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ScienceDirect
Energy Procedia 57 (2014) 3058 – 3070
Energy Procedia

2013 ISES Solar World Congress

Experimental Investigations on Mild Steel Compound Parabolic Reflector with Aluminum Foil as Selective Surface and Top Cover

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Abstract

The paper explores the experimental results of the prototype compound parabolic trough solar collector made of mild steel and its surface coated with an aluminum foil of thickness 10 micron as a selective surface. This prototype has been tested with top cover for instant water heating and steam generation application. The performance of collector has been evaluated with receiver coated with selective absorber coatings. This line focusing parabolic trough yields instantaneous efficiency of 60 % with top cover. Actual field experimentation for performance evaluation of prototype system has been done during month of April and May 2012 at Shivaji university, Kolhapur [Latitude: 16.42° N, Longitude: 74.13°W].For the hotter climatic conditions, this type of system may play an important role in industrial process heating applications.

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Keywords: Solar thermal technology; selective surfaces for collectors; Compound parabolic reflector; selective coating for absorber; solar collectors with top cover

1. Background

A solar thermal technology for industrial heating has been proven one of the excellent source of renewable heating. The proposed system described in this paper consists of a top covered & aluminium foil coated mild steel compound parabolic solar reflector. This collector has been tested with a copper receiver coated with a black copper as a selective absorber coating. The main aim of thermal performance evaluation is to find out the probable application of proposed system for industrial heating applications

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Selection and/or peer-review under responsibility of ISES.
doi:10.1016/j.egypro.2015.06.052

such as boiler water preheating, laundry cleaning etc. The reflector has been covered with top cover to minimize the heat losses from the receiver & to increase the life of reflecting surface. A brief literature relevant to the concern aspect has been cited.

Nomenclature

A	Area of collector (m^2)
C	Concentration ratio
C_p	Specific heat at constant pressure(J/kg K)
D	Diameter of absorber tube(m)
FR	Heat removal factor
F'	Collector efficiency factor
F	Focal length of parabola (m)
h_f	Convective heat transfer coefficient ($W/(m^{2\,0}C)$)
$H_{p\text{-}c}$	Convective heat transfer coefficient between tube to cover ($W/(m^{2\,0}C)$)
h_w	Convective heat transfer coefficient between cover to atmosphere ($W/(m^{2\,0}C)$)
I_b	Incident beam radiation (W/m^2)
k	Thermal conductivity ($W/(m^0C)$)
$\dot{m}$	Mass flow rate (kg/s)
L	Length of tube (m)
n	Number of day in year
Q_u	rate of useful heat gain (W)
Q_l	Heat loss (W/m)
S	Absorbed solar flux (W/m^2)
T_{in}	Inlet temperature of water (^oC)
T_{pm}	Mean temperature of tube surface (^oC)
T_a	Atmospheric temperature of water (^oC)
T_{out}	Outlet temperature of water (^oC)
T_c	Glass cover temperature (^oC)
U_l	Heat loss coefficient
V	Wind speed (m/s)
W	Width of trough (m)
α	Absorptance
β	slope (rad)
γ	Intercept factor
δ	Declination angle
ε	Thermal emittance
θ_i	Angle of incidence of central solar ray with collector aperture, [deg]
ρ	Specular reflectance
η_i	Solar collector instantaneous efficiency
$\tau\alpha$	Transmittance absorptance product of the receiver glass envelope
α	Solar altitude angle
Φ	Rim angle
ω	Hour angle
ϕ	Latitude angle
τ	Glass transmitivity
σ	Stefan Boltzmann constant (5.67 x 10-8 $W/(m^2\,^oC^4)$)
r_b	Conversion factor for beam radiation
D_o	Outer diameter of receiver tube (m)
D_i	Inner diameter of receiver tube (m)

T_{sky}	Sky Temperature (^oC)
Nu	Nusselt Number
A_{abs}	Area of receiver(m^2)

1.1 Literature Survey

Kamal Skeiker explains about the important parameter that affect the performance of a solar collector is its tilt angle with the horizon. This is because of the variation of tilt angle changes the amount of solar radiation reaching the collector surface. Kamal Skeiker has developed a mathematical model for estimating the solar radiation on a tilted surface, and to determine the optimum tilt angle and orientation (surface azimuth angle) for the solar collector in the main Syrian zones, on a daily basis, as well as for a specific period. The results expose that changing the tilt angle 12 times in a year maintains approximately the total amount of solar radiation near the maximum value that is found by changing the tilt angle daily to its optimum value. This achieves a yearly gain in solar radiation of approximately 30% more than the case of a solar collector fixed on a horizontal surface. [1] Oyedepo studied energy utilization pattern of the Nigeria. He investigated the possible areas of energy conservation in the major economic sectors like industry, transportation, office and residential buildings. Authors study reveals that there is inefficient utilization of energy in the major economic sectors of the country and this study also presented several energy conservation opportunities to cause energy savings and identified about six major areas through which energy conservation measures can effectively cause some savings in energy and allow for its stability. [2] Guangjie Gong et.al explains that, the vacuum solar receiver is the key component of a parabolic trough solar plant, which plays a prominent role in the gross system efficiency. This paper first establishes and optimizes a 1-D theoretical model at Mat lab program to compute the receiver's major heat loss through glass envelope, and then systematically analyzes the major influence factors of heat loss. With the laboratorial steady state test stand, the heat losses of both good vacuum and non-vacuum Sanle-3 receivers were surveyed by the authors. Authors comparison shows that the original 1-D model agrees with the ends covered test while remarkably deviating from end exposed test. Author's also developed a 3-D model by CFD software to further investigate the different heat transfer processes of receiver's end components. [3] Sagade et.al describes the experimental results of the prototype parabolic trough made of fiberglass reinforced plastic, and its aperture area coated by aluminum foil with reflectivity 0.86. Authors observed that this line focusing parabolic trough with mild steel receiver coated with black proxy material has been tested with glass cover and without glass cover yeilds instantaneous efficiency of 51 % and 39 % with glass cover and without glass cover respectively.[4] A solar thermal concentrator system has been proposed by Chung-Yu Tsai , Psang Dain Lin comprising a cylindrical heat-pipe receiver and a variable-focus-parabolic-trough (VFPT) reflector in which the focal length varies as a function of the vertical displacement of the incidence point relative to the horizontal centerline of the receiver. Authors analysed the light ray paths within the concentrator system using a skew-ray tracing approach. A method is then proposed for optimizing the geometry of the concentrator system in such a way as to optimize the uniformity of the irradiance distribution on the heat-pipe surface. Authors also demonstrated the validity of the proposed optimization method by means of ZEMAX/SolidWorks-Flow simulations. It has been shown that the optimized VFPT concentrator yields a significant improvement in both the irradiance uniformity and the heating efficiency compared to conventional cylindrical-trough and parabolic-trough concentrators. [5] Atul Sagade used parabolic dish collector for instant water heating application. This paper reveals comparative experimental analysis of effect of variation of natural and forced convection heat losses on performance of prototype parabolic dish water heater with coated and non coated receivers. Authors explained that instantaneous efficiency of 63% and 48 % has been achieved with coated and non coated receivers [6]. The breakage of the glass to metal sealing is the primary ongoing issue for the solar receiver tubes in parabolic trough solar power systems. Sealing failure leads to loss of the vacuum inside the tube which substantially reduces the collector efficiency. It is a technical difficulty to obtain good glass to metal seals with high mechanical strength and long-term temperature resistance during current receiver manufacturing. Dongqiang Lei et.al describes the development of the glass to metal seals in the parabolic trough receivers and presents a new method that uses the high-frequency induction heating to band a new borosilicate glass to the Kovar alloy ends. Authors used Kovar pre-oxidization experiments to measure the relation curves of Kovar oxidation weight gain during heating. The preoxidation of Kovar and the sealing process are guided by a series tests to measure the gas tightness, sealing strength, seal interface microstructure and thermal shock. Authors observed that the results show that excellent glass-to-Kovar sealing can be obtained with a Kovar oxidation weight gain of about 0.3–0.8 mg/cm^2. A new solar receiver was developed by the new sealing method by the authors [7]. The operation principle and design method of a new trough solar concentrator is presented by Tao Tao et.al in this paper. Some important design parameters about the concentrator are analyzed and optimized by the authors and their magnitude ranges are given. Some characteristic parameters about the concentrator are compared with that of the conventional parabolic trough solar concentrator. Authors also discussed the factors that have influence on the performance of the unit and it is indicated through the analysis that the new trough solar concentrator can actualize reflection focusing for the sun light using multiple curved surface compound methods. Authors also conclude that it has the advantages of improving the work performance and environment of high-temperature solar absorber and enhancing the configuration intensity of the reflection surface.[8] Atul Sagade et al. experimented on low cost FRP parabolic trough with aluminium receivers which are coated with black epoxy. Authors found that, the systems yields 52% and 38 % has been achieved with glass covered receiver and without glass covered aluminium receiver respectively.[9] Atul Sagade and N.N shinde worked out thermal performance of parabolic dish solar water heater with truncated cone shaped helical coiled receiver made up of copper and coated with nickel chrome at focal point. Instantaneous efficiency of 63.9% has been achieved with the system explained in this paper. [10] From the literature cited, it can be seen that the systems with top covered CPC trough collectors with different coating materials are under research and can prove a good application for industrial heating applications. Table 1 show the nomenclature used.

2. Experimental setup:

Experimental setup consists of following Equipment:-

1. Cold water storage tank with asbestos insulation of capacity 100 tr.
2. Support stand storage tank of height 6 feet.
3. Prototype of cylindrical parabolic trough with glass cover and absorber tube.
4. Support stand for CPC.
5. K-type thermocouple 8 quantity
6. USB enabled Data Logger to record temperature.
7. Lux meter calibrated for measurement of global radiation.
8. Anemometer to measure wind speed.

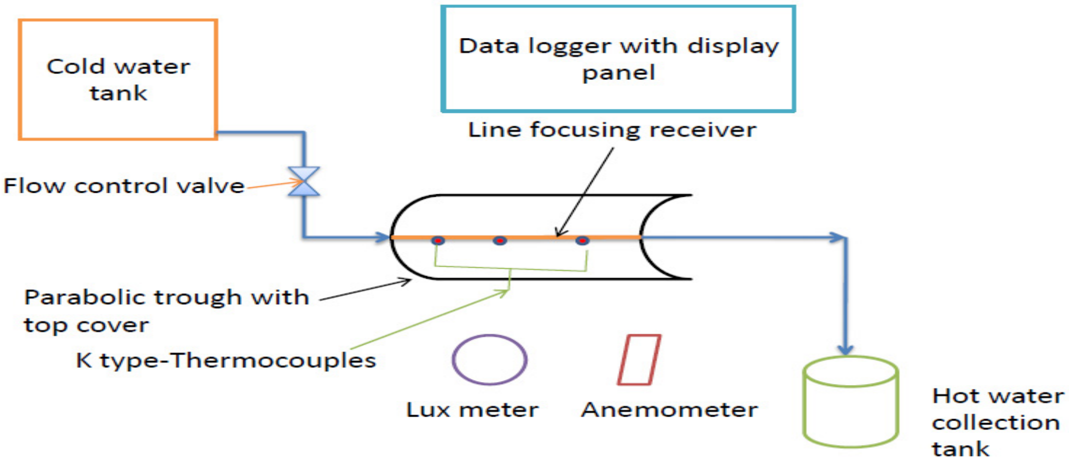

Fig.1 Experimental set up for thermal performance evaluation

Table 1 show system parameters of prototype system

Table1: System parameters

Items	**Values**
Collector Width	1.03 m
Collector length	1.82 m
Focal distance	0.221 m
Receiver inner diameter	0.017 m
Receiver outer diameter	0.019 m
Concentration ratio	16.94 ~ 17
Water flow rate	4 lph
Storage tank capacity	100 ltr.
Tank insulation material	Asbestos
Insulation Thickness	5 mm
Glass cover thickness	3 mm
Rim angle	134.52^{o}

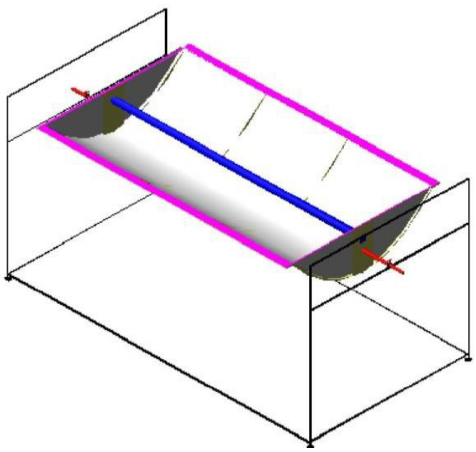

Fig.2 3D model of CPC

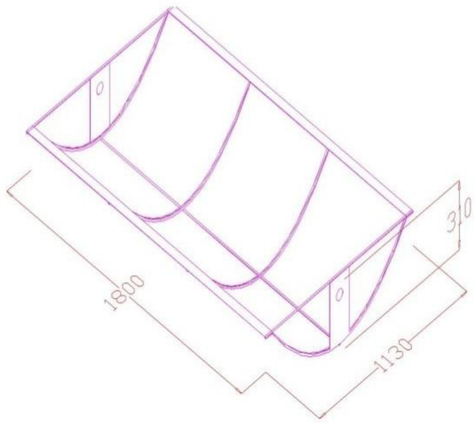

Fig.3 Explored view of CPC

3. Calculations

The performance of CPC is assumed for same radiation flux along the length of negligible heat drop across absorber tube and glass.The analysis of CPC is done by studying energy balance equation on an elementary slice dx of absorber tube, at a distance x from inlet, yields following equation 1 [11, 12, 13] for steady state

$$\frac{dq_u}{d_x} = I_b r_b(W - D_o)\rho\gamma(\tau\alpha)_b + I_b r_b D_o(\tau\alpha)_b - u_l \pi D_o\, (T_{pm} - T_a) \text{ --------- (1)}$$

The absorbed solar flux can be given by equation 2 [11, 12, 13]

$$S = I_b r_b \rho\gamma(\tau\alpha)_b + I_b r_b Do(\tau\alpha)_b \left(\frac{Do}{W-Do}\right) \text{ ---------------------- (2)}$$

Collector efficiency factor F' can be evaluated by equation 3 [11, 12, 13]

$$F' = \frac{1}{U_l \times \left[\frac{1}{U_l} + \frac{Do}{Di*h_f}\right]} \text{ ----------------------------- (3)}$$

Thus useful heat gain rate is given by equations 4 and 5[11, 12, 13]

$$Q_u = \dot{m} * Cp * (T_{out} - T_{in}) \text{ --------------------- (4)}$$

$$\text{Or } Q_u = Fr(W - Do) * L * \left[S - \frac{U_l}{C}(T_{in} - T_a)\right] \text{ ------------------- (5)}$$

Equation 4 and 5 is equivalent to Hottel-Whiller-Bliss Equation for flat plate collector.
Heat removal factor can by defined by equation 6, [11, 12, 13]

$$Fr = \frac{\dot{m}*Cp}{\pi*Do*U_l*L}\left[1 - e^{-\left\{\frac{F'*\pi*Do*U_l*L}{\dot{m}*Cp}\right\}}\right] \text{ ---------------------- (6)}$$

The instantaneous collection efficiency can be calculated as per equation 7 [11, 12, 13]

$$\eta_i = \frac{Q_u}{I_b*r_b*W*L} \text{ ------------------------------ (7)}$$

3.1 Calculation of Heat loss coefficient:-

It is necessary to evaluate heat loss coefficient for trough covered with glass cover and trough without glass cover. In case of CPC major heat loss takes place through top opening, bottom and side losses are very small quantities so they are neglected. Overall heat loss coefficient is nothing but the top loss coefficient for CPC.

3.2 Calculation of Heat loss coefficient trough with glass cover:-

Evaluation procedure of heat loss coefficient is very tedious and iterative. Based on large number of cases scientist have developed empirical correlation to calculate heat loss by simple equation. Empirical equation suggested by Malhotra et al.

$$U_l = [\frac{M}{\left(\frac{C}{T_{pm}}\right)-(\frac{Tpm-Ta}{M+f})^{0.252}} + \frac{1}{h_w}]^{-1} + [\frac{\sigma(T_{pm}^2+T_a^2)(T_{pm}+T_a)}{\left(\frac{1}{(£_p+0.0425M(1-£_p))}\right)-(\frac{2M+f-1}{£_c})} - M] \text{------ (8)}$$

Where,

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

$$C = \frac{204.429(cos\beta)^{0.252}}{L^{0.24}} \text{----------------- (10)}$$

$$h_w = 5.7 + 3.8v \text{----------------- (11)}$$

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

$320 < T_{pm} < 420K$, $260 < Ta < 310K$, $0.1 < £p < 0.95$, $0 < v < 10$ m/s, $1 < M < 3,0 < \beta < 90^0$

3.3 Convective Heat transfer coefficient:-

Value of Convective Heat transfer coefficient depends on properties of fluid and mean temperature absorber tube. Nusselt number is given by equation 11 [11, 12, 13] and Convective Heat transfer coefficient by equation 12 [11, 12, 13]

$$Nu = 3.66 \text{------------------------------ (11)}$$

$$h_f = Nu * \frac{\kappa}{Di} \text{---------------------------- (12)}$$

3.4 Calculation of Heat loss coefficient trough without glass cover:-

Heat losses are more in case of system without glass cover. Convection and radiation are the major causes of heat loss; it can be calculated by following equation 13 [11, 12, 13]

$$\frac{q_t}{A_{abs}} = h_w(Tc - Ta) + \sigma * \varepsilon_c * (Tc^4 - Tsky^4) \text{------------------- (13)}$$

Here U_l can be evaluated by equation 14 [11, 12, 13] and $T_{sky} = T_a-6$

$$U_l = \frac{{q_t}/{A_{abs}}}{(Tpm-Ta)} \text{-------------------- (14)}$$

4. Experimental Data collection

Experimental data has been collected with prototype compound parabolic trough solar collector made of mild steel and its surface coated with a aluminum foil of thickness 10 micron as a reflective surface. This prototype has been tested with top cover for instant water heating and steam generation application with water flow rate of 0.011kg/s. During experimental performance of collector, copper receiver has been coated with **black copper** as selective absorber coatings. The experiments have been carried out to check the reproducibility of the results. Table 2 shows the experimental data collected.

Table2: Experimental data

Time	Global Radiation (w/m^2)	Wind speed (m/s)	Temperature at various Location (°C)					Collector efficiency (%)	U_l (W/(m^{20}C))	Q_u (W)	Q_l (W/m)
			T_a	T_{in}	T_{pm}	T_{out}	T_c				
10.00	910.00	1.70	30.00	29	74.10	72.00	32.10	59.71	4.98	835.96	219.64
10.30	979.00	1.50	32.00	30	73.15	71.00	34.00	60.00	4.93	985.71	203.52
11.00	955.00	3.20	30.00	29	66.53	65.00	32.00	59.84	5.23	1014.97	191.07
11.30	962.00	4.00	31.00	30	68.70	68.00	33.00	59.52	4.56	1054.26	206.04
12.00	1003.00	5.00	36.00	32	67.80	65.00	32.00	59.55	5.67	1119.79	180.55
12.30	1009.00	1.20	33.00	31	73.35	72.00	36.00	60.30	4.86	1120.17	196.48
13.00	1007.00	3.00	34.00	32	74.98	72.50	37.00	59.52	5.43	1064.54	222.9
13.30	900.00	1.40	38.00	34	69.83	69.00	38.00	60.27	4.9	909.12	156.18
14.00	987.00	2.20	37.00	35	67.15	65.80	40.00	60.15	5.05	912.75	152.5
14.30	937.00	3.00	37.00	35	70.08	68.10	39.00	59.53	5.31	756.14	175.89
15.00	904.00	1.50	37.00	34	59.95	56.88	40.00	60.70	4.64	616.5	106.57
15.30	892.00	1.70	35.00	33	61.95	60.70	41.00	59.89	4.69	442.8	125.17
16.00	886.00	1.30	34.00	32	66.85	64.00	39.00	58.08	4.64	231.45	152.74

5. Results and discussions

For performance evaluation and design of the prototype system, a regression analysis, has been carried out with different design parameters such as wind velocity, solar radiation on collector, heat loss coefficient, total heat loss from the receiver, heat gained by water flowing through the receiver, collector efficiency. A simple relationship between parameters has been worked out using regression analysis.

5.1 Relation between overall heat loss coefficient with heat loss, heat gain and collector efficiency

Fig.4 shows the variation of heat loss, heat gain and collector efficieny as compared to overall heat loss coefficient. The regression equations indicates the trend of variation of the parameters.

$$\eta_i = 62.78 - 0.529(U_l) - 0.014(Q_l) + 0.00256(Q_u) \text{ ---------------------------------- (A)}$$

From eq.A it is clear that as the efficiency of the collector decreases by 1% with increase of overall heat loss coefficient by 0.53 W/(m^2 ^{0}C) and heat loss and increases as the heat gained by water increases. The equation (A) interprets that, heat loss through the receiver has negative effect on collector efficiency and heat gain. The positive association between collector efficiency and heat gain is reflected by the fact that higher heat gain also has high values on collector efficiency. If heat loss through the receiver increases by 0.014W, then collector efficiency will decrease by 1%.

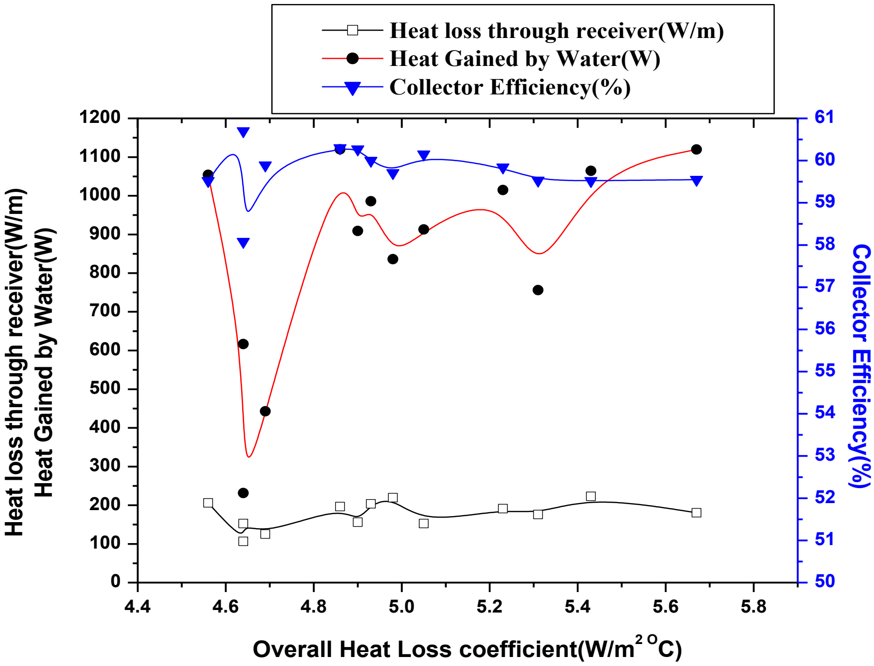

Fig. 4 variation of overall heat loss coefficient, heat loss, heat gain and collector efficiency

5.2 Relation between Wind velocity, overall heat loss coefficient on heat loss, total heat loss and collector efficiency

Fig.5 explores the effect of wind velocity on heat loss, overall heat loss coefficient and collector efficiency. From fig.5, it is clear that, the wind velocity has negative effect on collector efficiency. Eq.B interprets the positive effect of wind velocity on overall heat loss coefficient and heat loss. But the highly positive relationship between overall heat loss coefficient and heat loss indicates that, wind velocity is major adverse factor, affecting the efficiency as compared to other factors such as solar radiation and heat gain.

$$Q_l = 0.474 + 4.185\,(V) + 33.20\,(U_l) \text{ ------------------------------------ (B)}$$

Equation B indicates that when wind velocity increase by 4.18 m/s and overall heat loss coefficient increase by 33.20 W/(m^2 ^{0}C) then total heat loss will increase by 1W/m

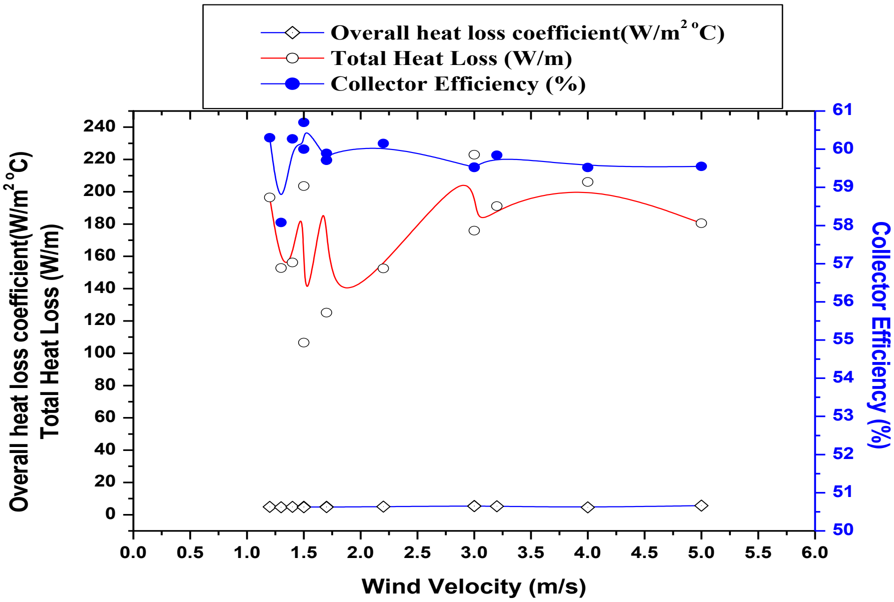

Fig.5 Variation of Wind velocity, overall heat loss coefficient, total heat loss and collector efficiency

5.3 Relation between receiver temperature, outlet water temperature, total heat loss and collector efficiency

From fig.6, it is observed that, the receiver temperature is one of the contributing factors affecting the collector efficiency. Regression eq.(C) explore the relation between collector efficiency, receiver temperature, outlet water temperature and heat loss.

$$\eta_i = 58.50 - 0.225(T_r) + 0.255(T_{out}) - 0.00771(Q_l) \text{ ------------------------------------ (C)}$$

Eq.C interprets that, collector efficiency has negative relation with the receiver temperature, because of the fact that, higher the receiver temperature, higher will be the heat losses through the receivers. From equation C it has been clear that, if receiver temperature increases by 0.225^{0}C and heat loss increase by 0.0077W then collector efficiency will decrease by 1%. Thus collector efficiency has negative relation with heat losses. Again collector efficiency has positive relationship with outlet water temperature outlet water temperature increases by 0.255^{0}C, then collector efficiency will increase by 1%. As receiver temperature increases, the outlet water temperature also increases.

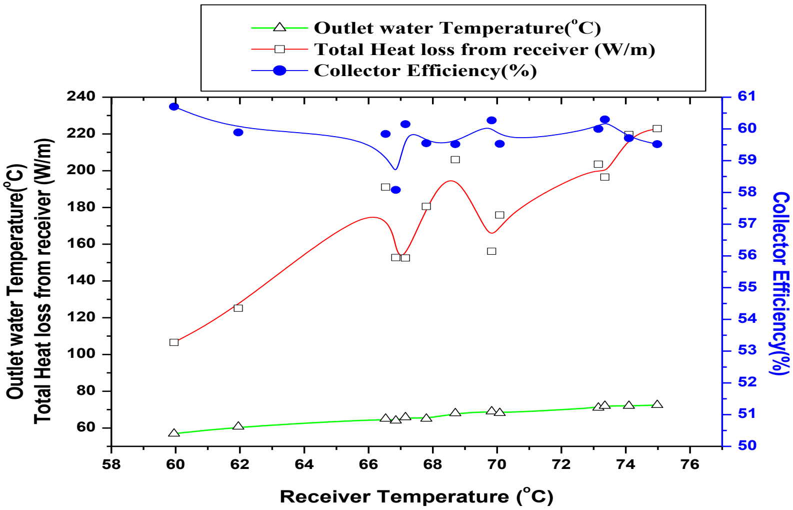

Fig.6 variation of receiver temperature, outlet water temperature, total heat loss and collector efficiency

5.4 Relation between wind velocity, solar radiation, heat gained by water and collector efficiency

Fig.7 shows the variation of wind velocity, solar radiation, heat gained by water and collector efficiency and eq.D interprets the relation between these parameters. It has been observed that, in case of designed CPC, solar radiation has negative relation with collector efficiency. It is due to the fact that, as radiation on collector increases, concentration at receiver increases and thus heat losses from receiver increases. As said earlier, wind velocity is major adverse factor, affecting the efficiency and has negative relation with efficiency. The model equation D predicts that collector efficiency will be decrease by 1%, when wind velocity increase by 0.25 m/s , solar radiation increase by 0.255W/m^2 . Equation D also interprets that when heat gained by water increases by 0.00228 W, then collector efficiency will increase by 1%.

$$\eta_i = 63.75 - 0.25(V) - 0.255(I_b) + 0.00228(Q_u) \text{ ------------------------------ (D)}$$

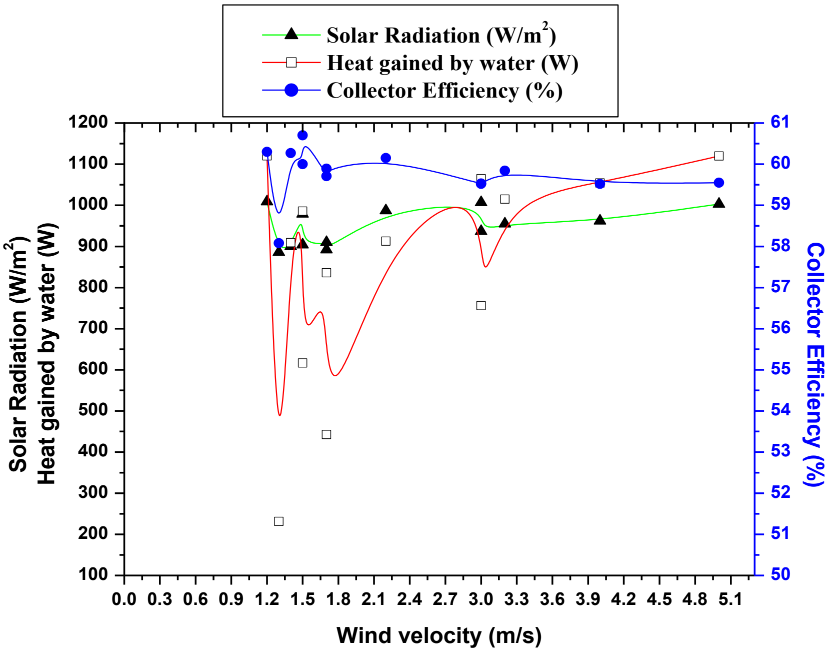

Fig.7 variation of wind velocity, solar radiation, heat gained by water and collector efficiency

6. Conclusions:

- From hotter climatic conditions, proposed system can be proven a one of the efficient source of industrial heating applications.
- Top covered CPC systems may increase the life of reflecting surfaces & receiver coatings with reduction of heat losses from the receivers.
- Theoretical regression modeling equations are satisfied by the actual experimental results; hence can be considered for design purpose & evaluation of thermal performance of the similar systems.
- The experimental results are presented and the validity of the model is examined by comparison of the theoretical results with experiments which demonstrates a good agreement.
- The proposed system yield a average temperature gradient of 35^0C throughout the day with maximum temperature gradient of 43^0C
- The proposed system yield a average efficiency of 59.8% with mass flow rate of 0.011Kg/S
- It has been seen that there is an average 20% heat loss per meter length of receiver. Average heat gained by water is 850W against average heat loss of 176W per meter.
- It has been observed that, the average 68^0C temperature is attained by the black copper coated receiver with maximum temperature of 76^0C

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