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Analytical performance investigation of parabolic trough solar collector with computed optimum air gap

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Abstract
Parabolic trough collectors have a wide range of industrial as well as domestic applications. This study analytically investigates the performance of non-evacuated absorber tube with glass cover of parabolic trough collector (PTC) in terms of overall heat loss from the absorber. The impact of different parameters such as diameter of absorber tube, mean temperature of absorber tube, wind velocity, emissivity of absorber and ambient temperature have been studied and find the optimum value of air gap. The optimum value of air gap has been computed considering one-dimensional steady state model, where heat loss due to convection plus radiation is equal to heat loss due to conduction plus radiation from absorber tube to glass cover under steady state. Optimum air gap is found to be approximately 7mm and 8 mm for an absorber tube of diameter in range of 1.2-3.18cm and 4.5-7.62cm, respectively. Corresponding to optimum air gap, minimum overall heat loss has been observed. Overall heat loss increases with increase in absorber temperature, wind velocity and emissivity of absorber, whereas decreases with increase in air temperature for different absorber tube diameters. Absorber tube with diameters in the range of 3.18-4.5cm gives better performance. From the obtained data, correlations have been developed, which can be further utilized for designing the PTC system for getting desired output.

Keywords: Parabolic trough collector; Absorber tube; Glass cover; Optimum air gap; Overall heat loss.

1. Introduction

Solar energy is a permanent, environmental friendly and sustainable energy source, which can play a vital role in fulfilling the escalating energy demand and save the depletion of fossil fuel resources. Among different types of solar thermal systems [1] parabolic trough collector (PTC) is receiving much attention, despite of the requirement of solar tracking. Generally, PTC are employed for a wide range of applications from domestic hot water production [2, 3] to steam generation for power [4, 5], industrial process heat generation [6, 7] and air conditioning.

The major advantages associated with PTC are low-pressure drop and achieved temperature about 300-400°C without significant loss in the efficiency of collector. Solar Electric Generation system (SEGS) plant at Kramer Junction in California clearly illustrated that the solar thermal power plants based on PTC are currently the most successful solar technology for electricity generation [8].

Parabolic trough collector (Figure 1) consists of a parabolic reflector to reflect solar energy in to an absorber tube, which is placed at its focal point. Solar radiation is mainly absorbed at the outer surface in form of heat and further transferred partially to working fluid inside the absorber tube by conduction through tube wall and convection from inner surface of the tube to flowing fluid. Remaining transferred by combined effect of radiation, convection or conduction to the inner surface of glass cover through air and then further by conduction from inner surface to outer surface of the glass cover. The heat dissipated to surrounding by two mechanisms, convection to surrounding air and by radiation to surrounding surfaces [9]. Thus, instantaneous thermal efficiency of a PTC is given by [10]:

$$\text{Thermal efficiency} = \text{Optical efficiency} - \frac{\text{Overall heat loss}}{\text{Normal radiation incident on collector aperture}} \quad (1)$$

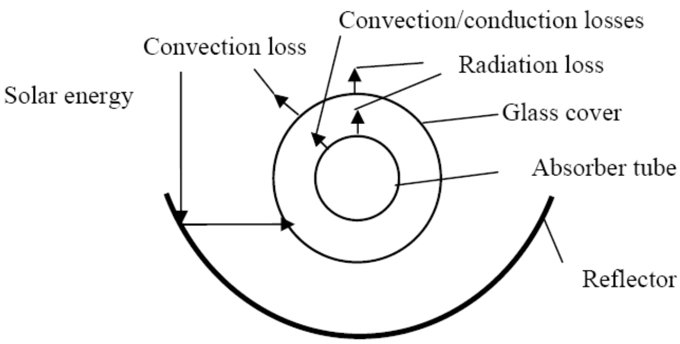

Figure 1. Configuration of Parabolic trough collector

From equation (1), it is clear that the overall heat loss through absorber tube is the prime factor to determine thermal efficiency of PTC system.
Under steady state condition, the overall heat loss from absorber tube is estimated by the correlations (2) or (3) [11-13],

$$Q_L = \frac{\pi \times D \times \sigma \times \left(T_{ab}^4 - T_c^4\right)}{\frac{1}{\in_{ab}} + \frac{D}{D_{ci}}\left(\frac{1}{\in_c} - 1\right)} + \pi \times D \times h_{cab-c} \times \left(T_{ab} - T_c\right) \quad (2)$$

$$Q_L = \frac{\pi \times D \times \sigma \times \left(T_{ab}^4 - T_c^4\right)}{\frac{1}{\in_{ab}} + \frac{D}{D_{ci}}\left(\frac{1}{\in_c} - 1\right)} + \frac{2\pi \times K_{air} \times \left(T_{ab} - T_c\right)}{\ln \frac{D_{ci}}{D}} \quad (3)$$

The overall heat loss from the glass cover is given by equation (4) as [13],

$$Q_L = \pi \times D_{co} \times \sigma \times \in_c \times (T_c^4 - T_{sky}^4) + \pi \times D_{co} \times h_{cc-a} \times (T_c - T_a) \quad (4)$$

Literature, suitably established that depending upon air gap (*y*) between absorber and glass cover, the overall heat loss can be computed using either equation (2) or (3) with equation (4). When convective heat transfer predominates the conduction heat transfer, then the overall heat loss (Q_L) is estimated using equations (2) and (4) otherwise equations (3) and (4) is used.
Out of different parameters, air gap is considered as the most crucial that affects the thermal performance of PTC. Treadwell [14] suggested 1cm annulus gap as the optimum gap between absorber and glass envelope, and based upon that the selection of glass tube sizes have been carried out. The sensitivity of PTC performance with change collector parameters and operating conditions has been carried out by Rabl et al. [15] and reported the optimal gap size of 0.7cm corresponding to an inner glass tube diameter of 3.9cm. Thomas and Thomas [10] presented design data for estimation of thermal loss in the receiver of parabolic trough concentrator at different parameters. The parameters considered for the analysis are:

outer diameter of absorber is 3.18 cm with emissivity 0.15, inner diameter of glass envelope is 5.5 cm having emissivity 0.90 and air gap of 1.16cm. Recently, Mohamad et al. [9] estimated theoretically the thermal performance of PTC and identified the heat losses. The above cited literature clearly enlightened the importance of air gap for optimal performance of PTC. Therefore, an attempt has been made in the present study to find the optimal air gap corresponding to different diameters of tube.

Certain assumptions (similar to Ref. [13]) have been made to carry out this investigation. These are as follows:

- Absorber tube and glass cover constitute a system of infinitely long concentric tubes.
- Flow of heat is one-dimensional.
- Negligible heat transfer via conduction in longitudinal direction.
- Temperature drop across the absorber tube and the glass cover is negligible. So, conduction through the glass cover is neglected.

In this investigation, two cases are assumed separately for finding overall heat loss from absorber to glass cover at optimum gap.

Case 1: heat loss takes place due convection and radiation, and

Case 2: heat loss takes place due to conduction and radiation.

Furthermore, the effects different design parameters and operating parameters on thermal performance in terms of overall heat losses have been discussed.

2. Estimation of optimum air gap and overall heat loss

In the present analytical investigation, a Matlab/Simulink model based on analytical expressions and their related parameters has been developed for calculation of optimum air gap and overall heat loss. The specifications of absorber tube and glass cover considered in the present investigation are reported in Table 1.

Table 1. Specifications of absorber tube and glass cover used in present study

Specifications	**Data**
Outside diameters of absorber tube, (*D*)	1.2, 2.2, 2.54, 3.18, 4.5, 5.08, 6.03, 7.62(cm) (Standard diameters of commercially available tubes)
Thickness of glass cover, (*t*)	0.002m
Emissivity of glass cover, ($\in_c$)	0.90
Emissivity of selective coating, ($\in_{ab}$)	0.15
Rim angle, (ϕ)	90°

The computation of optimum air gap and overall heat loss by using equations (2), (3) and (4) has been carried out as follows:

2.1 Overall heat loss between absorber tube and glass cover

Step 1: Initialization

a) Put the values of known parameters i.e. T_{ab}, T_a, $D, V_w, \in_{ab}$
b) Assumed initial guess value of air gap is 5mm.
c) Assumed value of T_c would be corrected till steady state is achieved using equations (2) and (4).
d) Further, the corrected value of T_c is kept constant and find the corrected value of air gap, where heat loss due to conduction and convection becomes equal. At this condition, the air gap is optimum and known as optimum air gap.
e) Using step (c) at optimum air gap and calculate T_c corresponding to optimum air gap.

Step 2: Calculation

a) Calculate mean temperature $\left(T_{mab-c} = \dfrac{T_{ab} + T_c}{2}\right)$.
b) At this mean temperature, calculate following properties of air between absorber and glass cover:

- Density(kg/m^3) using Ideal gas law at atmospheric pressure =101325N/m^2,
- Dynamic viscosity using the correlation given by Sutherland's [16],

$$\mu = \mu_\circ \left[\frac{T_{so} + C_o}{T_{mab-c} + C_o}\right]\left[\frac{T_{mab-c}}{T_{so}}\right]^{3/2} \tag{5}$$

- Thermal conductivity using the equation (6) [13],

$$K_{air} = 1.4\times10^{-11}T_{mab-c}^{3} - 5.3\times10^{-8}T_{mab-c}^{2} + 1.1\times10^{-4}T_{mab-c} - 0.0019 \tag{6}$$

- Kinematic viscosity and Prandtl number estimated from the above estimated values and the value of specific heat (C_p =1009 J/kg-K) is assumed to be constant at all T_{mab-c}.

c) Using above estimated values and data, Rayleigh number (R_a) can be calculated as [10].

$$R_a = \frac{g\times\beta\times(T_{ab} - T_c)\times Z^3\times \Pr}{\nu^2} \tag{7}$$

where, $\beta = \dfrac{1}{\dfrac{T_{ab} + T_c}{2}}$

d) Effective thermal conductivity has been estimated by Raithby and Holland correlation [17] as given below:

$$\frac{K_{eff}}{K_{air}} = 0.317(R_a^*)^{1/4} \tag{8}$$

where, $(R_a^*)^{1/4} = \dfrac{\ln\left(\dfrac{D_{ci}}{D}\right)\times R_a^{1/4}}{Z^{3/4}\left(\dfrac{1}{D^{3/5}} + \dfrac{1}{D_{ci}^{3/5}}\right)^{5/4}}$.

The limitations on using equation (8) are that R_a^* should be less than 10^7. The effective thermal conductivity K_{eff} cannot be less than the thermal conductivity K_{air}. Hence, the value of K_{eff}/K_{air} is taken as unity if equation (8) yields a value less than unity [13].

e) The value of h_{cab-c} can be computed as using equation (9) [13],

$$h_{cab-c} = \frac{2\times K_{eff}}{D\times\ln\left(\dfrac{D_{ci}}{D}\right)} \tag{9}$$

f) Substituting the calculated values of h_{cab-c} and other parameters in equations (2) and (3), to get the overall heat loss (Q_L) between absorber tube and glass cover at optimum air gap.

2.2 Overall heat loss between glass cover and surrounding air

Following steps have been followed:

Step 1: Same as above step 1.

Step 2: Calculation

a) Calculate mean temperature, $T_{mc-a} = \left(\frac{T_c + T_a}{2}\right)$

b) Use the correlations used in step 2 (b) from to estimate above properties of air on outside surface of glass cover at mean temperature, T_{mc-a} .

c) For the assumed value of wind velocity V_w , the Reynold number (Re) was computed using equation (10),

$$\mathrm{Re} = \frac{\rho_{air-c} \times D_{co} \times V_w}{\mu_{air-c}} \tag{10}$$

d) To compute Nusselt number, Hilpert's correlation has been used [13, 18]

$$Nu = C_1 \mathrm{Re}^n \tag{11}$$

where C_1 and n are constant having following values:

- for 40 < Re < 4000, C_1 = 0.615, n = 0.466;
- for 4000 < Re < 40000, C_1 = 0.174, n = 0.618;
- for 40000 < Re < 400000, C_1 = 0.0239, n = 0.805.

e) h_{cc-a} can be calculated using equation (12).

$$h_{cc-a} = \frac{Nu \times K_{air-c}}{D_{co}} \tag{12}$$

f) Substituting the calculated values of h_{cc-a} and other parameters in equation (4), to get the overall heat loss (Q_L) between glass cover and surrounding air at optimum air gap.

3. Results and discussions

The value of computed optimum air gap for different diameter of absorber tube and parameters are presented in Table 2.

Table 2. Optimum air gap for different diameter of absorber tube at T_{ab} = 250°C, T_a = 10°C, V_w = 1 m/s

D (cm)	y_{og} (mm)	D_{ci} (cm)	D_{co} (cm)	Q_{L-cd} (W/m)	Q_{L-cv} (W/m)	Q_L (W/m)
1.20	5.83	2.366	2.766	66.87	68	88.29
2.20	6.43	3.486	3.886	95.39	94.77	131.2
2.54	6.64	3.868	4.268	103.6	102.9	144.8
3.18	7.00	4.58	4.98	117.9	117.2	169.2
4.50	7.65	6.03	6.43	144.3	143.3	216.8
5.08	7.85	6.65	7.05	155.8	154.8	236.7
6.03	8.10	7.67	8.07	173.2	172.5	269.3
7.62	8.68	9.356	9.756	201.3	200.3	322

As the diameter of absorber tube increases, the optimum air gap increases slightly as observed from Table 2 and Figure 2. Optimum air gap is found to be approximately 7mm and 8mm for an absorber tube

of diameter in range of 1.2-3.18cm and 4.5-7.62cm, respectively for optimal performance. Heat loss due to conduction is nearly equal to heat loss due to convection at optimum air gap for given diameter of absorber tube is also clearly observed from Table 2. Further inspection of Figure 2 depicts the overall heat loss increases directly with increase in absorber tube diameter. Too larger diameter gives a higher intercept factor but simultaneously enhance the overall heat losses [19]. Therefore, the final selection of absorber tube diameter has been made on the basis of thermal analysis and a trade-off between increased interception of reflected solar energy and acceptably less heat losses.
Further, to reduce the overall heat loss, it is necessary to find the optimal air gap corresponding to different absorber tube diameters. That's the reason, which attributed the development of correlations presented in Table 3 for optimum air gap and overall heat loss with the variation of diameter of absorber tube. This procedure can also be utilized to compute design data and correlations at other parameters.
For estimating the accuracy of the developed correlations, error analysis has been carried out for the data obtained from the analytical expressions and the proposed correlations. It is varying in between 0.28% to 1.08% for optimum air gap and 0.17% to 1.31% for overall heat loss corresponding to the absorber tube diameter ranging from 1.2cm to 7.62cm.

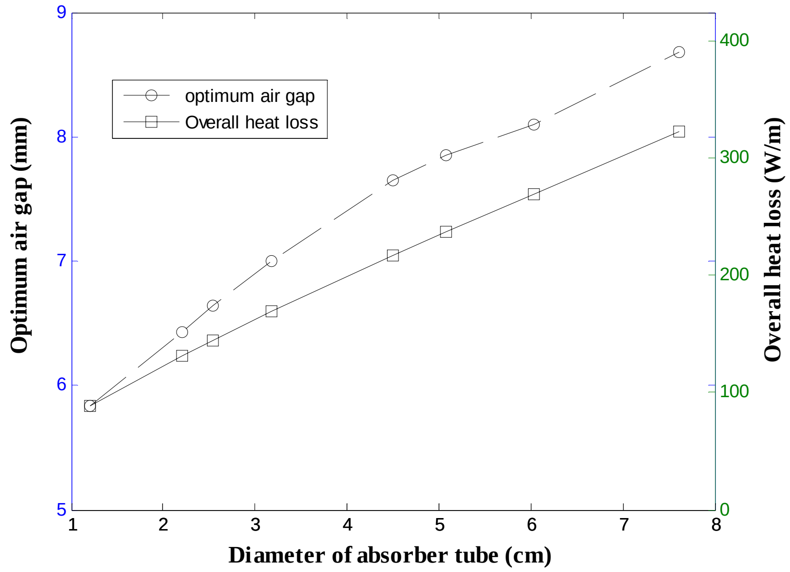

Figure 2. Variation of optimum air gap and overall heat loss with diameter of absorber tube at T_{ab} =250°C, T_a =10°C, V_w = 1 m/s

Table 3. Correlations for optimum air gap and overall heat loss with diameter of absorber tube

$y_{og} = -0.031D^2 + 0.71D + 5$
$Q_L = -0.82D^2 + 43D + 39$

Figure 3 shows the variation of overall heat with air temperature for different diameters of absorber tube. From the analysis of Figure 3, it is clear that with increase in temperature of air, overall heat loss decreases for a fixed absorber tube diameter at optimum air gap. As the diameter of absorber tube increases from 1.2cm to 7.62cm, the overall heat loss increases at a given air temperature.
From Figure 4, it has been clearly observed that the overall heat loss increases with increase in wind velocity for the given diameter of absorber tube with their computed optimum air gap. The overall heat loss also increases when diameter of absorber increases for same wind velocity.
The effect of emissivity of selective coating of absorber tube on overall heat loss has been shown in Figure 5. It reveals that with increase in emissivity of absorber tube, overall heat loss also increases from absorber to air for different diameters of absorber tube. Overall heat loss also increases when diameter of absorber increases for same emissivity of absorber surface. The range of emissivity considered in the present investigation is nearer to the range of emissivity of some selective surfaces which varies from 0.09 to 0.17 [20].

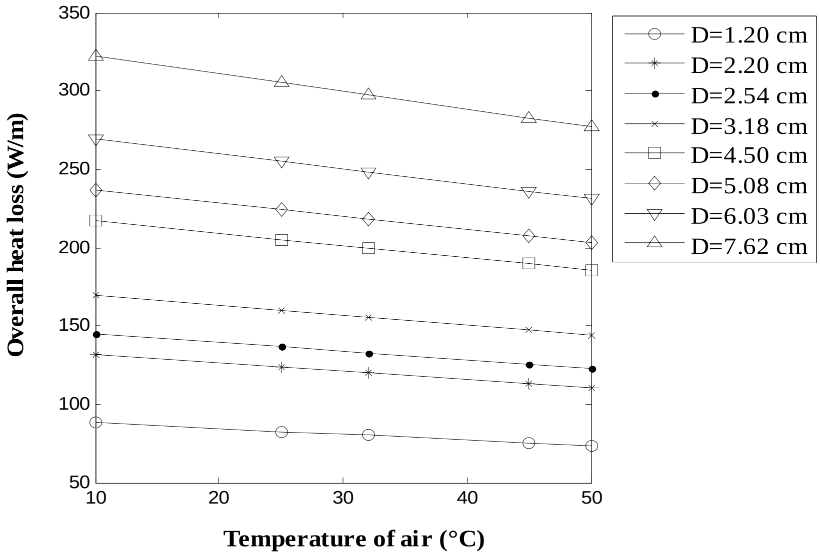

Figure 3. Variation of overall heat loss with temperature of air at T_{ab} =250°C, V_w =1 m/s

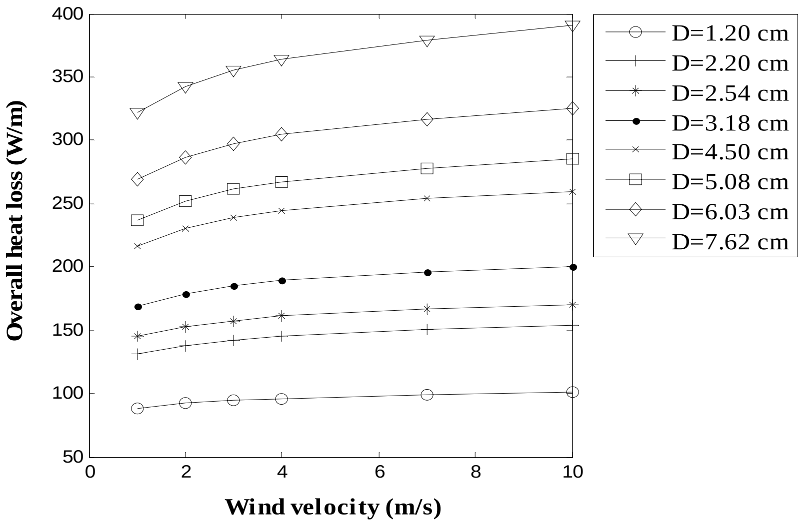

Figure 4. Variation of overall heat loss with wind velocity at T_{ab} =250°C, T_a =10°C

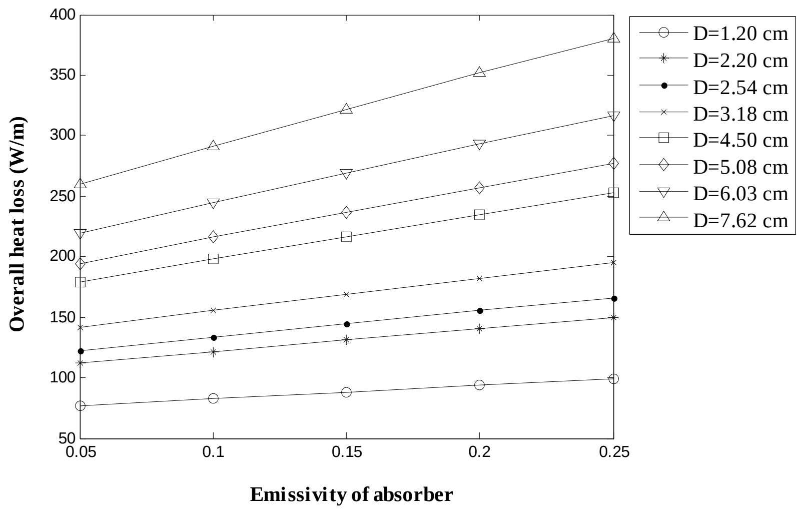

Figure 5. Variation of overall heat loss with emissivity of absorber at T_{ab} =250 °C, T_a =10°C, V_w =1 m/s

The temperature of absorber tube has significant effect on overall heat loss (Figure 6). Overall heat loss increases with increase in temperature of absorber tube. Overall heat loss also increases when diameter of absorber increases for same temperature of absorber.

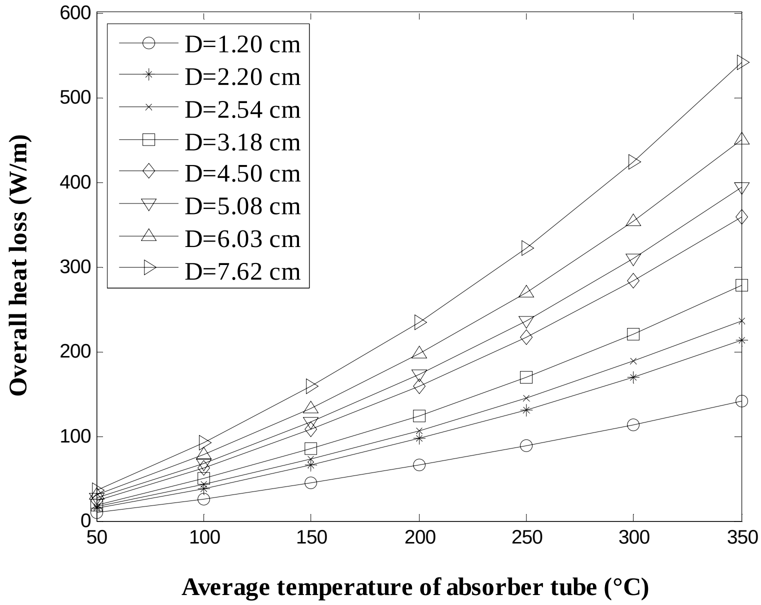

Figure 6. Variation of overall heat loss with average temperature of absorber tube at T_a=10 °C, V_w=1m/s

4. Validation of results

The correctness of the proposed model has been evaluated by comparing the results in terms of heat loss with the results of Thomas and Thomas [10] at D=3.18, D_{ci}=5.5 and D_{co}=6, Air gap (y)=11.6mm (Figure 7) The values of other parameters have been kept constant as per Ref. [10] to validate the results (T_{ab}=250°C, V_w=1m/s, $\in_{ab}$=0.15). From the inspection of Figure 7, it is clear that the proposed model gives close agreement (variation within ≤ 1.2 %) with Ref. [10]. The optimum air gap is found to be approximately 7mm for absorber tube diameter of 2.54cm, which is similar to that Ref. [15] and [16].

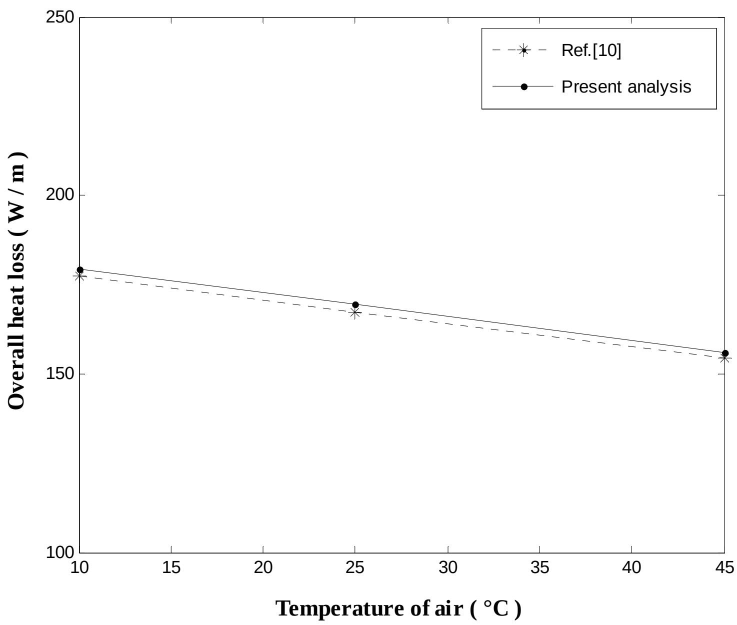

Figure 7. Variation of overall heat loss with temperature of air at D=3.18cm, D_{ci}=5.5cm and D_{co}=6cm, Air gap (y)=11.6mm

5. Conclusion

The present analysis and proposed correlations will be useful for computing optimum air gap in order to minimize overall heat loss and finally enhancing the performance of PTC system. Here, the procedure for finding optimum air gap and overall heat loss has been developed for non- evacuated absorber tube. At optimum air gap, non-evacuated configuration of PTC system performs better than at any other value of air gap, which results in improved efficiency as well as reasonable economic benefits. Overall heat loss increases with increase in absorber temperature, wind velocity, emissivity of absorber and decreases with increase in air temperature. From the overall analysis, it has been observed that the absorber tube diameter ranging from 3.18 to 4.5cm provides considerable overall heat losses keeping in view of interception of reflected solar energy at that absorber tube diameter. The benefits of such non-evacuated configuration at optimum air gap are simple design for installation and flexible for the developing countries.

Nomenclature

C_o	Sutherland's constant for air = 120 K
D	outer diameter of absorber tube (m)
D_{ci}	inner diameter of glass cover (m)
D_{co}	outer diameter of glass cover (m)
g	acceleration due to gravity = 9.81 (m/s^2)
h_{cab-c}	convective heat transfer coefficient between absorber tube and glass cover (Wm^{-2}K^{-1})
h_{cc-a}	convective heat transfer coefficient of outside surface of glass cover (Wm^{-2}K^{-1})
K_{air}	thermal conductivity of air between absorber tube and glass cover (Wm^{-1}K^{-1})
K_{eff}	effective thermal conductivity of air between absorber tube and glass cover (Wm^{-1}K^{-1})
K_{air-c}	thermal conductivity of air on outside surface of glass cover (Wm^{-1}K^{-1})
Pr	Prandtl number
Q_L	overall heat loss rate per unit length from absorber tube (W/m)
Q_{L-cd}	heat loss due to conduction from the absorber tube (W/m)
Q_{L-cv}	heat loss due to convection from the absorber tube (W/m)
Re	Reynolds number
R	gas constant for air = 287 J/kg-K
R_a^*	modified Rayleigh number
t	thickness of glass cover(m)
T_{ab}	average temperature of the absorber tube (K)
T_c	average temperature of glass cover (K)
T_a	ambient/air temperature (K)
T_{mab-c}	mean temperature between absorber and glass cover (K)
T_{mc-a}	mean temperature between glass cover and ambient air (K)
T_{sky}	sky temperature (K) = $T_a - 5$ [11]
T_{so}	reference temperature in Sutherland's formula = 291.15 K
V_w	wind velocity (m/s)
y	Air gap between absorber tube and glass cover(m)
y_{og}	optimum air gap by analytical expressions(m)
Z	Characteristic dimension (m) = $(D_{ci} - D)/2$

Greek symbols

$\in_{ab}$	emissivity of selective coating of absorber tube
$\in_c$	emissivity of glass cover
σ	Stefan-Boltzmann constant = 5.67×10^{-8} (Wm^{-2}K^{-4})
μ	dynamic viscosity at input temperature T_{mab-c} (N-s/m^2)
$\mu_\circ$	reference viscosity at T_{so} =18.27×10^{-6} (N-s/m^2)
ρ_{air-c}	density of air on outside surface of glass cover (kg/m^3)
μ_{air-c}	dynamic viscosity of air on outside surface of glass cover (N-s/m^2)
ν	kinematic viscosity of air between absorber tube and glass cover (m^2/s)

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