Experimental Investigation of One Glass Cover and Three Absorber Plates Two-Pass Solar Air Collector with Thermal Storage

Satya Prakash Mishra and Vipin Shrivastava*

Department of Mechanical Engineering, Lakshmi Narain College of Technology, Bhopal – 462021, Madhya Pradesh, India; satyapmis16@gmail.com, vipin_shrivastava07@rediffmail.com

Abstract

Objective: Improvement of the thermal efficiency of solar air collector is a major issue in researcher’s community. Therefore novel design of solar air collector is constructed. The objectives of this research paper are to obtained the results of novel solar air collector and compare it with conventional type solar air collector. Methods/Statistical Analysis: The experiments are carried out at the different mass flow rate of air on novel design of one glass cover and three absorber plates two-pass solar air collector (OGT_ApTPSAC) and one glass cover and two absorber plates single-pass solar air collector (OGT_ApSPSAC) operating on the same atmospheric conditions. The various thermal parameters such as heat gain, heat loss and thermal efficiency are determined with respect to time of the day. Findings: Major heat loss in any type of solar air collector is due to convection. To minimise this convection loss, vacuum is formed on top portion of novel designed solar air collector. Both solar air collectors taken for the comparison also has a unique feature of thermal storage. Comparative results show that the novel designed two-pass solar air collector is more thermally efficient than conventional single pass solar air collector. Application/improvements: Such type of solar air collector can be used in drying of fruits, crops; space heating for greenhouse, residential building etc.

Keywords: Heat Gain, Heat Loss, Thermal Efficiency, Thermal Storage, Two-Pass Solar Air Collector

Nomenclature

\( I \) Solar radiation (W/m\(^2\))
\( A_p \) Projected area (m\(^2\))
\( C_p \) Specific heat of working fluid (J/kg K)
\( D_e \) Equivalence diameter
\( h_c \) convective heat transfer coefficient (W/m\(^2\) K)
\( h_r \) Radiative heat transfer coefficient (W/m\(^2\) K)
\( k \) Thermal conductivity (W/m K)
\( \dot{m} \) Average mass flow rate (kg/s)
\( \text{Nu} \) Nusselt number
\( \text{Pr} \) Prandtl number
\( Q \) Heat transfer rate (W)
\( \text{Re} \) Reynolds number
\( T \) Temperature (K)
\( T_g \) Temperature of glass surface (K)
\( T_p \) Temperature of absorber plate (K)
\( T_a \) Temperature of ambient air (K)
\( U_t \) Collector top heat loss coefficient (W/m\(^2\)K)
\( U_b \) Collector back loss coefficient (W/m\(^2\)K)
\( \alpha \) Absorptivity
\( \epsilon \) Emissivity
\( \tau \) Transitivity
\( \rho \) Density (kg/m\(^3\))
\( \eta \) Thermal efficiency (%)
\( \sigma \) Stefan’s Boltzmann constant (W/m\(^2\) K\(^4\))

Subscript:

\( a \) ambient
\( c \) convective
\( f \) fluid
\( g \) glass
\( o \) outlet
\( i \) inlet

*Author for correspondence
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1. Introduction

Solar air collector converts the solar radiation to inside energy of the air. Solar air collectors are used in solar drying and heating applications\(^1\). The adoptability of the solar collector unit depends on various factors such as efficiency, fabrication cost, installation, operational cost and some other specific factors regarding specific uses. Extensive work to improve the performance of solar air collectors has been done. By extending the area, cost is not much changes but performance of new unit having large surface area is improve\(^2\). Substantial enhancement in the thermal efficiency is obtained in double-pass mode but convection losses are increased\(^3\). Use of two glass cover to reduce convective loss with two-pass mode provides better results instead of provide single-pass mode for same mass flow rate\(^4\). Similarly, higher outlet temperature has been observed when two absorber plate and two channels were used\(^5\). Various researchers used corrugated sheet to maximize the heat gain\(^6-8\). Attaching fins in absorber plate is another option which create zig zag motion in flowing fluid, thus higher thermal efficiency obtained\(^9,10\). Literature shows that two-pass solar air collector’s transfers more heat energy to the flowing fluid as compared to conventional solar air collectors. But it is also observed that major heat is lost by convection loss. Therefore the novel design was constructed. In which vacuum is created between the glass cover and first absorbing plate to reduce the convection losses and maximize the utilization of heat.

2. Material and Methods

2.1 Experimental Setup

Experimental set-up of novel OGT\(_{h,A}\)TPSAC is consisting of six essential parts, which are transparent glass cover, three absorber plates, and thermal storage material, air supplying unit, thermal storage and outer frame. Dimension of OGT\(_{h,A}\)TPSAC is 1m×1m×0.3m. Glass of thickness 0.004 m was used on top of the solar air collectors to create green house effect. Three black painted absorber plates of galvanised iron material having thickness 0.002 m were used to increase radiation absorptivity. Outer body and absorber plates were made up of polycarbonate sheet. An acrylic sheet of thickness 0.03 m was used to prevent side and bottom losses. One glass cover and three absorber plate two-pass solar collector consists of four different separating sections along the height of solar air collector. The first section is provided between the glass cover and first absorbing plate. Vacuumed is provided in this section to reduce the losses. Second and third sections are provided between first-second absorber plates and second-third absorber plate respectively. The fourth section is used for thermal storage material. The size of first section and other three sections are 0.06 m and 0.08 m respectively. In OGT\(_{h,A}\) A TPSAC, air pass through second and third section. Similarly, OGT\(_{w,A}\)SPSAC also constructed having three sections, in which inlet air pass only through second section. DC fans having electrical specification 12v and 0.16 amps are attached in inlet of duct. Obtained average mass flow rate (ṁ) on Day 1, Day 2, Day 3 and Day 4 was 0.022 kg/sec, 0.036 kg/sec, 0.046 kg/sec, and 0.056 kg/sec respectively. In thermal storage section 90 kg of black sand was used. Schematic view of OGT\(_{h,A}\)TPSAC and OGT\(_{w,A}\)SPSAC with thermal storage is shown in Figure 1(a) and 1(b) respectively.

![Figure 1. Schematic view (a) OGT\(_{h,A}\)TPSAC (b) OGT\(_{w,A}\)SPSAC.](image-url)
2.2 Instrumentation

Solar radiation on inclined surface of glass is measured by calibrated solar pyranometer (TM-207) having range 0-2000 W/m², sampling time 0.25 sec and accuracy ±10 W/m². Velocity of ambient air is measured with the help of calibrated hotwire 490 Testo anemometer having resolution and range are 0.1 m/s and 0.2–60 m/s respectively. Calibrated digital hygrometer HT-305 is used for measurement of temperature at inside, outside of solar air collector and for ambient temperature conditions. K type thermocouples ranges from -200°C to 1370°C with 0.1°C resolution and accuracy ±1°C are used to measure the temperature of different absorber plates and glass surface.

2.3 Experimentation

The experimental setup was constructed in Lakshmi Narain College of Technology, (23°15′N-77°25′E) Bhopal, India. All reading was taken from 10:00 AM to 21:00 PM for four consecutive days in the month of June 2016. The average mass flow rate of air during each consecutive day was 0.022 kg/sec, 0.036 kg/sec, 0.046 kg/sec, and 0.056 kg/sec. For all four days of experiment in ambient and weather conditions are nearly same.

2.4 Thermal Analysis

The solar radiation S (W/m²) falls on the absorber plate per unit area is equal to the difference between the incident solar radiation and the optical loss:

\[ S \cong 0.97 I_{d} \alpha_{g} \]  

(1)

Steady state energy balance equation for OGT_h,A_p,TPSAC and OGT_w,A_p,SPSAC are:

2.4.1 For the Glass Cover

\[ I_{d} \alpha_{g} = h_{c,ga}(T_{g} - T_{a}) + h_{r,ga}(T_{g} - T_{ps}) + h_{r,gs}(T_{g} - T_{a}) \]  

(2)

Where \( I_{d} \alpha_{g} \) is the solar radiation absorbed by a glass cover.

2.4.2 For First Absorber Plate and Inlet Fluid

\[ I_{d} \alpha_{1} = h_{c,1i}(T_{1} - T_{a}) + h_{c,1a}(T_{1} - T_{ps1}) + h_{r,1a}(T_{1} - T_{a}) \]  

(3)

2.4.3 For First Absorber Plate and Glass Cover

\[ I_{d} \alpha_{1} = h_{c,1}(T_{1} - T_{a}) + h_{c,1a}(T_{1} - T_{ps1}) + h_{r,1a}(T_{1} - T_{ps1}) \]  

(4)

Convective heat gain for inlet fluid is:

\[ Q_{c,i} = m_{f,i} \frac{dT_{f,i}}{dt} = h_{c,1i}(T_{f,i} - T_{a}) + h_{r,1i}(T_{f,i} - T_{ps1}) \]  

(5)

2.4.4 For Second Absorber Plate and Second Fluid

\[ h_{r,ps2}(T_{ps2} - T_{a}) + h_{c,ps2}(T_{ps2} - T_{ps1}) + h_{r,gs2}(T_{ps2} - T_{a}) \]  

(6)

2.4.5 For Third Absorber Plate

\[ h_{r,ps3}(T_{ps3} - T_{ps2}) \]  

(7)

The convective heat transfer coefficient for ambient air flowing outside of the glass cover (\( h_{c,ga} \)) is obtained by McAdams equation:

\[ h_{c,ga} = 5.7 + 3.8 V \]  

(8)

Where, V is the velocity of ambient air.

Radiative heat transfer coefficient between glass cover and sky (\( h_{r,ga} \)) is given by:

\[ h_{r,ga} = \frac{\sigma(T_{g}^2 + T_{a}^2)}{(1 - \epsilon_{g})} \]  

(9)

The convective heat transfer coefficient for air flowing inside the channel is:

\[ N_{u} = \frac{h_{c,ga}}{k_{at}} = 0.0334Re^{0.8}Pr^{0.28} \]  

(10)

Convective heat gain for outlet fluid is:

\[ Q_{c,o} = m_{f,o} \frac{dT_{f,o}}{dt} = h_{c,1o}(T_{f,o} - T_{a}) + h_{r,1o}(T_{f,o} - T_{ps1}) \]  

(11)

Thermal efficiency of solar air collector can be is:

\[ \eta = \frac{Q_{c}}{I_{d}A_{p}} = \frac{\sum C_{p}(T_{f} - T_{o})}{I_{d}A_{p}} \]  

(12)

Top heat loss coefficient (\( U_{t} \)) is:

\[ U_{t} = \frac{1}{h_{r,ga} + h_{c,ga} + h_{r,ga,1}} \]  

(13)

Mid heat loss coefficient (\( U_{m} \)) is:

\[ U_{m} = \frac{1}{h_{c,1o} + h_{c,1o} + h_{r,1o}} \]  

(14)

Bottom heat loss coefficient (\( U_{b} \)) of is:

\[ U_{b} = \frac{1}{h_{c,2o} + h_{c,2o} + h_{r,2o}} \]  

(15)

Conduction loss coefficient (\( U_{c} \)) is:

\[ U_{c} = \frac{1}{\frac{k_{s}}{k_{c} + k_{s}} + \frac{k_{s}}{k_{b}}} \]  

(16)
Overall heat loss coefficient of solar collector \((U_L)\) is the sum of heat loss through the top \((U_t)\), through mid \((U_m)\), through bottom \((U_b)\), in thermal storage \((U_c)\) and side loss \((U_s)\) of the collector given below:

\[
U_L = U_t + U_m + U_b + U_c + U_s \left( \frac{A_t}{A_s} \right)
\]  
(17)

Where \(A_s\) is the side area of solar air collector.

### 3. Results and Discussions

#### Graph between solar radiation and time of the day

Graph between solar radiation and time of the day is shown in Figure 2. Solar radiations from sun become more intense from the morning session to noon session. It is observed that intensity of solar radiation during consecutive days is higher at 1 PM. During peak hours solar radiation varies from 950-1050 W/m\(^2\). After reaching peak values in mid noon hour solar radiation gradually decreases. In the evening session, variations in the intensity of solar radiation are very less and after 7 PM radiation becomes approximately insignificant for heating purpose.

Variation in ambient temperature and time of the day is shown in Figure 3. As solar radiation increasing ambient temperature was also increasing. Ambient temperature was recorded maximum at 1 PM on each day of experimentation. It is due to geographic condition of location. The range of temperature at 01:00 PM varies from 40.5°C to 41.9°C. After reaching a peak value at 1 PM the ambient of air further gradually decreases. It is due to decrease of solar radiation. Variation in atmospheric temperature is less as compare to early session. Thus the slope of the graph between temperature and time of the day is linear from 6:00 PM to 9:00 PM. The minimum temperature obtained on all days of experiments is at 9:00 PM and is approximately equal to 25°C.
Figure 4. Variations in temperature of a glass cover, first absorber plate, second absorber plate and third absorber plate of OGT_A_TPSAC with time of the day for (a) day 1 (0.022 kg/s) (b) day 2 (0.036 kg/s) (c) day 3 (0.046 kg/s) and (d) day 4 (0.056 kg/s).
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Figure 4(a)-4(d) and Figure 5(a)-5(d) show the variation in temperature of different component of OGT_A TPSAC and OGT_A SPSAC with time of the day for all consecutive days of experiments. The temperatures of glass cover in both case increases from the morning session to mid-noon and obtain peak value nearly at 1 PM. The first absorber plate in both cases is in direct solar radiation; therefore, its temperature increases rapidly from morning 10 AM to 1 PM then after it is decreasing till end of the session. There is not much variation has been observed in temperature range of first absorber plate in consecutive days; for OGT_A TPSAC temperature range varies from 74-81°C, while range for OGT_A SPSAC is 70°C-81°C. It is due to same mass flow rate of air and ambient conditions. The temperature range for second and third absorber plate of OGT_A TPSAC is 63°C -67°C and 40°C-44°C respectively. While in OGT_A SPSAC, the temperature range for second absorber plate is 59°C-65°C. It is also observed that during night time, the temperature of the third absorber plate in OGT_A TPSAC and second absorber plate in case of OGT_A SPSAC is more compared to above absorber plates. It is due to releasing of stored heat from bottom portion.

Figure 5. Variation in temperature of glass cover, first absorber plate, second absorber plate of OGT_A SPSAC with time of the day for (a) day 1 (0.022 kg/s) (b) day 2 (0.036 kg/s) (c) day 3 (0.046 kg/s) and (d) day 4 (0.056 kg/s).
Figure 6(a) and 6(b) depict the variation in outlet temperature of air with time of the day in different days of experiments for OGT_{hA} TPSAC and OGT_{wA} SPSAC respectively. Inlet condition for all consecutive days of the experiment is approximately same. Obtained outlet temperature of air in both solar collectors is inversely proportional to the mass flow rate of air till evening hour. Outlet temperatures of air at different mass flow rates in both solar collectors are found to be higher at 1:00 PM. It is due to huge intensity of solar radiation. The maximum outlet temperature for two-pass and single pass condition was of 65.8°C and 59.1°C respectively. It is due to less contact surface area of circulation of air in single pass solar air collector. But afterwards temperatures become approximately same because of working of storage effect. The temperature difference between the inlet and outlet of OGT_{hA} TPSAC is varies from 6 PM to 8 PM and it was in the range of 7°C to 9°C.

Figure 7. Heat gain vs time of the day for (a) OGT_{hA} TPSAC (b) OGT_{wA} SPSAC.
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Figure 7(a) and 7(b) show the variation of heat gain in OGT_{h,A} TPSAC and OGT_{A,SPSAC} with time of the day. Heat gain increases with increasing in mass flow rate (ṁ) for both type of solar air collector. The value of heat gain during peak hours (i.e 10 AM to 01 PM) varies from 378.28 W to 516.23 W for OGT_{h,A} TPSAC and 269.78 W to 389.79 W for OGT_{A,SPSAC}. While on fourth day of experiment, the value of heat gain varies from 461.44 W to 636.62 W for OGT_{h,A} TPSAC and 382.62 W to 585.88 W for OGT_{A,SPSAC}.

Figure 8(a) and 8(b) show the variation in heat losses to ambient air with time of the day for OGT_{h,A} TPSAC and OGT_{A,SPSAC}. Heat losses from solar air collector preferable low as much as possible so that maximum heat energy can be transfer to ambient air for useful applications. Heat losses of OGT_{h,A} TPSAC and OGT_{A,SPSAC} are obtained by adding convective, radiative and conductive losses. On the first day, the value of heat losses varies from 360 W to 460 W for OGT_{h,A} TPSAC and 385 W to 513 W for OGT_{A,SPSAC}. While on the fourth day, the value of heat gain varies from 271 W to 405 W for OGT_{h,A} TPSAC and 301 W to 452 W for OGT_{A,SPSAC}. It means that increasing the value of convective heat transfer results more heat gain for the flowing air stream and reduces heat losses.

Figure 8. Heat loss vs time of the day for (a) OGT_{h,A} TPSAC (b) OGT_{A,SPSAC}.
Figure 9. Thermal efficiency vs time of the day for (a) OGT_{h, A}^p TPSAC (b) OGT_{w, p}^A SPSAC.

Variation in thermal efficiency with time of the day at different mass flow rate for OGT_{h, A}^p TPSAC and OGT_{w, p}^A SPSAC is shown in Figure 9(a) and 9(b) respectively. It is observed that during peak hours (10:00 AM to 01:00 PM), the thermal efficiency is proportional to the solar radiation intensity in both OGT_{h, A}^p TPSAC and OGT_{w, p}^A SPSAC. It is due to the increase of heat gain. But after 01:00 PM, the curve of thermal efficiency of OGT_{h, A}^p TPSAC and OGT_{w, p}^A SPSAC were decline. It is due to reduction in a heat gain and heat loss values. The thermal efficiency of OGT_{h, A}^p TPSAC and OGT_{w, p}^A SPSAC increases with consecutive days of experiment. The thermal efficiency of OGT_{h, A}^p TPSAC during peak hours is in the range of 47.76% to 64.24%. This is 7 to 8% higher than OGTwApSPSAC. During evening session (from 4 PM onwards) the abnormal changes has been observed in thermal efficiency curve of both OGT_{h, A}^p TPSAC and OGT_{w, p}^A SPSAC. These changes are due to thermal storage effect on solar air collectors.

4. Conclusion

When the mass flow rate of air is increased, the thermal efficiency of solar air collector also increases, but the outlet air temperature of air fluid stream decreases. During the evening time the thermal efficiency of both solar air collector increases. It was due to the effect of thermal storage. Maximum thermal efficiency of OGT_{h, A}^p TPSAC under higher mass flow rate condition is 64.24% compared to 59.12% obtained in OGT_{w, p}^A SPSAC. An obtained result firmly indicates that enhancing the heat transfer area and flow rate affects the thermal efficiency.

5. References


Appendix
Specific heat, thermal conductivity, dynamic viscosity and density of airstream are obtained by following empirical formula10.

\[
\begin{align*}
    c_p &= 0.9993 + 0.1494 T_m + 1.61 \times 10^{-4} + 1.61 \times 10^{-8} T_m^2 - 6.7361 \times 10^{-9} T_m^3 \\
    k_{air} &= 0.024 + 0.07673 \times 10^{-4} T_m \\
    \mu_{air} &= 1.718 \times 10^{-5} + 4.620 \times 10^{-8} T_m \\
    \rho &= \frac{353.44}{T_m + 273.15 p r}
\end{align*}
\]