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STUDIES ON THE THERMAL PERFORMANCE OF A SOLAR AIR HEATER

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ABSTRACT

In the present work, an attempt has been made to investigate the thermal performance of a flat plate solar air heater. Analytical Thermal model for the flat plate solar air heater is presented. Experimental observations were made in Indian (25º N, 81º E) climatic conditions. Maximum 10 % variation is found between the theoretical and experimental values of outlet air temperature of the solar air heater. Maximum instantaneous collector efficiency of 37% is obtained in the present study.

Key words: Solar collectors, Heat removal rate, Thermal efficiency, Exergy etc.

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1. INTRODUCTION

The rapid growth of population and industrialization are the basic reasons behind increasing energy demand. Conventional energy sources are used to meet this demand. Developing countries are heavily dependent on the fossil fuels for their energy needs. This causes depletion of fossil fuel resources and degradation of environment. Use of renewable energy sources may be alternative to the fossil fuels which also saves the environment. Therefore there is a great need to explore the renewable energy sources and to develop the renewable energy based technologies to meet our energy demand in the present context. Solar energy is the one most abundant renewable energy source and emits energy at rate of $3.8x10^{23}$kW, of which, approximately 1.8x 10^{14}kW is intercepted by the earth [1]. One of the most potential applications of solar energy is the supply of hot air for the drying of agriculture, textile, marine, products, heating a building to maintain a comfortable environment especially in the winter season [2]. Solar air heater is a device in which energy transfer is from a distant source of radiant energy to air. Solar air heaters can be used for many purposes, including crop drying, space heating and for re-generating agents [3-4]. Thermal performance of the solar air collectors depends on the material, shape, dimension and lay out of the collector. Performance improvement can be achieved using diverse materials, various shapes and different dimensions and layouts [5-8]. Thermodynamic analysis is an effective means to obtain precise and valuable information about energy efficiency and losses due to irreversibility in a real situation. The first law is widely used in the engineering practice and is the basis of the heat balance method of analysis that is commonly used in the engineering system performance analysis. The second law involves the reversibility or irreversibility of processes and is a very important aspect of the exergy method of energy systems analysis [9]. Energy analyses on their own incorrectly interprets some processes, e.g., environmental air, when isothermally compressed, maintains its energy equal to zero. Whereas the exergy of the compressed air is larger than zero. However, Exergy data are more practical and realistic in comparison to the respective energy values. Thus, the exergy analysis provides a more realistic view of process [10-11]. The exergy analysis establishes the theoretical limits of ideal operations, very convenient to determine the best collector for a specific application [12]. Several researchers [13-14] have undertaken experimental studies for thermodynamic analysis of solar air heaters. e.g., Karsli [15] determined the first and second law efficiencies of different types of air heating flat-plate solar collectors. Comakli and Yukesal [16] presented the experimental result of four type of solar air heater, only the exergy efficiencies of the collectors were calculated and comparison were made among them on the basis of the exegetic efficiencies. In the present work an attempt has been made to theoretically and experimentally analyze the performance of solar air heater based on the first and second law of thermodynamics.

2. EXPERIMENTAL SETUP

A solar air heater is fabricated, with galvanized iron absorber plate of size (1.21m x 0.92m), The thickness of the cover plate is 0.005m. A flat plate single pass solar air heater was constructed to obtain the thermal efficiencies. Here air is used as a working fluid; the observer plate is painted black in order to absorb the maximum insolation. An exhaust fan is provided at the inlet of the setup, since setup designed for forced convective mode. Insulation is provided at the base of the absorber plate which minimizes the heat loss through base of the system. The other two sides of the solar air heater are also insulated to prevent heat losses from the side walls. A desired flow rate is maintained through fan regulator, the temperature at the different point of the system is measured with the help of J-type thermocouple. Solar intensity and wind velocity are measured with solarymeter and anemometer respectively

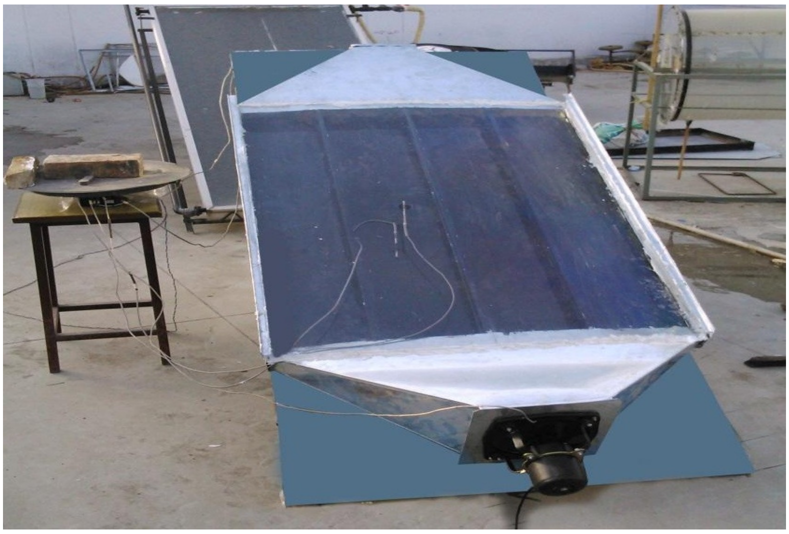

Figure 1 Photograph of the experimental set up of the solar air heater.

The experiment is performed in the Solar Energy laboratory of the Mechanical Engineering Department of SHIATS, Allahabad. The experiments were conducted in the month of May and observations are referred between the shine days of hours 8 am to 5:30 pm.

3. MATHEMATICAL MODELING OF FLAT PLATE SOLAR AIR HEATER

The photograph of the solar air heater under analysis is given in Fig. 1. The heat transfer in the solar air collector is studied by considering the overall energy balance between the components of the collector. Rate equations are used to estimate the convective and radiative heat transfer rates between the components. The model incorporates several empirical relations to calculate the various heat transfer coefficients used in the rate equations. [17].The assumptions used in developing the model are listed below:

- The absorber and back plate temperatures are assumed to be uniform throughout their respective surfaces.
- Air flow through the air heater/passage/surface is assumed homogeneous.
- The effect of pressure drop is negligible.
- Absorber is considered to be diffuse and gray for all absorbed and emitted radiations.
- Losses along the edges are neglected.
- There is no shading on the absorber plate.
- Dust effect on the cover is neglected.
- Temperature drop through the cover is neglected.

The above mentioned assumptions are consistent with studies conducted earlier.

4. ENERGY BALANCE ANALYSIS

For the present analysis, the energy balance equations for the three collector components – absorber plate, air and back plate – have been written by considering the solar input, the heat and mass flow rates across the collector, and thermal losses. The heat transfer modes and heat exchanges in the collector are presented in Fig 2. The energy input to the system is from the solar radiation received on the absorber surface. The net losses from the system are due to convection and radiation losses from the absorber surface and back plate. An energy balance on the absorber plate of area $(1\times\delta_x)$ is given by equation (1-28). [18].

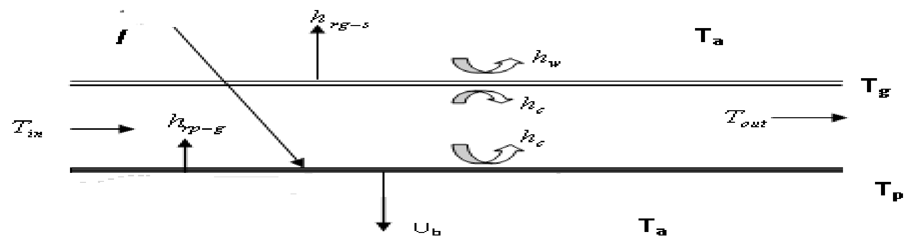

Figure 2 Energy balance on an absorber plate

$$S\,(\delta_x) = u_t\left(\delta_x\right)\left(T_p - T_a\right) + h_{c,p-a}\left(\delta_x\right)\left(T_p - T\right) + h_{r,p-b}\delta_x\left(T_p - T_b\right) \quad (1)$$

An energy balance equation for the air stream volume $(s\times1\times\delta_x)$ gives, where s=height:

$$\left(\frac{\dot{m}}{W}\right)C_p\left(\frac{dT}{dx}\delta_x\right) = h_{c,p-a}\left(\delta_x\right)\left(T_p - T\right) + h_{c,b-a}\delta_x\left(T_b - T\right) \quad (2)$$

An Energy balance equation for the back plate area $(1\times\delta_x)$ is obtained by the given equation:

$$h_{r,p-b}\delta_x\left(T_p - T_a\right) = h_{c,b-a}\left(\delta_x\right)\left(T_b - T\right) + u_b\delta_x\left(T_b - T_a\right) \quad (3)$$

Since u_b is much smaller, so $u_t , u_t \square u_L$, neglecting the back loss u_b, equation 3 T_b gives

$$T_b = \frac{h_{r,p-b}T_p + h_{c,b-a}T}{h_{r,p-b} + h_{c,b-a}} \quad (4)$$

Substituting the value of equation (4) in equation (1):

$$T_a\left(u_L + h\right) = S + u_L T_a + hT \quad (5)$$

Where, $S = \left(\tau\alpha\right)G_t$ (6)

$$h = h_{c,p-a} + \frac{1}{\left(\frac{1}{h_{c,b-a}}\right) + \left(\frac{1}{h_{c,p-b}}\right)} \quad (7)$$

Substituting equation (4) in Equation (2) gives:

$$hT_p = \left(\frac{\dot{m}}{W}\right) C_p \frac{dT}{dx} + hT \tag{8}$$

Equation (5) and equation (8) gives:

$$\left(\frac{\dot{m}}{W}\right) C_p \frac{dT}{dx} = F'\left[S - u_L\left(T - T_a\right)\right] \tag{9}$$

Where F'=collector efficiency factor for air collectors, given by:

$$F' = \frac{\frac{1}{u_L}}{\left(\frac{1}{u_L}\right)+\left(\frac{1}{h}\right)} = \frac{h}{h+u_L} \tag{10}$$

The initial condition of equation (9) are $T = T_i$ and $x = 0$. Therefore:

$$T = \left[\frac{S}{u_L} + T_a\right] + \frac{1}{u_L}\left[S - u_L\left(T_i - T_a\right)\right]\exp\left[-\frac{u_L F'}{\left(\frac{\dot{m}}{W}\right)C_p} x\right] \tag{11}$$

This equation gives the temperature distribution of air in the duct. The temperature of the air at the outlet for the collector is obtained from (11), using $x = L$ and considering $A_c = WL$, Therefore,

$$T_o = T_i + \frac{1}{u_L}\left[S - u_L\left(T_i - T_a\right)\right]\left[1 - \exp\left\{-\frac{A_c u_L F'}{\dot{m} C_p}\right\}\right] \tag{12}$$

Energy gain by the Air stream:

$$\frac{Q_u}{W} = \left(\frac{\dot{m}}{W}\right) C_p (T_o - T_i) = \frac{\dot{m} C_p}{A_c u_L}\left[S - u_L\left(T_i - T_a\right)\right]\left[1 - \exp\left\{-\frac{A_c u_L F'}{\dot{m} C_p}\right\}\right] \tag{13}$$

Heat removal factor F_R :

$$F_R = \frac{\dot{m} C_p}{A_c u_L}\left[1 - \exp\left\{-\frac{A_c u_L F'}{\dot{m} C_p}\right\}\right] \tag{14}$$

Using equation (14) in equation (13) gives:

$$Q_u = A_c F_R\left[S - u_L\left(T_i - T_a\right)\right] \tag{15}$$

Thermal Efficiency of solar collector

$$\eta_c = \frac{Q_u}{I_T A_p} \tag{16}$$

For calculating the radiative heat transfer:

$$h_{r,p-g} = \frac{\sigma\left(T_p + T_g\right)\left(T_p^2 + T_g^2\right)}{\left(\frac{1}{\varepsilon_p}\right)+\left(\frac{1}{\varepsilon_g}\right)-1} \tag{17}$$

To calculate the convective heat transfer coefficients, Reynolds's number should be determined:

$$R_e = \frac{\rho V D}{\mu} = \frac{\dot{m} D}{A_c \mu} \tag{18}$$

Where $A_c = s \times W$ the cross sectional area and D is the Hydraulic diameter of the channel:

$$D = 4\left[\frac{W \times s}{2W}\right] = 2s \tag{19}$$

After determination of Reynolds number, Convection heat transfer coefficients for turbulent flow can be calculated as follows:

$$h_{c,p-a} = h_{c,b-a} = \frac{K}{D} 0.0158 R_e^{0.8} \quad (20)$$

Exergy analysis: By applying exergy balance on solar air heater shown in Fig.1, exergy efficiency can be derived. Exergy balance on a flat plate solar air heater can be generally expressed as Chamoli (2013):

$$\sum \dot{E}_{in} - \sum \dot{E}_{out} - \sum \dot{E}_{loss} - \sum \dot{E}_{change} - \sum \dot{E}_{des} = 0 \quad (21)$$

The exergy input rate $\sum \dot{E}_{in}$ includes the exergy accompanying mass flow rate and the exergy of solar radiation and exergy out rate includes exergy mass flow. The $\sum \dot{E}_{gain}$ is the exergy accumulated by the fluid flow:

$$\sum \dot{E}_{gain} = \sum \dot{E}_{out,f} - \sum \dot{E}_{in,f} \quad (22)$$

Exergy efficiency is the ratio of gain exergy to solar radiation exergy.

$$\eta_{EX} = \frac{E_{gain}}{E_{in,r}} \quad (23)$$

A dimensional less parameter is defined as $\bar{\bar{E}} = \frac{\dot{E}}{\dot{E}_{in,r}}$, is the ratio of exergy to the incoming radiation exergy from the sun.

$$\eta_{EX} = \bar{\bar{E}}_{gain} = 1 - \sum \bar{\bar{E}}_{loss} - \sum \bar{\bar{E}}_{change} - \sum \bar{\bar{E}}_{des} \quad (24)$$

The inlet exergy rate includes the inlet exergy rate with fluid flow and the absorbed solar radiation exergy rate. The inlet exergy rate with fluid flow is given by:

$$\dot{E}_f = \dot{m} C_p \left[T - T_a - T_a \ln\left(\frac{T}{T_a}\right) \right] + \frac{\dot{m}\Delta p}{\rho} \quad (25)$$

Effect of pressure drop is neglected. According to Petela's theory the exact exergy received by solar radiation for a typical collector with surface area of A_p becomes: $\dot{E}_{in,r} = I_T A_p \eta_p$ (26)

η_p is Patella's efficiency of converting radiation energy (i.e. $I_T A_p$) in to work as Bejan (1988):

$$\eta_p = \frac{4T_a}{3T_s} + \frac{1}{3}\left(\frac{T_a}{T_s}\right)^4 \quad (27)$$

Considering the exergy at inlet and outlet then exergy efficiency is expressed as:

$$\eta_{Ex} = \frac{\dot{m}}{I_T A_p \eta_p} \left[C_p \left(T_o - T_i - T_a l_n \frac{T_o}{T_1} \right) \right] \quad (28)$$

5. RESULTS AND DISCUSSION

The Energy balance and rate equations discussed earlier were solved to find the exit air temperature, collector efficiency and heat delivered for a given set of input values, for a given time span. The intermediate values would include the estimation of the heat transfer coefficients and heat flux between the air heater components. For subsequent times, the initial temperatures are specified equal to temperatures values of the respective previous sections. The iterative process is continued until the end time is reached. As initial condition, the absorber plate, black plate and air are assume to be at ambient temperature. The results are discussed in the following section.

Table 1 Input parameters and their values used in the study

S.No	Input Parameter	Range	S.No	Input Parameter	Range
1	Glazing	Single glass	8	Transmttance-absoptance product,	0.84
2	Fluid	Air	9	plate thickness	0.002m
3	Bond conductance	Neglected	10	Absorber plate emittance, ε_p	0.92
4	Length of plate	L=1.21m	11	Glass cover emmitance, ε_g	0.92
5	Width of plate	W=0.92m	12	Stefan-Boltzman constant , σ	5.67×10^{-8} $W/m^2 K^4$
6	Depth of channel	s=0.05m	13	Mass flow rate $\dot{m}$	0.038 kg/s
7	Sun temperature, T_s	5760K	14	Heat lost coefficient u_L	$6.5 W/m^2$

Experiments were conducted in the summer seasons of Indian climatic conditions and in the premises of SHIATS Allahabad. Observations taken for a particular day are used in the study. Fig.3 shows the variation of solar intensity with respect to time of the day. Fig. 4 shows the variation of air temperature at the outlet of the heater. Theoretical instantaneous temperature values were obtained using the expression obtained through the mathematical modeling. Maximum 10 % variation is obtained between the theoretical and experimental values.

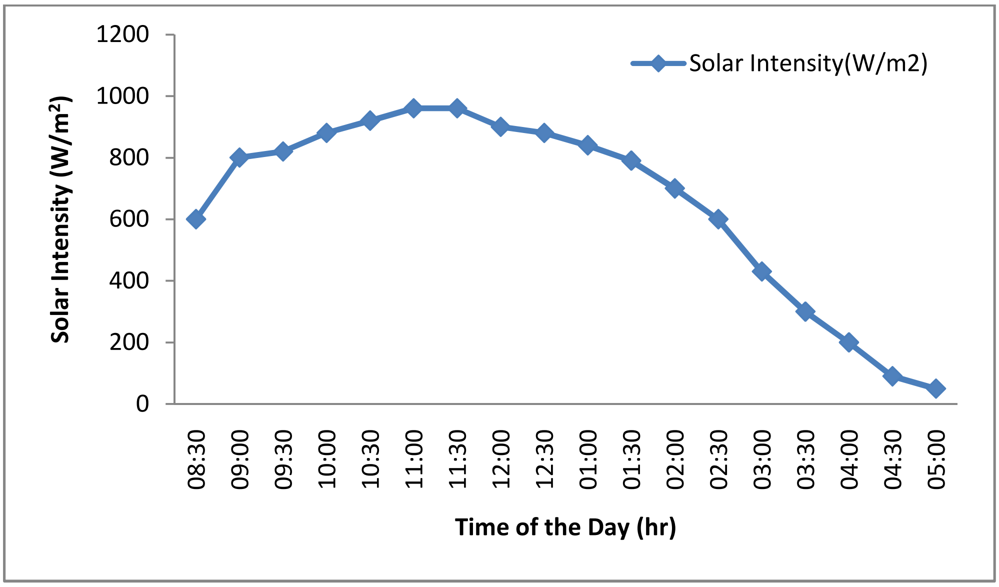

Figure 3 Variation of solar intensity with respect to (w.r.t.) time of the day

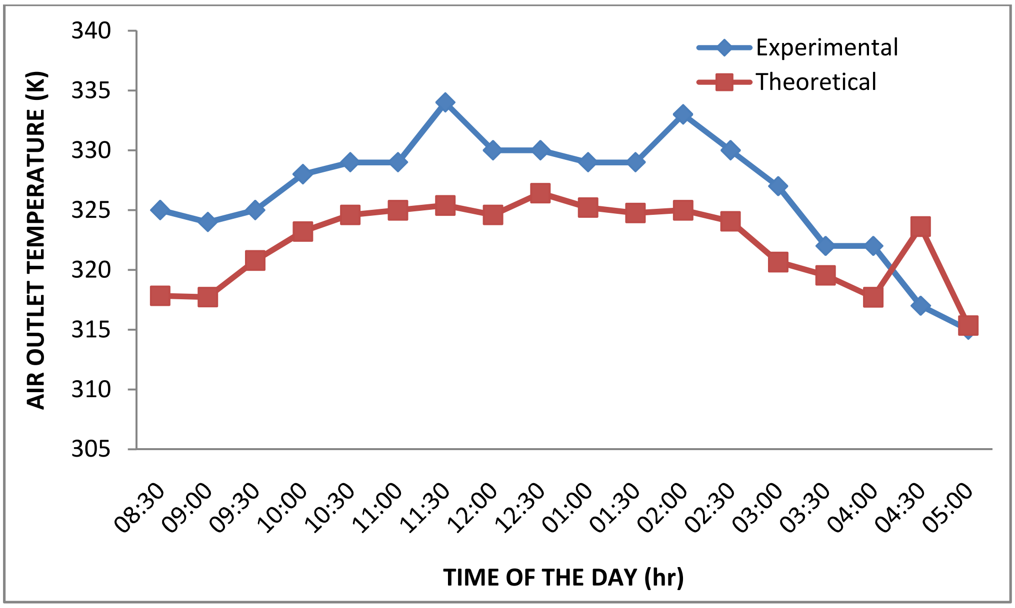

Figure 4 Variation of air outlet temperature of the system w.r.t. time of the day

Variation of collector efficiency is shown in the fig.5. As it is expected average collector efficiency comes out to be 28.21%, whereas 37.9% maximum instantaneous collector efficiency is obtained. Fig.6. Shows the variation of Petela efficiency with respect to the time of the day. As expected a straight line pattern with average value of 0.926 is obtained. Theoretical outlet air temperature is used to obtain theoretical exergy efficiency. Fig.6 shows the variation of theoretical and experimental exergy efficiency of the system. As solar intensity go on decreasing with respect to time of the day, experimental and theoretical exergy efficiencies are decreasing and reaching to a minimum at the end of the day. Exergy loss and distruction is found more in the solar air heater. Average experimental exergy efficiency of 0.74% and average theoretical exergy efficiency of 0.44% is obtained.

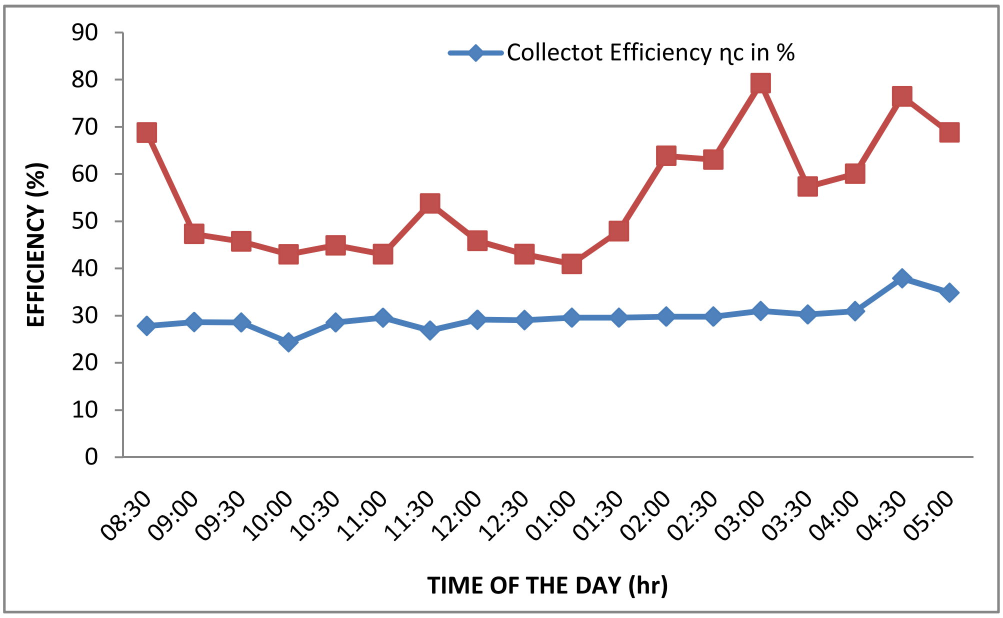

Figure 5 Variation of efficiencies (Collector and Thermal) of the system w.r.t. time of the day

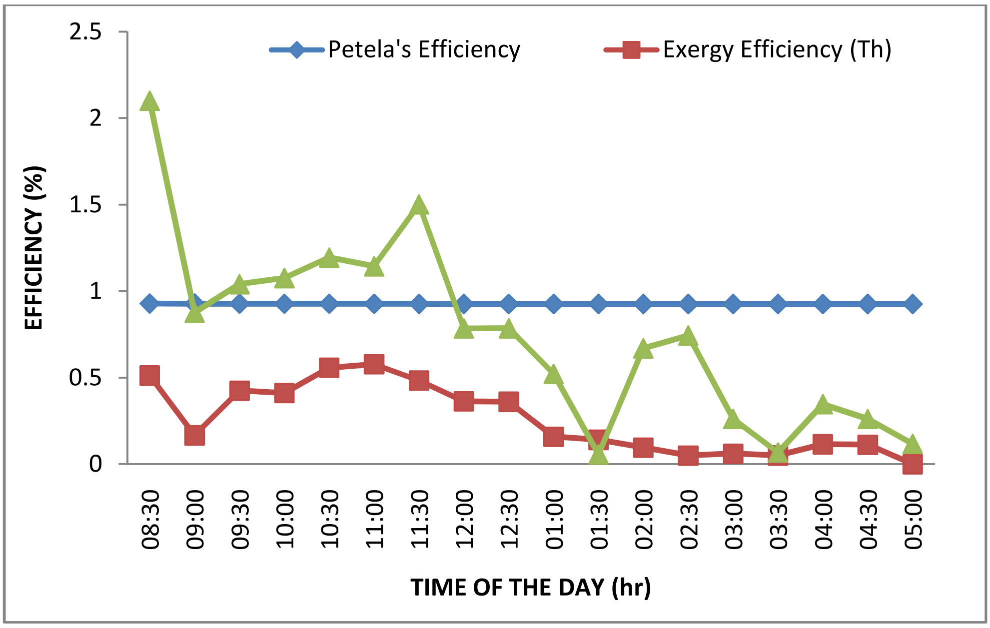

Figure 6 Variation of Exergy efficiencies theoretical (Th), and experimental (Ex) and Petela's efficiency of the system w.r.t. time of the day

6. CONCLUSION

This presents the details of a Mathematical model for a solar air heater using heat transfer expressions for the collector components and empirical relations for estimating the various Heat transfer coefficients. It predicts the thermal performance of a solar air heater over a wide range of operating condition namely solar intensity and inlet air temperature. Average collector efficiency of 28.21% and maximum instantaneous thermal efficiency of 79% is obtained. Exergy analysis shows the maximum loss and exergy destruction in the system. Average exergy efficiency of 0.75% is obtained in the present study.

6.1. Nomenclature

A Area (m^2)	h Heat transfer coefficient (W/m^2-K)
C_p Heat capacity of the fluid (K j /Kg k)	S Absorbed solar radiation (W/m^2)
F' Collector efficiency factor	R_e Reynolds's No.
F_R Heat removal factor	N_u Nusselt No
I_T Incident Solar radiation (W/m^2)	L Collector length (m)
U_L Over all heat loss coefficient (W/m^2-K)	W Collector width (m)
T Temperature (K)	s Depth of air channel (m)
Q Heat transfer rate (Watt)	$\dot{m}$ Mass flow rate (Kg/sec)
D Hydraulic diameter (m)	

6.2. Greek symbols

a absorptance	σ Stefan-Boltzman constant
η Efficiency	ε emmitance
τ Transmittance	μ Dynamic viscosity (Kg/m-s)
(τα) Effective product of transmittance-absoptance	k Thermal conductivity (W/m-k)

6.3. Subscripts

a ambient	s Sun ,Depth of channel
o outlet	g Glass
i inlet	E_x Exergy
exp exponential	c Convective
p plate , petla	r Radiative
c collector	t Total
u useful	b Back

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