EFFECT OF NON-UNIFORM TEMPERATURE DISTRIBUTION ON SURFACE ABSORPTION RECEIVER IN PARABOLIC DISH SOLAR CONCENTRATOR

by

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A flat surface absorption receiver was experimentally investigated with a parabolic dish solar concentrator in order to study the effect of receiver temperature distribution on heat gain and losses. The addition of specially designed metal fins in the inner surface of the receiver surface side the receiver transfers the incident heat flux to heat transfer fluid. The receiver surface temperature increased with increase in concentration ratio, intensity of beam radiation, and ambient temperature, but decrease with wind speed. The absorptivity of black coated mild steel of 0.85 and also the 0.15 emissivity of mild steel reduced the heat loss from the surface and improved heat gain to heat transfer fluid. The temperature gradient between the receiver periphery and centre is around 150 °C. Fluid flow direction like straight and curved paths have been discussed for effective heat absorption and reduced operational duration. The thermal efficiency and operational duration were determined for a flow rate of 80 litres per hour through the receiver. Water flow though the curved path was observed with improved thermal efficiency of 3.8% and 20% reduction in operational duration when compared to the vertical flow through the receiver at same flow rate.

Key words: *parabolic dish collector, surface absorption, dish receiver*

Introduction

The demand for utilization of solar concentrating collectors for supplying domestic and industrial thermal needs in the temperature range of 150-350 ºC is steadily increasing due to higher cost of conventional fossil fuel energy and non-polluting renewable energy. The concentration ratio is the ratio of collector aperture to receiver aperture. Heat flux on the receiver and useful heat gain by a heat transfer fluid (HTF) in concentrating collectors relies on the geometric concentration ratio. For a particular concentration, the higher surface temperature in the receiver not only results in a higher thermal output but also increases the heat loss from the open surface of the receiver. Glass cover has been used to block direct contact of wind with the receiver surfaces and the convection losses can be reduced considerably up to 36%. There is a little penalty of 2-4% concentration with glass transitivity and use of thin glass cover compensates the transitivity issues. The heat loss is directly proportional to the temperature difference between the surface and ambient conditions whereas the heat gain is directly proportional to the receiver inner surface temperature and HTF temperature. Heat loss of a solar receiver has been studied by many researchers in the past few decades. The paraboloidal dish with receiver

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at a fixed focus was introduced by Scheffler and this technology is useful in serving domestic and industrial needs in medium temperature range. Munir *et al.* [1] described the design principle and mathematical construction calculations for parabolic curve and its elliptical frame with respect to equinox. They constructed an 8 m^2 Scheffler type solar concentrator for medium temperature applications up to 300 °C. The installed parabolic dish collector (PDC) was set as its axis of rotation is equal to the angle of latitude at the site. Fuqing *et al.* [2] analysed the use of quartz glass cover and heat loss from receiver using a two 2-D, steady-state and laminar simulation model with implicit solver adopted for combined natural convection and surface radiation in the Rayleigh number in the order of more than 10^9. The total heat flux of the receiver with quartz glass cover at 0° inclination was reduced to 36% of that for uncovered receiver. Tao *et al.* [3] analysed numerically using Monte Carlo ray tracing method and finite volume method for the coupling phase change heat transfer process of solar dish reflector and the non-uniform heat flux on tube surface which resulted in a non-uniform temperature distribution at the inner surface. The authors assumed isothermal inner and adiabatic outer surfaces. They solved the governing equations in FLUENT 6.3 using a Boussineq approximation for momentum equation. The simulation results showed that the quartz glass cover largely reduced the natural convection and surface radiation heat losses. Different receiver aperture configurations of cavity receivers were investigated numerically by Sendhil Kumar and Reddy [4] and they optimized the ratio of the inner surface area and focal image area as eight for less natural convection heat loss. The front focal area and heat loss are directly proportional and a reduction in this area results in less heat loss from the receiver in solar parabolic dish concentrator system. Ashmore and Taole [5] examined a cylindrical cavity receiver in an SK-14 parabolic dish experimental set-up with the provision of testing different receivers using energy and exergy analyses. They demonstrated a cylindrical cavity receiver with an optical efficiency of 52% and heat loss factor of the receiver 4.6 W/K. Safa *et al.* [6] experimentally evaluated the energy coming into the focus of a solar parabolic concentrator using four types of absorbers: flat plate, disk, water calorimeter, and solar heat exchanger. The concentrated solar heat flux on the four different receivers was analysed and the absorbed energy by the receiver, mean concentration ratio, energy, and exergy efficiency were determined experimentally. The energy and exergy were determined as 71% and 55% for the solar PDC.

Wang *et al.* [7] used the Monte Carlo ray tracing method to study heat distribution on the fluid media receiver for thermal performance analysis. They obtained the maximum temperature of 1372 and 1287 K for non-uniform heat flux and uniform heat flux distribution boundary conditions, respectively. Jianfeng *et al.* [8] investigated the solar receiver of a parabolic trough collector using non-uniform heat transfer model with the energy balances among HTF, glass cover, absorber, and surrounding. The non-uniformity in the parabolic trough receiver increased from 442.67 W/m^2 to 808.47 W/m^2 for increase in average absorber temperature above surroundings from 150 K to 380 K. Wang *et al.* [9] analysed numerically the complex fluid flow, heat transfer, and thermal stress in a parabolic trough solar collector with non-uniform solar flux for velocity of fluid flow, inlet temperature, and direct normal irradiance. The circumferential temperature difference from 22 K to 94 K made the thermal stress distribution in the receiver. Wang *et al.* [10] numerically analysed the use of an elliptic-circular shaped glass cover for the parabolic trough solar receiver in the place of circular glass cover and reduced the peak heat flux by 32.3%. The method of increasing allowable flux density in central receivers was discussed by Liao *et al.* [11]. Roldan and Monterreal [12] devised a model for prediction of temperature distribution on solar volumetric receiver in a solar furnace. The smaller tube diameter, thin walls and higher velocity of flow result higher heat flux on the receiver around 1 MW/m^2 in molten salt receiver than 0.34 MW/m^2 in water/steam receiver. Datle and Shinde [13] designed and tested a 16 m^2 Scheffler parabolic dish with water at 2 bar pressure. The surface temperature was observed to be around 138 °C to 235 °C with overall efficiency of 57.41%. Rupesh *et al.* [14] tested the thermal performance of an 8 m^2 Scheffler reflector using water boiling test for the solar cooking application. The cooking pot capacity of 20 litres water was tested at the focal point of the parabolic dish concentrator with overall efficiency of 21.61%. The cooking pot acted as both absorber and storage. The heat loss and performance study on the Scheffler type dishes have been investigated in the past decades and the temperature distribution on the receiver plays a vital role in the thermal conversion efficiency from concentrated solar energy to the useful heat of the working fluid. The close-range photogrammetry and ray tracing method were used to study the image of concentrated rays on the absorber surfaces. Thermal imaging camera was also used to visualize the temperature distribution on the focus of the receiver [15-17]. When an uneven temperature distribution was experienced by the receiver, it lead to higher heat losses as well as thermal stresses on the receiver material and it may reduce the effective life of the receiver considerably by creep and expansion stresses.

The method of determining heat flux and temperature distributions on the receiver and heat losses were reported in the previous literature for design of receiver for the maximum heat transfer. Rate of heat absorption is an important operating parameter in solar concentrating collectors and the non-uniformity in temperature distribution with flow path design was not reported in previous literatures. In this work, the effects of non-uniform temperature distribution of the receiver in a parabolic dish concentrator with associated heat losses have been studied and also the proposed HTF flow path in the receiver is also reported with a considerable improvement in thermal performance of the modified receiver.

Experimental work

The parabolic dish concentrator is fabricated with hardened steel in an elliptical frame with dimensions of x- and y-axis as 4 m and 6 m, respectively. Around 800 numbers of solar grade mirrors (Saint Gobain make, Metalia G031 of size 180 mm by 100 mm, thickness of 3 mm) are placed in the collector frame. The reflectivity of the mirror is 0.9 in the wavelength band 0.3 to 3μm. The focal length of the receiver from the centre of the parabolic dish reflector is 2.5 m. The diameter of the receiver is 406 mm, width of the absorber is 150 mm and the receiver is made of mild steel (Fe410, Grade IS 2052:2006, thermal conductivity of 16.27 W/mK) with a plate thickness of 5 mm and also 27.5 mm thick glass wool insulation is provided on the receiver side and back surfaces. The water flow direction is upwards, inside the receiver and a storage tank of 110 litres capacity is attached to the HTF circuit. Two receiver configurations were tested for the improvement in thermal efficiency by effectively utilizing the non-uniform temperature distribution inside the receiver. Figure 1 shows the schematic layout of two receivers without back cover in order to show the path of fluid flow. Straight path receiver was fitted with five rectangular fins having 2.5 mm thickness and height of 8 cm. Circular path receiver also fitted with similar fin dimensions spirally. Both the receivers were fitted with

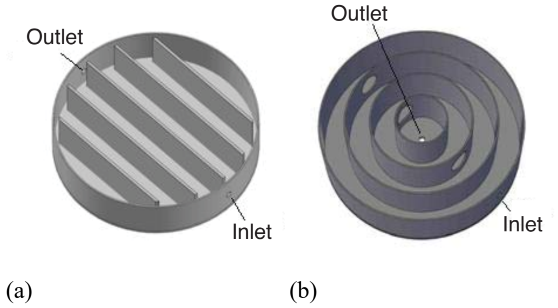

Figure 1. Schematic of receiver flow path configurations (a) straight path and (b) spiral path

inlet and outlet pipe of 12.5 mm diameter. In the straight path receiver, the HTF enters the receiver from bottom pipe and leaves the receiver at the top after passing through the linear path inside the receiver. In the spiral path receiver, the HTF enters the receiver from bottom and passes through the circular path and leaves axially from the receiver.

The geometric concentration ratio of the selected solar collector is 120 and the effective concentration ratio varies between 80 and 110 based on tracking day of the year. The incident surface is open to ambient and all the other sides are well insulated. The Sun tracking of the parabolic dish is done by Programmable Logic Controllers at two-way axis as east to west and seasonal mode to maintain fixed focus for solar declination angle. The seasonal mode is done once in three days towards zenith to south with telescopic clamp mechanism. The water is circulated to the receiver using a 0.37 kW centrifugal pump. The piping consists of 0.125 m diameter steel pipe with glass wool insulation. The fixed focal distance of the receiver is 2.5 m.

The receiver is tilted 13° towards south from the vertical axis so that the surface of the receiver to be maintained normal to the polar axis of the reflector to capture maximum concentrated energy. The piping circuit is fitted with a rotameter (0 to 150 liter per hour, ±1% accuracy) and flow control valve to adjust the fluid flow rate to the receiver. The schematic diagram of the experimental set-up is shown in fig. 2. The photographic view of the parabolic dish test facility is shown in fig. 3.

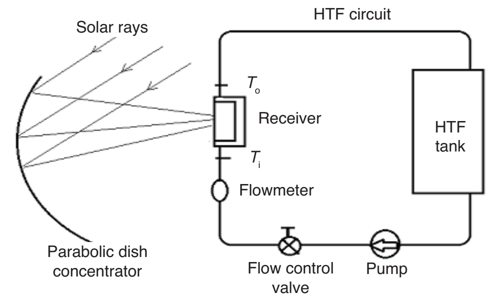

Figure 2. Schematic layout of the parabolic dish system

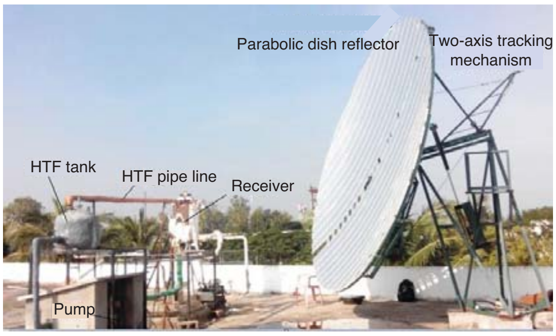

Figure 3. Experimental test facility of PDC

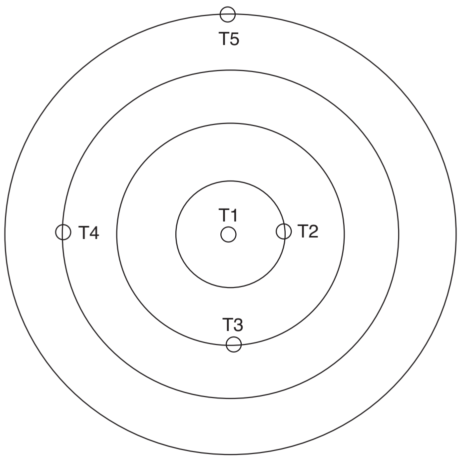

Figure 4. Thermocouple locations in the receiver

The solar radiation, wind speed, and ambient temperature were measured during the test periods at the site using Kipp and Zonen pyranometer with shaded ring (±3% accuracy), cup type anemometer (0.3 to 30 m/s, ±0.1 m/s accuracy), and K-type thermocouples (±0.5 ºC accuracy), respectively. The temperatures on the surface of the receiver were measured at five predetermined points (T1, T2, T3, T4 and T5) using thermocouples, as shown in fig. 4. The thermocouple probes were fixed in the 2.5 mm deep grooves on the inner surface of the absorber plate. The inner and outer surface temperatures of the receiver are assumed to be same that of the temperature at the mid plane of the receiving surface.

Results and discussion

The outdoor experiments were conducted on the solar receiver on sunny days in September, 2014 to examine the temperature achieved on the receiver surface with and without HTF. The receiver surface reached the highest temperature within half an hour and it mainly depends on the solar beam radiation intensity. The temperature at the centre of the receiver was higher than the extreme end of the receiver surface due to the variation in the concentration along the surface area of the receiver. The exposed surface area was subjected to wind convective and radiation heat losses. The temperature distribution observed on the receiver surface was not uniform throughout the incident surface and it was useful when HTF differential temperature is high with reduced flow passed from the lower temperature region to higher temperature regions. Temperature readings were taken during the heating experiments and the heat losses from the surface of the receiver without solar concentration on the receiver. The linear interpolation and extrapolation technique was used to calculate the temperature at the intermediate points in the x- and y-directions. A maximum temperature of 468 ºC was observed at 3:05 p. m. from fig. 5. The maximum temperature of the receiver during the testing with circulation of water through the receiver was about 260 ºC at the centre and 180 ºC at the periphery of the receiver.

The same test conducted on three more sunny days and similar trends of thermal gain were observed and the wind velocity was in the range of 0 to 2 m/s. The properties of air have been taken at the mean film temperature from eq. (1) by considering the average receiver wall temperature and ambient temperature. The natural and forced convection heat losses were calculated using Grashoff and Reynolds numbers with Nusselt correlations using eqs. (2 and 3). From eq. (4), the radiation heat loss can be determined for the average receiver wall temperature. Useful heat gained by HTF can be found using eq. (5) and thermal efficiency of the receiver can be determined during the load test with water to reach its boiling point using eq. (6).

From fig. 6, it is seen that the maximum average receiver wall temperature observed was around 350 ºC at 3.00 p. m. The wall temperature was increasing even though there was a drop in solar radiation from 2.00 p. m. onwards till 3.15 p. m. and then it decreased due to drop in solar beam radiation. The wind velocity was a second important parameter (well below 2 m/s at the site) for the receiver wall temperature and the ambient temperature was around 35 ºC during the test periods.

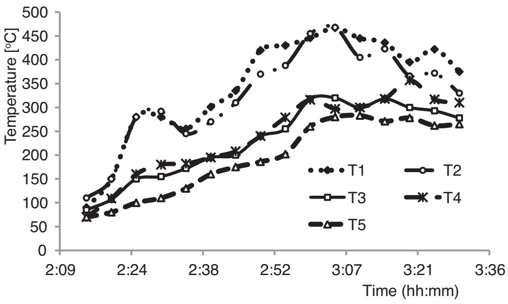

Figure 5. Variation of receiver surface temperature

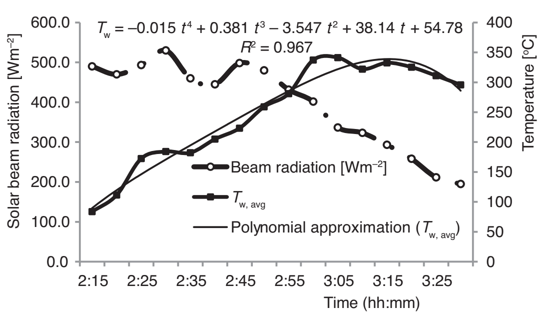

Figure 6. Effect of solar radiation on average receiver wall temperature

From fig. 7, it is seen that the maximum temperature observed at the centre and the periphery of the receiver was 450 ºC and 257 ºC, respectively**.** Zero non-dimensional radial distance is the centre of the receiver and 1 and –1 are the extreme (x or y) and (–x or –y). The same test was done on September 15th, 22nd, and 23rd, 2014, the maximum temperatures that were observed were 410 ºC, 385 ºC at the centre of the receiver and 210 ºC, 207 ºC at the periphery of the receiver, respectively.

Figure 8 shows the convection and radiation heat losses from the receiver surface against time during transient cooling. The heat losses due to natural, forced convection and radiation decreased with time. The forced convection dominated after 6 p. m. due to increase in wind speed. The average wind speed was recorded around 1.5 m/s at the site. The variation of ambient temperature was observed between 29 ºC and 35 ºC. The beam radiation intensity varied between 300 W/m^2 and 650 W/m^2 during the test periods. The temperature drop was found to be very low after 4:30 p. m. because of the low emissivity (0.2) of mild steel.

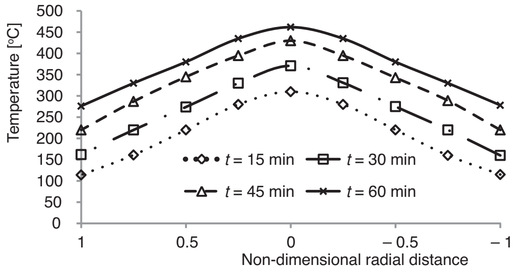

Figure 7. Variation of receiver surface temperature from the centre to periphery

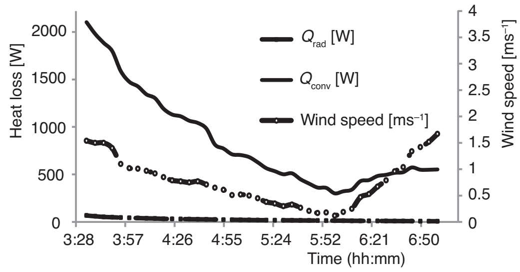

Figure 8. Convection and radiation heat losses from the receiver surface

Fluid properties are calculated at the film temperature as given in eq. (1):

$$T_f = \frac{T_\mathrm{w} + T_\mathrm{a}}{2} \tag{1}$$

The Nusselt correlation for the natural convective heat loss from the dish receiver is expressed in eq. (2):

$$\mathrm{Nu}_\mathrm{free} = 0.27\,\mathrm{Ra}^{1/4} \tag{2}$$

The Nusselt correlation for the forced convective heat loss from the dish receiver is expressed in eq. (3):

$$\mathrm{Nu}_\mathrm{forced} = 0.644\,\mathrm{Re}^{1/2}\mathrm{Pr}^{1/3} \tag{3}$$

The radiation heat loss from the receiver surface is expressed as eq. (4):

$$\mathrm{Q}_\mathrm{radiation} = \sigma A_r \varepsilon (T_\mathrm{w}^{\;4} - T_\mathrm{a}^{\;4}) \tag{4}$$

The useful heat gain by the water in the receiver is given by eq. (5):

$$Q_u = \dot{m} C_p (T_\mathrm{o} - T_\mathrm{i}) \tag{5}$$

Thermal efficiency of the receiver is defined as the ratio of useful heat gained by HTF to the incident solar energy on the PDC and is given by eq. (6):

$$\eta = \frac{Q_u}{(A_c I_b)} \tag{6}$$

Receiver wall temperature with respect to time during heating in minutes is given by eq. (7):

$$T_\mathrm{w} = 54.78 + 38.14t - 3.547t^2 + 0.381t^3 - 0.015t^4 \tag{7}$$

The rate of decrease of receiver surface temperature during transient cooling is estimated using eq. (8):

$$T_w = 265.5 - 8.147t + 1.506t^2 \tag{8}$$

The hourly total heat loss is may be calculated using eq. (9), which is a polynomial fit of the total heat loss in fig. 10:

$$Q_{loss} = 156 - 2.335t + 0.015t^2 \tag{9}$$

Figure 9 shows the hourly heat losses from the different regions of the receiver surface from 3:35 p. m. to 4:35 p. m. when the parabolic concentrator has been kept out of focus of the receiver and it depicts the comparisons of the losses at four different imaginary temperature zones, namely from the centre to 5 cm, from 5-10 cm, from 10-15 cm, and 15-20 cm. The slope was slightly steep in the first thirty minutes due to higher surface temperature and its corresponding convection and radiation heat losses. The heat loss was almost same from all the regions after an hour. The temperature was found to vary from 276 ºC to 81 ºC. The average receiver wall temperature during heating and transient cooling can be determined using eqs. (7) and (8), respectively, under the normal operating conditions. Equation (9) is the polynomial fit which can be useful to find the heat loss during the transient cooling with respect to time from its maximum temperature.

The flow regime on the flat plate collector is laminar because of its Reynolds number lies less than $5 \cdot 10^5$.

Figure 10 shows the total hourly heat loss from the receiver surface during transient cooling. The outer periphery had more heat loss due to the larger surface area, whereas the inner region at the centre had minimum heat loss due to smaller surface area primarily by convection heat transfer.

Figure 11 indicates the hourly total heat loss from the receiver surface during the test period without solar concentrated radiation.

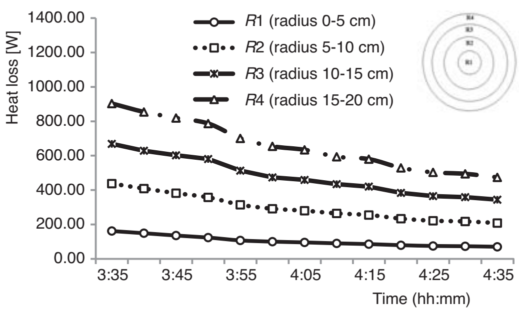

Figure 9. Hourly heat loss from different regions of the receiver

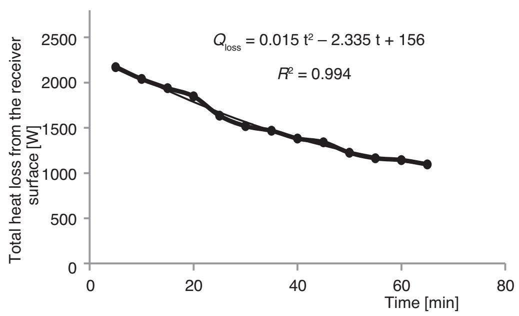

Figure 10. Total hourly heat loss from the receiver

The receiver surface was exposed to ambient conditions. The receiver surface temperature takes long time of three hours due to the hot air presence in the receiver. The maximum heat loss occurred from the outermost regions due to the more exposed surface area of the receiver even though there is more difference between receiver surface temperature and ambient temperature. The maximum and minimum of average surface temperatures were 350 °C and 210 °C without HTF during stagnation test whereas 200 °C and 145 °C with the circulation of

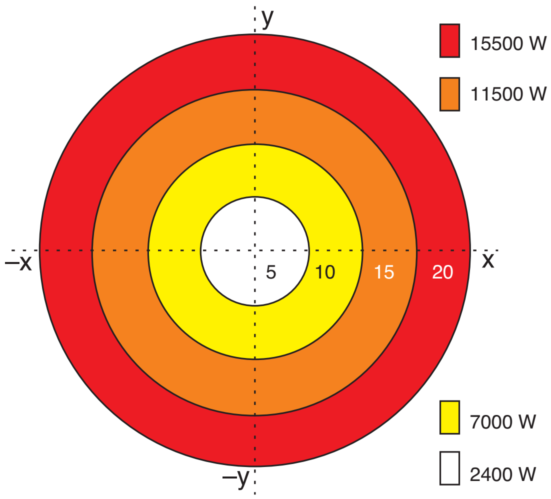

Figure 11. Hourly total heat loss from the different zones of the receiver

water, respectively. The average receiver surface temperature was around 280 °C during the test without HTF and 140 °C with HTF. The average receiver surface temperature was reduced by 100 °C due to the circulation of HTF in the receiver when compared to the receiver at stagnation conditions. The overall efficiency of the receiver with 80 liter per hour is 63.3% for the straight path receiver whereas 67.1% for the curved flow path receiver during the boiling test of water of 110 litres. The water boiling test is considered as the limiting case in this study, in order to avoid two phase flow inside the piping (steam may lead to loss in mass of water) and also to heat the same quantity of water in both receiver configurations. Operational time taken by water to reach to 100 °C from ambient temperature is 135 minutes for the straight flow path receiver and 110 minutes for the curved flow path receiver. The compared results are based on the readings during the solar beam radiation of 600 W/m^2 to 700 W/m^2 at the site. The non-uniform temperature distribution is effectively utilized in the modified receiver.

The accuracy of solar radiation measurement is ±3%, anemometer is ±0.1 m/s, and temperature measurements are ±0.5 °C. The measurement uncertainty has been calculated using root means square method as 4.7%. The uncertainty of results was found within the permissible value.

Conclusions

The non-uniform temperature distribution was observed on the receiver of the solar PDC. The temperature difference from centre ($x = 0$, $y = 0$) to periphery of the receiver ($x = 20$ cm, $y = 20$ cm) is around 150 °C during with and without circulation of water through the receiver. The receiver surface temperature is directly proportional to the solar irradiance whereas it is inversely proportional to wind speed. The major findings are as follows.

- The high heat loss occurred at the region closer to periphery due to the more exposed surface area even though the centre had higher temperatures.
- In forced circulation, the fluid flows from bottom to top of the receiver and the effective heat absorption is affected due to the fluid flow in lower temperature region at the exit of the receiver. This issue has been significantly reduced by circular path inside the receiver.
- The useful heat gain by the fluid is higher in the circular motion due to more residence time of water inside the receiver. The HTF circulated in a circular path from outer region to inner region showed 3.8% improve thermal efficiency with 20% reduction in operational duration.

The experimental results are useful for designing integrated receiver storage for PDC. The study of flow regimes with optimization of mass flow rate will be carried out in the future.

Nomenclature

A_r – surface area of the receiver, [m^2]
I_b – solar radiation, [Wm^{-2}]
lph – litres per hour, [lh^{-1}]
Q – heat, [W]

t – time, [s]
T – temperature, [ºC]

Greek symbols

ε – emissivity, [–]
σ – Stefan-Boltzmann constant, $5.67 \cdot 10^{-8}$ [$Wm^{-2}K^{-4}$]

Subscripts

a – ambient
i – inlet
loss – leakage, lost
o – outlet
w – receiver wall

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