Thermal performance of double-pass counter flow and double-parallel flow solar air heater with V-grooved absorber plate

Ali M. Rasham¹³ and Basim A. R. Al-Bakri²

¹Department of Energy Engineering, University of Baghdad, Al-Jadriya, Baghdad, Iraq.
²Department of Aeronautical Engineering, University of Baghdad, Al-Jadriya, Baghdad, Iraq.
³E-mail: a.rasham@coeng.uobaghdad.edu.iq

Abstract. The performance of double-pass counter flow (model A) and double-parallel flow (model B) solar air heater with V-grooved absorber plate are numerically investigated. The mathematical models are written based on energy balance equations for each element of both models. The matrix inversion method is used to numerically solve the mathematical models for solar intensity \( (I = 350 - 950 \text{ (W/m}^2)) \), ambient temperature \((T_a = 20 \text{ (°C)})\), mass air flow rate range of \((\dot{m} = 0.02 - 0.09 \text{ (kg/s)})\), and opening angle of V-grooved absorber plate of \((\varnothing = 60°)\). The results are presented in terms of: Air temperature rise, useful heat gain, thermal efficiency, and thermo-hydraulic efficiency. Obviously when air mass flow rate increases, model (A) produces the highest air temperature rise, useful heat gain, and thermal efficiency. As air mass flow rate increases, the thermo-hydraulic of model (B) also increases while thermo-hydraulic efficiency of model (A) decreases from air mass flow rate \((\dot{m} = 0.05, 0.06, \text{and} 0.07 \text{ (kg/s)})\) for solar intensities \((350, 650, \text{and} 950 \text{ (W/m}^2))\) respectively. In terms of thermal efficiency, the model (A) is more efficient than that of model (B) by \((26.8470 \%, 27.8890 \%, \text{and} 28.2496 \%)\) for solar intensities \((350, 650, \text{and} 950 \text{ (W/m}^2))\) respectively. In terms of thermo-hydraulic efficiency, model (A) is more efficient than that of model (B) by \((12.1046 \%, 20.6830 \%, \text{and} 25.1215 \%)\) for solar intensities \((350, 650, \text{and} 950 \text{ (W/m}^2))\) respectively.

1. Introduction
The ordinary or flat plate solar air hearts have been investigated in the literature. However only recently different constructing is investigated. The construction of the absorber plate with V-grooved is considered. Astonishing enhancement is achieved due to the large heat transfer area although the project area equal to the collector area. There are two configurations of arranging the V-grooved absorber plate: perpendicular to the air flow as by [1-3], or parallel to the air flow as by [4-6]. [7] presented a comparative study between two types of the absorber plate of solar air collector: flat absorber and V-grooved plate absorber. The comparison was accomplished at the same air passage with the same
pressure drop and the same mass air flow rate per unit cross sectional area of the collector. The heat transfer coefficient for V-grooved absorber plate was high than that for flat absorber plate by about 47-300% due to the flow regime. [8] studied the thermal performance of double-pass solar air collector with v-grooved absorber plate. The simulation results were compared with experimental data from the literature to verify the mathematical model. 7% difference in the thermal efficiency was identified. They also found that some parameters have significant effect on the performance of the collector as: air mass flow rate and inlet air temperature. [9] analytically investigated the thermal performance of double-pass solar air heater with recycling air. Flat and v-grooved absorber plate was examined. They found that the optimal recycle air ratio, mass flow rate, the absorptivity and emissivity where applied at the maximum thermal efficiency. [10] experimentally investigated the thermal performance of flat and v-corrugated absorber plate solar air collector with and without thermal storage material. Wax of different mass was used as thermal storage material. They found that the instantaneous and daily efficiencies increased with mass flow rate increased and outside temperature decreases, the solar air heater could work for a long time after sun set. The daily efficiency increased by 10.8-13.6% when using the wax. [11] investigated the exergetic efficiency of double -pass solar air collector with v-grooved absorber plate. The exergetic efficiency was optimized using four independent variables as: the length between two adjacent glazing’s, height of v-grooved, area of the heater and air mass flow rate. It was found that the temperature difference between sun and the absorber palate has the most significant effect on the internal exergetic loss. [12] experimentally implemented a study of the energy and exergy efficiencies of double-pass solar air collector with flat and V-grooved absorber plate. It was found that the average energy and exergy efficiencies were 43-60% and 6-12% respectively. Higher thermal performance identified for the V-grooved absorber plate as compared with flat absorber plate. [13] experimentally studied the thermal performance of V-grooved and flat plate absorber solar air heater in terms of optical and thermal efficiencies. They utilized three models in the analyses: The Quasi-dynamic model, the dynamic heat transfer model and transfer function model. It was revealed that the optical efficiency was about 15.8% and the thermal efficiency was about 10.7%.

The current research provides a comparative study to evaluate the thermal and thermo-hydraulic performance of solar air heater with V-grooved absorber plate for two configurations double-pass counter flow and double-parallel flow to determine which of the two designs is more efficient for solar energy applications. The comparison of the performance is accomplished in terms of thermal and thermo-hydraulic efficiencies, useful heat gain and air temperature rise to determine which of the two models is feasible for solar energy applications.

2. Thermal analysis of V-grooved absorber plate solar air heater

Figure 1 and Figure 2 show the schematic representation of the key notations of double-pass (model A-counter flow) and double-flow (model B- parallel flow) solar air collector. The following assumptions are considered as: (1) The performance of the solar air heater operates under steady-state, (2) One-dimensional conduction heat transfer is adopted, (3) The solar air heater is well thermally insulated from the edges and bottom side, (4) Fully developed forced convection heat transfer is assumed, (5) There is no air leakage, (6) The shading of V-grooved absorber plate is neglected, (7) The physical properties are assumed to be constant of fluid and materials, and (8) The temperature of the air flow is varied linearly along the heater.

2.1. Thermal analysis of V-grooved solar air heater

2.1.1. Energy balance of double-pass and double-flow V-grooved absorber plate solar air heater.

The energy balance under steady state conditions for the glass cover, air flow in upper duct, the V-grooved absorber plate, air flow in lower duct, and the back plate are written respectively as:

\[ A_g \ h_w(T_g - T_a) + A_g \ h_{rgs}(T_g - T_s) + A_g \ h_{cgfu}(T_g - T_{fu}) = A_g \ l \ \alpha_g + A_a \ h_{rpg} (T_p - T_g) \]  

Here \( A_a = A_c / \sin(\theta/2) \).


\[ A_g \, h_{c\,gf} \, (T_g - T_{fu}) + A_a \, h_{c\,pfu} \, (T_p - T_{fu}) = Q_{fu} = \dot{m} \, C_{fu} \, (T_{fu \, o} - T_{fu \, i}) \]  

(2)

\[ A_a \, h_{c\,pfu} \, (T_p - T_{fu}) + A_a \, h_{r\,pg} \, (T_p - T_g) + A_a \, h_{c\,pfl} \, (T_p - T_{fl}) + A_a \, h_{r\,pb} \, (T_p - T_b) = A_g \, l \, \tau_g \, \alpha_p \]  

(3)

\[ A_a \, h_{c\,pfl} \, (T_p - T_{fl}) + A_b \, h_{c\,bfl} \, (T_b - T_{fl}) = Q_{fl} = \dot{m} \, C_{fl} \, (T_{fl \, o} - T_{fl \, i}) \]  

(4)

\[ A_b \, h_{c\,bfl} \, (T_b - T_{fl}) + A_b \, U_b \, (T_b - T_a) = A_a \, h_{r\,pb} \, (T_p - T_b) \]  

(5)

2.1.2. System description and boundary conditions.

According to the air flow direction, two models of double-pass V-grooved absorber plate solar air heater were used in the present study. In Figure 1 (model A), air enters from the upper duct and is reversed to exit from the lower duct (counter flow). Whereas in Figure 2 (model B), air enters at the beginning of the upper and lower ducts and exits the end of the ducts (parallel flow).

The boundary conditions for model (A) are written as: for upper duct, at \( x = 0 \), \( T_{fu \, i} = T_a \). For the lower duct, at \( x = L \), \( T_{fl \, i} = T_{fu \, o} \). While, the boundary conditions for model (B) are written as: for upper duct, at \( x = 0 \), \( T_{fu \, i} = T_a \). Also, for the lower duct, at \( x = 0 \), \( T_{fl \, i} = T_a \).

Figure 1. Schematic diagram of double-pass counter flow solar air heater (model A).

Figure 2. Schematic diagram of double-parallel flow solar air heater (model B).
Table 1. Specifications of V-grooved absorber plate solar air heater.

| Feature                              | Value        | Feature                              | Value        |
|--------------------------------------|--------------|--------------------------------------|--------------|
| Length of solar air collector        | 2.0 (m)      | Emissivity of glass covers           | 0.94 (–)     |
| Width of solar air collector         | 1.0 (m)      | Transmittance of glass cover         | 0.90 (–)     |
| Height of air duct                   | 0.051 (m)    | Bottom loss coefficient              | 1.0 (Wm⁻². °C) |
| Angle of equilateral triangles       | (60°)        | Wind speed                           | 1.0 (m/s)    |
| Width of absorber plate              | 2.0 (m)      | Inlet air temperature                | 20 (°C)      |
| Absorptivity of absorber plate       | 0.95 (–)     | Insulation thermal conductivity      | 0.025 (Wm⁻¹.K) |
| Emissivity of absorber plate         | 0.90 (–)     | Fan efficiency                       | (70 %)       |
| Emissivity of back plate             | 0.94 (–)     | Motor efficiency                     | (90 %)       |

2.2. The Hydraulic Diameter and Reynolds number

After some mathematical manipulations, the hydraulic diameter for V-shaped corrugation of equilateral triangle absorber plate at any opening angle (∅) can be formulated as:

\[ D_{h-\text{u,l}} = \left[ \frac{2H_d \sin \left( \frac{\sqrt{3} \phi}{2} \right)}{1+\sin \left( \frac{\sqrt{3} \phi}{2} \right)} \right] \]

(6)

For model (B), the mass flow rate is divided equally for upper and lower ducts. Thus, the Reynolds number for the upper or lower triangular ducts for models (A) and (B) are respectively written as:

\[ Re_{\text{u,l}} = \frac{mD_{h-\text{u,l}}}{\mu (A_{t-\text{u,l}})} \]

(7)

\[ Re_{\text{u,l}} = \frac{\left( \frac{\ln}{2} \right) D_{h-\text{u,l}}}{\mu (A_{t-\text{u,l}})} \]

(8)

Here, the area of triangular upper and lower ducts \(A_{t-\text{u,l}}\) is proposed in the present study as:

\[ A_{t-\text{u,l}} = \left( H_d^2 \tan \left( \frac{\sqrt{3} \phi}{2} \right) \times N_t \right) \]

(9)

2.3. Convection Heat transfer coefficients

The coefficients of forced convection heat transfer for the upper and lower triangular ducts of models (A) and (B) are respectively estimated as:

\[ h_{c,\text{gu}} = h_{c,\text{pfu}} = K_a \frac{\mu}{D_{hu}} Nu_{\text{u,fc}} \]

(10)

\[ h_{c,\text{pn}} = h_{c,\text{bfn}} = K_a \frac{\mu}{D_{hi}} Nu_{\text{l,fc}} \]

(11)

The Nusselt numbers in equations (10 and 11) for V-grooved ducts are estimated as:

\[ Nu_{\text{u,l,fc}} = 2.821 + 0.126 Re_{\text{u,l}} \left( \frac{H_d/2}{L} \right), \quad \text{if } \left( Re_{\text{u,l}} < 2800 \right) \]

(12)

\[ Nu_{\text{u,l,fc}} = 1.9 \times 10^{-6} Re_{\text{u,l}}^{0.74} + 225 \left( \frac{H_d/2}{L} \right), \quad \text{if } \left( 2800 < Re_{\text{u,l}} < 10^4 \right) \]

(13)

\[ Nu_{\text{u,l,fc}} = 0.0302 Re_{\text{u,l}}^{0.74} + 0.242 Re_{\text{u,l}}^{0.74} \left( \frac{H_d/2}{L} \right), \quad \text{if } \left( 10^4 < Re_{\text{u,l}} < 10^5 \right) \]

(14)
2.4. The analysis of useful gain and thermal efficiency of V-grooved absorber plate Solar Air Heater

The useful heat gain of model (A) is written as:

$$Q_{us} = \dot{m} C_{fav} (T_{fl o} - T_{fu i})$$ (15)

While the useful energy gain for the upper and lower ducts of model (B) are respectively written as:

$$Q_{us,u} = \dot{m} C_{fu} (T_{fu o} - T_{fu i})$$ (16)

$$Q_{us,l} = \dot{m} C_{fl} (T_{fl o} - T_{fl i})$$ (17)

Finally, the thermal efficiency for the adopted models (A) and (B) is written as:

$$\eta_t = \frac{Q_{us}}{A_c I}$$ (18)

2.5. Thermo-hydraulic analysis of V-grooved absorber plate Solar Air Heater

The power of the fan required to pump an amount of air into the solar air heater is estimated as

$$P_{fan} = \frac{P_{flow}}{\eta_{fan} \eta_{motor}}$$ (19)

The pumping power of air flow is calculated as:

$$P_{flow} = \frac{\dot{m} \Delta p}{\rho}$$ (20)

The pressure drop across the air flow for upper or lower triangular ducts is estimated as [1]:

$$\Delta p_{duct-u,l} = 2 \rho f_{u,l} V_{u,l}^2 \left( \frac{L}{D_{h-u,l}} \right)$$ (21)

The friction factors for upper or lower triangular ducts are calculated as:

$$f_{u,l} = \begin{cases} 13.33 + 0.65 \left( \frac{H_d/2}{L} \right), & (\text{if } (Re_{u,l} < 2800)) \\ 3.2 \times 10^{-4} Re_{u,l}^{0.34} + 2.94 Re_{u,l}^{-0.19} \left( \frac{H_d/2}{L} \right), & (\text{if } (2800 < Re_{u,l} < 10^4)) \\ 0.0733 Re_{u,l}^{-0.25} + 0.51 \left( \frac{H_d/2}{L} \right), & (\text{if } (10^4 < Re_{u,l} < 10^5)) \end{cases}$$ (22)

(23)

(24)

For the $\left(180^\circ\right)$ close return bend, the pressure drop is estimated as:

$$\Delta p_{bend} = 0.5 \rho V_u^2$$ (25)

The thermo-hydraulic efficiency of double-pass V-grooved absorber plate solar air heater for both models estimated as:

$$\eta_{th} = \frac{(Q_{us}-P_{fan})}{A_c I}$$ (26)
2.6. Validation of mathematical model
Several designs of V-grooved absorber plate solar air heater were investigated in the literatures theoretically and experimentally. Most of these experimental works were achieved for single-pass solar air heater. Whereas, some experimental work were carried out for double-parallel flow. A validation for the mathematical model is achieved numerically in the present study. Good agreement was found when comparing the experimental thermal efficiency data determined by [1] with the predicted thermal efficiency from present analysis of model (B) with a percentage error of 1.65% as shown in Figure 3.

![Figure 3](image_url)

**Figure 3.** Comparison of our results with the experimental results of [1].

3. Results
Thermal and thermo-hydraulic performance for two adopted models (A) and (B) of double-pass V-grooved absorber plate solar air heater are studied in the present research. In order to cover the fluctuation in the intensity of solar radiation, three values of solar radiation have been adopted in the present study (J = 350 – 950 (W/m²)). In the present study, the ambient temperature adopted was (T_a = 20 °C), and air mass flow rate was varied from (m = 0.02 – 0.09 (kg/s)). The results were presented in terms of: Air temperature rise versus air mass flow rate, the useful energy gain versus air mass flow rate, the thermal efficiency versus air mass flow rate, the thermo-hydraulic efficiency versus air mass flow rate, and the air temperature in the ducts along the solar air heater. Generally, model (A) produces highest air temperature rise, useful heat gain, thermal efficiency, and thermo-hydraulic efficiency due to the long air passageway. As the intensity of solar radiation increases, the air temperature rise, useful gain, thermal efficiency, and thermo-hydraulic efficiency also increases.

Figure 4 illustrates the air temperature rise of double-pass V-grooved absorber plate solar air heater with the examined range of air mass flow rate. For both models (A) and (B), the air temperature rise increases with the decrease of air mass flow rate and increases of solar intensities (J = 350 – 950 (W/m²)). Due to the longer air passage of model (A) than of model (B), the model (A) shows the highest air temperature rise. The percentage increase in air temperature rise for model (A) compared with model (B) is presented as an improvement factor (I_f) in Table 2. Generally, if the solar radiation falls on the solar heater, the air heating at a small air mass flow rate is greater than it is at a large air mass flow rate. Consequently, the largest of percentage improvement in temperature rise occurs at
lowest air mass flow rate while the lowest percentage improvement occurs at highest air mass flow rate. Clearly, the improvement factor ($I_f$) increases gradually with increases of solar intensity, Table 2.

![Figure 4](image1.png)

**Figure 4.** Air temperature rise $\Delta T$ (°C) versus air mass flow rate $\dot{m}$ (kg/s).

Figure 5 shows the behaviour of useful heat gain of double-pass V-grooved absorber plate solar air heater versus the examined range of air mass flow rate. Obviously, the useful heat gain increases with increasing of air mass flow rate and solar radiation intensities ($I = 350 - 950$ (W/m$^2$)) for both models (A) and (B). Table 2 shows the improvement factor ($I_f$) as percentage of increase in useful heat gain for model (A) compared with model (B). Also, at lowest air mass flow rate the largest percentage improvement in useful heat gain occurs at highest air mass flow rate and vice versa.

![Figure 5](image2.png)

**Figure 5.** Useful gain $Q_{us}$ (W) versus air mass flow rate $\dot{m}$ (kg/s).
Figure 6 displays the behaviour of thermal efficiency versus air mass flow rate. Clearly, the thermal efficiency increases with increasing of air mass flow rate and solar intensities \( (I = 350 – 950 \text{ W/m}^2) \) for both models. From Figure 6, the model (A) produces higher thermal efficiency compared with model (B) and the improvement factor \((I_f)\) of model (A) is recorded in Table 2. At the highest value of air mass flow rate and solar intensity, the thermal efficiency was (73.93 % and 67.32 %) for model (A) and model (B) respectively.

![Figure 6. Thermal efficiency \( \eta_t \) (–) versus air mass flow rate \( \dot{m} \) (kg/s).](image)

Figure 7 presents the behaviour of thermo-hydraulic efficiency of V-grooved absorber plate solar air heater versus air mass flow rate of models (A) and (B) for solar intensities \((I = 350 – 950 \text{ W/m}^2)\). Due to the shortest air passageway and no bend in model (B), the behaviour of thermo-hydraulic efficiency increases with the increase of air mass flow rate and solar intensities \((I = 350 – 950 \text{ W/m}^2)\). In contrast, the behaviour of thermo-hydraulic efficiency of model (A) decreases from air mass flow rate of \((\dot{m} = 0.05, 0.06, \text{ and } 0.07 \text{ kg/s})\) for solar intensities (350, 650, and 950 \text{ W/m}^2) respectively. The improvement factor \((I_f)\) as percentage of increase in thermo-hydraulic efficiency for model (A) compared with model (B) recorded in Table 2.
Thermo-hydraulic efficiency $\eta_{th}(\cdot)$ versus air mass flow rate $m_\dot{\cdot}$ (kg/s).

Table 2. The improvement factor ($I_f$) of model (A) V-grooved absorber plate solar air heater compared with model (B).

| Solar Intensity ($Wm^{-2}$) | ($I_f$ %) in air temperature rise | ($I_f$ %) in useful gain | ($I_f$ %) in thermal efficiency | ($I_f$ %) in thermo-hydraulic efficiency |
|-----------------------------|----------------------------------|--------------------------|-------------------------------|----------------------------------------|
| 350                         | 26.8465                          | 26.8465                  | 26.8470                       | 12.1046                                |
| 650                         | 27.8858                          | 27.8860                  | 27.8890                       | 20.6830                                |
| 950                         | 28.2375                          | 28.2372                  | 28.2496                       | 25.1215                                |

Figure 8 and Figure 9 exhibit the behaviour of average air temperature in upper and lower duct along the solar air heater of models (A) and (B) for solar intensity ($I = 650$ (W/m²)). The average air temperature in the ducts is plotted for lowest and highest mass flow rate ($m_\dot{\cdot} = 0.02$ (kg/s)) and ($m_\dot{\cdot} = 0.09$ (kg/s)) in order to show the air heating effect. For model (A), the air temperature rise were (20.9230, and 5.3190 °C) for air mass flow rate ($m_\dot{\cdot} = 0.02$, and 0.09 (kg/s)) respectively. While the air temperature rise of model (B) were (11.2722, and 4.4479 °C) for air mass flow rate ($m_\dot{\cdot} = 0.02$, and 0.09 (kg/s)) respectively. According to outlet air temperature from Figure 8 and Figure 9, the solar air heater of model (A) is more efficient compared with model (B).
4. Conclusion
The thermal behaviour of double-counter and double-parallel flow V-grooved absorber plate solar air collector was numerically investigated, and the result was compared. In the present study, a comparison between models (A) and (B) were achieved in terms of improvement factor to determine which of the models is more efficient for solar energy applications. The results clearly showed that the model (A) is the best in terms of thermal and thermo-hydraulic performance. Consequently, model (A) is recommended for solar energy applications. The thermal efficiency of model (A) was higher by (26.8470 %, 27.8890 %, and 28.2496 %) compared to model (B) for solar intensities (350, 650, and 950 (W/m²)) respectively. Whereas, model (A) provided (12.1046 %, 20.6830 %, and 25.1215 %) higher thermo-hydraulic efficiency for solar intensities (350, 650, and 950 (W/m²)) respectively. Additionally, the useful heat gain was about (26.8465 %, 27.8860 %, and 28.2372 %) higher for model (A) than that of model (B) for solar intensities (350, 650, and 950 (W/m²)) respectively. While, (26.8465 %, 27.8858 %, and 28.2375 %) increases in air temperature rise of model (A) was recorded compared to model (B) for solar intensities (350, 650, and 950 (W/m²)) respectively.

5. Appendices

| Symbol | Description | Symbol | Description |
|--------|-------------|--------|-------------|
| $A_a$  | Area of absorber plate, $(m^2)$ | $Q_{us}$ | Useful gain of SAH (W) |
| $A_b$  | Area of back plate, $A_b = A_c$, $(m^2)$ | $Re_{u,l}$ | Reynolds number of upper and lower ducts ($-$) |
| $A_c$  | Area of solar air collector $(m^2)$ | $T_a$ | Ambient air temperature ($^\circ$C) |
| $A_g$  | Area of glass cover, $A_g = A_c$, $(m^2)$ | $T_{fu}$ | Arithmetic mean of air temperature in upper duct ($^\circ$C) |
\( A_{D,\text{up}} \) Area of upper and lower ducts \((m^2)\) 

\( C_{f,av} \) Average of air heat capacity \((J/kg.\, K)\) 

\( C_f \) \( u \) heat capacity of air at upper duct \((J/kg.\, K)\) 

\( C_f \) \( l \) heat capacity of air at lower duct \((J/kg.\, K)\) 

\( D_{h,\text{up}} \) Hydraulic diameter of upper and lower ducts \((m^2)\) 

\( f_{u,l} \) Friction factor of upper and lower ducts \((-)\) 

\( H_d \) Height of V-groove absorber plate \((m)\) 

\( h_{c,\text{gf}} \) Forced convection heat loss between glass cover and air in upper duct \((W/m^2.\, K)\) 

\( h_{c,\text{pf}} \) Forced convection heat loss between absorber plate and air in upper duct \((W/m^2.\, K)\) 

\( h_{c,\text{pf}l} \) Forced convection heat loss between absorber plate and air in lower duct \((W/m^2.\, K)\) 

\( h_{c,\text{bf}l} \) Forced convection heat loss between back plate and air in lower duct \((W/m^2.\, K)\) 

\( I \) Solar radiation intensity \((W/m^2)\) 

\( K \) Constant in eq. (25), \((K = 2.2)\) 

\( K_a \) Thermal conductivity of air flow \((W/m.\, K)\) 

\( l_c \) Collector length \((m)\) 

\( \dot{m} \) Mass flow rate \((kg/s)\) 

\( \text{Nu}_{\text{fc}} \) Nusselt number of forced convection \((-)\) 

\( \text{Nu}_{\text{u-fc}} \) Nusselt number of forced convection for air in upper duct \((-)\) 

\( \text{Nu}_{\text{l-fc}} \) Nusselt number of forced convection for air in lower duct \((-)\) 

\( P_{\text{fan}} \) Fan power \((W)\) 

\( P_{\text{flow}} \) Power of air flow in upper and lower ducts \((W)\) 

\( Q_{\text{n}} \) Heat rate of air in lower duct \((W)\) 

\( Q_{\text{fu}} \) Heat rate of air in upper duct \((W)\) 

\( T_{\text{fu}i} \) Air temperature at inlet of upper duct \((^\circ C)\) 

\( T_{\text{fu}o} \) Air temperature at outlet of upper duct \((^\circ C)\) 

\( T_{\text{fl}} \) Arithmetic mean of air temperature in lower duct \((^\circ C)\) 

\( T_{\text{fl}i} \) Air temperature at inlet of lower duct \((^\circ C)\) 

\( T_{\text{fl}o} \) Air temperature at outlet of lower duct \((^\circ C)\) 

\( T_g \) Glass cover temperature \((^\circ C)\) 

\( T_b \) Back plate temperature \((^\circ C)\) 

\( V_{u,l} \) Air velocity in upper and lower ducts \((m/s)\) 

\( W_a \) Width of absorber plate \((m)\) 

\( W_c \) Width of solar collector \((m)\) 

\( \alpha_g \) Absorptivity of the glass covers \((-)\) 

\( \alpha_p \) Absorptivity of absorber plate \((-)\) 

\( \eta_{\text{fan}} \) Fan efficiency \((-)\) 

\( \eta_{\text{motor}} \) Motor efficiency 

\( \eta t \) Thermal efficiency of SAH \((-)\) 

\( \eta_{\text{th}} \) Thermo-hydraulic efficiency of SAH \((-)\) 

\( \tau_g \) Transmittance of the glass covers \((-)\) 

\( \Delta p \) Pressure drop of air through the SAH \((Pa)\) 

\( \Delta p_{\text{ducks}} \) Pressure drop of air through the upper and lower ducts of SAH \((Pa)\) 

\( \Delta p_{\text{bend}} \) Pressure drop of air through the bends of SAH \((Pa)\) 

**Greek symbols**

\( \alpha_g \) Absorptivity of the glass covers \((-)\) 

\( \alpha_p \) Absorptivity of absorber plate \((-)\) 

\( \eta_{\text{fan}} \) Fan efficiency \((-)\) 

\( \eta_{\text{motor}} \) Motor efficiency 

\( \eta t \) Thermal efficiency of SAH \((-)\) 

\( \eta_{\text{th}} \) Thermo-hydraulic efficiency of SAH \((-)\) 

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\( \Delta p_{\text{ducks}} \) Pressure drop of air through the upper and lower ducts of SAH \((Pa)\) 

\( \Delta p_{\text{bend}} \) Pressure drop of air through the bends of SAH \((Pa)\)

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