Impact of Top Cover on Thermal Performance of CPC Solar Air Heater

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Abstract - *Solar air heater is one of the useful technologies in agriculture sector. In hot climatic countries like India solar air heater can play vital role in drying of different foods, crops as well as different industrial process. In the present work Cylindrical Parabolic Solar Collector (CPC) was tested using selectively coated (Black Copper) copper receiver. The main objective of present study is to assess the impact of the top cover on CPC. The glass of 5mm thickness was used as a top cover on the CPC. Experimental results show that an instantaneous efficiency 60% can be achieved using top cover.*

Keywords: **Solar air heater, cylindrical parabolic solar collector, Selective receiver coating, solar concentrating collector with top cover.**

NOMENCLATURE

A - Area (m^2)
A_r- Area of receiver
A_c- Aperture area of collector
CPC – Cylindrical Parabolic Collector
C- Concentration ratio
C_p - Specific heat at constant pressure (J/kg K)
D- Diameter of receiver (m)
F_R - Heat removal factor
F' - Collector efficiency factor
F- Focal length of parabola (m)
h_f-Heat transfer coefficient on the inside surface of the receiver (W/(m^2K))
h_{p-c}-Natural Convection heat transfer coefficient (W/(m^2K))
h_w-Forced convection heat transfer coefficient (W/ (m^2K))
h_{total}-Total convective heat transfer coefficient (W/(m^2K))
h_{rad}-Radiative heat transfer coefficient (W/(m^2K))
I_b - Incident beam radiations (W/m^2)
I_t –Total beam radiations (W/m^2)
K- Thermal conductivity (W/ (m K))
L- Length of receiver (m)
W- Width of trough (m)
ṁ- Mass flow rate (kg/s)
σ -Stefan Boltzmann Constant = 5.67 x 10-8 (W/ (m^2 K^4))
n- Number of day in year
Q_u -Useful heat gain for without top cover (W)
$(Q_u)_c$ - Useful heat gain for with top cover (W)
Q_l - Heat loss (W)
dq_u – useful heat gain rate
S- Absorbed solar flux (W/m^2)
T_{in} - Inlet temperature of air (°C)
T_{pm} –Mean temperature of receiver (°C)
T_a – Ambient temperature of air (°C)
T_{out} - Outlet temperature of air (°C)
T_c - Top cover temperature (°C)
T_{sky} – Temperature of sky = (T_a- 6) (°C)
U_l – Overall heat loss coefficient for without top cover (W/(m^2K))
$(U_l)_c$ - Overall heat loss coefficient for with top cover (W/(m^2K))
Gr- Grashoff Number for natural convection
Nu- Nusselt Number for Convective heat transfer
Re –Reynolds number
Pr –Prandtl number
g - Acceleration due to gravity (m/s^2)
V- Wind speed (m/s)
ɣ - Intercept factor
α – Absorptivity of receiver
τ – Transmissivity of top glass cover
$(\tau\alpha)_{av}$-Average transmissivity_abosrptivity product
ε - Thermal emissivity of receiver
P-Specular reflectivity of the concentrator surface
η-Solar Collector instantaneous efficiency (%)

1. INTRODUCTION

Solar air heaters have been investigated all over the world for drying of different food items as well as different industrial heating processes. Many of them have been proposed to study the design improvements, alternatives geometries, heat loss characterization, drying of specific crops and many more. Thus solar air heating is one of the attractive fields of research in hot climatic countries. Number of studies available in the literature highlights different solar drying technologies.

The reviews of the various designs, and operational principles; Low cost solar drying technologies, wide variety of practically realized designs, of solar energy drying systems was presented by Ekechukwu and Norton [1] Chua and Chou [2] Patel and Brahmbhatt [3] explicated the numerical simulation for computing the thermal performance of an air heater with a truncated cylindrical parabolic concentrator (CPC) having a flat one-sided receiver and experimental validation of a cylindrical parabolic concentrator (CPC) are presented. Yadav et al. [4] reported the performance of parabolic trough air heater to enhance the heat transfer using different type of reflectors. Tchinda [5] presented a mathematical model for computing the thermal performance of an air heater with a truncated cylindrical parabolic concentrator having a flat one-sided absorber. The effects of the air mass flow rate, the wind speed and the collector length on the thermal performance of the air heater are investigated. Nasir [6] studied the performance of the solar air heater having a double-flat-plate collector with galvanized square pipes which are assembled into a parabolic cylindrical trough solar collector. Hamid Beckan [7] reported designs of solar air heaters such as the matrix air heater; Plastic air heater and the transpired air heater with absorption characteristics and temperature profiles in matrices made by stacking copper wires screens. Patel and Patel [8] enlighten experimental investigation of parabolic trough collector with different materials. Mohammad [9] has reported an analysis of solar air heater which is operated in the two- pass mode and has a matrix in the second channel. Experimental investigation on an air heater with a wire mesh screen matrix have been reported by Varshney and Saini [10] Bansal et al. [11] have developed all plastic solar air heaters fabricated from flexible plastic sheets. The potential of solar heaters for the drying process of a few cash crops were also assessed in detail by Bansal [12].The design and operation of solar agricultural dryers which have been used to dry a variety of crops all over the Caribbean region were described by Headley [13]. A solar dryer, which consisted of a solar air heater and a drying chamber, was developed by Tiris et al.[14] Shanmugam and Natarajan [15] reported the thermal performance of desiccant integrated FPC based solar dryer to investigate under the hot and humid climatic conditions of Chennai, India to dry the green peas. Chaudhary et al. [16] explained an energy and exergy analysis of solar drying of jackfruit leather in a solar tunnel dryer. Akpinar and Bicer [17]; Kadam and Samuel [18]; Tayeb [19]; conducted experiments on forced convective FPC based solar dryers to dry different food items such as cauliflower, corn, peanuts and food grains. Several techniques such as twisted tape [20]; with rectangular winglet vortex generator [21]; packed bed of spherical capsules with a latent heat storage [22]; V-shaped baffles [23]; longitudinal fins [24]; multi arc shaped ribs [25]; roughened absorber plate [26] have been used to enhance the heat transfer in the different designs of solar air heaters/dryers. Novel designs of solar air heaters/dryers have been investigated by several researchers with the motive of increase in thermal efficiency [27, 33, 34, 35]; efficient thermal storage [28, 29]; double pass heat augmentation techniques [30, 37]; hybrid heating [31, 32, 36, 39]. Patel et al. [38] developed spiral solar air heater and reported the thermal performance. Varun et al. [40] developed indirect type solar dryer and evaluated its performance for natural convection and forced convection modes of drying. Sagade et al. [41, 42, 43] reported the thermal performance of CPC based solar water heater with top cover and different selectively coated receivers.

From the discussed literature the author have not found evidences of experimentations on the CPC using top glass cover for air heating applications. Therefore in the present work focuses on the impact of top glass cover on the thermal performance of CPC air heater. A metallic tube of Copper coated with Black Copper (hereafter denoted as BC) as a selective receiver coating is used as line focusing receiver. Ambient air is used as a working fluid which is heated by using solar energy and further use for various applications in industry as well as agriculture.

2. EXPERIMENTAL WORK

Fig. 1 shows the test set up.

At the start of experiments the mass flow rate of air flowing through the receiver adjusted at (m= 0.00075m/s) and kept constant throughout the experiment. Solar radiation falls on the CPC and concentrates on the line focusing receiver. Thus the surface temperature of receiver increases and it turn the air flowing through receiver get heated.

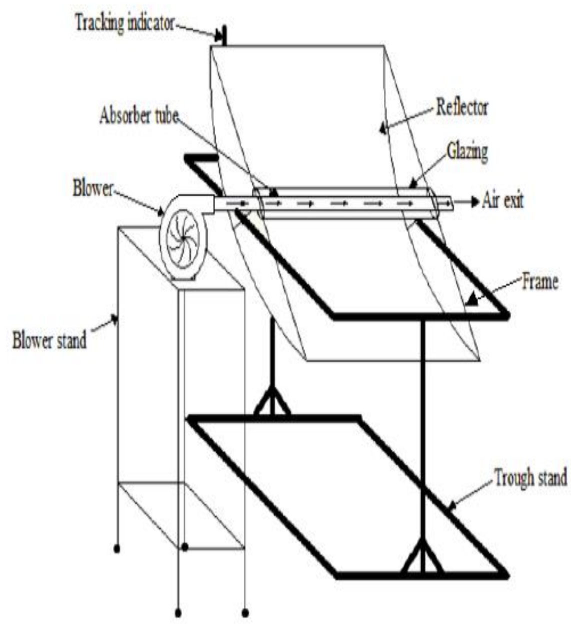

Fig.1. Schematic Diagram of Experimental Setup

Solar radiations were measured by pyranometer (Dyna lab India). Temperature of ambient air (T_a), inlet air (T_{in}), outlet air (T_{out}) and surface temperature of receiver was measured using the J type thermocouple. The surface temperature of receiver is measured at four locations on the line and the average value (T_{pm}) has been used in the calculations. Wind velocity was measured using hand held anemometer. All the measuring instruments were wired to the data logger (Unilog, India).

Specifications of line focusing CPC and other system components are enlisted in table 1.

Table 1: Parameters of CPC-receiver system and top cover

Sr No.	Particulars	Symbol	Value
1	Width of CPC	W	1.030 m
2	Length of CPC	L	1.820 m
3	Depth of CPC	D	0.300 m
4	Focal length of CPC	F	0.221 m
5	Outer dia of receiver	D_o	0.019 mm
6	Inner dia of receiver	D_i	0.0175 mm
7	Mass flow rate of air	$\dot{m}$	0.00075
8	Reflectivity of material	ρ	0.85
9	Transmisivity of cover	T	0.85
10	Rim angle	Ψ	90°
11	Absorptivity of Black Cu	α	0.94
12	Emissivity of Black Cu	ε	0.06

The experiments were carried out from 11.00 a.m.to 3.00 p.m. at (16˚E, 74˚N) under test conditions as $I_t \geq 500$ W/m^2, 20˚C ≤ T_a ≤ 40˚C and wind speed ≤ 3 m/s. Manual tracking was provided as per necessity. It is to be noted that inaccurate tracking of CPC will lead to incorrect estimation of thermal performance of CPC. It is to be measured separately keeping the results of experimental values presented here. The absolute error of 1% in the solar radiations measurement 0.5% in temperature and 1m/s in wind measurement were possible. It is assumed that,
i) Change in the temperature of air is expected to have minimum impact on the work fluid; ii) The measurement errors of different parameters (I_t, T_a) will be replicated in the estimation of overall heat transfer but their impact on the overall heat transfer process is expected to be negligible; iii) flow is fully developed; iv) Properties of air have been estimated at the mean temperature, $T_{mean} = \frac{T_{in}+T_{out}}{2}$; v) heat transfer process is evaluated under steady state condition; vi) impact of friction is neglected.

3. BASIC THEORY

3.1 Performance of CPC without top cover

The performance of CPC is analyzed using the energy balance equation on an elementary slice *dx* of the receiver, at a distance *x* from inlet, yields following equations.
Total heat gained by the collector is given by equation (1) [42].

$$\frac{dq_u}{dx} = I_b r_b \rho \gamma\, (\tau\alpha)_b + I_b r_b D_o\, (\tau\alpha)_b \left(\frac{D_o}{W-D_o}\right) - U_l \pi D_o (Tpm - T_a)$$
---------- (1)

Heat transfer coefficient on the inside Surface of the receiver (h_f) is given by equation, (2) [42].

$$f = Nu\, \frac{K}{D_i} \text{ ----------- (2)}$$

Where,

$$Nu = \frac{hD_i}{K}$$

The Reynolds number (Re) for air is given by equation (3) [42]

$$Re = \frac{v\, D_i}{\vartheta} \text{ .... (3)}$$

Where,

$$v = \frac{\dot{m}}{(\frac{\pi}{4} {D_i}^2 \rho)} \text{......... (a) [42]}$$

In the present case the value of Re for air is (3072.48), For the cylinder having circular cross section the value of Nusselt number is find out by following co-relations- (from Zukauskas 1972 and Jakob, 1949) [38] from report, and given by equation no. (4)

$$Nu = 0.023\, Re^{0.8}\, Pr^{0.4} \text{ ...... (4)}$$

3.1.1 Calculation of heat loss coefficient (U_l) from the receiver for without top cover

In the case, CPC–receiver system is open to sky; therefore the natural and forced convective heat losses are expected to be higher. The natural convective heat losses occurs because of air flows at the right angle as well as in the cross direction across the receiver. Also, the wind velocity at the experimental location causes forced convective heat losses from the receiver.

Thus the natural convective heat loss coefficient (h_{p-c}) from the receiver is estimated using equation (5) [42].

$$h_{p-c} = Nu \left(\frac{K}{D_o}\right) \text{ (W/(m}^2\text{K)) -------- (5)}$$

For cylinder, if $10^4 < GrPr < 10^9$ then, there is Laminar flow.
For this case Nusselt's number for natural convection is calculated by equation (6) [41]

$$Nu = 0.53\, (GrPr) \text{ --------------- (6)}$$

Where,
Prandtl Number (Pr)-

$$Pr - \frac{\mu C_p}{K} \text{ ------------------------ (b)}$$

Grashof's Number (G_r)

$$G_r = \frac{{D_o}^3(\beta g \Delta t)}{\vartheta^2} \text{ ------------------(c)}$$

Where,
Volumetric Coefficient of Expansivity (β)

$$\beta = \frac{1}{(T_{mean}+273)} \text{ (1/K) ------- (d)}$$

In case of, without top cover the receiver tube is open to ambient, thus the flow of wind causes forced convective heat losses from the receiver.
Therefore the forced convective heat transfer coefficient (h_w) is given by equation (7) [41]

$$h_w = N_u\left(\frac{K}{D_o}\right)\ (W/(m^2K)) \text{ --------------------(7)}$$

Total convective heat transfer coefficient is given by equation (8) [41].

$$h_{total} = h_{p\text{-}c} + h_w\ (W/(m^2K)) \text{ -----------------(8)}$$

Calculation of Radiative Heat Loss coefficient (h_{rad}) from the Receiver
For the line focusing receiver of CPC, the radiative heat loss coefficient is determined using equation (9) [41].

$$h_{rad} = \frac{\sigma\epsilon(T_{pm}^4 - T_{sky}^4)}{T_{pm} - T_{sky}}\ (W/(m^2K)) \text{ ---------- (9)}$$

Therefore overall heat loss coefficient in case of without top cover (U_l) is given by equation (10)

$$U_l = (h_{total} + h_{rad})\ (W/(m^2K)) \text{ ---------- (10)}$$

3.2 Performance of CPC with top cover

In case of CPC with top cover, the equation (1) can be modified as equation (10)

$$\frac{dq_u}{dx} = I_b r_b \rho \gamma\ (\tau\alpha)_b + I_b r_b D_o\ (\tau\alpha)_b\ \left(\frac{D_o}{W - D_o}\right) - (U_{l(C)})(\pi D_o(Tpm - T_a)) \text{------------ (10)}$$

3.2.1 Calculation of overall heat loss coefficient from the receiver with top cover

When CPC is covered with top cover, it is assumed that the impact of wind velocity is nullified. Thus, the empirical equation (11) suggested by Malhotra et al. [42] [42] is used to estimate overall heat loss coefficient

$$U_l = \left[\frac{M}{\left(\frac{C}{T_{pm}}\right) - \left(\frac{T_{pm} - T_a}{M + f}\right)^{0.252}} + \frac{1}{h_w}\right]^{-1} + \left[\frac{\sigma(T_{pm}^2 + T_a^2)(T_{pm} + T_a)}{\left(\frac{1}{\epsilon_{p+0.0425M(1-\epsilon_p)}}\right) - \left(\frac{2M + f - 1}{\epsilon_c}\right)} - M\right] \text{ ---------- (11)}$$

Where,

$$f = \left(\frac{9}{h_w} - \frac{30}{h_w^2}\right)\left(\frac{T_a}{316.9}\right)(1 + 0.091M)$$

$$C = \frac{204.429(\cos\beta)^{0.252}}{L^{0.24}}$$

$$hw = 5.7 + 3.8xv$$

L= spacing,M=1 (No. of covers)

Assumptions- $320 < T_{pm} < 420K$, $260 < T_a < 310K$, $0.1 < \varepsilon_p < 0.95$, $0 < v < 10$ m/s, $1 < M < 3$, $0 < \beta < 90^0$

3.3 Total heat loss from the CPC- receiver system (Q_l)

Total heat loss from the CPC- receiver system (Q_l) in both the cases i.e. with and without top cover can be estimated using equation (12) [42]

$$Q_l = A_r\ (U_l/U_{lC})(T_{pm} - T_a)\ (W) \text{ ---------- (12)}$$

3.4 Collector efficiency factor (F')

Collector efficiency factor is given by equation (13)

$$F' = \frac{1}{U_l\left[\frac{1}{U_l} + \frac{1}{D_i}\frac{D_o}{h_f}\right]} \text{ ---------- (13)}$$

3.5 Heat removal factor (F_R)

Heat removal factor is given by equation (14)

$$F_R = \frac{\dot{m}\,C_p}{\pi\ D_o\ U_l\ L}\left[1 - e^{-\left\{\frac{F'\pi\ D_o U_l L}{\dot{m}\ x C_p}\right\}}\right] \text{---------- (14)}$$

3.6 Useful heat gain (Q_u)

Useful heat gain is given by equation (15)

$$Q_u = F_R\ (W - D_o)\ L\left[S - \frac{U_{l/(U)C}}{C}(T_{in} - T_a)\right]\ (W) \text{ ------------ (15)}$$

3.7 Instantaneous efficiency (η_i)

Instantaneous efficiency is given by equation (16)

$$\eta_i = \frac{Q_u}{I_b r_b\ W\ L}\ (\%) \text{ ----------------------- (16)}$$

4. RESULTS AND DISCUSSIONS

The CPC-Receiver system is experimented with and without top cover to reach the conclusion.

4.1 Variation of total heat loss from the receiver and CPC efficiency

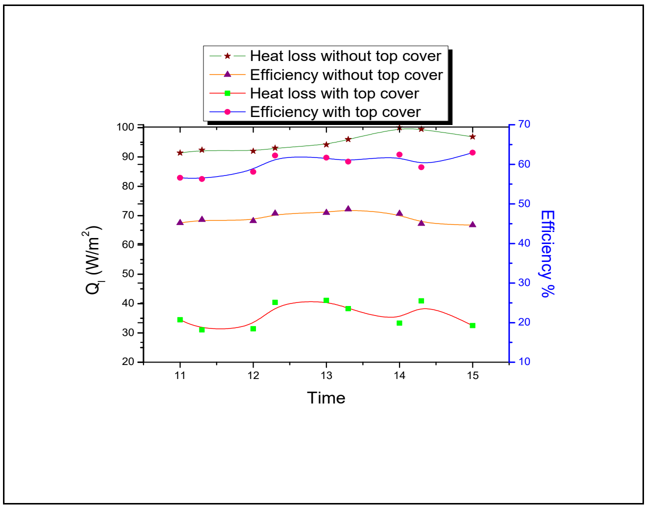

Fig.2: Variation of heat loss and efficiency with time (with and without top cover)

Fig 2 show variation of heat loss and efficiency with time for with and without top cover. In case of CPC- receiver, when top cover is not used the natural & forced convective heat transfer coefficient can be calculated using equation no. (5) and (7) respectively. The air flows at right angle as well as in the cross direction across the receiver. The mean value of h_{pc} and h_w is estimated to be ,6.42 ± 0.32 W/(m^2-K) and 11.08 ± 1.25W/(m^2 –K),respectively and the mean values of total convective heat loss coefficient is (17.40) W/(m^2-K).Similarly the mean value of h_{rad} 0.0015 ± 0.00035 W/(m^2-K). Thus, the mean value of overall heat loss coefficient (U_l) 22.31± 3.10 W/ (m^2-K). Therefore the mean value of total heat loss (Q_l) from CPC receiver system is 91.74 ±5.24 W. Therefore the experimental determined value of thermal efficiency of CPC receiver system is 46.44 ±1.44 %.On the other hand, when top cover is used it is seen that the impact of wind at the receiver is minimized. This reduces natural and forced convective heat losses from the CPC receiver system. Thus the decrease is seen in the mean value of overall heat loss coefficient as 2.03 ± 0.54 W/ (m^2-K) in turn it reduces the value of $(Q_l)_c$ with mean value as 21.80 ± 5.22 (W). It is seen
that the mean value of thermal efficiency of CPC receiver system with top cover is increased to 60.56 ± 2.89%.therefore a clear impact of top cover can be seen on the thermal performance of proposed system.

4.1 Variation of useful heat gain by air and outlet air temperature

Fig.3 depicts effect of useful heat gain by air flowing through the receiver on temperature gradient. Temperature gradient is defined as the temperature difference between outlet and inlet temperature of air flowing through receiver and gains heat.

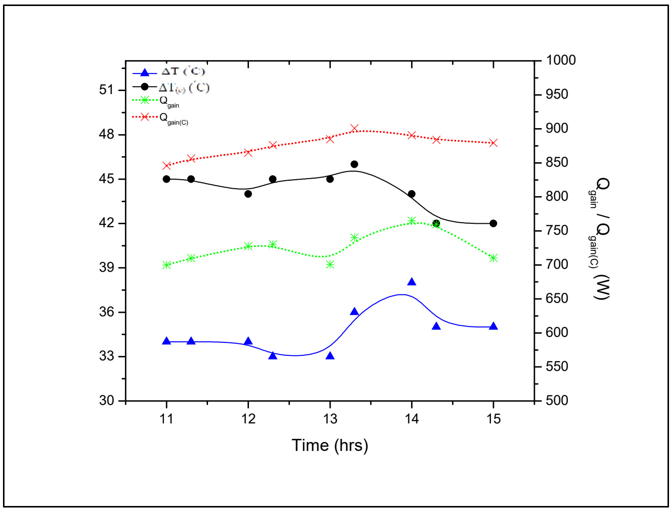

Fig.3. Effect of useful heat gain by air on temperature gradient (with and without top cover)

From fig. 3 it can be seen that with top cover useful heat gain starts to increases as incident solar radiations on the collector increases and reaches the maximum value around the ±30 minutes of solar noon and then starts to decline. When top cover is not used the variation in the useful heat gain can be clearly seen from the fig.3 .It is because natural and forced convective heat losses from the receiver .the mean values of Q_u and $Q_{u(c)}$ are estimated to be (value) respectively. The effect of useful heat gained by air reflects in the form of increase in the temperature gradient in case of top cover .the mean values of ΔT and $\Delta T_{(c)}$ are (value) respectively. Thus the impact of top cover can be clearly seen on useful heat gain and temperature gradient for the receiver- CPC system proposed in the present case.

4.1The plot of Efficiency vs $(T_f\text{-}T_a)/I_t$

In the present case the value of mass flow rate of air kept constant and the efficiency curve of CPC is determined.

The plot of efficiency values against parameter $(T_f - T_a)/I_t$, is Plotted for with and without top cover (Fig.4 and 5) which is seen to be a straight line.
The intercept on the y axis being [F' $(\tau\alpha)_{av}$ A_r /A_c] & the slope being (F'U_l A_r /A_c).Here the collector efficiency is obtained from 11.00 am to 3.00 pm and the value of I_t ≻ 500 W/m^2, and ± 120 min. of solar noon at a given location. In case of concentrating solar collectors, the beam radiation is dominant parameter and also the angle of incidence of beam radiations is always small. Therefore the term $(\tau\gamma)_{av}$ in the parameter [F_R $(\tau\gamma)_{av}$ A_r / A_c] is effectively the transissivity – absorptivity product for normal incidence beam radiation. For The CPC- receiver system described in the present work the value of parameter (T_f -T_a) /I_t is 26.52 ± 3.56 and the thermal efficiency is seen to be 46.44 ± 1.44 % and 60.56 ± 2.89 % respectively with and without top cover.

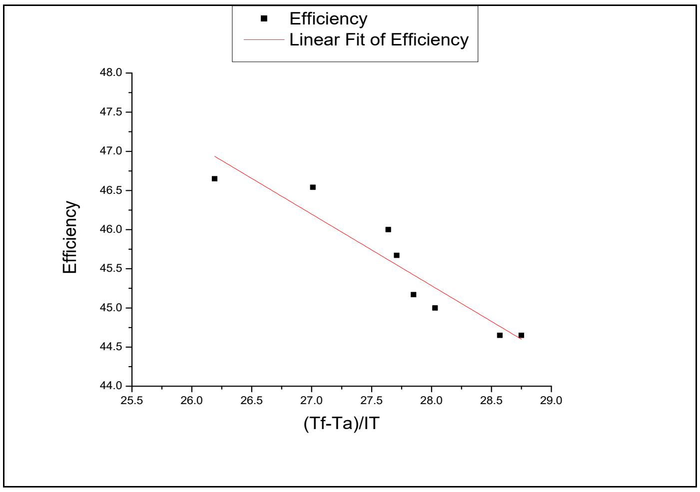

Fig.4. Efficiency vs $(T_f-T_a)/I_t$ for without top cover

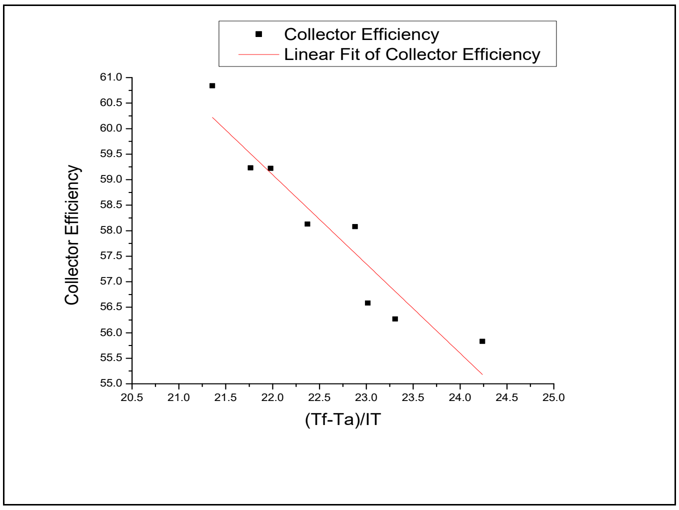

Fig.5. Efficiency vs $(T_f-T_a)/I_t$ for with top cover

5. CONCLUSION

In the present work a CPC- receiver system is presented for air heating applications for hotter climatic conditions. The proposed system yields an average temperature gradient of 40.8 °C and 34.66 °C respectively with and without top cover. Also, the maximum gradient of 46°C and 38°C respectively with and without top cover .also, the average outlet air temperatures reached with and without top cover are 79.44°C and 70.22°C respectively. Thus the impact of top cover is clearly visible. Therefore the proposed system can be efficiently used for drying of different kinds of food items as well as industrial process heat applications.

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