Thermal Analysis of Double-Pass Solar Air Collector with Different Materials of Absorber Plate and Different Dimensions of Air Channels

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Abstract:*Numerical simulation for thermal performance of double-pass solar air heater with counter flow was investigated in the present study. Thermal efficiency of double-pass solar air collector was estimated for different materials of absorber surface and different dimensions of upper and lower air channels. The optimum thermal efficiency of double-pass solar air heater was identified for a wide range of the solar radiation intensities (600-1000 W/m^2) and mass flow rates (0.01-0.10 kg/s). A code of MATLAB program was built to simulate the mathematical model of double-pass solar air heater. The results showed that the thermal efficiency of double-pass solar air collector of Velesat black plastic absorber plate was better than of Aluminum absorber plate. A comparison between the thermal efficiency of double-pass solar air collector of Velesat black plastic absorber plate and Aluminum absorber plate was analyzed and discussed. A validation of MATLAB code was checked via a comparison with the previous literature. Generally, the thermal efficiency was increased when the mass flow rates and solar radiation intensities increased. The optimum thermal efficiency was occurred at the upper air channel (D$_1$=0.04m), lower air channel (D$_2$=0.08m) for the Velesat black plastic absorber plate for solar radiation intensity (I=1000 W/m^2) and mass flow rate (m$^.$=0.10 kg/s).*

Keywords: Solar collector, double-pass solar air collector, thermal performance, and solar energy applications

1. Introduction

The solar air heater is the simplest form of flat plate solar collector in which the working medium is air. The principle usually followed is to expose a dark surface to solar radiation, so that radiation is absorbed. A flat plate collector used for heating the air generally knows as a solar air heater. A solar air heater is a special type of heat exchanger that transforms solar radiation energy into internal energy of the transport medium air. Day by day industries and societies used the solar systems because of its eco-friendly nature, less maintenance cost and long life products. Solar air heater may be divided into two main categories. The first category is related to the air channel flow configuration. Previously, various air channel flow configurations were constructed to increase the system thermal efficiency. The various configurations were presented in four sub-titles under this category. Single flow, single pass, double flow single pass, single flow double pass and single flow recycled double pass. The second category is related to the air channel design. The air channel design affects the system efficiency significantly. For that reason different design, configuration can be used in the solar collectors.

Thethermal performance and cost benefit analysis of double-pass solar collector with and without fins was studied [1].The theoretical model using steady state analysis was developed and compared with the experimental results.

Steady state energy balance equations for the finned double-pass solar collector were developed [2]. The mathematical model was solved using the matrix inversion method. The predicted results were in agreement with the results obtained from the experiments. The predictions and experiments were observed at the mass flow rate ranging between 0.03 kg/s and 0.1 kg/s, and solar radiation ranging between 400 W/m**2** and 800 W/m**2**.

The thermal performance for solar air heaters was predicted via part of a developed internet-based system [3]. 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.

Various designs of solar collectors have been the subject of many theoretical and experimental investigations;these designs must be able to reduce the heat losses from the collector to increase the operating temperature and collector efficiencies of the system. Therefore, the performance and entropy generation of the double-pass flat plate solar air heater with longitudinal fins are studied numerically [4], A transient 3-D mathematical model for double-pass solar collector with porous media in the lower channel has beendeveloped [5].

A mathematical model and solution procedure for predicting the efficiency of double-pass solar collector was presented [6]. It is composed of steady state form solution was obtained to determine the outlet temperature for energy balance equations, The predicted results agreed with the results obtained from the experiments in the solar laboratory.

The thermal efficiency of a counter-flow double-pass solar air heater studied experimentally and developed the theoretical thermal model to describe its thermal behavior [7].Experimental outdoor tests were carried out during a winter period, with maximum solar irradiance on the collector plane around 1050 W/m**2**. Experimental results and the predictions of the theoretical model were found to be in good agreement.

The double-pass solar collector with porous media in the lower channel provides a higher outlet temperature compared to the conventional single-pass collector [8].The porous media has been arranged in different porosities to

increase heat transfer, area density, and the total heat transfer rate. The study concluded that the presence of porous media in the second channel increases the outlet temperature, therefore increases the thermal efficiency of the systems. The analysis of the thermal performance of the double pass solar air heater having a collector angle of inclination 60° was experimentally investigated [9].Comparative study of solar air heater with baffles and with longitudinal fins was presented [10].A working model of the active type, solar air heater with the capacity of thermal storage was created [11].A review of the performance of double pass solar air heater was presented [12-14].

2. DescriptionsofDouble-Pass Solar Air Collector

A double-pass solar air collector of counter flow was used in the present study was shown in **Fig. 1**. The length and width of solar air collector were used $(L = 2.4m, W = 1.2m)$[1]. A part of the solar radiation impinging on the glass cover was absorbed by glass cover itself, another part was reflected to surrounding and the remainder was transmitted to the absorbent surface. A Velesat black plastic and Aluminum were used as an absorbent surface for the solar air collector with different properties summarized in **Table. 1** to identify the optimum thermal performance. Different dimensions of upper and lower channel were used to predict the optimum performance and these dimensions were summarized in **Table. 2**. The ambient temperature, wind velocity adopted in the present study were $(T_a = 25\ °C,$ and $V = 1\ m/s)$. The air enters the inlet of upper channel at an initial temperature $(T_i = 27\ °C)$ and get out from the lower channel with an outlet temperature. The insulation material used to reduce the conduction losses through the bottom side of solar air collector. Several applications were used for double-pass solar air collector such as solar dryer, solar cooking, space heating, and thermal industrial processes.
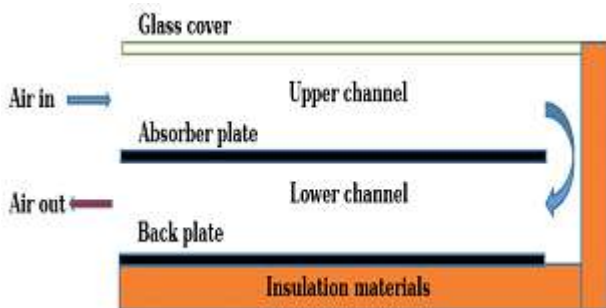
Figure 1: Double-pass solar air collector of the present study

Table 1: Specifications of materials used in the present study.

Material	Absorptivity	Emissivity	Transmissivity
Aluminum	0.95	0.90	-
Velesat black plastic	0.96	0.85	-
Glass cover	0.05	0.80	0.92

Table 2: Dimensions of upper and lower channel depth.

Channels	Upper channel depth	Lower channel depth
Channels-1	0.04 m	0.08 m
Channels-2	0.05 m	0.10 m
Channels-3	0.06 m	0.12 m

3. Mathematical Model

The solar air heater model and the cross section view showing the various heat transfer coefficientsof the double-pass solar collector studied in the present work has been considered in **Fig. 2**. The collector consists of a glass cover plate, a blackened absorber plate, and a back plate. Simplified steps were used to analyze the heat transfer for the air flow to calculate the amount of energies, the outlet temperatures, and the efficiency.

The calculations were based on the following assumptions: (a) study state. (b) All convection heat transfer coefficients in the channels and flowing air are equal and constant. (c) The thermal conductivity of the absorber plate is constant. (d) Uniform useful heat gain along the length of the collector.
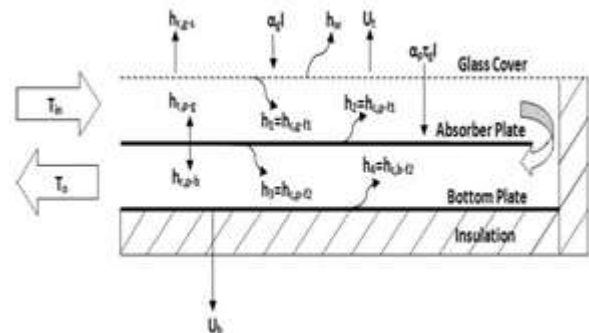
Figure 2: Schematic of heat transfer coefficients and cross section view of a double-pass solar collector

The steady state energy balance equations for double-pass solar air collector can then be written as follow:
At the glass cover:

$$U_t(T_g - T_a) + h_1(T_g - T_{f1}) = h_{rpg}(T_p - T_g) + \alpha_g I \qquad (1)$$

Here (U_t) is the top losses coefficient $(W/m^2.°C)$, (T_g) is the glass cover temperature $(°C)$, (T_a) is the ambient temperature $(°C)$, (T_{f1}) is the mean air temperature in the upper channel $(°C)$, (T_p) is the absorber plate temperature $(°C)$, (I) is the solar radiation (W/m^2), (α_g) is the absorptivity of glass cover, (h_1) is the convection heat loss from glass cover to air in the upper channel $(W/m^2.°C)$, and (h_{rpg}) is the radiation heat losses from absorber plate to glass cover $(W/m^2.K)$.

For the first air flow in the upper channel:

$$Q_1 = h_1(T_g - T_{f1}) + h_2(T_p - T_{f1}) \qquad (2)$$

Here (Q_1) is the heat transfer rate in the upper channel (W), and (h_2) is the convection heat loss from absorber plate to air in the upper channel $(W/m^2.°C)$.

For the absorber plate:

$$h_{rpg}(T_p - T_g) + h_2(T_p - T_{f1}) + h_{rpb}(T_p - T_b)$$
$$+ h_3(T_p - T_{f2}) = \alpha_p \tau_g I$$

$$(3)$$

Here (T_b) is the back plate temperature (°C), (h_3) is the convection heat loss from absorber plate to air in the lower channel $(W/m^2.°C)$, (α_p) is the absorptivity of absorber plate, and (τ_g) is the transmissivity of the glass cover.

For the second air flow in the lower channel:

$$Q_2 = h_3(T_p - T_{f2}) + h_4(T_b - T_{f2}) \qquad (4)$$

Here (h_4) is the convection heat loss from back plate to air in the lowerchannel $(W/m^2.°C)$, and (T_{f2}) is the mean air temperature in the lower channel (°C).

For the back plate:

$$h_{rpb}(T_p - T_b) = h_4(T_b - T_{f2}) + U_b \qquad (5)$$

Here (h_{rpb}) is the radiation heat loss from absorber plate to back plate $(W/m^2.K)$, and (U_b) is the bottom loss coefficient $(W/m^2.°C)$.

For the short collectors, the mean air temperature would equal the arithmetic mean and can be estimated as,

$$T_{f1} = [(T_{f1,o} + T_i)/2] \qquad (6)$$
$$T_{f2} = [(T_{f2,o} + T_{f1,o})/2] \qquad (7)$$

Here (T_i) is the air temperature at inlet of upper channel (°C), $(T_{f1,o})$ is the air temperature at exit of upper channel (°C), and $(T_{f2,o})$ is the outlet air temperature from the lower channel (°C).

The thermal efficiency of the solar air collector represent the ratio of useful gain to the incident solar energy on the collector area. It can be calculated by the equation below:

$$\eta_{th} = \frac{m^° C (T_o - T_i)}{A_c I} \qquad (8)$$

Here $(m^°)$ is the mass flow rate (kg/s), (A_c) is the collector area (m^2), (T_o) is the outlet air temperature (°C), and (η_{th}) is the thermal efficiency (dimensionless).

The physical properties of air are assumed to vary linearly with temperature and governed by the relations below:

a) Specific heat
$$C = 1.0057 + 0.000066(T - 27) \qquad (9)$$
Here (C)is the specific heat of air (J/kg. K).

b) Density
$$\rho = 1.1774 - 0.00359(T - 27) \qquad (10)$$
Here (ρ) is the air density (kg/m^3).

c) Thermal conductivity
$$k = 0.02624 - 0.0000758(T - 27) \qquad (11)$$
Here (k) is the thermal conductivity of air $(W/m.°C)$

d) Viscosity
$$\mu = [1.983 - 0.00184(T - 27)] \times 10^{-5} \qquad (12)$$
Here(μ) is the air viscosity (m^2/s).

The wind heat transfer coefficients represent the convection heat transfer between the glass cover and surrounding air and its value depended on the wind velocity. It can be formulated as:

$$h_w = 2.8 + 3.3\,V \qquad (13)$$

Here (V) is the wind velocity (m/s), and (h_w) is the wind loss coefficient $(W/m^2.°C)$.

The radiation loss heat transfer coefficients from glass cover to the sky canbe expressed as:

$$h_{r,g-s} = \frac{\sigma\varepsilon_g(T_g+T_s)(T_g^2+T_s^2)(T_g-T_s)}{(T_g-T_a)} \qquad (14)$$

Here $(h_{r,g-s})$ is the radiation heat loss coefficient $(W/m^2.K)$, (σ) is the Stefan Boltzmann constant $(W/m^2.°C^4)$, (ε_g) is the emissivity of glass cover (dimensionless), and (T_s) is the sky temperature (°C).

The radiation loss heat transfer coefficients from absorber plate to glass cover can be calculated as:

$$h_{r,p-g} = \frac{\sigma(T_p+T_g)(T_p^2+T_g^2)}{(1/\varepsilon_p)+(1/\varepsilon_g)-1} \qquad (15)$$

Here (ε_p) is the emissivity of the absorber plate (dimensionless).

The radiation loss heat transfer coefficients from absorber plate to back plate are estimated as:

$$h_{r,p-b} = \frac{\sigma(T_p+T_b)(T_p^2+T_b^2)}{(1/\varepsilon_p)+(1/\varepsilon_b)-1} \qquad (16)$$

Here (ε_b) is the emissivity of the back plate (dimensionless).

The sky temperature can be written as:

$$T_s = 0.0552\,T_a^{1.5} \qquad (17)$$

The convective heat transfer coefficients are calculated using following relations:

$$h = \frac{k}{D_h}\,Nu \qquad (18)$$

Here (D_h) is the hydraulic diameter (m), and (Nu) is the Nusselt number (dimensionless).
For laminar flow region, Nusselt number for $(Re < 2300)$

$$Nu = 5.4 + \frac{0.0019[Re\,Pr\,(D_h/L)]^{1.71}}{1+0.00563\,[Re\,Pr\,(D_h/L)]^{1.71}} \qquad (19)$$

Here (Re) is the Reynolds number (dimensionless), (Pr) is the Prandtl number (dimensionless), and (L) is the collector length (m).

For transition flow region (2300 <Re< 6000):

$$Nu = 0.116(Re^{(2/3)} - 125)Pr^{(1/3)}\left[1 + \left(\frac{D_h}{L}\right)^{(2/3)}\right]\left(\frac{\mu}{\mu_w}\right)^{0.14} \qquad (20)$$

Turbulent flow region (Re> 6000):
$$Nu = 0.018\,Re^{0.8}Pr^{0.4} \qquad (21)$$

The Reynolds number and hydraulic diameter are written as:

$$Re = \frac{m^° D_h}{A_c\mu} \qquad (22)$$

$$D_h = \frac{4\,Wd}{2(W+d)} \qquad (23)$$

Here (W) is the collector width (m), and (d) is the channel depth (m).

4. Results and Discussion

In the present study, the thermal performance of the double-pass solar air collector was presented via estimated the thermal efficiency of the double-pass solar air collector. The influence of the type of absorbent surface material and depth of air channels on the thermal performance of the solar air collector was studied.

Velesat black plastic absorber plate and Aluminum absorber plate of different specifications of emissivity and transmissivity were used and analyzed. Comparisons of thermal efficiency were used for the absorbent surfaces mentioned above of double-pass solar air collector. Also, air channels were used at different depths to identify the optimum thermal efficiency of double-pass solar air collector.

Several ranges for solar radiation intensity (600, 800, and 1000 W/m^2) and mass flow rates (0.01-0.10 kg/s) were adopted and analyzed in the present study. The results were presented in the forms, the thermal efficiency with mass flow rate, and thermal efficiency with solar radiation intensity, and temperature distribution with mass flow rates for different materials and optimum channels dimensions. The results showed that the thermal efficiency was influenced by the depth of the upper and lower channels besides the material of the absorber plate.

A comparison of the thermal efficiency of double-pass solar air collector of Aluminum absorber plate in the present study with [1] was achieved and showed a good agreement as shown in **Fig. 3**.

Figs.(4-6) illustrate the thermal efficiency for different absorber plates of Velesat black plastic and Aluminum absorber plates with mass flow rate ranges (0.01-0.10 kg/s) at the depth of the channel of ($D_1 = 0.04\ m$, and $D_2 = 0.08\ m$), ($D_1 = 0.05\ m$, and $D_2 = 0.10\ m$), and ($D_1 = 0.06\ m$, and $D_2 = 0.12\ m$) for upper and lower channels respectively. A comparison between the thermal efficiency of Velesat black plastic and Aluminum absorber plates was achieved and analyzed for different ranges of mass flow rates, solar radiation intensities, and channels depth. The material for the absorber plate of a solar collector in addition to the depth of the upper and lower channelwas an important parameter that enhanced the heat transfer rate of solar air collector.

The absorptivity and emissivity of the absorber plate were taken into consideration for the Velesat black plastic and Aluminum. In general, the thermal performance of double-pass solar air heater with Velesat back plastic absorber plate was better than the double-pass solar air heater with Aluminum absorber plate as shown from **Figs. (4-6)**.

A part of solar radiation falling on the solar air collector is absorbed by the glass cover and another part is reflected from the glass surface and the remaining part is transmitted to the absorbent surface. The air entering the double-pass solar air collector draws the absorbed thermal energy from the absorbent surface along the solar air collector as it passes from the upper channel to the lower channel. As shown in the **Figs. (4-6)**, as the mass flow rates increased, the thermal efficiency of the solar air collector increased due to enhanced in heat transfer rate. Obviously, the double-pass solar air heater of upper and lower channels depth ($D_1 = 0.04\ m$, and $D_2 = 0.08\ m$) showed the better performance than the others. This is because the flow of air in the narrow channel is faster according to the law of mass conservation. As a result, the heat transfer is better compared to the larger channel. Consequently, the optimum dimensions for the upper and lower air channels depth was ($D_1 = 0.04\ m$, and $D_2 = 0.08\ m$).

For the optimum dimensions of the upper and lower channels, the overall average of thermal efficiency was (0.5787, 0.5834, and 0.5866) for Velesat black plastic at solar radiation intensities (600, 800, and 1000 W/m^2) respectively. Whereas the overall average of thermal efficiency was (0.5690, 0.5737, and 0.5768) for Aluminum at solar radiation intensities (600, 800, and 1000 W/m^2) respectively.

For the optimum dimensions of upper and lower channels, the temperature distribution of glass cover, air flow in the upper channel, absorber plate, airflow in the lower channel, back plate, and outlet air were presented and revealed. **Figs. (7-9)** show the temperature distribution aforementioned with mass flow rates for solar radiation intensities (600-1000 W/m^2). It is noticeable that when the mass flow rate increased, the temperature of the glass cover, airflow in the upper channel, absorber plate, airflow in the lower channel, back plate, and outlet air was decreased. **Figs. (10-12)** shows a comparison between the thermal efficiency of Velesat black plastic and Aluminum with solar radiation intensities at the depth of the channel of ($D_1 = 0.04\ m$, and $D_2 = 0.08\ m$), ($D_1 = 0.05\ m$, and $D_2 = 0.10\ m$), and ($D_1 = 0.06\ m$, and $D_2 = 0.12\ m$) for upper and lower channels respectively with mass flow rate of ($\dot{m} = 0.1\ kg/s$). For any solar radiation intensity, the double-pass solar air heater of Velesat black plastic absorber plate showed up the better performance than the of Aluminum absorber plate. While **Fig. 13**illustrates the average of thermal efficiency for double-pass solar air heater at solar radiation intensity (1000 W/m^2) at the depth of the channel of ($D_1 = 0.04$ m, and $D_2 = 0.08$ m), ($D_1 = 0.05$ m, and $D_2 = 0.10$ m), and ($D_1 = 0.06$ m, and $D_2 = 0.12$ m) for upper and lower channels respectively. Clearly, the thermal efficiency of the double-pass solar air heater of Velesat black plastic absorber plate showed up the better performance than the of Aluminum absorber plate. As mentioned before, the thermal efficiency was the best for the channel-1 of ($D_1 = 0.04$ m, and $D_2 = 0.08$ m), followed by channel-2 of ($D_1 = 0.05$ m, and $D_2 = 0.10$ m), and then channel-3 of ($D_1 = 0.06$ m, and $D_2 = 0.12$ m).

5. Conclusions

The thermal performance of double-pass solar air collector of different dimensions for upper and lower channels and different materials for the absorbent surface at a range of mass flow rates and solar radiation intensities was investigated in the present study. The main conclusions are as follows:

1) The absorber plate material of double-pass solar air collector and the depth of air channels were the main factors affecting thermal efficiencyof double-pass solar air collector.
2) Generally, the thermal efficiency of double-pass solar air collector was increased when the mass flow rate of working fluid (air) increased.

3) Also, the thermal efficiency of double-pass solar air collector was increased when the solar radiation intensity increased.
4) Clearly, the thermal efficiency of double-pass solar air collector of Velesat black plastic absorbent surface was better than of Aluminum absorbent surface due to absorptivity and emissivity properties.
5) It is worth mentioning that the optimum thermal efficiency occurred at the depth of ($D_1 = 0.04$ m, and $D_2 = 0.08$ m) for upper and lower channels due to the enhancement in heat transfer rate.
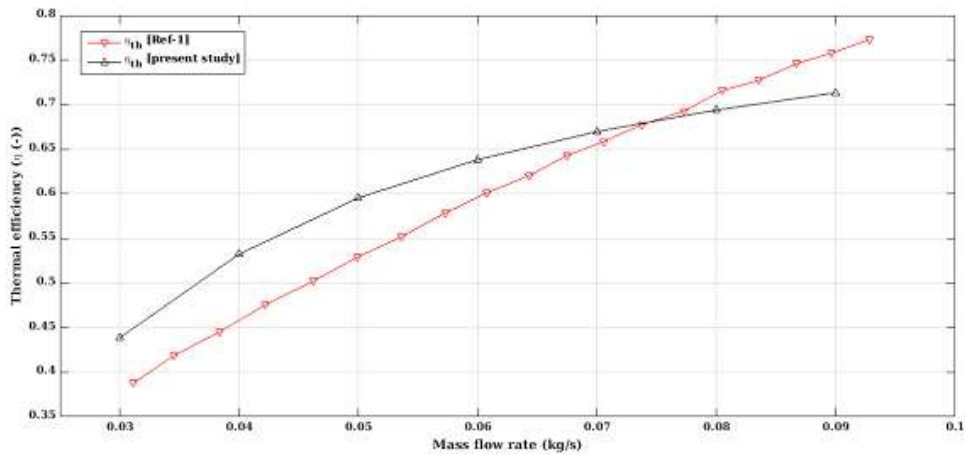
Figure 3:A comparison of thermal efficiency between the present study and [Ref-1]

(a)
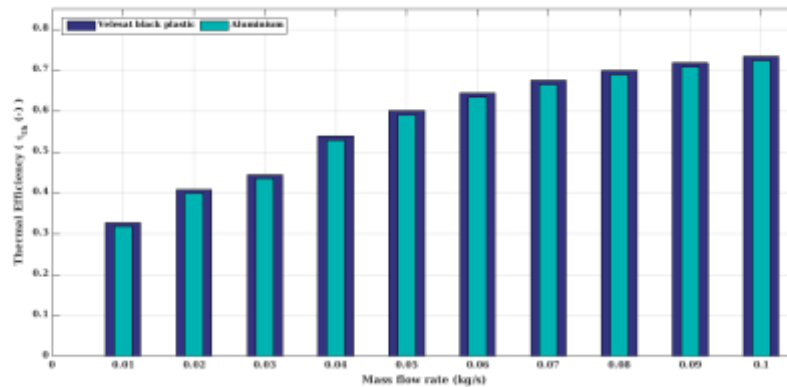
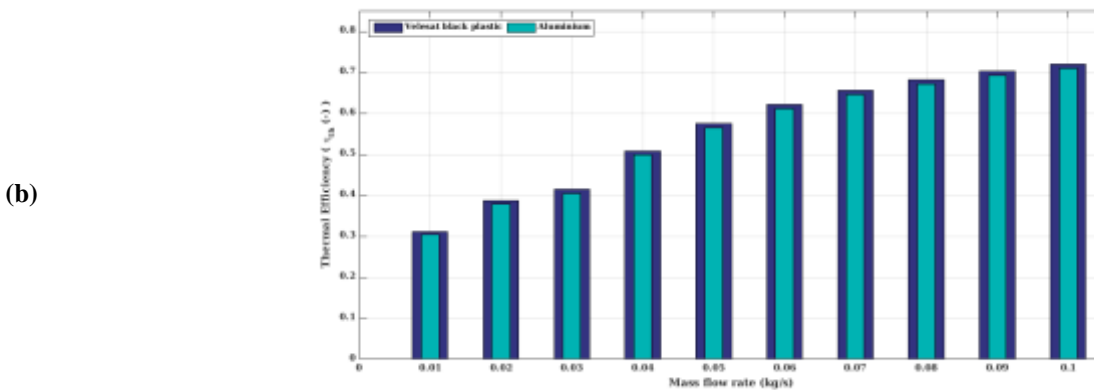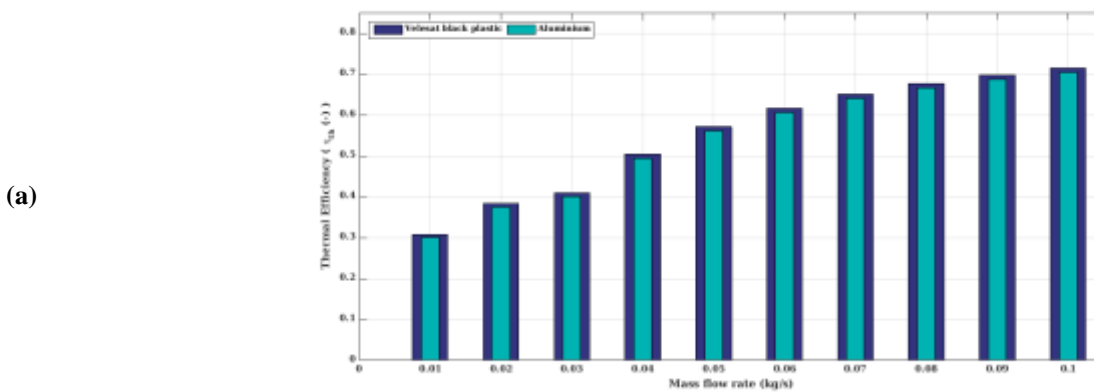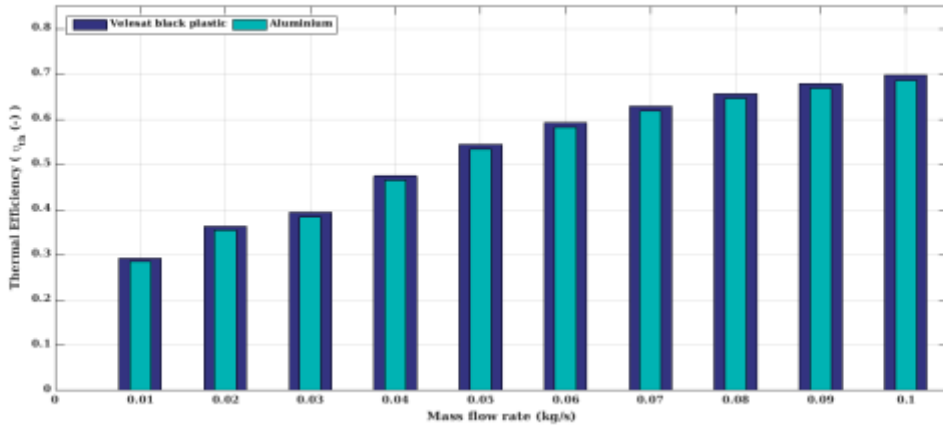**(b)**
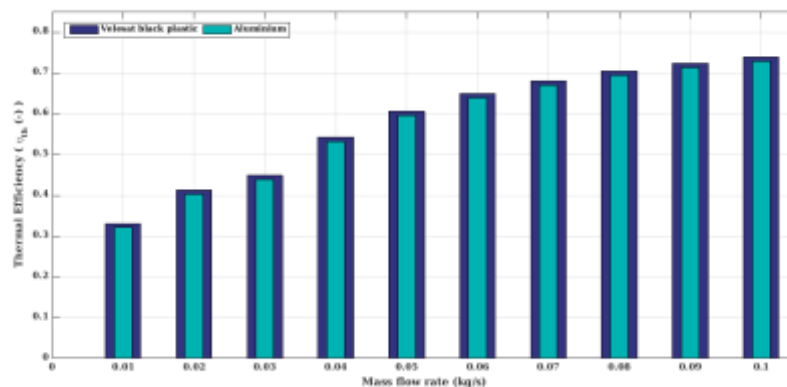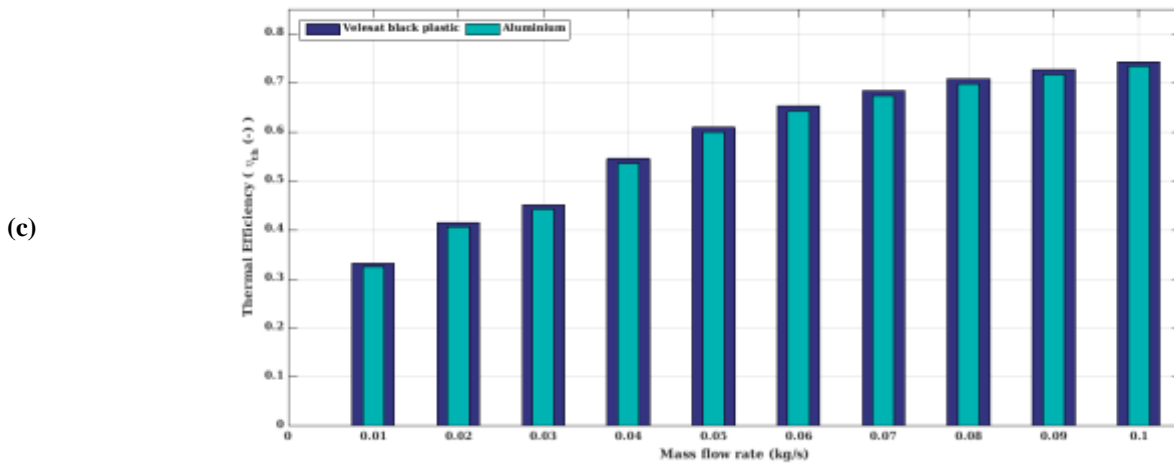
Figure 4: Thermal efficiency of double-pass solar air collector at: (a) I = 600 W/m^2, (b) I = 800 W/m^2, and (c) I = 1000 W/m^2 for channels-1.
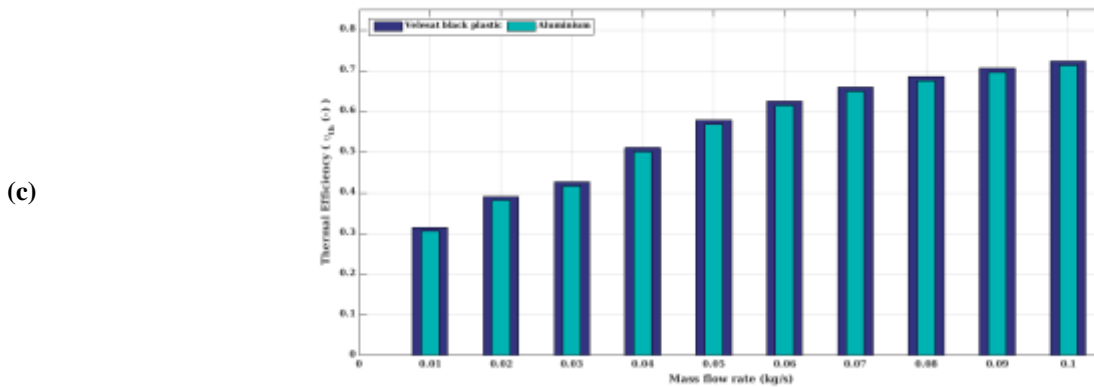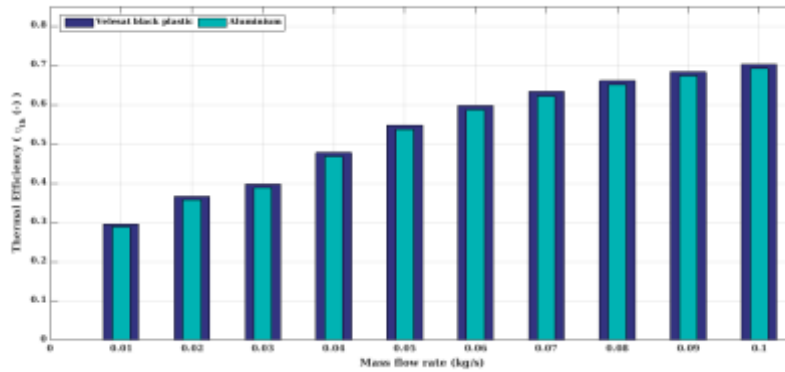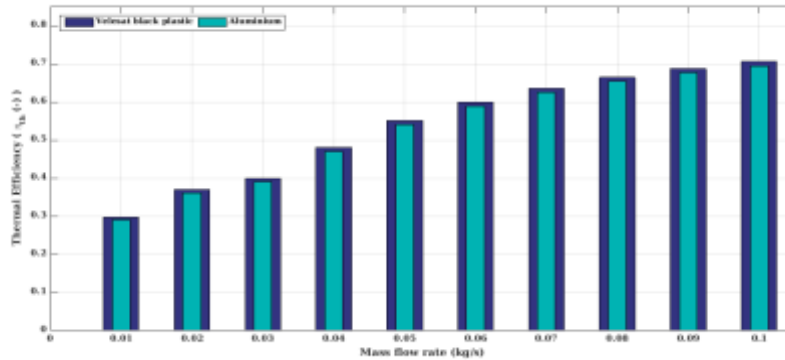

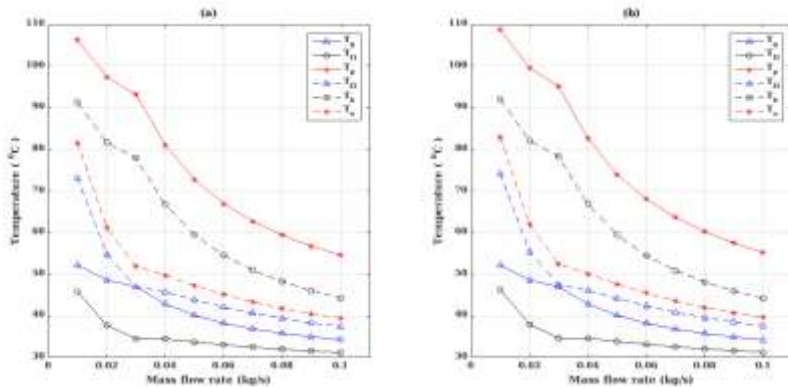**Figure 5**: Thermal efficiency of double-pass solar air collector at: (a) I = 600 W/m^2, (b) I = 800 W/m^2, and (c) I = 1000 W/m^2 for channels-2.

(a)

(b)

(c)

Figure 6: Thermal efficiency of double-pass solar air collector at: (a) $I = 600$ W/m^2**, (b)** $I = 800$ W/m^2**, and (c)** $I = 1000$ W/m^2 **for channels-3.**

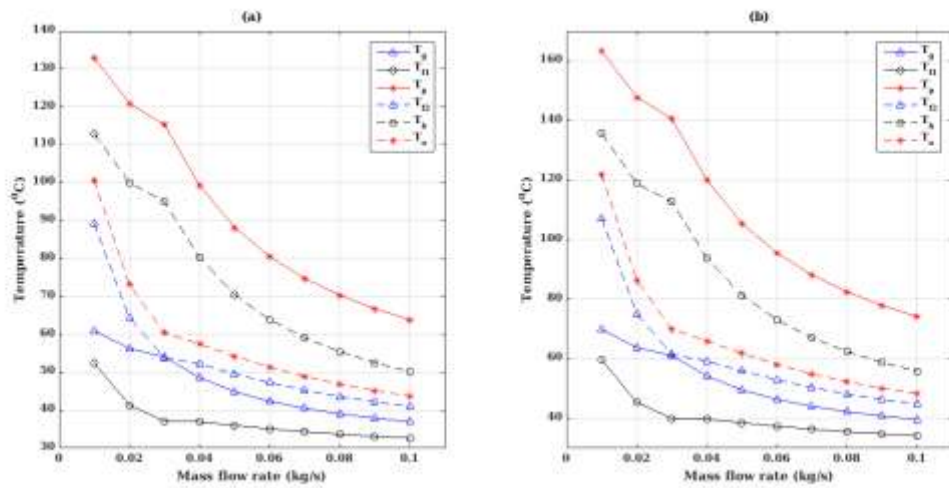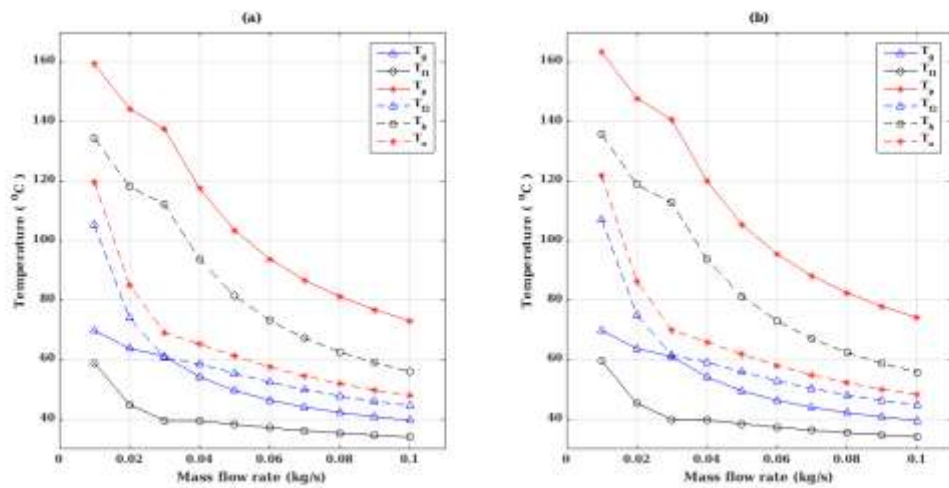**Figure 7:** Temperature distribution with mass flow rates at solar radiation intensity (600 W/m^2) for: (a) Aluminum, and (b) Velesat black plastic.

Figure 8: Temperature distribution with mass flow rates at solar radiation intensity (800 W/m^2) for: (a) Aluminum, and (b) Velesat black plastic.
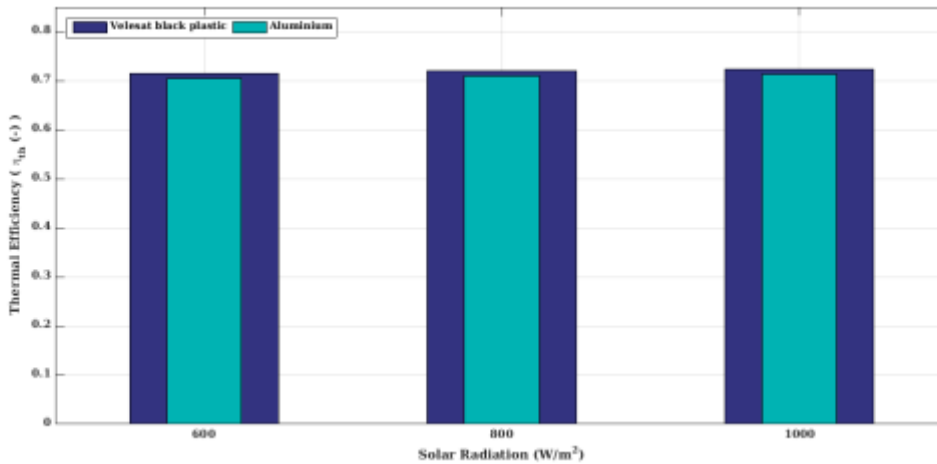

Figure 9: Temperature distribution with mass flow rates at solar radiation intensity (1000 W/m^2) for: (a) Aluminum, and (b) Velesat black plastic.
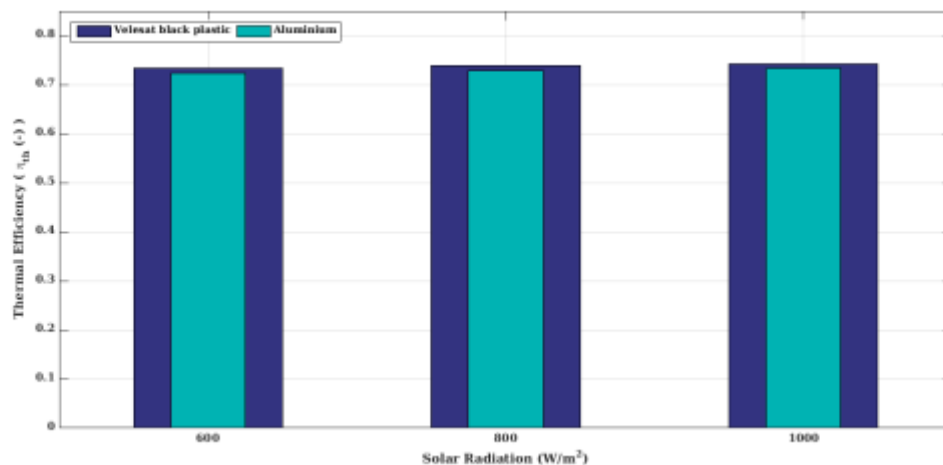
Figure10: Comparison between the thermal efficiency of Velesat black plastic and aluminum with solar radiation intensities at mass flow rate of ($\dot{m} = 0.1$ kg/s).
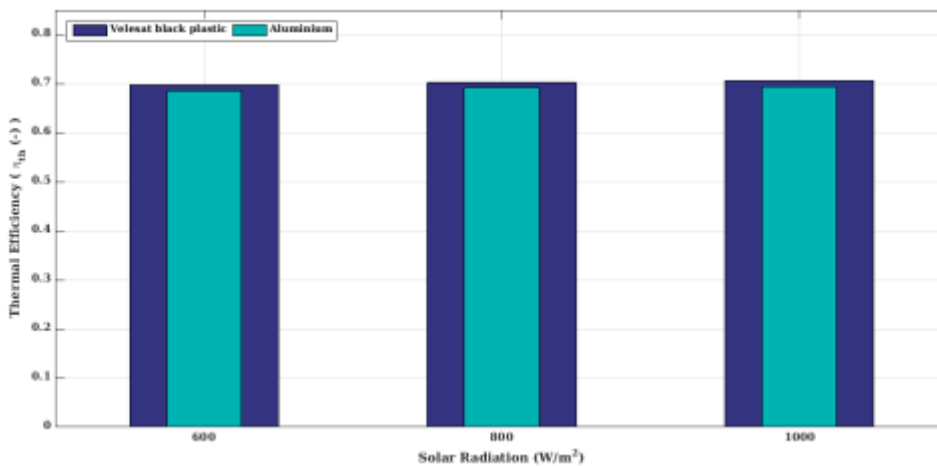
Figure 11: Comparison between the thermal efficiency of Velesat black plastic and aluminum with solar radiation intensities at mass flow rate of ($\dot{m} = 0.1\ \mathbf{kg/s}$).
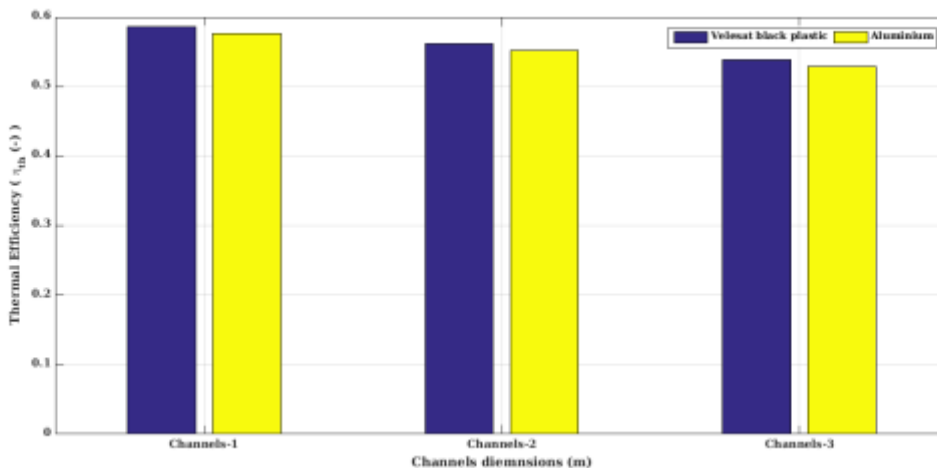

Figure 12: Comparison between the thermal efficiency of Velesat black plastic and aluminum with solar radiation intensities at mass flow rate of ($\dot{m} = 0.1\ \text{kg/s}$).

Figure 13: Comparison of thermal efficiency at solar radiation intensity ($1000\ \text{W/m}^2$) for Velesat black plastic and Aluminum absorbent surfaces at different channels depth.

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