the above Eq. 1-5 can be presented in a 5×5 matrix form. The above matrices may be displayed as (Fudholi et al., 2011)

\[
[A][T] = [B]
\]  

(22)

Where:

\[
S_1 = U_s T_s + \alpha_s I
\]  

(24)

\[
S_2 = -(2mC/wL) T_i
\]  

(25)

\[
S_3 = \alpha_s \tau_s I
\]  

(26)

\[
S_4 = -S_5
\]  

(27)

\[
S_5 = -T_i U_b
\]  

(28)

\[
S_6 = h_i + h_{mg} + U_i
\]  

(29)

\[
S_7 = -[h_i + h_{mg} + (2mC/wL)]
\]  

(30)

\[
S_8 = h_2 + h_i + h_{mg} + h_{pb} + \frac{N}{A_l} \left( 2kA_dL_{h1} \right)^{1/2} \tanh MH
\]  

(31)

\[
S_9 = -[h_i + N \left( 2kA_dL_{h1} \right)^{1/2} \tanh MH]
\]  

(32)

\[
S_{10} = -S_9
\]  

(33)
Incorporating these relations in Eq. 2 and 4 and making some algebraic rearrangements, the mean temperature vector may be determined with Excel by matrix inversion form.

$$[T] = [A]^{-1}[B]$$

The newly computed temperatures are then compared with the previously assumed ones and computed is repeated until all consecutive mean temperatures differ by less than 0.01°C. In the present case, a sufficient convergence for $T_g$, $T_{f1}$, $T_p$, $T_{f2}$ and $T_b$ are achieved in 4-6 iterations.

RESULTS AND DISCUSSION

The physical properties of air are assumed to vary linearly with temperature (°C) (Alfegi et al., 2009)

specific heat:

$$C_p = 1.0057 + 0.000066(T - 27)$$

density:

$$\rho = 1.1774 - 0.00359(T - 27)$$

thermal conductivity:

$$k = 0.02624 + 0.0000758(T - 27)$$

viscosity:

$$\mu = [1.983 + 0.00184(T - 27)]10^{-5}$$

The useful gain by the solar collector to solar radiation with values of fluid inlet and outlet temperature and the fluid mass flow rate is given as follows:

$$Q_u = \dot{m}C(T_s - T_i)$$

where, $C$ is the specific heat of the fluid. The efficiency of the collector is given by:

$$\eta = \frac{Q_u}{A_i I} = \frac{\dot{m}C(T_s - T_i)}{A_i I}$$

$$\eta = F_o (\tau a) - F_o U_l \frac{(T_e - T_i)}{S}$$

Where:

$A_i$ = The area of collector
$I$ = The solar radiation incident on the collector
$F_o$ = Heat removal factor referred to outlet temperature of solar collector
$U_l$ = Collector total loss coefficient

The convective heat transfer coefficients are computed accordingly, such as:

$$h_w = 2.8 + 3.3V$$

where, $h_w$ is the convection heat transfer coefficient due to wind and $V$ is the wind velocity:

$$h_{mn} = \frac{\sigma(T_s^2 + T_o^2)(T_s + T_o)}{\frac{1}{e_p} + \frac{1}{e_g} - 1}$$

$$h_{rg} = \frac{\sigma (T_s^2 + T_o^2)}{T_s^2 - T_o^2}$$

there $T_s$ is the sky temperature:

$$T_s = 0.0552(T_a)^{1.5}$$

$$U_l = \left(\frac{1}{h_w + h_{mn}}\right)^{-1}$$

The convective heat transfer coefficients are calculated using following relations:

$$h = \frac{k}{D_h Nu}$$

Where:

$Nu$ = Nusselt number
$D_h$ = The equivalence diameter of the channel

Nusselt number for laminar flow region (Re<2300), transition flow region (2300<Re<6000) and
turbulent flow region respectively are (Basria et al., 2007; Fudholi et al., 2011):

\[
\text{Nu} = 5.4 + \frac{0.00190 \left[ \text{RePr} \left( \frac{D_h}{L} \right) \right]^{0.71}}{1 + 0.00563 \left[ \text{RePr} \left( \frac{D_h}{L} \right) \right]^{0.37}}
\]

(51)

\[
\text{Nu} = 0.116 \left( \text{Re}^{0.5} - 125 \right) \text{Pr}^{0.14} \left[ 1 + \left( \frac{D_h}{L} \right) \right]^{0.17} \left( \frac{U}{\text{Pr}} \right)^{0.14}
\]

(52)

\[
\text{Nu} = 0.018 \text{Re}^{0.4} \text{Pr}^{0.4}
\]

(53)

Where:

\( \text{Pr} = \text{Prandtl} \)

\( \text{Re} = \text{The Reynolds number:} \)

\[
\text{Re} = \frac{\dot{m} D_h}{\lambda_\mu}
\]

(54)

\[
D_h = \frac{4wd}{2(w + d)}
\]

(55)

The collector efficiencies increase with flow rate, efficiency increase is about 30% at mass flow rate of 0.04-0.08 kg sec\(^{-1}\). The optimum efficiency is about 70% lies between the mass flow rates 0.07-0.08 kg sec\(^{-1}\).

To determine the physical characteristics of the collector, one represents effectiveness with efficiency curve, i.e. efficiency versus the reduced temperature parameters \((T_o - T_a)/S\) in Fig. 9-11. As seen in the figure shows the efficiency curve decrease with increase of the reduced temperature parameters. The curve obtained is a straight line. It will results where the slope is equal to \(F_o U_L\) and the y-intercept is equal to \(F_o(\tau a)\). The respective efficiency equation and the physical characteristic of the collector are presented in Table 1-2.

The model is validated by comparing with the experimental. It can be clearly seen from figures or table that the error on calculating the thermal efficiency are about 6.47%, 6.84% and 6.23% for \(I = 423 \text{ W m}^{-2}\), \(I = 572 \text{ W m}^{-2}\) and \(I = 788 \text{ W m}^{-2}\), respectively. The model gives fair prediction with an average error of 6.5%. This may be due to error in the initial conditions, as well as the thermal conductivity of the fin material.

The effect of solar radiation on the efficiency of experimental study is shown in Fig. 15.

**Table 1:** Efficiency, loss factor and efficiency equation of double-pass solar air collector of theoretical study (from Fig. 9, 11 and 13)

| \(S \) (\text{W m}^{-2}) | \(F_o(\tau a)\) | \(F_o U_L\) | Efficiency equations | \(R^2\) |
|-------------------------|----------------|-------------|----------------------|-------|
| 423                     | 0.67           | 2.3         | \(y=-2.3x + 67.4\)   | 0.95  |
| 572                     | 0.71           | 2.6         | \(y=-2.6x + 70.9\)   | 0.95  |
| 788                     | 0.74           | 2.9         | \(y=-2.9x + 73.6\)   | 0.96  |

**Table 2:** Efficiency, loss factor and efficiency equation of double-pass solar air collector of experimental study (from Fig. 10, 12 and 14)

| \(S \) (\text{W m}^{-2}) | \(F_o(\tau a)\) | \(F_o U_L\) | Efficiency equations | \(R^2\) |
|-------------------------|----------------|-------------|----------------------|-------|
| 423                     | 0.73           | 4.1         | \(y=-4.1x + 72.7\)   | 0.85  |
| 572                     | 0.78           | 5.8         | \(y=-5.8x + 77.8\)   | 0.95  |
| 788                     | 0.84           | 6.9         | \(y=-6.9x + 84.1\)   | 0.97  |

Fig. 6: Variation of efficiency with mass flow rate for \(I = 423 \text{ W m}^{-2}\)

Fig. 7: Variation of efficiency with mass flow rate for \(I = 572 \text{ W m}^{-2}\)

Fig. 8: Variation of efficiency with mass flow rate for \(I = 788 \text{ W m}^{-2}\)
Fig. 9: Efficiency versus (To-Ta)/\(I\) of theoretical for \(I = 423\) W m\(^{-2}\)

Fig. 10: Efficiency versus (To-Ta)/\(I\) of experimental for \(I = 423\) W m\(^{-2}\)

Fig. 11: Efficiency versus (To-Ta)/\(I\) of theoretical for \(I = 572\) W m\(^{-2}\)

Fig. 12: Efficiency versus (To-Ta)/\(I\) of experimental for \(I = 572\) W m\(^{-2}\)

Fig. 13: Efficiency versus (To-Ta)/\(I\) of theoretical for \(I = 788\) W m\(^{-2}\)

Fig. 14: Efficiency versus (To-Ta)/\(I\) of experimental for \(I = 788\) W m\(^{-2}\)

Fig. 15: The effect of solar radiation on efficiency

At the solar radiation at 423-788 W m\(^{-2}\) and mass flow rate 0.087 kg sec\(^{-1}\) increase efficiency is about 11%.

**CONCLUSION**

Performance curves of double-pass solar air collector with finned absorber in lower channel have been obtained. These include the effects of mass flow rate and solar radiation on efficiency of the solar collector. The efficiency of the collector is strongly dependent on the flow rate. It increases with flow rate. The optimum efficiency is about 70% lies between the mass flow rates 0.07-0.08 kg sec\(^{-1}\). The average error
on calculating the thermal efficiency is about 7%. The efficiency is increased proportional to mass flow rate and solar radiation.

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