Steady state simulation on combination of glazed and unglazed flat plate solar collectors for air heating

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Abstract. Unglazed flat plate solar collectors usually have lower costs and can have an advantage in situations where heat loss is low. In this study a steady state simulation of the combination of glazed and unglazed flat plate solar collectors for air heating was performed for various conditions of heat transfer coefficient (3, 12 and 21 W/m²°C), inlet temperature (30, 45, 60 and 75°C) and solar irradiation (350, 600 and 850 W/m²). The efficiency of combined system of the glazed and unglazed solar collector area was better compared to the system which consists entirely of glazed at a low coefficient of heat transfer (3 W/m²°C), except for high inlet temperature (more than 60°C) and low solar irradiation (350 W/m²°C). For the combination of 6m² unglazed and 9m² glazed solar collector, the higher percentage of flow rate through unglazed collector slightly increases efficiency except at high inlet temperatures (> 60°C) for high heat transfer coefficient (12 and 21 W/m²°C) and at low solar irradiation (350 W/m²). For air heating operations which are maintained in equilibrium conditions the temperatures of the air produced were slightly higher than that of the collector inlet.

1. Introduction
Solar energy is an abundant energy source especially in tropical areas. By using simple conversion equipment (such as flat plate solar collector) the temperature produced is usually very suitable for agricultural products drying purposes. Many flat plate collector-based dryers directly use air as drying medium, such as those developed by Karsli [1] or Abene et al [2]. Application of this collector type for high moisture content products can still be performed, but the daily solar irradiation level must be high during the drying process [3]. However in general, solar dryers developed rarely use adequate collector size and heat storage system so that in unfavorable weather the drying process cannot be carried out continuously. Especially, in solar dryers that rely on natural convection, the drying rate is very dependent on weather conditions so that the products’ quality produced may be poor [4]. The continuity of drying process can only be achieved when the process is supported (hybrid) with relatively large amounts of additional fuel.

The use of water as a heat collecting fluid from collectors makes the installation more flexible. In addition, water can be used for heat storage media. Fudholi et al. [5] concluded that solar drying systems with water-based solar collectors presented higher performance and stable output temperatures. The excess solar energy captured can be stored and then utilized at lower temperatures for a longer time. However, because drying of agricultural products generally uses heated air as a drying medium, the heat from the water must be transferred to the drying air. Nitipraja and Nelwan [6] tested the use of water as a thermal storage system for rice drying and this system was very potential to be developed further.

In general, solar collectors have transparent covers. The use of transparent covers on solar collectors has advantages as well as disadvantages. The main advantage is reducing heat loss from the
absorber to the environment. At high temperature difference of the collector and the ambient, the performance of the glazed type flat collector is usually better. The disadvantage is that the portion of solar radiation reflected to the surrounding from the collector is larger. In addition, the reduction of transmissivity of the transparent cover at a sufficiently large incident angle (> 45°) is quite significant [7]. Therefore at low difference temperature levels and lower convection heat transfer coefficients, the unglazed flat collector type could provide benefits because the amount of radiation that can be transmitted to the absorber is greater. Some test results show that unglazed type collectors have quite good performance [8-11].

Because each type collector has advantages and constraints, the combination of the two types could provide advantages in certain conditions of solar irradiation, the coefficient of heat transfer and ambient air temperature. Nelwan et al. [12] have designed and tested semi-closed flat plate solar collectors with air as working fluids for various percentages of glazed parts at various slope levels and obtained a collector with 60% glazed provided the best performance.

Water is used as a heat collecting fluid and the heat is subsequently transferred to the air through a heat exchanger. The objective of this study is to simulate a combination of glazed and unglazed flat plate solar collector systems and a combination of flow in the collector systems for various of solar irradiation, convective coefficient and water inlet temperature magnitudes for air heating under steady state conditions.

2. Material and Method

2.1. Collector design

Figure 1 shows a cross-section schematic of the solar collector design for glazed type solar collector. The glazed and unglazed type flat plate solar collectors investigated in this study differ only in the use of transparent covers. The absorber was made of a 1.2 mm black-painted aluminum plate with a size of 3000 mm x 950 mm. The water pipes used were copper pipes with inner diameter of 11.8 mm and outer diameter of 12.8 mm. The pipe was attached to the absorber plate using thermal glue and was clamped where the distance between the pipes was 15 cm. At the bottom a plate was used to protect the adhesive from the friction of the insulator used. While the glazed collector uses 1.2 mm width polycarbonate as the transparent cover, the unglazed type did not use transparent cover at all.

![Figure 1. Schematic diagram of a cross section of a glazed solar collector design.](image)

2.2. Model on the collector

2.2.1. Heat loss coefficient. The model used for simulations on collectors was based on models from Duffie and Beckman [13]. The thermal balance in the collector included solar irradiation and heat exchange from the top, bottom and sides. However, because the area of the side wall was much lower than the top and bottom, heat loss from the side was ignored.

The convective heat transfer coefficient from the top ($U_{T1}$) of the collector through convective mode can be expressed in the following equation:

$$U_{T1} = \frac{1}{\frac{C}{T_{pl}} \times \left( T_{pl} - T_a \right) \left( N_{cover} + F \right)} + \frac{1}{h_{wind}}$$

(1)
where the coefficients F, C and E are
\[
F = \left(1 + 0.089 \times h_{\text{wind}} - 0.1166 \times h_{\text{wind}} \times \varepsilon_{\text{plat}}\right) + \left(1 + 0.007866 \times N_{\text{Cover}}\right)
\]
\[
C = 520 \times \left(1 - 0.000051 \times 45^2\right)
\]
\[
E = 0.43 \times \left(1 - \frac{100}{T_{\text{plat}}}\right)
\]

\(N_{\text{cover}}\) denotes the number of transparent covers, \(T_{\text{plat}}\) denotes the absorber temperature (K), \(T_a\) denotes the ambient temperature (K), \(\varepsilon_{\text{plat}}\) denotes the emittance of absorber plate, \(h_{\text{wind}}\) denotes the surface heat transfer coefficient due to the wind (W/m\(^2\)K).

The radiative heat transfer coefficient from the top \((U_{T2}\) in W/m\(^2\)K), can be expressed as:
\[
U_{T2} = \frac{\alpha \left(T^2_{\text{plat}} + T^2_a\right)}{B_1 + B_2 - N_{\text{cover}}}
\]

where
\[
B_1 = \frac{1}{\varepsilon_{\text{play}} + 0.00591 N_{\text{Cover}} \times h_{\text{angin}}}
\]
\[
B_2 = \frac{2 N_{\text{cover}} + F - 1 + 0.133 \varepsilon_{\text{plat}}}{\varepsilon_{\text{cover}}}
\]

\(\varepsilon_{\text{cover}}\) denotes the emittance of the transparent cover.

The overall heat transfer coefficient from the bottom of the collector \((U_B\) in W/m\(^2\)K), is expressed as:
\[
U_B = \frac{1}{1/(k/\Delta x) + 1/h_{\text{wind}}}
\]

where \(k\) and \(\Delta x\) is thermal conductance (W/mK) and thickness (m) of the collector insulation.

The overall heat transfer coefficient \((U_B\) in W/m\(^2\)K) of the collector is the total of all heat transfer coefficients, i.e.: 
\[
U_L = U_{T1} + U_{T2} + U_B
\]

### 2.2.2. Useful energy

The useful energy \((Q_u)\) of the collector is expressed as:
\[
Q_u = F' \left[(\pi \alpha) I - U_L (t_f - t_a)\right]
\]

where F' is the efficiency factor defined as
\[
F' = \frac{1}{U_L} \left(L + D \left(\frac{1}{U_L D + L \eta F} + \frac{1}{C_p} + \frac{1}{\pi D_i h_{\text{fin}}}\right)\right)
\]

\(L\) and \(D\) are the distance (m) between the pipes and the pipe diameter, respectively. \(C_p\) is the thermal resistance at the junction (W/mK), \(h_{\text{fin}}\) is the inner wall coefficient of heat transfer (W/m\(^2\)K) and \(D_i\) is the inner diameter (m), \(t_f\) is the temperature of the incoming fluid (K), \(t_a\) is the ambient air temperature (K) and \(\eta F\) is the efficiency of the fin.

By definition the loss factor \((F_R)\) is as follow:
\[
F_R = \frac{\dot{m} c_p}{A_{CT} U_L} \left(1 - e^{-\left(\frac{U_L A_{CT} F'}{\dot{m} c_p}\right)}\right)
\]

then the useful energy can be expressed as:
\[
Q_u = F_R A_{CT} \left[(\pi \alpha) I - U_L (t_{\text{fin}} - t_a)\right]
\]

where I is the solar irradiation (W/m\(^2\)) and \(A_{CT}\) is the collector area (m\(^2\)).
The efficiency of the solar collector ($\eta$) is the ratio between useful energy and radiation received or expressed as:

$$\eta = \frac{Q_u}{I_{CT}}$$

(8)

2.3. Simulation

Figure 2a shows the schematic of water flow through collectors I to V. In order to simulate the combination of glazed and unglazed collectors, the scenarios in table 1 was used.

**Table 1. Scenario for simulation of combination glazed and unglazed collector**

| Scenario | Unglazed | Glazed |
|----------|----------|--------|
| 1        | I        | II-V   |
| 2        | I-II     | III-V  |
| 3        | I-III    | IV-V   |
| 4        | I-IV     | V      |
| 5        | -        | I-V    |
| 6        | I-V      | -      |

For simulation of flow combination through glazed and unglazed collector, the collector system was arranged in schematic shown in figure 2b where collector I and II are the unglazed collectors while collector III-V are the glazed collectors. Valve settings of $K_1$, $K_2$, and $K_3$ was used to combine the magnitude of the flow rate through the two types of collectors, which is shown in table 2.

Simulations were carried out to determine the temperature of water and air, collector efficiency, the rate of useful energy, as well as the rate of heat transfer to the air. The parameters varied in the simulation include water inlet temperature, convection heat transfer coefficient and solar irradiation.

The air flow rate was 0.25 kg/s the area of each collector was 3 m$^2$ and the transmittivity product cover-absorptivity plate was 0.8. The inlet water temperatures of the source used were 30, 45, 60 and 75°C, the heat transfer coefficients were 3, 12 and 21 W/m$^2$·°C and the solar irradiations were 350, 600 and 850 W/m$^2$.

The rate of energy for heating the air ($Q_{ud}$ in kW) by a heat exchanger is expressed as:

$$Q_{ud} = C_{min}(t_o - t_s)\varepsilon$$

(9)

$c_{min}$ is the lower heat capacity rate (kW/K) of fluid (in this case: air) passing through the heat exchanger, and to is water outlet temperature (K), and the equation for the effectiveness of the heat exchangers used is

$$\varepsilon = 1 - \exp\left(\frac{-N\cdot C\cdot n}{C\cdot n} - 1\right)$$

(10)

where coefficient n and C are respectively:

$$n = NTU^{-0.22}$$

$$C = \frac{C_{min}}{C_{max}}$$

NTU is the number of transfer unit, $C_{max}$ are the higher heat capacity rate (W/K) of fluid passing through the heat exchanger.
Figure 2. Collector system arrangement for: (a) glazed and unglazed area combination, (b) flow rate combination through glazed and unglazed collector.

### Table 2. Scenario for simulation of flow rate combination through glazed and unglazed collector.

| Scenario | Unglazed (kg/s) | Glazed (kg/s) |
|----------|----------------|--------------|
| 1        | 0.05           | 0.2          |
| 2        | 0.1            | 0.15         |
| 3        | 0.15           | 0.1          |
| 4        | 0.2            | 0.05         |
| 5        | 0.25           | 0            |
| 6        | 0              | 0.25         |

3. Results and Discussion

3.1. Variation in the area of the collector combination

Figure 3 shows the simulation results for the average temperature of the exit water, useful energy, heat transferred to the air and collector efficiency. The ambient air temperature was assumed to be 28°C. At the heat transfer coefficient of 3 W/m²·°C, the collector efficiency becomes lower at a greater percentage of the glazed solar collector area.
Figure 3. Combination of Glazed-Unglazed collectors at $h=3\,\text{W/m}^2\cdot\text{oC}$, inlet temp (a) inlet temp =30$^\circ$C and 45$^\circ$C and (b) 60$^\circ$C and 75$^\circ$C (c) Combination of Glazed-Unglazed collectors at $h=21\,\text{W/m}^2\cdot\text{oC}$, inlet temp =30$^\circ$C and 45$^\circ$C; (d) Combination of Glazed-Unglazed collectors at $h=21\,\text{W/m}^2\cdot\text{oC}$, inlet temp =60$^\circ$C and 75$^\circ$C.

At the low heat transfer coefficient, the use of transparent covers only slightly affected the heat loss compared solar irradiation reduction due to the transparent covers. However, at high inlet
temperatures (> 60°C), the greater percentage of glazed solar collector area will cause the collector’s efficiency to increase to a low level for low solar irradiation (350 W/m²) (Fig. 3b).

The overall efficiency of the glazed collector is significantly higher at medium and high heat transfer coefficient. The more percentage area of unglazed collector, the lower the efficiency, especially for high inlet temperatures. At low inlet temperatures (30°C), solar irradiation does not affect the collector efficiency, but at high inlet temperatures low solar irradiation provided the negative efficiency value. This means that the heat loss from the collector is greater than that entering the collector. In general, for high heat transfer coefficient, inlet temperature is the dominant factors of collector efficiency. In addition, the use of a cover clearly provides a benefit where the efficiency of the whole collector of glazed is significantly higher than that of the collector using a combination of glazed and unglazed.

3.2. Combination of flow rates

Figure 4 show that the results of simulation showed that the combination of flow rates provided a slight difference in efficiencies. There is a tendency that the efficiency of the collector will increase to the percentage of the rate of flow through the unglazed collector. In this simulation, the total flow of water entering the unglazed collector would enter the glazed collector, so increasing the percentage of the flow rate through the unglazed collector increased the total heat from the collector. However, in conducting operation at conditions of high inlet temperatures (> 60°C), high heat transfer coefficients (12 and 21 W/m²·°C) and low irradiation levels (350 W/m²) (shown in Fig. 4d), the collector efficiency decreased with the percentage of fluid flow through the unglazed collector. This means that flowing the water through the previous unglazed collector provided more heat loss.
3.3. The air temperature in equilibrium operation

Table 3 shows some system performance parameters for the operation of the collector system for heating air in equilibrium conditions, i.e. the heat entering the collector system is the same as the heat used for air heating at several levels of inlet temperature (T_{in}). When the water inlet temperature was higher, the airflow rate that can be heated was lower. It can be seen that the temperature of the air produced is slightly higher than the inlet temperature. However, when the inlet temperature is high, operations to produce lower air temperatures can still be carried out with the consequence of heat deficit (more heat comes out than incoming heat). The heat that can be utilized while maintaining thermal equilibrium is very significantly decreased at high inlet temperatures.

| T_{in} (°C) | Eff (%)  | m_{air} | T_{out} (°C) | T_{air} (°C) | Q (W)  |
|------------|----------|---------|--------------|--------------|--------|
| 35,0       | 49,23    | 0,28    | 37,471       | 37,34        | 2583,42|
| 40,0       | 45,03    | 0,17    | 42,260       | 42,25        | 2363,95|
| 45,0       | 40,80    | 0,11    | 47,048       | 47,05        | 2142,17|
| 55,0       | 32,30    | 0,06    | 56,621       | 56,62        | 1695,66|

3.4. Operating application

The operation of combined unglazed and glazed collectors in areas with high wind speeds especially at high inlet temperatures still provides significantly lower efficiency compared to the system consisting of all glazed collector. However, in areas with high variations of wind speed and solar irradiation, the combination system could provide advantages.
In order to utilize the advantages, a control system should be implemented to regulate the flow of hot water (as conducted by Hao et al. [14]) based on the simulation results above. Bypassing (flowing the fluid without passing through the collector at all) is often necessary when the use of heat for heating air need to be performed but the potential for solar irradiation collected by the collector is very low.

4. Conclusion
The efficiency of the combination of the glazed and unglazed solar collector was better than that of the solar collector consisting entirely of glazed at a low heat transfer coefficient (3 W/m²·°C), except in conditions at high inlet temperatures (more than 60°C) and low solar irradiation (350 W/m²). For a combination of 6m² unglazed and 9m² glazed solar collectors, increase of the percentage flow rate through unglazed collectors only slightly increased collector efficiency except at high inlet temperatures (> 60°C), high heat transfer coefficient (12 and 21 W/m²·°C) and low irradiation rates (350 W/m²). For air heating operations which are maintained in equilibrium conditions the temperature of the air produced was slightly higher compared to the temperature of the collector inlet.

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