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Experimental Study of Heat Loss from Receivers of Solar Collectors under Different Conditions

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***Abstract*— Heat loss from receivers in solar thermal energy systems is a major factor that can significantly affect efficiency and consequently cost effectiveness of the system. Therefore, it is necessary to understand the exact nature of heat losses from receivers. Heat losses from a parabolic trough collector (PTC) / compound parabolic collector (CPC) primarily depend on the temperature of fluid contained in the receiver, wind speed, surface area of the receiver and orientation of the reflector.**

In this paper, experimental study of overall heat losses from receivers are reported for different laboratory conditions such as controlled wind speeds, different receiver temperatures, and also different enclosures for receiver pipes. Experimental measurements give the overall heat loss and from which the quantitative estimates of convective, conductive and radiative losses were made. Experimental observations for the convective heat loss from bare pipe and from that with glass tube enclosure were found to agree with the reported correlations in the literature. Further, from these experimental results we have developed correlations for these geometries for different receiver temperatures and wind speeds. The results also show that 'insulated mirror cap' enclosure would be a cost effective method for reducing heat loss.

Keywords*— *Solar thermal, heat loss, solar collector receivers, non-evacuated receivers, mirror cap receivers.

1. INTRODUCTION

Solar energy is a clean and the promising renewable source of energy having huge potential to fulfil the future energy requirement. CPC, PTC, Dish and Solar tower are well developed and widely deployed solar thermal technologies. The receiver, i.e. Heat Collecting Element (HCE) is a crucial part in the solar thermal technologies. It is designed to improve efficiency of conversion of concentrated sunlight to thermal energy.

In recent years, considerable research activity is directed to the development of solar collectors generating higher temperatures, useful for Industrial Process Heat (IPH) applications and electricity generation. Among all the solar thermal technologies, parabolic trough collector (PTC) is a technically mature and commercially competitive technology. The market potential for industrial process heat application of solar concentrators is considerable (Jadhav et al. 2013). In India approximately 50% of the total thermal energy usage is in the range of temperatures 100- 300°C. Presently India's annual energy consumption is about 600MTO and this figure is increasing day by day. Hence, the challenges and opportunities for this solar application are self-evident. In particular for use in IPH applications up to 200ºC, CPC technology is relatively superior as compared to the other technologies in certain aspects. Particularly because; it does not require an elaborate tracking mechanism and captures a part of diffused radiation.

In solar thermal systems, heat loss is a major factor significantly reducing efficiency and increasing cost of the system. The overall heat loss occurs due to convection, radiation and conduction. The size, shape, surface texture characteristics of the coating, temperature, and enclosure geometries are important parameters that affect thermal performance of the receiver. An order of magnitude estimate indicates that the major loss from a receiver pipe is due to convection from the bare or enclosed surfaces.

For reducing heat losses, the receiver pipes can be enclosed in an evacuated glass tube and coated with a special selective coating. It increases efficiency of the system, so also the cost. Also, it is difficult to maintain a vacuum in the receiver tube and there are greater chances for braking. Therefore, it is essential to develop alternatives which give better performance, are easy to handle and also economical.

So, it becomes necessary to measure heat losses from receiver pipe to the surroundings with different types of enclosures when a wind of known velocity is maintained outside, and when the annular gap is non-evacuated.

Extensive experimental studies have been reported in the published literature on natural convection heat transfer from horizontal cylinders. Table 1 shows a summary of experimental works on heat loss published in the past few years, whereas table 2 shows previous analytical/ numerical work published in the literature. It can be seen that substantial

Table 1: Analysis of previous *experimental work* published in the literature

Author, year	Experimental set up	Results	Comments
(Ratzel AC, Hickox and Gartling 1979)	The Sandia Laboratories receiver assembly was used. Heat given by a heater element inside the receiver tube. Input power was maintained to within ±1.0 W during testing.	Total heat loss reductions of 10 to 50 % may be obtained if conduction heat loss is limited. Natural convection was studied considering non-uniform temperature distributions and eccentric cylinders. Highly nonuniform temperature distributions affect a natural convection process between concentric cylinders and large eccentricities cause increase in natural convection heat transfer.	Study for vacuum, other gases in the annuals between receiver pipe and glass tube. Effect of external wind speed was studied
(Faik Abdul Wahab Hamad 1989)	Two Al cylinders (length 90 cm) formed the annulus with 44 mm as the outer diameter of the inner cylinder 72 mm as the inner diameter of the outer cylinder. Inner was heated by 900 W electric heater. To avoid the effects of air currents and to reach the steady state condition, the apparatus was placed in a closed room. Experiments conducted to investigate the influence of angle of inclination on natural convection heat transfer through air layer bounded by two cylinders closed at their ends.	Heat transfer taking place in the transition region between conduction and laminar convection is: $1300 < Ra_L < 8000$. Nu increases with Ra for θ equal to 0, 30, 45 ° & constant for θ equal to 60°. It decreases for the vertical condition. Variation of HTC is small with Ra number, so that the enclosed annular space should be considered a good insulating method for heat transfer by natural convection. The angle of inclination has a small effect on HTC through the annulus. Optimum angle for maximum heat transfer is 30° to horizontal.	Study for non-evacuated annuals at different angles of inclination. Other receiver geometries and external wind speed not considered.
(Kitamura, Kami-iwa, and Misumi 1999)	The cylinder was of acrylic resin pipe 4 mm thick and stainless steel foil heaters 30 mm thick. The heaters were glued on the outer surface of the cylinders and were connected in series. The cylinders were heated with uniform heat flux and their diameters were varied from 60 to 800 mm to enable experiments over a wide range of Rayleigh numbers. Length of cylinder 500mm. Water-repellent foam rubbers 20 mm thick thermally insulates the inner surface of the cylinder to reduce the conduction heat loss.	Three-dimensional flow separations occur first at the trailing edge of the cylinder when Ra_D beyond 2.1 x 10^9, and the separation points shift upstream with increasing the Rayleigh numbers. These separations become a trigger to the turbulent transition. Transitional and turbulent flows appear downstream of the separations at higher Rayleigh numbers. Also, they occupy a small portion of the cylinder surfaces, even at the maximum. Rayleigh numbers. The local heat transfer coefficients were also measured. The results show that the coefficients are increased markedly in the transitional and turbulent regions.	Study for non-evacuated receiver only for different diameters. Other receiver geometries and external wind speed not considered.
(Misumi, Suzuki, and Kitamura 2003)	The cylinders were heated with uniform heat flux and their diameters were varied from 200 to 1200 mm. A wide range of modified Rayleigh numbers, $Ra_D = 1.0 \times 10^8$ to 5.5×10^{11}. The flow fields around the cylinders were visualized with smoke to investigate the turbulent transition mechanisms. The results show that three-dimensional flow separations occur first at the trailing edge of the cylinder when Ra_D exceeds 3.5×10^9, and the separation points shift upstream with increasing Rayleigh numbers.	The local heat transfer coefficients were also measured. The results show that the coefficients are increased significantly in the transitional and turbulent regions compared with the laminar coefficients. Moreover, the present results for air were compared with previous results for water and the effects of Prandtl number on the flow and heat transfer were discussed.	Study for receiver having different diameters Other receiver geometries and external wind speed not considered.

(Nada 2007)	Two copper tubes of diameters 37and 75 mm, respectively, the length is 400 mm. Dimensionless annulus gap widths (L/Do) 0.23, 0.3 and 0.37 and inclination 0 º, 30 º, 60º. Cooling water circulated around annulus outer tube at high flow rate. Adjusted heater power to obtain a certain Rayleigh number, experiments run for a long period (3-4 hr) until steady state condition was achieved. Repeat for different inclination angles; and five different Rayleigh numbers varying from 5 x 10^4 to 5 x 10^5.	Increasing the annulus gap width increases the heat transfer rate; the heat transfer rate slightly decreases with increasing the inclination of the annulus from the horizontal, increasing Ra increases the heat transfer rate for any L/Do and at any inclination. Correlations of the heat transfer enhancement due to buoyancy driven flow in an annulus has been developed in terms of Ra, L/Do and α	Study for air filled two concentric copper tube receivers at different inclination, not for another receiver geometries and wind speed.
(Lüpfert et al. 2008)	4 m long receiver is placed and heated up by 4 m long quartz heating element. The heater is surrounded by a copper tube 50 mm diameter, to reduce temperature gradients. The absorber ends are insulated with a mineral wool of 60 mm thickness. The temperature of the absorber tube, copper tube, glass envelope, and ambient is monitored with 14thermocouples.	Measured and analysed the temperature of the glass envelope of evacuated receivers and to model overall heat losses and emissivity coefficients of the receiver. For solar parabolic trough plants operating in the usual 390°C temperature range, the thermal loss is around 300 W/m receiver length.	Experiments only related to glass tube receiver geometries.
(Burkholder 2009)	HCEs are 4.06 m long at 25°C (4.08 m at 400°C) with a stainless steel absorber inner/outer diameter of 6.6/7.0 cm. To test HCE heat loss, two 2.17 m long, 5.4 cm outer diameter copper pipes with internal heaters are inserted into the ends of an HCE The copper pipe evens out the temperature distribution generated by three internal electric resistance heaters. Two of the heaters are 3-cm-long, stainless-steel-sheathed, coiled cable heaters whose surfaces contact the interior of the copper pipe. These heaters are referred to as "coil heaters" in the remainder of this report. The third heater is a 2.12 m (2.01 m heated-length) inconel-sheathed cartridge heater suspended along the cylindrical axis of the copper pipe using inconel spacers.	The uncertainty associated with average temperatures and temperature differences is about ±1°C, and for heat loss, it's ±10 W/m Thus a simple estimation of HCE heat loss in parabolic trough solar fields corresponds to the heat loss that occurs at an absorber temperature of 350°C. The heat loss of the 2008 PTR70 is 150 W/m. Studies of some previous generations of the PTR70 found heat losses between 200 and 270 W/m at an absorber temperature of 350°C. The black chrome receivers installed at the SEGS plants lost more than 350 W/m at this temperature. The HCE heat loss has been decreased significantly.	Experiments only related to vacuum tube receiver geometries.
(S.Ozgur Atayılmaz et .al. 2010)	Two copper cylinders with different diameters. Bare cylinders (4.8 & 9.45mm) and wrapped cylinders (9 & 12.8 mm) and 1m length. Room dimensions 4000×4900× 2550mm. Air velocity 0 to 0.25 m/s, Ambient temp T=10 °C to 40 °C. Outer surface temp. 10 °C to 60 °C Experiments are performed in the different uniform wall and ambient temperature. Steady state 5 hr later. Insulating material made of XPS are placed on endpoints of the cylinder the conduction heat loss is neglected, heat transfer from the horizontal cylinder surface by convection	Average Nu numbers were calculated and compared with known correlations on natural convection heat transfer from a horizontal cylinder in the specified range of Rayleigh number by using experimental data. Experimental and numerical results fall in ±30% band. Nu numbers increase with increasing Ra numbers. Numerical and experimental data points show similar trends.	Study for air filled two concentric copper tube receivers at different wind speed and pipe temperature.

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Table 2: Analysis of previous *numerical work* published in the literature

Author, year	Experimental set up	Results
(S.C. Mullick S.K. Nanda 1982)	Absorber with a concentric glass cover. Developed solution for this geometry without the requirement of an iterative solution for the simultaneous equations.	The semi empirical equation predicts the heat loss factor to within ± 1.2% of the value obtained by the actual solution of the simultaneous equations, in the range of variables wind velocity 0.5m/s to 10m/s, absorber temperature 20 °C to 200 °C above ambient air temperature and emittance of coating, 0.1 to 0.95.
(Bhowmik et al. 1985)	Absorber with a concentric glass cover. Under steady state the total radiative and convective heat loss from the absorber at temperature TI to the transparent glass cover at T2 equals that from the glass cover to the ambient air at Ta.	This equation predicts the overall heat loss factor, HUF, to within ± 5% of the value obtained by exact solution of the simultaneous equations, in the range of variables--absorber temperature, 60°C to 220°C, emittance 0.1 to 0.95, and wind velocity, 1.5 m/s to 10 m/s.
(Mazumder et.al. 1986)	Tubular receivers with an evacuated space between the absorber and concentric glass cover to suppress convection heat loss are employed as absorbers of linear concentrators in the intermediate temperature range.	A correlation for heat loss factor of evacuated receivers as a function of the variables absorber temperature, emittance, diameter and wind loss coefficient. The correlation predicts the heat loss factor to within + 1.5% of the value obtained by exact solution of equations. Range of variables: wind loss coefficient, 10-60 W/m^2 °C; emittance, 0.1-.95; absorber temperature, 50-200°C.
(Patil et al. 2014)	Assuming non evacuated receiver pipe temperature varying along the surface of receiver. Sinusoidal and square wave functions are employed in modelling, since actual temperature distributions on solar receiver pipes are combinations of these two functions.	An hour angle increases from 0 to 90, heat loss decreases by 20% in case of sinusoidal temperature distribution and 24% in case of square wave temperature distribution. The effect of radius ratio (RR) on heat loss has been studied, 1.375 is the critical radius ratio, for which heat losses from the receiver are minimum

work has been carried out for different types of receiver geometries, and for different external wind speeds. Problems exist for glass tube enclosed receivers and vacuum tube enclosed receivers such as high cost, higher chances of breakage. Mirror cap receiver geometry, introduced and studied by us in this work has some additional benefits explained below.

Tables 1 & 2 show that the published work in literature mainly focuses on the geometry where the receiver pipe is enclosed in a glass tube, and effect of external wind has mostly not been considered. In case of solar collectors, external wind is an unavoidable factor. Also other types of enclosures for receiver pipe were not considered. The objective of the present study is to improve performance of the collector by modifying receiver assembly to reduce the overall losses. For this purpose, heat losses from the receivers with different types of enclosures at different pipe temperatures and for different wind speeds are studied. Experiments were performed in controlled laboratory conditions and at uniform surface temperatures of the receiver. Different geometrical configurations comprise bare pipe, pipes enclosed with glass tube and with mirror caps explained in the next section. The experimental results for bare pipe and pipe enclosed in a single glass tube are compared with the empirical correlations available in the literature.

2. EXPERIMENTAL

2.1. Experimental setup

The experimental study of heat losses by a receiver pipe of a solar collector on the actual site of the solar field is not easy. This is because, the environmental conditions are not steady; and one cannot control them. Experiments were, therefore, set up in the laboratory, where the parameters of the surrounding conditions could be controlled.

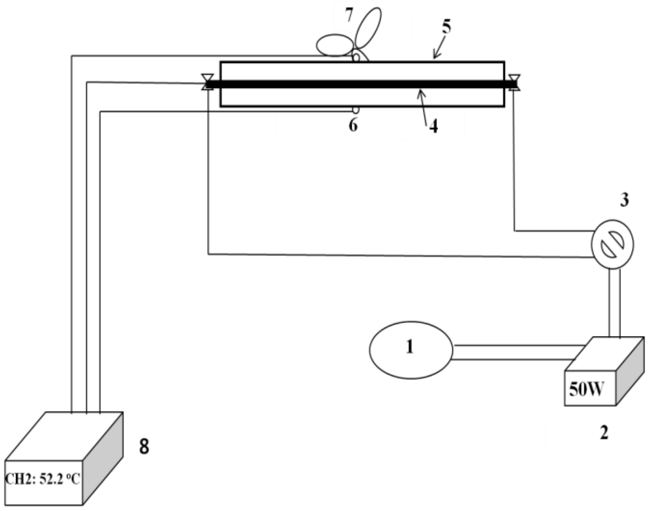

Figure 1: Schematic diagram of experimental setup (1: Ac mains, 2: Wattmeter, 3: Varaic, 4: Heater, 5: Pipe, 6: Thermocouples, 7: Fan and 8: Temperature Data logger)

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The schematic of the experimental setup is shown in Fig. 1. An electric heating rod (4) was placed inside the centre of a MS pipe (5) of OD 0.032m and electric supply (1) was given through a watt meter (2) and a voltage regulator (3). Wattmeter measured the power given to the heater, which was varied as required. The temperature was recorded (8) at various locations, three locations (6) on the receiver, three locations on the glass tube or mirror cap, one location on the glass plate and one thermocouple was used for recording ambient temperature. A pedestal fan (7) was kept near the pipe. When the pipe reached steady state, power supplied to maintain its temperature equalled the total heat loss from the pipe. A similar method was employed earlier (Lupfert and Riffelmann)[21] and NREL in the tests of Solel UVAC receiver and Schott PRT-70 receiver (Burkholder and Kutscher) [22].The measuring instruments used to measure various parameters were, temperature data logger with storage module, thermocouples, wind anemometer, and watt meter. The pedestal fan was used to adjust wind speeds across the length of the pipe to 0 m/s, 1m/s, 2m/s, and 2.5m/s.

In this experimental set up we have used four different types of enclosures.

i) In one setting, a bare metallic pipe of length 0.5m and OD 0.032m with solkote coating was heated with an electrical heater placed inside the pipe. Wattage supplied to the heater was varied to attain different steady state temperatures.

ii) A glass tube having 0.054m OD enclosed the receiver pipe and ends of the annulus between the glass tube and pipe were sealed with the help of silicon 'O' rings which could sustain up to 200°C. Thus the annulas gap between pipe and glass tube was 9mm. The arrangement (fig. 2 (a)) is similar to the receivers of many PTC and CPC type of collectors

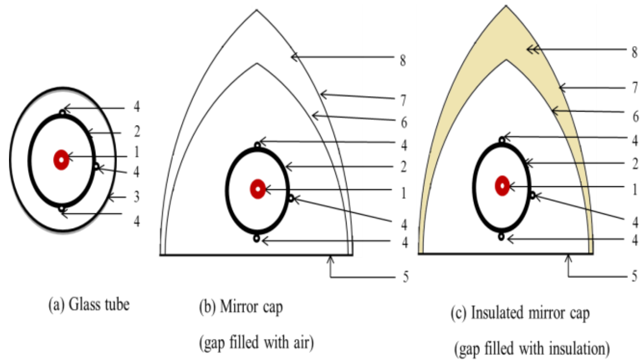

Figure2: Schematic of receiver geometry (1: Heater, 2: Receiver pipe, 3: Glass tube, 4: Thermocouples, 5: Glass plate, 6: Inner reflecting wall, 7: Outer wall, 8: air gap, 9: Insulation)

iii) A mirror cap is a parabolic shaped and double walled cavity (fig. 2(b)). Inner wall (A) is reflecting and faces the receiver pipe. Consequently, this wall reflects back the thermal radiations emitted by the pipe and thus reduces radiative heat losses. Also, the 'capturing' of reflected radiation improves because the radiations reflected towards the receiver pipe, but missed by it become incident on the inner reflecting wall and are reflected back to the receiver pipe. An air gap (B) is maintained between the inner wall and the outer one, which reduces conductive and convective heat losses. Alternatively the gap is filled with insulating material. This slightly increases the weight and cost of the mirror cap; but heat loss is reduced further.

Some solar thermal technologies (Fresnel reflector, modified CPC, etc.) reflected rays fall on the receiver pipe from below. The mirror cap is placed above the pipe. Hence the cap is effective in reducing heat losses and trapping even 'missed' radiation. It also prevents radiative and convective losses from the receiver.

iv) An insulated mirror cap was also used in this work to study the reduction in heat loss from the receiver (fig. 2(c)), where the gap between two walls of the mirror cap was filled using insulating material like glass wool.

v) Effect of external winds flowing at different speeds on heat losses from receivers was also studied. A pedestrian fan generating wind with a maximum measured speed of 3m/s was used for this purpose. The wind was blown perpendicular to the length of the pipe- bare pipe, that enclosed with single glass tube and that enclosed in a mirror cap, and an insulated mirror cap. Different wind speeds were achieved by adjusting the power supplied to the fan, and its distance from the pipe. Wind speeds were recorded at the center and at both ends of the pipe in each experiment and the mean value was taken as the average wind speed.

2.2. Experimental procedure:

The set up consisted of a MS pipe of 0.5 m length with 0.032m OD, which was coated with a solar selective coating. The pipe was heated using electric heater until a steady state temperature was reached. The experiments were performed by reaching different steady state temperatures by controlling wattage supplied. The experiments were repeated for, no wind and for different wind speeds. In each case the steady state temperature, the wattage and the wind speed were recorded.

The same procedure was followed for glass tube enclosed pipe, mirror cap enclosed pipe and insulated mirror cap enclosed pipe. Electric heater was the heating source that balanced during steady state heat loss from the bare pipe, or pipe with glass tube enclosure, or with mirror cap or with insulated mirror cap enclosures.

2.3. Temperature Measurement

The pipe surface temperatures were recorded with IM 2000-16KS data logger via K-type thermocouples having a resolution of 0.1°C and an accuracy of 0.3% over the temperature range 0°C to 200oC. Three thermocouples were strategically located on the pipe surface as shown in figure 2 (a), 2 (b) or 2 (c). To check for uniform surface temperature of the pipe and its stability, preliminary heating tests were conducted. Equilibrium conditions were approached after 4-5 hours of heating and were verified by subsequently recording temperatures at 1 minute time intervals. Equilibrium was taken to be established when the variation in temperature readings over a twenty-minute period was contained within 0.75 % of the temperature of the pipe at the beginning of that time period. The fluctuations in individual temperature readings were less than 0.2 % for the twenty-minute period. The same procedure was adopted for glass tube enclosed pipe, mirror cap enclosed pipe and insulated mirror cap enclosed pipe.

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2.4 Emissivity Measurement

All objects at all temperatures emit thermal radiations, which depend on the emissivity of the object. It is a dimensionless number between 0 (for a perfect reflector) and 1 (for a perfect black body).

The empirical correlations in the literature are valid only for convective heat loss; and experimental results give the total heat loss. So, radiative heat loss needs to be subtracted for comparison. To calculate radiative heat loss, emissivity of coating on the pipe should be known. To know the emissivity of (solar selective coating) solkote, following experiments were performed. Two receiver pipes of identical dimensions and of same material were chosen; one coated with solkote and the other wrapped tightly with aluminium foil having good reflectivity. Three thermocouples were connected at different locations to measure temperatures. Both pipes were kept for heating until steady state was reached (4-5 hours heating). Then heating was stopped and pipes were allowed to cool. Temperatures of both pipes were recorded

at regular interval of time. We then calculated rate for cooling for both pipes at the same temperature.

Rate of heat loss from reflecting pipe

$$M_r S_r \left[\frac{d\theta}{dT}\right]_r = 0.04\sigma A_r (T_r^4 - T_0^4) + Convective\ loss \quad (1)$$

Where the emissivity of aluminium foil is 0.04 from website [24] .

Rate of heat loss from black pipe

$$M_s S_s \left[\frac{d\theta}{dT}\right]_s = \varepsilon\sigma A_s (T_s^4 - T_0^4) + Convective\ loss \quad (2)$$

Convective losses in both the cases must be equal at the same temperatures. Equating above equations we get emissivity of solkote as,

$$\varepsilon = \frac{M_s S_s \left[\frac{d\theta}{dT}\right]_s + 0.04\sigma A_r (T_r^4 - T_0^4) - M_r S_r \left[\frac{d\theta}{dT}\right]_r}{\sigma A_s (T_s^4 - T_0^4)} \quad (3)$$

Using equation (3), Emissivity of solkote at different temperatures was calculated. It was observed that the emissivity of solkote varied with temperature and the average emissivity was about 0.27 for the particular temperature range. This value of emissivity was used to calculate the radiative heat loss from the bare pipe.

2.5 Data analysis

Steady state is reached, when the power supplied to the cylinder equals the power lost by conduction, convection and radiation.

$$Q = Q_{cond} + Q_{conv} + Q_{rad} \quad (4)$$

Using supports of non-conducting materials in the experimental set up, the conductive heat loss is reduced to negligible levels. Thus, eq. (3.15) reduces to,

$$Q_{conv} = Q_e - Q_{rad} \quad (5)$$

Here we define modified heat transfer coefficient similar to a regular one except that temperature difference is the difference between the surface temperature of pipe and surrounding.

Q_conv=h_m A_p (T_c-T_a) (6)

Heat loss from the pipe surface by radiation is given by:

For pipe

$$Q_{rad} = \sigma\varepsilon A(T_c^4 - T_a^4) \quad (7)$$

For single glass annulus

$$Q_{rad} = \frac{\sigma A_c (T_c^4 - T_a^4)}{1/\varepsilon_c + (A_c / A_g)(1/\varepsilon_g - 1)} \quad (8)$$

Nusselt number is defined as

Nu= hD⁄k …… (9)

All the air physical properties (ρ, μ, v and κ) were evaluated at the average mean temperature.

3. RESULTS AND DISCUSSION

The objective of the experimental study was to study the heat loss from various types of receiver geometries and to find the way to minimize it. Hence, a number of experiments were performed on horizontal receiver pipe enclosed with different enclosures, i.e. bare pipe and glass tube enclosed, mirror cap enclosed and insulated mirror cap enclosed pipe. Also the correlations in literature are compared with our experimental results, assuming the temperature of the pipe to be uniform in all cases.

3.1 Bare pipe

Convective heat losses from bare pipe at different pipe temperatures and with no wind across the pipe for the receiver temperatures up to 175ºC using analytical correlations published in the literature are calculated. Table 3 shows correlations used to calculate convective heat loss from a receiver at zero wind. Using experimental data, we calculated convective heat loss for the same temperature range. Results obtained using correlations were compared with experimental results (fig 3). It is seen that there is good agreement of experimental results with the analytical correlations up to temperatures100oC above ambient temperature.

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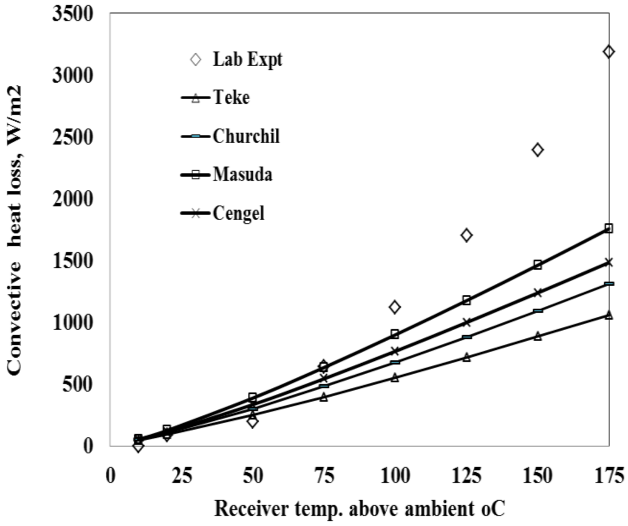

Figure3: Convective heat loss from bare receiver at zero wind

Convective heat losses from bare pipe at different pipe temperatures and with the wind across the pipe for the same receiver temperatures using analytical correlations published in the literature are calculated. Table 4 shows correlations to calculate convective heat loss from receiver due to forced convection. Using experimental data, we calculated convective heat loss for the same temperature range. The performances of bare pipe in the presence of wind were also studied at different wind speeds i.e.1 m/s, 2 m/s and 2.5 m/s. As wind velocity

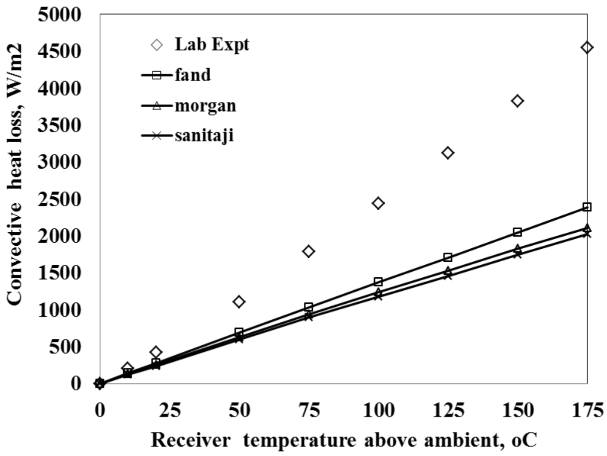

Figure4: Convective heat loss from a bare receiver at 1m/sec wind speed

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Table 3: Summary of Numerical studies on heat transfer in bare pipe

Author	Range of Pr	Range of Ra	Correlations for Nusselt number
1. (Cengel 2002)	--	Ra<=10^{12}	$Nu = 0.6 + (0.387 * Ra^{1/6})/(1+(0.559/\Pr)^{9/16})^{8/27})^2$
2. (Churchill Bernstein 1976)	0.7	10^{-6}-10^{9}	$Nu = 0.36 + 0.518Ra^{0.25}[1+(0.559/\Pr)^{9/16}]^{4/9}$
3. (Masuda 1967)	0.7	10^{-6}-10^{9}	$Nu = 0.36 + 0.048Ra^{0.125} + 0.52Ra^{0.25}$
4. (Atayılmaz and Teke 2009)	0.7	7.4×10^{1}-3.4×10^{3}	$Nu = 0.954Ra^{0.168}$

Table 4: Summary of Numerical studies on heat transfer in bare pipe due to forced convection

Author	Range of Pr	Range of Ra	Correlations
1. (R.M. Fand 1972)	0.7	10^{-2} to 10^{5}	$Nu = 0.184 + 0.324 * \mathrm{Re}\wedge 0.5 + 0.291 * \mathrm{Re}\wedge x$ $x = 0.247 + 0.0407 * \mathrm{Re}\wedge 0.168$
2. (Morgan 1990)	0.7	5×10^{3}-5×10^{4}	$Nu = 0.148 * \mathrm{Re}\wedge 0.633$
3. (Sanitjai and Goldstein 2004)	0.7	10^{3}-10^{4}	$Nu = 0.446\,\mathrm{Re}\wedge 0.5\,\Pr\wedge 0.35 + 0.528((6.5e\wedge \mathrm{Re}/5000)\wedge$ $-5 + (0.031\mathrm{Re}\wedge 0.8)\wedge -5)\wedge -1/5 * \Pr\wedge 0.42$

Table 5: Summary of correlations obtained to calculate heat loss for various geometries suggested by authors

Geometry	Correlation *
1. Bare pipe receiver	$Loss(\frac{W}{m^2}) = \left[\left(8.96dT + 0.083dT^2\right) + \left(13.12dT - 0.0151dT^2\right)v + \left(-1.423dT - 0.0041dT^2\right)v^2\right]$
2. Glass tube receiver	$Loss(\frac{W}{m^2}) = \left[\left(6.6dT + 0.033dT^2\right) + \left(5.94dT - 0.034dT^2\right)v + \left(-1.25dT + 0.009dT^2\right)v^2\right]$
3. Mirror cap receiver	$Loss(\frac{W}{m^2}) = \left[\left(7.3dT + 0.002dT^2\right) + \left(2.78dT - 0.0002dT^2\right)v + \left(-0.41dT + 0.0001dT^2\right)v^2\right]$
4. Insulated mirror cap receiver	$Loss(\frac{W}{m^2}) = \left[\left(5.04dT + 0.004dT^2\right) + \left(0.22dT - 0.001dT^2\right)v + \left(0.73dT - 0.0009dT^2\right)v^2\right]$
* Correlations valid for receiver temperatures above the ambient in the range 0 to 175°C and wind velocity between 0 to 2.5 m/sec	

increases, the heat loss from pipe also increases. Results by correlations were compared with experimental results for 1m/s wind speed which are shown in fig 4. Here, the experimental results are marked by above calculated ones. Whereas, figure 5 (a) shows that the effect of different wind speeds on the heat loss.

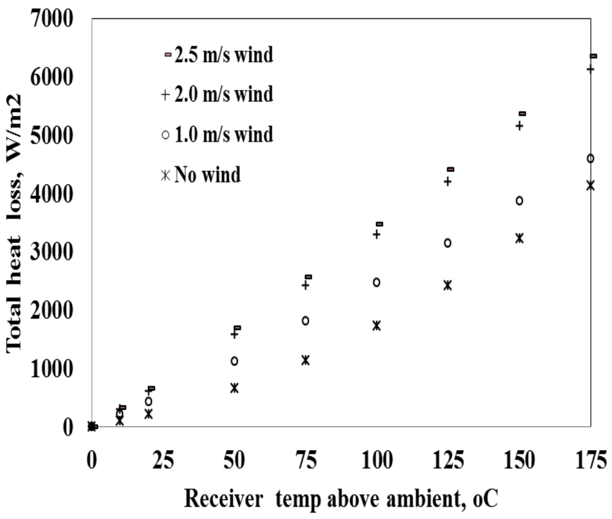

Figure5 (a): Effect of wind speed on the heat loss from bare pipe

3.2 Pipe enclosed in a glass tube

Convective heat losses were calculated from pipe enclosed in a glass tube at different pipe temperatures with and without wind for the temperatures up to 175°C. The results obtained are shown in fig 5(b). It shows that heat loss increases with increasing wind speed and also glass tube enclosed receiver geometry shows less heat loss in comparison with bare pipe

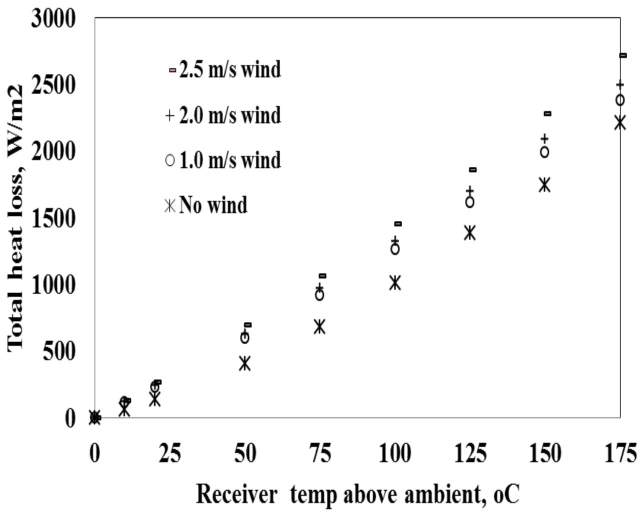

Figure5 (b): Effect of wind speed on the heat loss from pipe enclosed with a glass tube

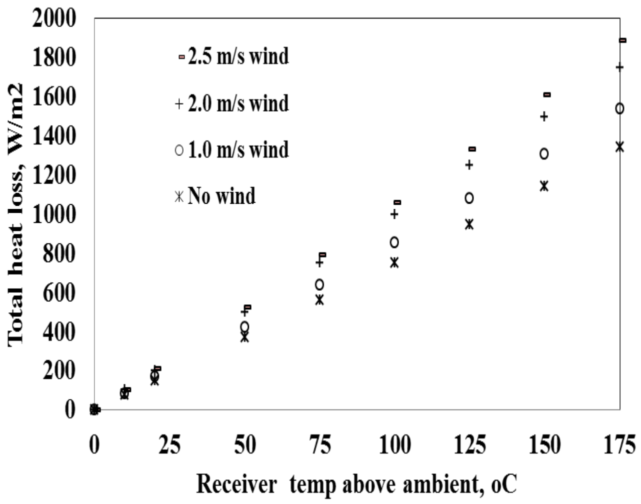

Figure5 (c): Effect of wind speed on the heat loss from pipe enclosed with mirror cap

3.3 Pipe enclosed in mirror cap without insulation

Heat losses from mirror cap enclosed receiver at different pipe temperatures with and without wind across the pipe for the receiver temperatures up to 175°C using experimental data are shown in fig 5(c). It shows that, the nature of heat loss from mirror cap enclosed receiver geometry is same as glass tube enclosed receiver but quantitatively less.

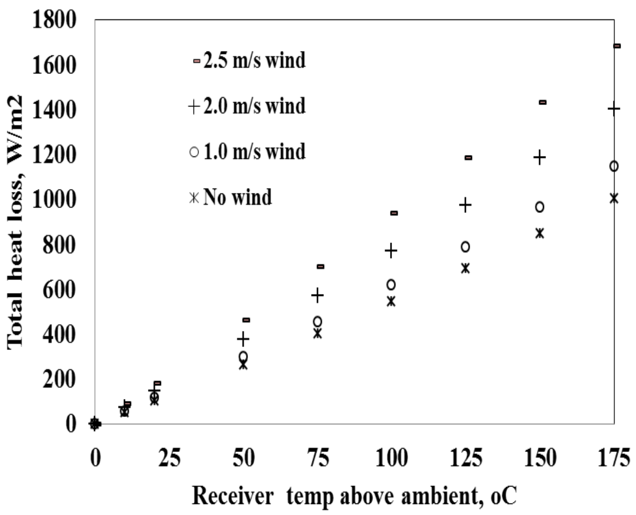

Figure5 (d): Effect of wind speed on the heat loss from pipe enclosed with insulated mirror cap

3.4 Pipe enclosed in mirror cap with insulation

Figure 5 (d) shows that reduction in heat loss from insulated mirror cap enclosed receiver at different pipe temperatures, with and without wind across the pipe for the receiver for temperatures up to 175oC. The heat loss from such a receiver drastically reduces as compared to all other geometries.

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3.5. Effect of Enclosures:

The heat loss from horizontal cylindrical pipe can be minimized by enclosing it with a glass tube, mirror cap and insulated mirror cap. The annular gap between pipe and glass tube is filled with air. Experiments were conducted for 0, 1, 2 and 2.5m/s wind speed. Fig: 6(a) and (b) shows how the heat loss increases for all geometries with wind for pipe temperature 50oC and 150ºC respectively. Fig. 7 shows heat losses at zero wind from ahorizontal receiver with various enclosures; with single glass tube, mirror cap and insulated mirror cap, whereas, fig.8 shows heat losses from the pipe for same cases at 2.5 m/sec wind speed.

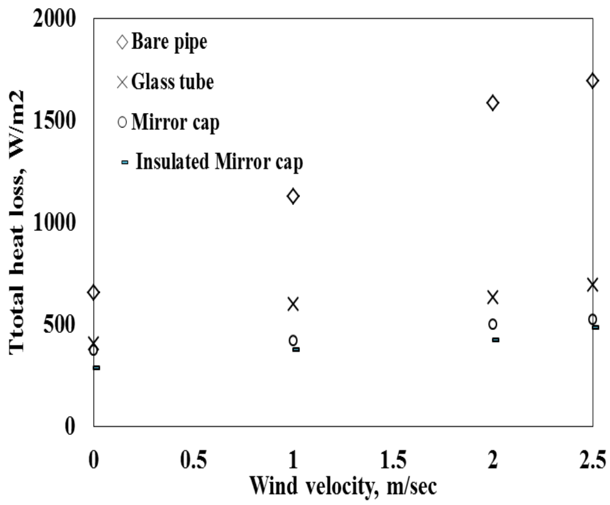

Figure 6 (a): Heat loss for wind speed and for different cases receiver temperature 50°C above ambient

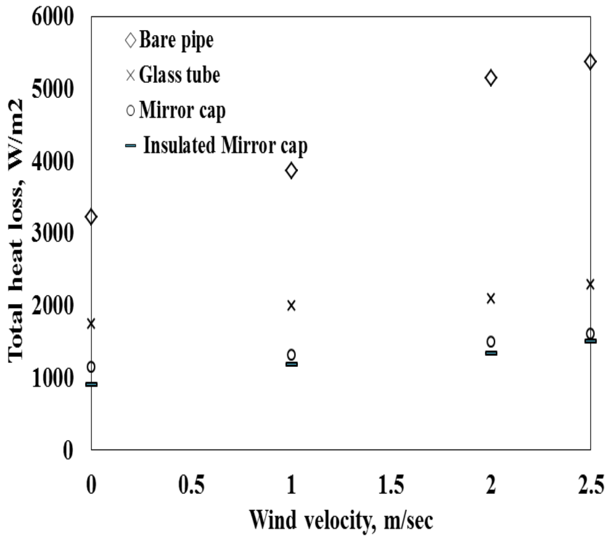

Figure 6 (b): Heat loss for wind speed and for different cases receiver temperature 150ºC above ambient

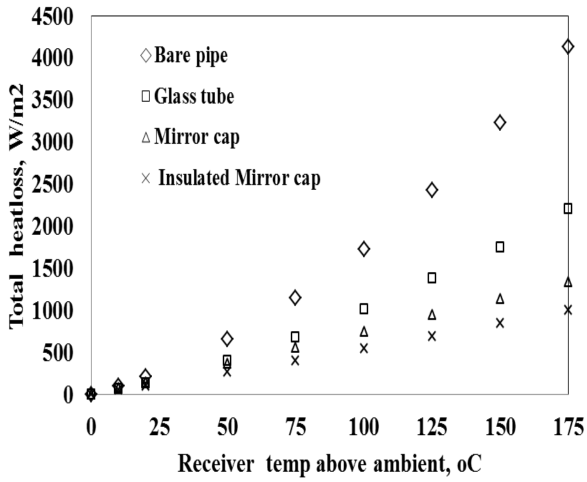

Figure 7: Zero wind Heat loss for different geometries

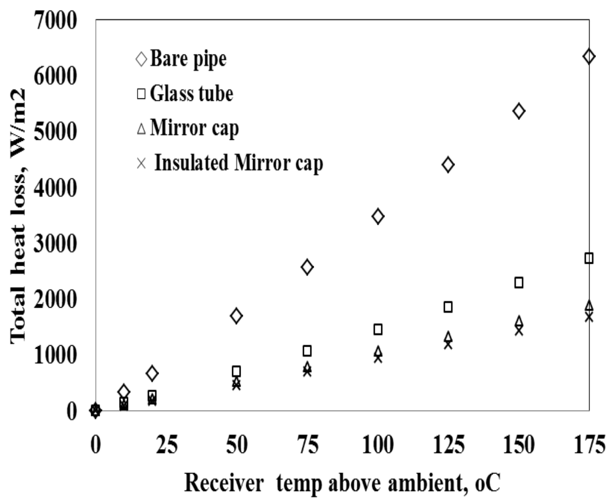

Figure 8: 2.5m/s wind Heat loss for different geometries

In all cases the heat loss from receiver increases with temperature as is expected. We calculated % of heat loss in cases of all enclosures, where heat loss from the bare pipe was treated as the reference level 100% (fig 9). In case of insulated mirror cap and at zero wind speed and at receiver temperature 100oC the heat loss is 31.4 %, whereas, for the same temperature and at 2.5 m/s wind speed it becomes 27% of the reference level. Fig 9 (d) shows that, at higher wind speed mirror cap with air in the gap and insulation in the gap shows similar results in their percentage heat losses.

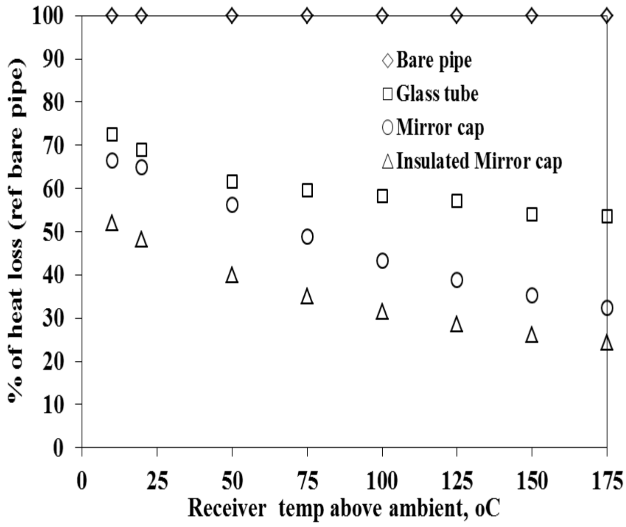

Figure 9 (a): Percentage of heat loss with reference to that from bare pipe for no wind

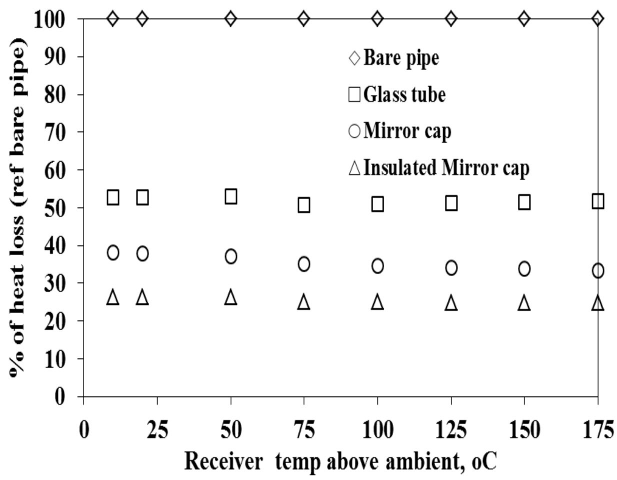

Figure 9 (b): Percentage of heat loss with reference to that from bare pipe for 1.0 m/sec wind

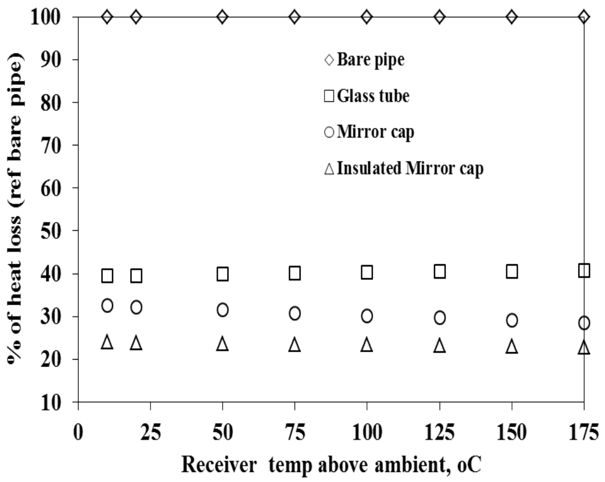

Figure 9 (c): Percentage of heat loss with reference to that from bare pipe for 2.0 m/sec wind

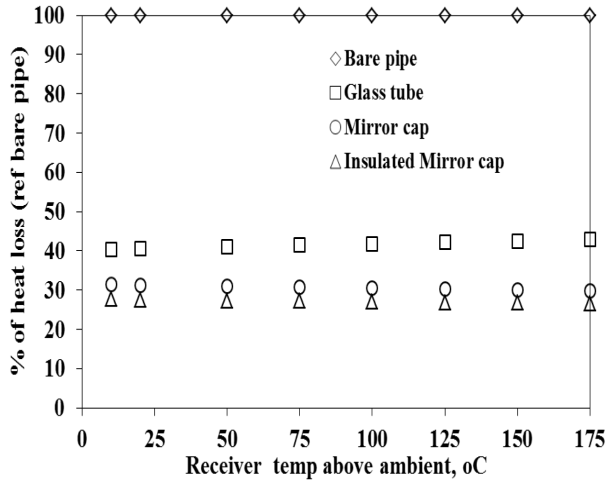

Figure 9 (d): Percentage of heat loss with reference to that from bare pipe for 2.5 m/sec wind

3.6. Correlation obtained using experimental results

By using the experimental results we developed empirical correlations to calculate heat loss at different receiver temperatures and wind speeds. Table 5 below shows correlations obtained to calculate convective heat loss from receiver at different temperatures and different wind speeds. These correlations are valid for receiver temperatures up to 200oC and wind speed up to 2.5 m/s.

4. CONCLUSIONS

An experimental and numerical study has been performed for heat losses from bare pipe, glass tube enclosed pipe, mirror cap enclosed pipe, and insulated mirror cap enclosed pipe. Experimental evidence, coupled with analytical results from simple models lead to the following conclusions.

(1) The emissivity of solkote is about 0.27 and it is nearly same as that reported (0.25) by the manufacturer (solkote).

(2) The presence of wind shows results similar to those obtained using various correlations in literature. As wind velocity increases, the heat loss from receivers increases. The nature of results obtained using correlations developed by Fand,Morgan and Sanitaji are same with the experimental results, quantitatively experimental results higher than the analytical results.

Out of all receiver geometries as bare pipe receiver, glass tube receiver, mirror cap receiver and insulated mirror cap receiver the insulated mirror cap enclosure has comparatively less heat loss. From fig: 10, it is clear that the % of heat loss from receiver pipe with insulated mirror cap enclosure is significantly reduced. Thus, an insulated mirror cap is the best option to reduce heat loss. Also, it has advantages of ease of handling and installation and cost effectiveness. Also, its chances of breakage are comparatively less as compared to glass tube enclosures or evacuated tube enclosure.

At higher temperature experimentally measured heat loss is more than that predicted by empirical relations. But it may also be noticed that at higher temperatures different relations give divergent values of heat loss. Similar results are obtained for glass tube enclosed receiver at zero wind speed.

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Nomenclature

Mr Mass of AL foil coated pipe (kg)

Sr Specific heat capacity of MS pipe (J/kg/K)

Ar Area of AL foil coated pipe

σ Stefan Boltzmann constant (6.67 x10-8 W/m2/K4)

Ms Mass of solkote pipe (kg)

As Area of solkote pipe (m2)

D Outer diameter of pipe (m)

Dgi Inner diameter of glass tube (m)

Q Heat loss (W/m2)

ε Pipe selective coating emissivity

Qcond Heat loss from pipe by conduction (W)

Qconv Heat loss from pipe by convection (W)

Qrad Heat loss from pipe by radiation (W)

hm Modified heat transfer coefficient (W/m2K)

Ac Surface area of pipe (m2)

Ta Ambient temperature

Tc Surface temperature of pipe (oC)

ϵ_g Glass tube emissivity

ϵ_c Pipe surface emissivity

Ag Surface area of glass tube (m2)

k Thermal conductivity of air (W/m-K)

Nu Nusselt Number

h Heat transfer coefficient (W/m2K)

Re Reynolds Number

Ra Rayleigh Number

Pr Prandtl Number

REFERENCES

[1] Abdul, Faik, and Wahab Hamad. 1989. "Experimental Study of Natural Convection Heat Transfer in Inclined Cylindrical Annulas." 6(5): 573–79.

[2] Atayılmaz, Ş. Özgür, and İsmail Teke. 2009. "Experimental and Numerical Study of the Natural Convection from a Heated Horizontal Cylinder." International Communications in Heat and Mass Transfer 36(7):731–38.

[3] Atayılmaz, Ş. Özgür, and İsmail Teke, 2010. "Experimental and Numerical Study of the Natural Convection from a Heated Horizontal Cylinder Wrapped with a Layer of Textile Material." International Communications in Heat and Mass Transfer 37(1): 58–67.

[4] Bhowmik, N C, Mullick S C, Hauz Khas, and The Hottel. 1985. "Calculation of Tubular Absorber Heat Loss Factor." 35 (3): 219–25.

[5] Burkholder, F, and C Kutscher. "Heat Loss Testing of Schott ' S 2008 PTR70 Parabolic Trough Receiver Heat Loss Testing of Schott ' S 2008 PTR70 Parabolic Trough Receiver." (May 2009).

[6] Churchill S W, Bernstein M.. 1976. "A Correlating Equation for Forced Convection From Gases and Liquids to a Circular Cylinder in Crossflow." J. Heat Transfer 99 (2): 300–306.

[7] Abdul, Faik, and Wahab Hamad. 1989. "Experimental Study of Natural Convection Heat Transfer in Inclined Cylindrical Annulas." 6 (5): 573–79.

[8] Atayılmaz, Ş. Özgür, and İsmail Teke. 2009. "Experimental and Numerical Study of the Natural Convection from a Heated Horizontal Cylinder." International Communications in Heat and Mass Transfer 36 (7): 731–38.

[9] Atayılmaz, Ş. Özgür, and İsmail Teke, 2010. "Experimental and Numerical Study of the Natural Convection from a Heated Horizontal Cylinder Wrapped with a Layer of Textile Material." International Communications in Heat and Mass Transfer 37 (1): 58–67.

[10] Bhowmik, N C, Mullick S C, Hauz Khas, and The Hottel. 1985. "Calculation of Tubular Absorber Heat Loss Factor." 35 (3): 219–25.

[11] Burkholder, F, and C Kutscher. "Heat Loss Testing of Schott ' S 2008 PTR70 Parabolic Trough Receiver Heat Loss Testing of Schott ' S 2008 PTR70 Parabolic Trough Receiver." (May 2009).

[12] Churchill S W, Bernstein M.. 1976. "A Correlating Equation for Forced Convection From Gases and Liquids to a Circular Cylinder in Crossflow." J. Heat Transfer 99 (2): 300–306.

[13] Cengel, Yunus A. 2002. Heat Transfer ; 2nd Edition,Tata Mcgraw Hill.

[14] Jadhav A S, et al. 2013. "Performance Analysis of a Novel and Cost Effective CPC System." Energy Conversion and Management 66: 56–65.

[15] Kitamura, K, F Kami-iwa, and T Misumi. 1999. "Heat Transfer and Fluid Flow of Natural Convection around Large Horizontal Cylinders." 42: 4093–4106.

[16] Lüpfert, Eckhard et al. 2008. "Experimental Analysis of Overall Thermal Properties of Parabolic Trough Receivers." Journal of Solar Energy Engineering 130(2): 021007.

[17] Masuda, T. Tsubouchi and H. 1967. "Heat Transfer by Natural Convection from Horizontal Cylinders at Low Rayleigh Numbers." Report of the Institute of High Speed Mechanics 19: 205–19.

[18] Misumi, Toshiyuki, Koji Suzuki, and Kenzo Kitamura. 2003. "Fluid Flow and Heat Transfer of Natural Convection around Large Horizontal Cylinders: Experiments with Air." Heat Transfer?Asian Research 32(4): 293–305.

[19] Morgan, V T. 1990. "The Forced Convective Heat Transfer from Insulated Circular Cylindrical Conductor in Air." Int. Comm. Heat Mass Transf 17: 367–76.

[20] Nada S A. 2007. "Experimental Investigation of Natural Convection Heat Transfer in Horizontal and Inclined Annular Fluid Layers." Heat and Mass Transfer 44(8): 929–36.

[21] Mullick S C, Nanda SK. 1982. "Heat Loss Factor for Linear Solar Concentrators." Applied Energy: 1–13.

[22] Patil R G., Kale D M,. Panse S V, and Joshi J B. 2014. "Numerical Study of Heat Loss from a Non-Evacuated Receiver of a Solar Collector." Energy Conversion and Management 78: 617–26.

[23] Fand R M, Keswani K K. 1972. "A Continuous Correlation Equation for Heat Transfer from Cylinders to Air in Crossflow for Reynolds Numbers from 10 −2 to 2 × 105." International Journal of Heat and Mass Transfer 15(3): 559–62.

[24] Ratzel AC, Hickox, C E, and D K Gartling. 2015. "Techniques for Reducing Thermal Conduction and Natural Convection Heat Losses in Annular Receiver." (February 1979): 108–13.