Thermal Performance of Parabolic Dish Water Heater with Helical Coiled Receiver

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Thermal performance of parabolic dish water heater with helical coiled receiver

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Abstract: Solar parabolic dish type collectors are being tested for their thermal performance using different designs of cavity receivers for diverse applications such as water heating, industrial process heat, thermal power generation and many more. In the present work, thermal performance of parabolic dish water heater is assessed for small scale industrial process heat using helical coiled truncated cone receiver which is coated with black chrome as a selective coating. It is seen that, the proposed system yields an average instantaneous efficiency of 63% when the flow rate of water is 0.0056 kg/s.

Key Words: Solar water heating systems, instant water heating, solar receivers, industrial heating, selective receiver coatings.

Nomenclature

Symbol/Abbreviation	Name of the Parameter
Pr	Prandalt's number
Re	Reynold's Numbar
Gr	Grashoff's Number
$\boldsymbol{u_l}$	overall heat loss coefficient (W/(m^2 ^{0}C))
$\boldsymbol{Q_{loss}}$	Useful heat gain by water (W/m)
$\boldsymbol{v}$	Wind velocity (m/s)
$\boldsymbol{\eta_c}$	Collector efficiency (%)
$(\boldsymbol{Q_{useful}})$	Useful heat gain (W/m)
$(\boldsymbol{T_r})$	Average receiver temperature (^{0}C)
$(\boldsymbol{I_b})$	Solar Radiation (W/m^2)
$(\Delta\boldsymbol{T})$	Temperature gradient (^{0}C)
T_{fi}	Inlet water Temperature (^{0}C)
T_{fo}	Outlet water Temperature (^{0}C)
T_{air}	Ambient Temperature (^{0}C)

1. Introduction

In India, different concentrating collectors are now in application to deliver a medium to high temperature heat for industrial heating and (60 - 70 %) of all the energy consumed in industry is in the form of thermal energy. Thus, there is a great potential for utilizing solar energy for industrial process heat and domestic heating applications, especially in tropical countries like India where solar radiation is abundant. The demand for domestic heating is constant throughout the year and hence the capacity utilization of solar systems for these purposes can be very high. In several industries, 100 % processes, heat required is below 180ºC which can easily and economically be supplied by solar devices. Thus, this low temperature process heat requirement in industries makes the solar heating systems quite attractive. This paper aimed to explain the performance of novel truncated cone receiver for parabolic dish collector. Different researchers working on CSP for industrial heating and receiver geometry for parabolic dish systems and their brief literature is discussed in upcoming section. Kumar and Reddy [1] explained a numerical investigation to study the natural convective heat loss from three types of receivers for a fuzzy focal solar dish concentrator, namely cavity receiver, semi-cavity receiver and modified cavity receiver. Authors reported comparative study to predict the natural convection heat loss from the cavity, semi-cavity and modified cavity receivers. Larsen et al. [2] studied the heat loss of a linear absorber with a trapezoidal cavity and a set of pipes used for a linear Fresnel reflecting solar concentrator at laboratory scale. Authors observed that around 91% of the heat transferred to outdoors occurs at the bottom transparent window, for a pipe temperature of 200 °C. Wang and Siddiqui [3] designed a three-dimensional model of parabolic dish-receiver system with argon gas as a working fluid to simulate the thermal performance of a dish-type concentrated solar energy system. Authors explained an impact of the aperture size, inlet/outlet configuration of the solar receiver and the rim angle of the parabolic dish on the performance of proposed system. They concluded that, the aperture size and different inlet/outlet configuration have a considerable impact on the receiver wall and gas temperatures, but the rim angle of the parabolic dish has negligible influence on the thermal performance of the system. Hahm et al. [4] described the performance of a cone concentrator combined with a solar cavity receiver and they compared its performance with a single cavity receiver. Authors explored that, the cone concentrator suffers from a high amount of rejected rays if the exit aperture is too small and larger exit aperture increases the thermal losses of the cavity. Fang et al. [5] reported a combined calculation method for evaluating the thermal performance of the solar cavity receiver. Using this method, the thermal performance of a solar cavity receiver and a saturated steam receiver was simulated under different wind environments. Authors explained that, change in the wind angle or velocity affects the air velocity inside the receiver. Harris and Lenz [6] discussed the Power profiles produced in cavities of varying geometry with concentrators of varying rim angle. They found that variation in concentrator rim angle and cavity geometry affects the cavity power profile without a large effect on system efficiency. Authors concluded that the described methodology can be used to optimize concentrator/cavity design variables. Humphrey and Jacobs [7] investigated free-forced laminar flow convective heat transfer from a square cavity in a channel with variable inclination. They studied the influence on heat transfer of cavity orientation, channel inclination, flow direction and entrance profile for flow conditions. Authors concluded that, the stable stratification of the flow in downward-facing cavity geometry was responsible for reducing the rate of heat transfer relative to an upward-facing geometry under equivalent flew conditions. Reddy and Sendhil Kumar [8] reported the numerical study of combined laminar natural convection and surface radiation heat transfer in a modified cavity receiver of solar parabolic dish collector using simulation model for combined natural convection and surface radiation. Authors investigated the influence of operating temperature, emissivity of the surface, orientation and the geometry on the total heat loss from the receiver. Sagade [9-10] described the effect of convective heat loss and effect of variation of mass flow rate on the thermal performance of parabolic dish solar water heater. He concluded that, the system performance affected adversely because of convective heat loss. Also the decreases in mass flow rate leads to increase in heat loss and decease in collector efficiency. Plenty of similar literature is available on cavity receivers, but avoided to discuss here because of space limitations.

From the literature discussed in above section, it is clear that the researchers evaluated thermal performance of cavity receiver geometries such as trapezoidal, square, conical, semi cavity and modified cavity and different design parameters such as, rim angle, concentration ratio, receiver inclination, cavity orientation. The aim of present study is to report the thermal performance of a helical coiled truncated cone shape receiver with a small helix angle to enhance the efficiency of water heating and possible application in industrial heating, low pressure steam generation and water heating applications in domestic sector. A helical coiled truncated cone shape receiver is coated with black nickel chrome as a selective receiver coating and covered with glass cover and mounted/ coupled with a parabolic dish. A regression analysis technique is used to evaluate a simple relationship between the thermal performance parameters and the performance measurement.

2. Test set up and system parameters

The prototype performance was evaluated under standard test condition of $I_b \geq 700$ W/m^2 and 20^0C $\leq T_{air} \leq 40^0$C with water flow rate of 0.0056 kg/s. Fig.1 shows the experimental set up for the performance evaluation and table 1 indicates the dimensions of the parabolic dish – helical coil receiver system.

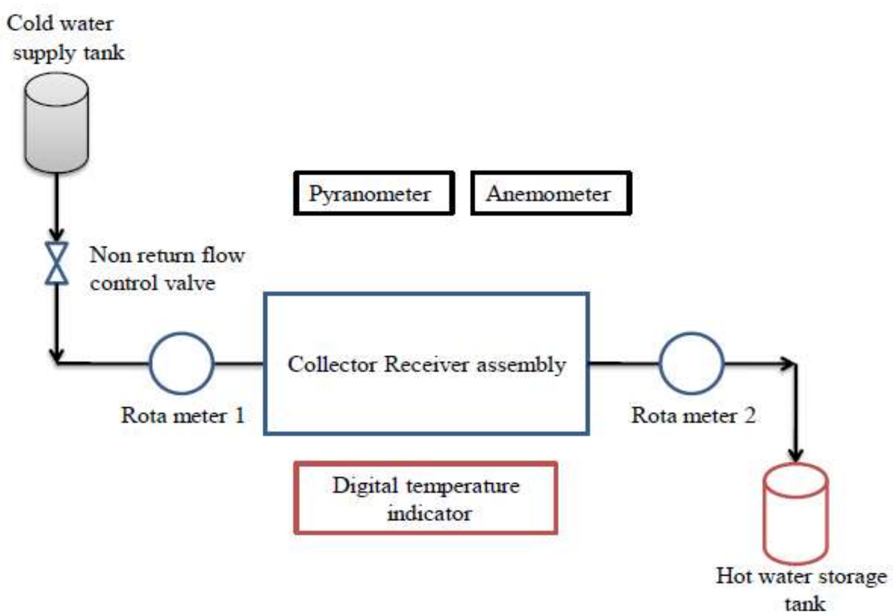

Fig.1. Schematic of Test setup

Table 1: Dimensions of the parabolic dish–helical coiled receiver system

Parameter	Value
Absorptivity(α)-Transmitivity (τ) product for receiver coating	0.93
Absorptivity(α)-Transmitivity (τ) product for copper	0.7
Aperture area of parabolic dish collector (A_c)	1.54 m^2
Depth of Dish (D)	0.38 m
Diameter of Parabolic dish (d)	1.4 m
Diameter of receiver at bottom(D_{rb})	0.135 m
Diameter of receiver at top (D_{rt})	0.095 m
Effective length of receiver coil (e_l)	3.96 m
Emissivity(ξ_c) of receiver coating	0.14
Emissivity (ξ_{cu}) of copper	0.725
Focal length of dish (f)	0.32 m
Mean Diameter of receiver (D_{mean})	0.115 m^2
Reflectivity of Parabolic dish mirrors (r)	0.86
Surface area of parabolic dish collector (A_s)	1.92 m^2
Surface area of receiver (A_{rec})	0.2357m^2
Thermal conductivity (K) of copper	384 W/(m k)
Thickness (T) of mirrors of Parabolic dish	2mm

3. Calculations

The experiment was repeated for four times to check the reproducibility of the results. The equations used for the calculation of various thermal performance parameters were described below. For simplifying the calculations, it is assumed that,

a. Heat transfer is steady state process.
b. Solar radiation on the collector receiver system is constant at the instant.
c. Axial heat transfer is neglected.
d. Specific heat of water remains constant with respect to temperature.

3.1 Useful Heat gain by the water

Useful heat gain by water is given by eq.1 [13]

$$Q_{useful} = E_{opt} - Q_{tl} \text{ (W)} \tag{1}$$

Where,
E_{opt} = optical radiation trapped receiver (W)
Q_{tl} = rate of heat loss from receiver (W)

3.2 Calculation of Heat Losses from the Receiver

Thermal losses from solar open cavity receivers include convective and radiative losses to air.
For focal plane i.e. cavity receiver overall heat loss is given by eq.2 [13]

$$Q_{loss} = A_{rec} \times u_l \times (T_r - T_{air}) \text{ (W)} \tag{2}$$

Where,
T_{air} = Temp of air surrounding a receiver (°C)
T_r = Average receiver temp (°C)
u_l = Overall heat loss coefficient
Overall heat loss coefficient (u_l) is given by eq.3 [13]

$$u_l = \left[\frac{1}{[hct+hr]}\right]^{-1} \text{(W/(m}^2\text{°C))} \tag{3}$$

Where,

Radiative heat transfer coefficient (h_r) is given by eq.4 [13]

$$h_r = \frac{\xi c \times \sigma\,(T_r^4 - T_{air}^4)}{T_r - T_{air}} \text{ (W/(m}^2\text{°C))} \tag{4}$$

Total convective heat loss coefficient (h_{ct}) can be given by eq.5 [13]

$$h_{ct} = h_{cn} + h_{cf} \quad (W/(m^2 °C)) \tag{5}$$

To estimate natural convective heat loss, following correlation developed by eq.6 [11]

$$Nu = 0.21\, G^{1/3} (1+Cos(\theta_{rec}))^{3.2} (T_{mean}/T_{air})^{-1.5} \tag{6}$$

Where,
G_r = Grashof's number for natural convection
T_{mean} = mean temperature of water in receiver coils at natural convection (°C)
θ_{rec} = receiver inclination angle = 90^0

Forced convection loss coefficient (h_{cf}) is developed by Ma [12] and can be calculated by eq. (7)

$$h_{cf} = f(\theta_{rec})\, v^{1.401} \, (W/(m^2 °C)) \tag{7}$$

Instantaneous efficiency of collector can be calculated by eq. (8) [13]

$$\eta_c = \left(\frac{Q_{useful}}{I_b}\right) \times 100\% \tag{8}$$

4. Uncertainty analysis

The errors occurred in the measuring instruments are calculated in this section. Thermocouples, pyranometer, cup type wind sensor, data logger and rotameter are used for measuring temperature, solar intensity and wind velocity, recording of continuous data and mass flow rate respectively. The minimum error occurred in any instrument is equal to the ratio between its least count and minimum value of the output measured. The accuracies of various measuring instruments used in the experiments are given in table 2.

Table 2: Uncertainty Analysis

Instrument	Accuracy	Uncertainty
Pyranometer	± 2 W/m^2	1.2%
Thermocouples	± 1^0C	1.3%
Wind sensor	±1m/s	2%
Data logger	± 1 units	0.5%
Rotameter	± 0.5 LPH	2.15%

5. Results and discussion

5.1 Effect of wind velocity on overall heat loss coefficient and collector efficiency

Fig.2 indicates Variation of overall heat loss coefficient (u_l) and total heat losses (Q_{loss}) from receiver with wind velocity(v). Regression eq. (A) interprets that, when wind velocity increases by 1.04 m/s and heat loss increases by 0.0113 $Wm^{-20}C^{-1}$, heat loss coefficient will proportionally increases by 1 $W/(m^{2\,0}C)$

$$u_l = 2.74 + 1.04\,(v) + 0.0113(Q_{loss}) \tag{A}$$

With glass covered coated receiver, it is observed that, heat losses are increased, when air gets heated due to rise in receiver temperature, air conductivity increases and heat losses also increases. But the main cause for heat loss is temperature gradient between receiver and ambient and wind velocity at instant. Therefore there is an increase in convective and radiative heat losses from receiver. An average receiver temperature was considered to provide accurate receiver loss predictions. The higher value of the linear regression coefficient of determination (R^2) is 0.92 which shows the best fit of regression relation between the parameters.

5.2 Effect of heat loss on collector efficiency and heat gain

Fig.3 Explores effect of heat loss on collector efficiency (η_c) and heat gain (Q_u). Useful heat gained by water is a function of solar radiation, receiver temperature total heat loss and wind velocity at instant. The system which yields more water temperature and will show better system performance. Equation (B) shows the relationship between collector efficiency with total heat losses and heat gain by water. Equation (B) indicates that, when total heat loss, increases by 0.09 W/m, collector efficiency decrease by 1%. It also interpret that, when heat gain of water increase by 0.018 W/m proportionally collector efficiency increase by 1%. The value of the linear regression coefficient (R^2) is 0.945 which shows good linear relationship between the parameters. This model interprets that, the efficiency of collector decreases as the heat loss increases, which is confirmed by the negative slope coefficient.

$$\eta_c = 64.97 - 0.09\,(Q_{loss}) + 0.018\,(Q_{useful}) \tag{B}$$

5.3 Effect of receiver temperature on temperature gradient and collector efficiency

From fig.4 it is observed that average receiver temperature (T_r) is majorly affected by solar radiation (I_b) and wind velocity at the instant. Receiver temperature increases sharply as solar radiation increases. Equation (C) shows the relation of receiver temperature and temp gradient (ΔT) with collector efficiency. Eq.C interprets that, receiver temperature has negative impact on collector efficiency. When receiver temperature increases by 0.233^0C, proportionally collector efficiency decreases by 1%. When temperature gradient increases by 0.47^0C, collector efficiency decreases by 1%.

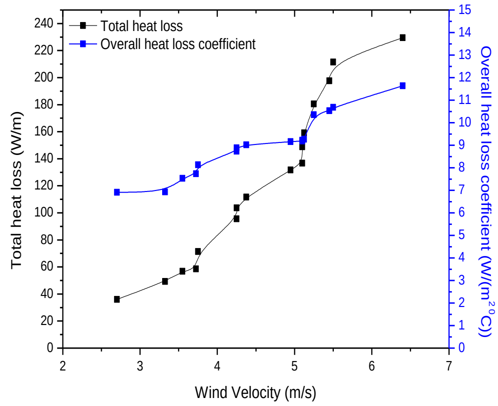

Fig.2. Effect of wind velocity on overall heat loss coefficient and collector efficiency

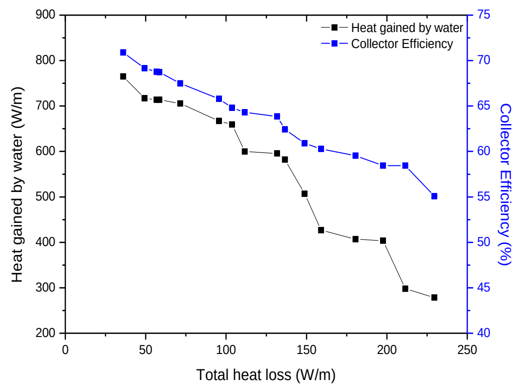

Fig.3. Effect of heat loss on collector efficiency and heat gain

$$\eta_c = 74.01 - 0.233(T_r) + 0.47\,(\Delta T) \tag{C}$$

The linear regression coefficient (R^2) for this linear model is 0.85 which shows good linear relationship between the collector efficiency, receiver temperature and temperature gradient. This model interprets that, the efficiency of collector decreases as the receiver temperature increases, which is shown by the negative slope coefficient and collector efficiency increases as temperature gradient increases which is indicated by positive slope coefficient

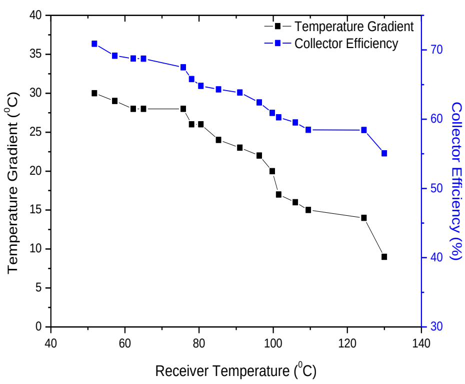

Fig.4. Effect of receiver temperature on temperature gradient and collector efficiency

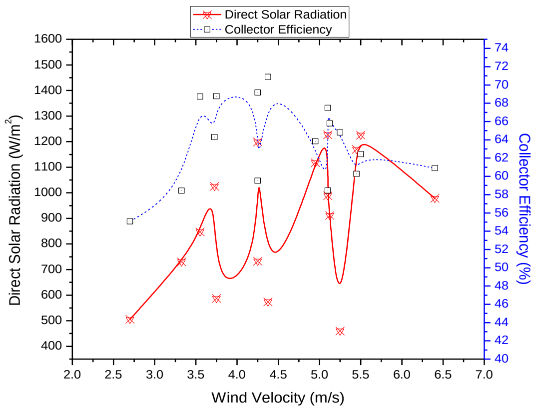

Fig.5. Variation of beam solar radiation and collector efficiency with wind velocity

5.4 Effect of beam solar radiation and wind velocity on collector efficiency

Fig.5 shows the variation of beam solar radiation and collector efficiency with wind velocity. Collector efficiency is a function of two environmental parameters solar radiation and wind velocity and out of which beam solar radiation has a negative effect on collector efficiency beyond certain limit.

$$\eta_c = 74.53 - 0.35(v) - 0.009(I_b) \quad \text{(D)}$$

This is shown by equation D. Equation D predicts that, collector efficiency will decrease by 1%, when wind velocity increases by 0.35 m/s and very little increase in solar radiation 0.009 W/m. When wind velocity and solar radiation shows no increase then collector efficiency will be equal to intercept of equation D.

5.5 Effect of overall heat loss coefficient and receiver temperature on collector efficiency

Equation E predicts that overall heat loss coefficient and receiver temperature has negative effect on collector efficiency.

$$\eta_c = 84.16 - 1.15(u_l) - 0.107(T_r) \quad \text{(E)}$$

When overall heat loss coefficient and receiver temperature increases by 1.15 W/(m^{20}C) and 0.107^0C respectively, proportionally collector efficiency will decrease by 1% because collector efficiency is a function of parameters such as solar radiation, surface reflectance, receiver absorptance, atmospheric conditions, and wind velocity. The regression coefficient (R^2) for this model is 0.81 which interpret satisfactory relationship between the overall heat loss coefficient and receiver temperature on collector efficiency. This model predicts that, the efficiency of collector decreases because of both parameters which is shown by the negative slope coefficients of overall heat loss coefficient and receiver temperature. Fig.6 indicates effect of overall heat loss coefficient and receiver temperature on collector efficiency.

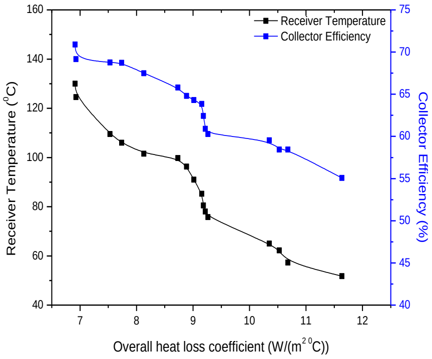

Fig.6. Effect of overall heat loss coefficient and receiver temperature on collector efficiency

6. Conclusions

From the field experiments, it is seen that, when receiver is coated black and covered with glass cover, system performance get enhanced. From the experimentation, it is clear that, the efficiency of the system achieved with considerable flow rate of 0.0056 Kg/s flowing through the helical coiled receiver which was glass covered and coated with black nickel chrome is 63%. Total heat loss from the receiver adversely affects the overall efficiency of the system. Actual field experiments confirm the same. There is average rise of 40% in outlet water temperature with average outlet water temperature of 52^0C. Average increment in receiver temperature was 35 % from initial receiver temperature and average receiver temperature was noted as 88^0C. It also seen that, the theoretical regression modeling equations are satisfied by the actual experimental results; hence can be considered for design purpose and evaluation of thermal performance of the similar systems with different permutations and combinations. The experimental results are presented in the current work and the validity of the model is examined by comparison of the theoretical results with experimental results; and it demonstrates a good agreement between two results. It is to be noted here that, the results described in this paper are valid for water and similar fluids only.

It is concluded that, the described parabolic dish-helical coiled receiver system can prove a good alternative for flat plate and evacuated tube water heaters and could be implemented effectively for low temperature small scale industrial process heat applications. It is further recommended to perform the comparative study of different receiver designs with the helical coiled receiver which can be used with the parabolic solar dish water heaters to select the best receiver.

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