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NUMERICAL STUDY AND THERMAL PERFORMANCE OF THE FLAT PLATE SOLAR AIR HEATERS WITH AND WITHOUT THERMAL STORAGE

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ABSTRACT

The heat transfer characteristic and performance of the flat plate air heater with and without thermal storages are studied experimentally. The mathematical models described that the heat transfer of the plate solar air heater are derived from the energy equation. The implicit method of finite difference scheme is employed to solve these models. The effect of the thermal conductivity of the thermal storage on the heat transfer characteristic and performance is considered. The results obtained from the model are validated by comparison with experimental data. There is a reasonable agreement between the present model and experimental data.

Keywords: flat plate solar air heater, thermal storage, performance.

1. INTRODUCTION

To obtain maximum amount of solar energy of minimum cost the flat plate solar air heaters with thermal storage have been developed. Solar air heater is type of solar collector which is extensively used in many applications such as residential industrial and agricultural fields. The heat transfer characteristics of solar air heater have been widely studied. Veh and lin [1] theoretically and experimentally studied the effect of collector aspect ratio of the collector on that plate upward baffled solar air heaters. Fath [2] studied the performance of the simple design solar air heater. The conventional flat plate absorber is presented by a set of tubes filled with a thermal energy average material org [3] predicted the thermal performance of four common types of single pass solar air heater. Yadav *et al*., [4] analyzed the parametric studied of a suspended flat plate solar air heater. Ahonad *et al*., [5] investigated a packed bed solar air heater. The effect of these parameters on the thermal performance has been investigated and the results have been compared with those flat plate collectors. Al Kamil and Al - Gharab [6] experimentally and theoretically studied the effect of various parameters such as temperature, solar intensity and air flow rate on the performance of a flat plate solar air heater. Ekechukwu and Norton [7] reviewed the various design and the performance evaluation techniques of flat plate solar energy air heating collector for low temperature solar energy crop drying applications. Gao *et al*., [8] numerically studied in natural convection inside the channel between the flat plate solar cover and the sine wave absorber in a cross corrugated solar air heater. Recently Naphon and Kongtragoel [9] applied the mathematical model for predicting the heat transfer characteristic and performance of the various configuration of flat plate solar air heater. P. N. Sarsavadia [10] has developed a solar assisted drier and evaluation of energy requirement for the drying of onion. V. Shanmugam and E. Natarajan [11] conducted experimental of regenerative desiccant integrated solar drier with and with out reflective mirror, Buckola *et al*., [12] have been studied evaluation of mixed mode solar drier. G. N. tiwari [13] analyzed the design parameter of a shallow bed solar crop drier with reflector. N. S. Thakur [14] has been studied heat transfer and function factor correlation for packed bed solar air heater. T. W. Cheng [15] have been studied influences of recycle on performance of baffled double - pass flat plate solar air heater with internal fins affected A. Zamorodian *et al*., [16] have been optimized a semi continuous solar drier for cereals. Shared Kumar and R. P. Saini conducted CFD based performance analysis of a solar air heater [17] duct provided with artificial reoughness. Hikmut Esen *et al*., [18] have been developed ANN and WNN approaches for modeling of a solar air heater. Wenferng Gao *et al*., [19] conducted analytical and experimental studies on the thermal performance of cross corrugated and flat plate solar air heaters. S. Kumar and A. Augustus Leon [20] have been developed mathematical modeling and thermal performance analysis of unglazed transpired solar collectors. As described above, there are many studies on the heat transfer characteristics and performance of the solar air heater with different thermal storage one still limited. The objective of this paper is to study theoretically and experimentally on the heat transfer characteristics and performance of solar air heater with and without thermal storage materials.

2. MATHEMATICAL MODELING

The basic physical equations used to describe the heat transfer characteristics are developed conservation of energy. The method is based on that of Naphon and Kongtragool [21] with the following assumptions.

- flow of air study.
- inside and outside corrective heat transfer coefficient is constant along the length of solar air heater.

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- thermal conductivity of the storage material is constant along the length of solar air heater.

2.1 For top glass cover

$$I\alpha c = hc(Tc - Ta) + hfl(Tc - Tfl) + hr,Cp(Tc - Tp) + hr,ac(Tc - Ta) \quad (1)$$

Where I is the solar intensity, αc is the absorptivity of the glass cover, h_a is the heat transfer coefficient between the ambient and the glass, Tc is the glass cover temperature, Tp is the absorber plate temperature, Ta is the ambient temperature, T_{fc} is the fluid temperature in a channel and hr, Cp and h_r,a_c are the relative heat transfer coefficient between the glass cover and the absorber plate, the glass. Cover and ambient follows:

$$h_{r1}, c_p = \frac{[Tc^2 + Tp^2][Tc + Tp]}{[1/\varepsilon_c + 1/\varepsilon_p - 1]} \quad (2)$$

$$h_{r1}, a_c = \frac{\sigma[T_c^{\ 2} + T_{a2}][T_c + T_a]}{(1/\varepsilon_c - 1)} \quad (3)$$

2.2 For air stream

(a) Without thermal storage materials

$$m.c_p \frac{dT_{f1}}{dx} = h_{f1c}(T_c - T_{f1}) + h_{f1p}(T_p - T_{f1}) \quad (4)$$

(b) With thermal storage materials

$$mc_p \frac{dT_{f1}}{dx} = k_{th} \frac{d^2 Tf_1}{dx^2} h_{f1c}(T_c - T_{f1}) + h_{f1p}(T_p - T_{f1}) \quad (5)$$

Where m is the mass flow rate of fluid per unit width, C_p is the specific heat transfer coefficient the glass cover and working fluid and h_{f1c} is the heat transfer coefficient between the glass cover and working fluid and h_{f1p} is the heat transfer coefficient between the absorber plate and working fluid.

2.3 For absorber plate

$$I.\alpha_p.\tau_c = h_{f1p}(T_p - T_{f1}) + h_r, c_p(T_p - T_c) + h_r, p_b(T_p - T_b) \quad (6)$$

Where α_c the absorptivity of the absorber plate, τ_c is the transmitivity of the glass cover.

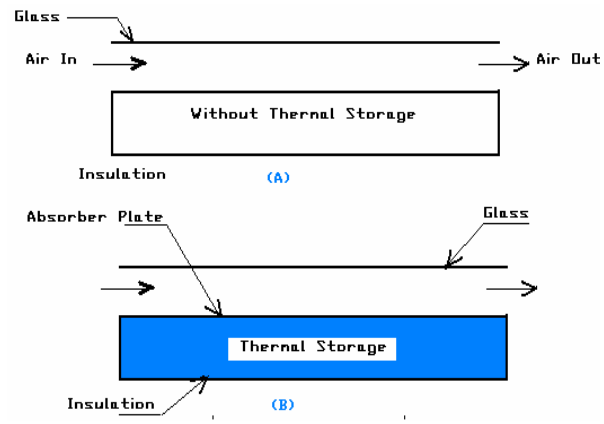

Figure-1. Solar air heater (A) without (B) with thermal storage.

2.4 For bottom plate

$$o = h_{f1b}(T_b - T_{f1}) + h_{rpb}(T_b - T_p) + v_a(T_b - T_a) \quad (7)$$

Where v_a is the overall heat transfer coefficient, T_b is bottom plate temperature, K_p is the thermal conductivity of the thermal storage material.

3. CALCULATION METHOD

The heat transfer characteristic of solar air heater for the model is described by equation (1) to (6). The solar air can be divided into number sections as Figure-2. The calculation is performed section by section along solar air heater length. In order to solve the model the convective heat transfer coefficient for air flowing over the outside surface of the top glass cover and inside heat transfer coefficient are needed. The convective heat transfer coefficient for air flowing over the outside surface of the glass cover is proposed by Mc dams follows:

$$h_a = 5.7 + 3.8v \quad (8)$$

Where h_a is the convective heat transfer coefficient and V is the wind velocity.

Niles *et al*., [23] proposed convective heat transfer coefficient in the channel as follows:

$$N_u = \frac{h_i D_e}{k} = 0.0333 \text{ Re}^{0.8} \text{ Pr}^{1.3} \quad (9)$$

Where N_u is Nusselt number, R_e is the reynold number, P_r is prandtl number, D_e is equivalence diameter of channel as follows:

$$D_e = \frac{4W.H}{(2W + 2H)} \quad (10)$$

Where W is the width of solar air heater, H is the height of the channel.

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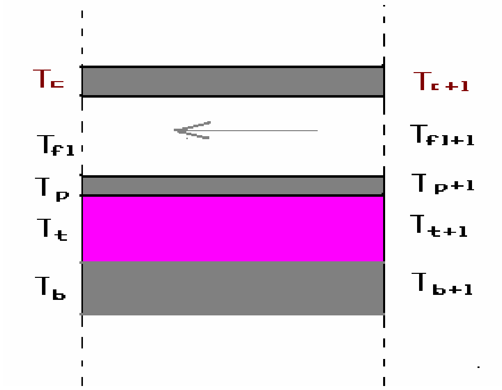

Figure-2. Simulation approach.

In addition the solar air heater configuration and properties of working fluid, as well as operating conditions are also needed. The implicit method of finite scheme is employed to solve these models. Iteration process is described as follows:

- The outlet working fluid temperature is assumed.
- Equation (1) to (6) are solved simultaneously to obtain the glass temperature Tc, working temperature Tf, absorber plate temperature Tp, bottom plate temperature Tb.
- The heat transfer rate Q is calculated.
- The calculation of working fluid temperature at the outlet of channel is compared with the inlet working fluid temperature. If the difference with in 10-6, the calculation is ended and not, computations are repeated until convergence is obtained.

4. RESULTS AND DISCUSSIONS

In the following sections, results of heat transfer characteristic of the solar air heater with and without thermal storage are presented, in order to validate the limited available experimental date.

Numerical calculations are earned out by employing the values of the relevant parameters also taken into account are as follows:

$$\alpha_p = 0.95, \varepsilon_p = 0.8, \tau_c = 0.94, \alpha_c = 0.05, \varepsilon_c = 0.94, v = 1, \frac{w}{m^2 k} \text{ and } v = 3 m/s$$

Figures 3 and 4 show the variation of thermal efficiency of solar air heater with and without thermal storage.

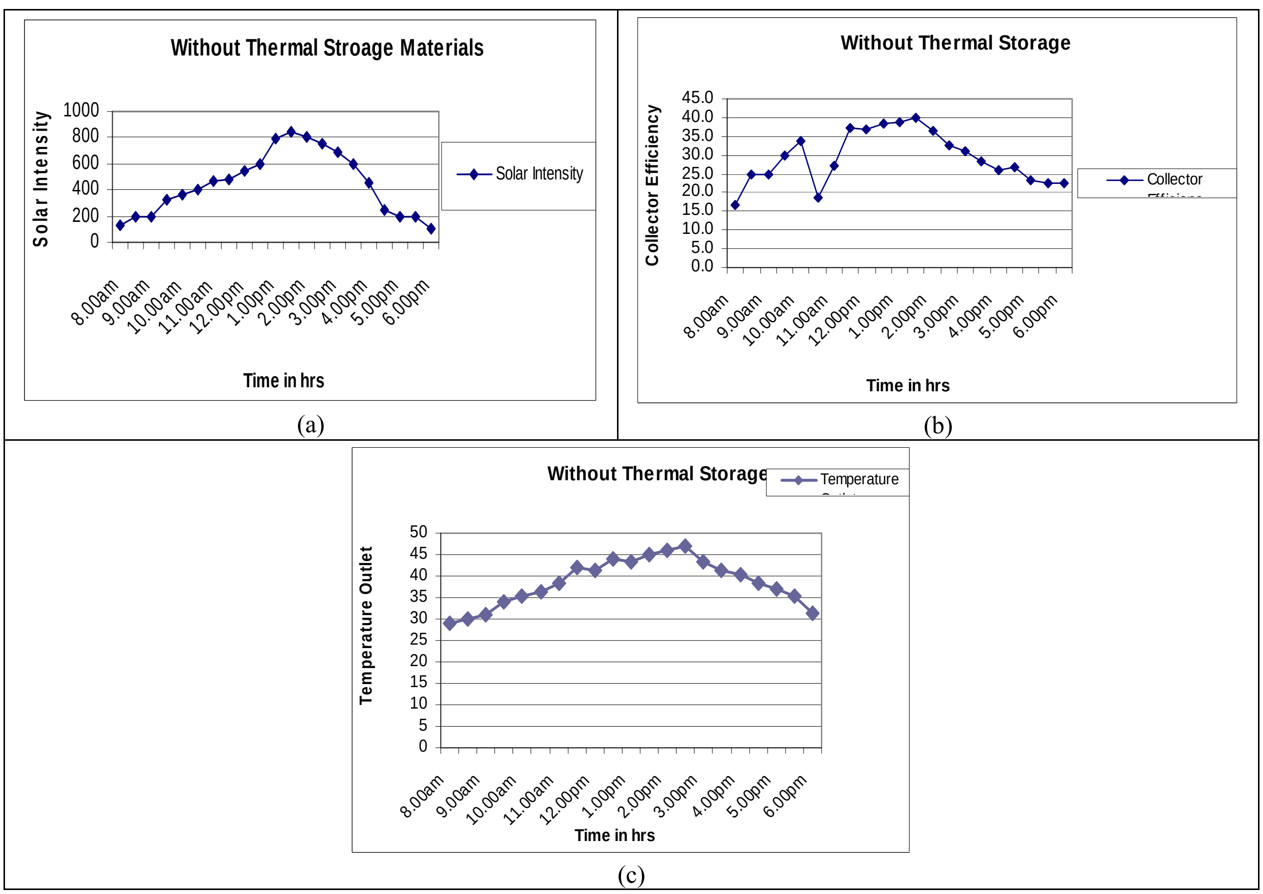

Figure-3. (a) Solar intensity, (b) Collector efficiency and (c) Temperature outlet varying with respect to time for solar air heater without thermal storage materials.

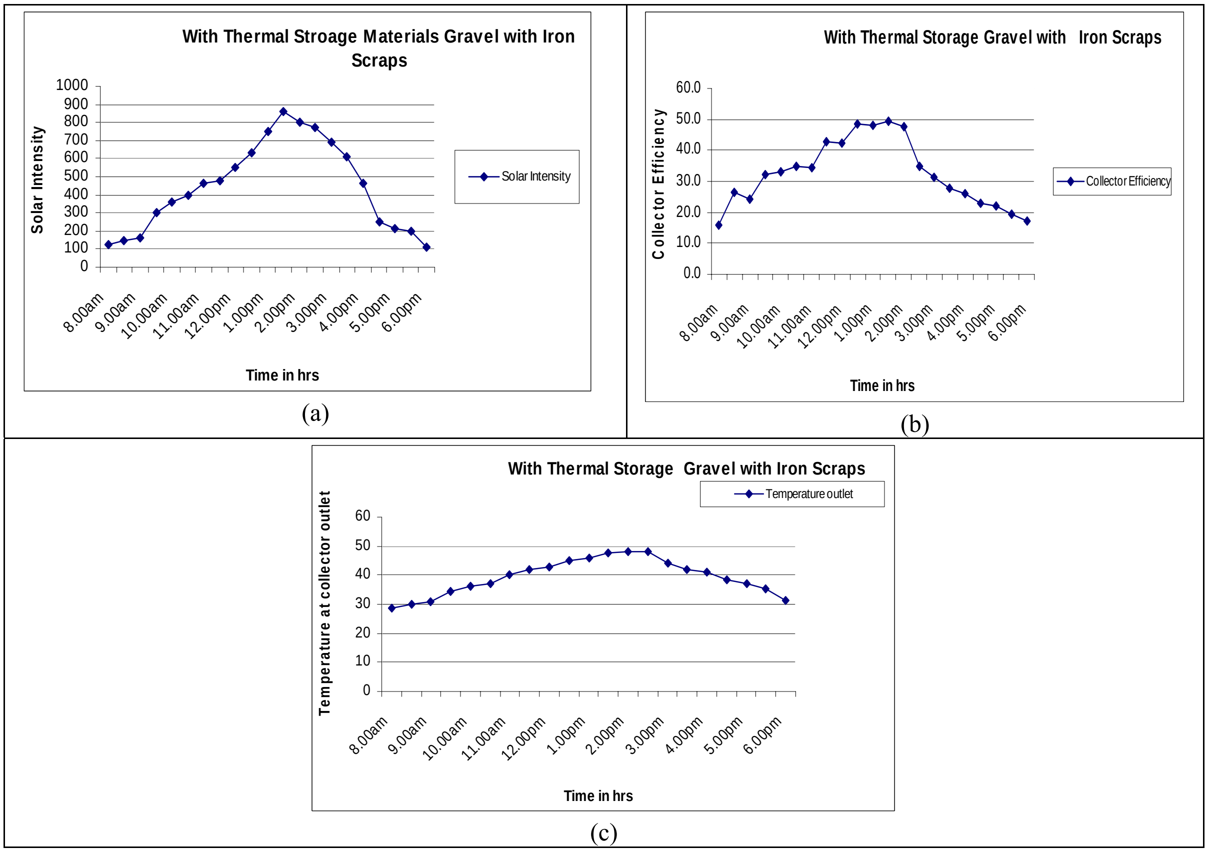

Figure-4. (a) Solar intensity, (b) Collector efficiency and (c) Temperature outlet varying with respect to time for solar air heater without thermal storage materials.

5. CONCLUSIONS

This study presents the mathematical model for predicting the heat transfer characteristic and performance of the air heater with the thermal storage material. The solar air heater with the thermal storage material gives 5-10% higher collector efficiency than that of without thermal storage material. The thermal conductivity of thermal storage material has significant effect o thermal performance of the solar air heater. The model is validated by comparing with experimental data.

Nomenclature

A_c = area of solar air heater (m_2)
D_e = equivalence diameter (m)
h_a = heat transfer coefficient between ambient and glass (W/m^2 K)
h_{fc} = heat transfer coefficient between fluid and glass (W/m^2 K)
h_{fp} = heat transfer coefficient between fluid and glass (W/m^2 K)
h_i = inside heat transfer coefficient (W/m^2 K)
$h_{r,pc}$ = radiative heat transfer coefficient between glass and absorber (W/m^2 K)
$h_{r,ac}$ = radiative heat transfer coefficient between glass and ambient (W/m^2 K)
I = solar intensity (W/m^2)
L = length of solar air heater (m)
N_u = Nusselt number
TS = thermal storage material
R_e = Reynold's number
T_b = bottom plate temperature (K)
T_f = fluid temperature (K)
U_a = overall heat transfer coefficient (W/m^2 K)
V= wind velocity (m/s)
h = collector efficiency
C_p = specific heat of fluid (kJ/kg K)
H = height of channel (m)
K_{ts}= thermal conductivity of thermal storage materials (W/m^2 K)
m = mass flow rate (kg/s)
P_r = Prandtl Number
Q = heat transfer rate (W)
T_a = ambient temperature (K)
T_c = glass temperature
T_p = absorber plate temperature
W = width of solar air heater (m)

Greek symbols

α_c = absorptivity of glass
τ_c = transmitivit of glass
ε_c = emissivity of glass

α_p = absorptivity of absorber plate
ε_p = emissivity of absorber plate

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