Modelling, dynamic simulation and parametric studies of
double pass solar air heater for solar drying application

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Abstract. Solar air heater (SAH) is an integral component of many solar energy systems which converts the incident solar irradiation into thermal energy by air flowing through it. The present study addresses the modelling and simulation of dynamic performance of a double-pass SAH and predicts the effect of different geometric and operating parameters on the performance of air heater. Mathematical models are developed from the first principles to simulate the dynamic performance of the air heater. The number of glazings, nature of absorber plate and aperture height are the different geometric parameters and mass flow rate is the operating parameter considered to conduct the parametric study in the present work. Simulation predictions infer that the optimum performance of the air heater is obtained for double glazing V-corrugated absorber geometry when aperture height and mass flow rate are 0.05m and 0.025kg/s, respectively. The peak plate temperature, exit temperature and efficiency of the heater are predicted for optimum condition as 353.47 K, 346.23 K and 85.50%, respectively. Further, the performance of the heater for different mass flow rates is simulated.

1. Introduction

The current situation of rising energy demand and environmental issues necessitates alternatives to the utilisation of non-renewable energy sources that pollute the atmosphere. Solar energy is one of the promising options for solving the conventional energy shortage and environmental issues. One of the efficient methods to utilise solar energy is to transform the solar energy into thermal energy by employing solar air heaters in solar thermal systems. SAHs are typical examples for this. SAHs are used in various applications such as process heating like desalination, crop drying, laundry, prevention of ice formation, rusting of equipment, other drying processes and space heating. Due to lower heat transfer rate between flowing air and absorber plate and high heat loss, the efficiency of SAH is low. As a result, several performance improvement approaches to increase the rate of heat transfer between the absorber plate and the flowing air have been suggested. According to the literature, the SAH performance can be enhanced by varying the absorber plate geometry by adding fins/baffles, V-corrugated surfaces, porous mesh matrix and incorporating artificial roughness to the absorber geometry. Several studies imply that by varying the number of glazing and number of passes also will improve the performance.

Many researchers have modelled different solar air heater configurations and numerically predicted the temperature variations of their components by employing fundamental principles. Karim et al.[1] developed a numerical model to predict the performance of DPSAH with V-corrugated absorber
under steady state condition. Singh et al.[2] numerically studied the thermal performance of a double glazed single pass SAH with wavy fin type absorber geometry under transient conditions. They predicted the maximum outlet temperature of heater as 85.4°C, 73.34°C and 64.9ºC when mass flow rates are 0.005kg/s, 0.0075 kg/s and 0.01kg/s, respectively. Rajesh Kumar et al.[3] formulated a model incorporating herringbone corrugated fins to the absorber plate to enhance the performance and it was observed the thermal efficiency of the air heater improves from 36.2% to 56.6% with a fin pitch of 2.5cm and mass flow rate of 0.026 kg/s. While modelling a single pass SAH with trapezoidal corrugated absorber plate mathematically, Ondieki et al. [4] reported that the system efficiency is higher when channel depth is smaller. The optimum values of efficiencies were obtained when the collector length is between 1 m and 2 m. From the study carried out by Karim and Hawlader [5], V-corrugated absorber geometry is more efficient than flat plate and finned absorber geometry configurations.

After reviewing several numerical simulation studies on different types of SAHs, It was found that the dynamic simulation of DPSAH with V-corrugated absorber geometry was not reported in the literature. The objectives of the current investigations are: (1) modelling and dynamic simulation of thermal performance of DPSAH with V-corrugated absorber geometry and (2) to investigate the impact of various geometrical parameters such as number of glass covers, aperture height as well as operating parameters like mass flow rate on the performance of SAH and to get the configuration for optimum performance.

2. Mathematical model and problem description

2.1 Problem description

Two different configurations of double pass SAH are considered in the current study. Configuration 1 consists of single and double glazing with flat plate as absorber geometry, while configuration 2 consists of similar components except V-corrugated absorber plate replacing flat plate. The schematic representation of a DPSAH with double glazing showing energy interactions is presented in Fig.1. It was assumed that the SAH was provided with proper insulation on right and left sides as well as bottom side below the second pass air flow passage as shown in figure 1. The duct between the absorber plate and lower glass cover allows the air to flow through it. The main aim of using V-corrugated absorber geometry is to increase the heat transfer rate between absorber plate and flowing air.

![Figure 1. Energy interactions in Double pass - double glass SAH](image)

Table 1 and Table 2 represent the physical properties and geometric details, respectively for the system considered for the performance investigation.
Table 1. Physical properties of solar air heater

| Input parameters          | Value      |
|--------------------------|------------|
| Absorptivity of glass covers | 0.1        |
| Transmissivity of glass covers | 0.89       |
| Absorptivity of absorber plate | 0.9        |
| Emissivity of absorber plate | 0.9        |
| Emissivity of glass covers | 0.89       |
| Specific heat of glass covers | 840 kJ/kgK |
| Specific heat of absorber plate | 500 kJ/kgK |
| Density of glass covers   | 2500 kg/m³ |
| Density of absorber plate | 7850 kg/m³ |

Table 2. Geometrical details of solar air heater

| Geometric parameters       | Value  |
|----------------------------|--------|
| Length of the heater       | 1.8 m  |
| Width of the heater        | 0.8 m  |
| Distance between glass covers | 0.02 m |
| Thickness of glass covers  | 4 mm   |
| Thickness of absorber plate | 2 mm   |

2.2 Mathematical model

The following are the assumptions considered in the mathematical modelling [6]:

- Each component of SAH is homogeneous and isotropic.
- The air flow temperature has a significant variation in stream wise direction compared to cross stream direction.
- Variation of relative humidity is neglected.
- No thermal losses from the lateral faces of SAH.
- For long wavelength radiation, the sky is considered as a black body.

The following equation was used to represent the input profile of solar irradiation falling on the heater with time [6].

\[ G(t) = G_{\text{max}} \times \sin \frac{3.14t}{43200}, \quad 0 < t < 43200 \text{ s} \]  

where, \( G_{\text{max}} \) is the maximum solar irradiation at noon.

The radiation profile obtained from the above function is 0 W/m² at 6:00AM and 6:00PM, and a peak of 800 W/m² at 12:00PM. In the current study, the first configuration model of single pass single glazing (type 1) SAH has five components, namely, glass cover, upper air channel, absorber plate, lower air channel and back plate. For the double pass double glazing (type 2) arrangement, two new components, lower glass cover and air gap are added. So, the total number of components present in the model is 5 and 7 for type 1 and type 2 of the first configuration, respectively. Along the stream
wise direction, the SAH is divided into “n” number of nodes. Therefore, the model consist of a total 5xn and 7xn nodes depending upon the case in a configuration considered.

3. Energy balance equation for solar heater components

For a control volume, by neglecting the energy generation term, the transient thermal energy balance equation is written as:

\[
\frac{dE}{dt} = E_{in} - E_{out} \tag{2}
\]

where, \( \frac{dE}{dt} \) is the rate of accumulation of thermal energy in a component, \( E_{in} \) is the sum of the rates of thermal energies entering the component, \( E_{out} \) is the sum of the rates of thermal energies leaving the component.

The following are the energy balance equations written for configuration 1.

**Upper glass cover:**

\[
\rho_{ag}\, V_{gl}\, C_{ag} \frac{dT_{gl}}{dt} = \left[ h_{cag-gu}(T_{ag} - T_{gl}) + h_{cag-gl}(T_{gl} - T_{gu}) + h_{tag-gu}(T_{ag} - T_{gu}) + \alpha_g G \right] W \Delta Z \tag{3}
\]

**Lower glass cover:**

\[
\rho_{gl}\, V_{gl}\, C_{gl} \frac{dT_{gl}}{dt} = \left[ h_{cag-gl}(T_{ag} - T_{gl}) + h_{cag-air}(T_{air1} - T_{gl}) + h_{rgh-gl}(T_{gu} - T_{gl}) + \tau \alpha_G \right] W \Delta Z \tag{4}
\]

**Air gap:**

\[
\rho_{ag}\, V_{ag}\, C_{ag} \frac{dT_{ag}}{dt} = \left[ h_{cag-gu}(T_{gu} - T_{ag}) + h_{cag-gl}(T_{gl} - T_{ag}) \right] W \Delta Z \tag{5}
\]

**Air channel 1:**

\[
\rho_{air1}\, V_{air1}\, C_{air1} \frac{dT_{air1}}{dt} = \left[ h_{cgl-air1}(T_{gl} - T_{air1}) + h_{cp-air}(T_{p} - T_{air1}) \right] W \Delta Z - m_c \rho \frac{dT_{air1}}{dz} \tag{6}
\]

**Absorber plate:**

\[
\rho_{p}\, V_{p}\, C_{p} \frac{dT_{p}}{dt} = \left[ h_{cp-air1}(T_{air1} - T_{p}) + h_{cp-air2}(T_{air2} - T_{p}) + h_{rpg-gl}(T_{gl} - T_{p}) + \tau \alpha G \right] W \Delta Z \tag{7}
\]

**Air Channel 2:**

\[
\rho_{air2}\, V_{air2}\, C_{air2} \frac{dT_{air2}}{dt} = \left[ h_{cp-air2}(T_{p} - T_{air2}) + h_{cb-air2}(T_{b} - T_{air2}) \right] W \Delta Z - m_c \rho \frac{dT_{air2}}{dz} \tag{8}
\]

**Base plate:**

\[
\rho_{b}\, V_{b}\, C_{b} \frac{dT_{b}}{dt} = \left[ h_{cat-o-air2}(T_{air2} - T_{b}) + h_{rpg-b}(T_{p} - T_{b}) + U_b (T_{ae} - T_{b}) \right] W \Delta Z \tag{9}
\]

4. Numerical solution procedure

To find the temperature distribution of all elements, the energy balance equations are discretized using a finite difference scheme and solved. In order to solve the derivative of component temperature with time, forward difference scheme is employed as:

\[
\frac{dT_m}{dt} = \frac{T_{m+1}^{t+\Delta t} - T_{m}^{t}}{\Delta t} \tag{10}
\]

In order to solve the derivative of flowing air changing with position, the backward difference scheme is employed as:

First pass
\[
\frac{dT_{air1}}{dz} = \frac{T_{air1,j} - T_{air1,j-1}}{\Delta z} \\
\text{Second pass:} \\
\frac{dT_{air2}}{dz} = \frac{T_{air2,j} - T_{air2,j+1}}{\Delta z}
\]

4.1 Discretized equations for different components

The following discretized equations are formulated for temperatures of different components using equations (10) to (12).

**Upper glass cover:**
\[
T^{t+\Delta t}_{gul,j} = \frac{T^{t}_{gul,j}}{X_1 \Delta t} + \frac{A_1}{X_1} T^{t+\Delta t}_{air1,j} + \frac{A_2}{X_1} T^{t+\Delta t}_{glu,j} + \frac{A_3}{X_1} T^{t+\Delta t}_{ag,j} + \frac{A_4}{X_1} G^{t+\Delta t}_{r}
\]

**Air gap between glass covers:**
\[
T^{t+\Delta t}_{aglr,j} = \frac{T^{t}_{aglr,j}}{X_2 \Delta t} + \frac{A_5}{X_2} T^{t+\Delta t}_{gul,j} + \frac{A_6}{X_2} T^{t+\Delta t}_{glr,j}
\]

**Lower glass cover:**
\[
T^{t+\Delta t}_{glr,j} = \frac{T^{t}_{glr,j}}{X_3 \Delta t} + \frac{A_7}{X_3} T^{t+\Delta t}_{air1,j} + \frac{A_8}{X_3} T^{t+\Delta t}_{air2,j} + \frac{A_9}{X_3} T^{t+\Delta t}_{glr,j} + \frac{A_{10}}{X_3} T^{t+\Delta t}_{p,j} + \frac{A_{11}}{X_3} G^{t+\Delta t}_{r}
\]

**Air channel 1:**
\[
T^{t+\Delta t}_{air1,j} = \frac{T^{t}_{air1,j}}{X_4 \Delta t} + \frac{A_{12}}{X_4} T^{t+\Delta t}_{air1,j} + \frac{A_{13}}{X_4} T^{t+\Delta t}_{p,j} + \frac{A_{14}}{X_4} T^{t+\Delta t}_{air1,j-1}
\]

**Absorber plate:**
\[
T^{t+\Delta t}_{p,j} = \frac{T^{t}_{p,j}}{X_5 \Delta t} + \frac{A_{15}}{X_5} T^{t+\Delta t}_{air1,j} + \frac{A_{16}}{X_5} T^{t+\Delta t}_{air2,j} + \frac{A_{17}}{X_5} T^{t+\Delta t}_{p,j} + \frac{A_{18}}{X_5} G^{t+\Delta t}_{r}
\]

**Air channel 2:**
\[
T^{t+\Delta t}_{air2,j} = \frac{T^{t}_{air2,j}}{X_6 \Delta t} + \frac{A_{19}}{X_6} T^{t+\Delta t}_{air2,j} + \frac{A_{20}}{X_6} T^{t+\Delta t}_{b,j} + \frac{A_{21}}{X_6 \Delta z} T^{t+\Delta t}_{air2,j+1}
\]

**Base plate:**
\[
T^{t+\Delta t}_{b,j} = \frac{T^{t}_{b,j}}{X_7 \Delta t} + \frac{A_{22}}{X_7} T^{t+\Delta t}_{air2,j} + \frac{A_{23}}{X_7} T^{t+\Delta t}_{p,j} + \frac{A_{24}}{X_7} T^{t+\Delta t}_{ae,j}
\]

The temperature of all the nodes of the different components of the heater is considered to be ambient temperature initially. At every time step, the boundary condition of air at the entry to the heater is equal to inlet air temperature.

The initial condition is,
\[ T_{air}(0) = 294.74 \text{ K}, \] for all the nodes of every component.

The boundary condition is,
\[ T_{air} (I, t) = 294.74 \text{ K} \]

4.2 Performance parameters

The instantaneous daily useful heat gain of the SAH is calculated by[2]:
\[
Q_u(t) = mc_{p,air}(T_{out,f} - T_{int})
\]
The daily thermal efficiency is calculated as[2]:
\[
\eta_{th} = \frac{\int_{t}^{t+2} mc_{p,air}(T_{out,f} - T_{int}) dt}{\int_{t}^{t+2} G_A dt}
\]
5. Results and discussion

5.1 Input profile for ambient temperature and solar irradiation

Codes are developed to simulate the thermal performance of proposed solar air heater configurations with flat plate and V-corrugated absorber geometry using MATLAB programming language. Figure 2 shows the ambient temperature and solar irradiation profile of Kozhikode on 01 February 2021. The ambient temperature varies from 294.7K to 305.69K, reaching its peak temperature at 01:56 PM. The incident solar irradiation values range from 0 to 820 W/m\(^2\) reaching its peak at 12:00 noon. Equations (22) and (23) are the correlations used for the solar irradiation and ambient temperature profiles, respectively.

\[
G(t) = 820 \times \sin \frac{3.14 t}{43200}, \quad 0 < t < 43200 \text{ s} \tag{22}
\]

\[
T_{ae} = (-5 \times 10^{-5})t^2 + (0.0468)t + 294.742 \tag{23}
\]

5.2 Thermal performance of configuration 1

Simulation results of different types of SAHs under configuration 1 along with the effects of different geometrical parameters is presented in this section. The glass cover and plate temperature variations are simulated for a mass flow rate of 0.025kg/s and an aperture height of 0.05m with different number of glazings and the simulation results of the same are presented in figure 3. From the figure, it is observed that the trend of plate and glass temperatures increases when the number of glazing changed from single to double. This is due to the fact that the total heat loss from the SAH is reduced when two glass covers are employed.
Figure 3. Variation of plate and glass cover temperatures with time

Figure 4. Comparison of exit temperature variation with time for different glass covers

Figure 4 shows the variations of exit temperatures with time for different cases in configuration 1. The exit air temperature starts increasing from 6:00AM and attains a peak temperature of 338.11 K around mid-day at the exit of the second pass with double glass covers. The heat transferred to the plate reduces as solar radiation decreases with time, which leads to decrease in outlet temperature around 6:00PM. The maximum outlet temperature for single glass cover first pass, single glass cover second pass, double glass cover first pass and double glass cover second pass are predicted as 321.34 K, 331.53 K, 326.81 K and 338.11 K, respectively.

Figures 5 and 6 show the variation of collector component temperatures of SAH with aperture height, for a mass flow rate of 0.025kg/s. Figures 7 and 8 show that as the aperture height of the SAH is reduced, the exit air temperature and thermal efficiency increase. The maximum outlet temperature and efficiency are observed as 338.11 K and 68.03%, when mass flow rate of 0.025 kg/s, aperture height 0.05m and number of glass covers are 2. Thus, the optimal value of aperture height and number of glass covers were found out by studying the effect of geometrical parameters.
5.3 Thermal performance of configuration 2

In this section, the thermal performance of configuration 2 (SAH with V-corrugated absorber) is presented. The purpose of using V-corrugated absorber plate is to improve the heat transfer rate between absorber plate and flowing air, resulting in enhanced performance of SAH. Figure 9 represents the schematic diagram of V-corrugated absorber plate. The optimal geometrical parameters i.e. double glazing and aperture height of 0.05m obtained for configuration 1 are employed to evaluate the thermal performance for configuration 2.

![Figure 9. V-Corrugated absorber plate](image)

Variation of average temperature of different components of SAH with V-corrugated absorber geometry are simulated and shown in figure 10. The maximum average temperature of each component is occurring at 12:50 PM in the day. Maximum temperature values of $T_{gl}$, $T_{gl}$, $T_{air1}$, $T_{p}$, $T_{air2}$ and $T_b$ are simulated to be 321.15 K, 331.53 K, 317.48 K, 363.28 K, 333.43 K and 341.14 K, respectively.

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**Figure 7.** Comparison of exit temperatures for different aperture heights

**Figure 8.** Comparison of efficiency for different aperture heights

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5.3 Thermal performance of configuration 2

In this section, the thermal performance of configuration 2 (SAH with V-corrugated absorber) is presented. The purpose of using V-corrugated absorber plate is to improve the heat transfer rate between absorber plate and flowing air, resulting in enhanced performance of SAH. Figure 9 represents the schematic diagram of V-corrugated absorber plate. The optimal geometrical parameters i.e. double glazing and aperture height of 0.05m obtained for configuration 1 are employed to evaluate the thermal performance for configuration 2.

![Figure 9. V-Corrugated absorber plate](image)

Variation of average temperature of different components of SAH with V-corrugated absorber geometry are simulated and shown in figure 10. The maximum average temperature of each component is occurring at 12:50 PM in the day. Maximum temperature values of $T_{gl}$, $T_{gl}$, $T_{air1}$, $T_{p}$, $T_{air2}$ and $T_b$ are simulated to be 321.15 K, 331.53 K, 317.48 K, 363.28 K, 333.43 K and 341.14 K, respectively.
Figure 10. Variations of different component temperatures with time

The outlet air temperature of both the passes against time is plotted in figure 11. The maximum outlet temperature for first pass and second pass are 335.8K and 346.23K respectively, both occurred at 12:50 PM. The variation of air temperature along the length of the air heater for different nodes is plotted in figure 12. Bottom part shows the variation of flowing air temperature in first pass and top part shows the variation in second pass.

Figure 11. Variations of outlet temperature with time

Figure 12. Variation of air temperature along the length

Figure 13 and 14 depict the outlet temperature variation and useful heat gain in the SAH for different mass flow rates: 0.017kg/s, 0.025kg/s, 0.034kg/s. When the mass flow rate is reduced, the exit air temperature increases but the heat gain by the SAH decreases.
Variations of exit air temperature and efficiency with time are simulated for different types in two configurations and the results of the same are presented in figure 15 and figure 16. DPSAH with double glazing and V-corrugated absorber geometry configuration was found to have the maximum thermal performance with maximum outlet temperature at 346.23K and maximum efficiency at 85%.

5.4 Comparison with literature
The simulation results of SAHs with flat plate and V-corrugated absorber plates are validated by comparing with the results of Krishnananth et al. [7] and Sebaii et al. [8], respectively. Experimental results of Krishnananth et al. [7] for double pass single glass cover flat plate SAH and Sebaii et al. [8] for double pass double glass cover V-corrugated SAH are used for validation. Ambient temperature and solar radiation correlations are developed as per [7] and [8] to validate the simulated results with published results. From figure 17 and figure 18, it can be noticed that the present results are in fair agreement with reasonable accuracy when compared with the published ones. The maximum outlet temperature for the flat plate heater was reported as 325.6 K at a mass flow rate of 0.02 kg/s with the
peak solar radiation of 900 W/m² and it is predicted to be 328.2 K using similar geometrical and flow parameters in present study. For V-corrugated configuration, the maximum outlet temperature was recorded as 335.5 K, at a mass flow rate of 0.0203 kg/s and maximum solar radiation of 1000 W/m² and the simulation predicts it as 338.4 K. Results show that both configurations are with ±10% of variations with the published ones. Therefore, it has been considered that the proposed numerical model can be used to predict the thermal performance of DPSAH with bare plate and V-corrugated absorber geometries.

Conclusion

The mathematical models were developed for two configurations of double pass SAH and the performance of both configurations were simulated with the codes developed using MATLAB programming language. Temperature distribution of each component of the SAH, the air temperature at every location of heater, air outlet temperature, efficiency and heat gain of the heater were simulated for both the configurations. The predicted results were validated with experimental data from the literature, and it was found that the model was able to predict the thermal performance of the heater accurately.

The parametric study conducted shows that, number of glazings, aperture height, mass flow rate and nature of absorber geometry have significant effect on the thermal performance of the heater. The optimal performance was predicted when aperture height is 0.05 m at a mass flow rate of 0.025 kg/s employing two glass covers. For bare plate and V-corrugated plate configurations, the maximum exit temperatures were observed as 338.11 K and 346.23 K, maximum efficiencies were observed as 68.03% and 85.50%, and the maximum plate temperatures were observed as 363.28 K and 353.47 K, respectively.

Appendix A - Coefficients of discretized equations

\[
A_1 = \frac{h_{\text{amb}}}{\rho_{\text{gu}} \delta_{\text{gu}} c_{\text{gu}}} \\
A_2 = \frac{h_{\text{rgu}}}{\rho_{\text{gu}} \delta_{\text{gu}} c_{\text{gu}}} \\
A_3 = \frac{h_{\text{cag}}}{\rho_{\text{gu}} \delta_{\text{gu}} c_{\text{gu}}} \\
A_4 = \frac{\alpha}{\rho_{\text{gu}} \delta_{\text{gu}} c_{\text{gu}}} \\
A_5 = \frac{h_{\text{cag}}}{\rho_{\text{ag}} \delta_{\text{ag}} c_{\text{ag}}} \\
A_6 = \frac{h_{\text{cag}}}{\rho_{\text{ag}} \delta_{\text{ag}} c_{\text{ag}}} \\
A_7 = \frac{h_{\text{cag}}}{\rho_{\text{ag}} \delta_{\text{ag}} c_{\text{ag}}} \\
A_8 = \frac{h_{\text{car1}}}{\rho_{\text{gl}} \delta_{\text{gl}} c_{\text{gl}}}
\]
\[ A9 = \frac{h_{rgu-gl}}{\rho_{gl}\delta_{gl}C_{gl}} \quad A10 = \frac{h_{rpgl}}{\rho_{gl}\delta_{gl}C_{gl}} \quad A11 = \frac{h_{cgl-air1}}{\rho_{a1r1}\delta_{air1}C_{air1}} \quad A12 = \frac{h_{cp-air1}}{\rho_{a1r1}\delta_{air1}C_{air1}} \]

\[ A13 = \frac{mc_p}{\rho_{a1r1}V_{air1}C_{air1}} \quad A14 = \frac{h_{cpl-air1}}{\rho_p\delta_pC_p} \quad A15 = \frac{h_{cp-air2}}{\rho_p\delta_pC_p} \quad A16 = \frac{h_{cgl-p}}{\rho_p\delta_pC_p} \]

\[ A17 = \frac{h_{rpb}}{\rho_p\delta_pC_p} \quad A18 = \frac{\tau^2\alpha}{\rho_p\delta_pC_p} \quad A19 = \frac{h_{cpl-air2}}{\rho_{a2r1}\delta_{air2}C_{air2}} \quad A20 = \frac{h_{cb-air2}}{\rho_{a2r1}\delta_{air2}C_{air2}} \]

\[ A21 = \frac{mc_p}{\rho_{a2r1}V_{air2}C_{air2}} \quad A22 = \frac{h_{cb-air2}}{\rho_p\delta_pC_p} \quad A23 = \frac{h_{rpb}}{\rho_p\delta_pC_p} \quad A24 = \frac{U_b}{\rho_p\delta_pC_p} \quad B1 = \frac{\tau\alpha}{\rho_{gl}\delta_{gl}C_{gl}} \]

\[ X1 = \frac{1}{\Delta t} + A1 + A2 + A3X2 = \frac{1}{\Delta t} + A5 + A6 \quad X3 = \frac{1}{\Delta t} + A7 + A8 + A9 + A10 \]

\[ X4 = \frac{1}{\Delta t} + A11 + A12 + A13 \quad X5 = \frac{1}{\Delta t} + A14 + A15 + A16 + A17 \]

\[ X6 = \frac{1}{\Delta t} + A19 + A20 + A21 \quad X7 = \frac{1}{\Delta t} + A22 + A23 + A24 \]

**Appendix B – Heat transfer correlations**

The convective heat transfer coefficient between ambient air and glass cover is [9],

\[ h_{w} = 5.67 + 3.86v_{ext} \quad (B.1) \]

The radioactive heat transfer between sky and glass cover [6],

\[ h_{rguc} = \frac{\varepsilon_{gu} \sigma(T_{gu}^4 - T_{sky}^4)}{T_{gu} - T_{sky}} \quad (B.2) \]

The sky temperature is evaluated as [6],

\[ T_{sky} = 0.0522T_{ae}^{1.5} \quad (B.3) \]

Radioactive heat transfer coefficient between absorber plate and lower glass cover[6],

\[ h_{rpgl} = \frac{\sigma(T_{pgl}^4 + T_{gl}^4)(T_{gl} + T_{gl})}{(T_{pgl} + T_{gl})} \quad (B.4) \]

Natural convection heat transfer coefficient for air gap[9],

\[ h_{cgugl} = \frac{k_{ag}N_{uag}}{D_h} \quad (B.5) \]

Where,

\[ N_{uag} = 1 + 1.44 \left[ 1 - \frac{1708}{Ra \cos \theta} \right]^{2} + \left[ \frac{1}{2} \left( 1708 \sin(1.8\theta) \right)^{1.8} \right] + \left[ \frac{Ra \cos \theta^{0.3}}{5830^{0.3}} - 1 \right] + \left[ \frac{Ra \cos \theta^{0.3}}{5830^{0.3}} - 1 \right] \quad (B.6) \]

Radioactive heat transfer coefficient between upper and lower glass cover[6],
\[ h_{rugu_l} = \frac{\sigma(T_{gu}^2 + T_{gl}^2)(T_{gu} + T_{gl})}{\left( \frac{1}{T_{gu}} + \frac{1}{T_{gl}} \right)} \] (B.7)

The forced convective heat transfer coefficients for air between lower glass cover, absorber plate and base plate is calculated as follows:

For flat plate absorber geometry[9]:

(a) Laminar flow system \( Re \leq 2100 \)

\[ Nu = 4.9 + \frac{0.0606(RePrD_h)^{1/2}}{1 + 0.0999(RePrD_h)^{0.7}Pr^{0.7}} \] (B.8)

(b) Turbulent flow system \( Re > 2100 \)

\[ Nu = 0.0158Re^{0.8} \] (B.9)

For V-corrugated absorber geometry[1]:

(a) For \( Re < 2800 \)

\[ Nu = 2.821 + 0.126 Re \frac{H_a}{l} \] (B.10)

(b) For \( 2800 < Re < 10^4 \)

\[ Nu = 1.9 \times 10^{-6} Re^{1.79} + 225 \frac{H_a}{l} \] (B.11)

(c) For \( 10^4 < Re < 10^5 \)

\[ Nu = 0.0302 Re^{0.74} + 0.242 Re^{0.74} \frac{H_a}{l} \] (B.12)

The convective heat transfer coefficient between the flowing air and lower glass cover and convective heat transfer coefficient between the flowing air and absorber plate are considered as equal[6].

References

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