Experimental Investigation on Heat Loss from Solar Scheffler Receiver

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Abstract— **Convection heat loss occurs in receivers of high concentrating solar concentrators, Solar Scheffler, downward focusing systems and solar towers. In most applications, it can contribute a significant fraction of total energy loss, and hence it is an important determining factor in system performance. Solar concentrators are especially located in an open environment with wind flow on the receiver surface. This wind flow has the major effect on heat loss when the flow direction is parallel to the receiver plane or at an angle. All the concentrators are needed to be track during the operation and hence the position of the receiver is changing continually. In case of the Scheffler applications Scheffler's are provided with tracking mechanism while receivers are fixed. The angle between the wind flow and the receiver axis will then play an important role in the heat loss. In this study lab experimental setup was developed to predict the heat loss from receiver with experimental simulation. To investigate convection lose from receiver, an electrically heated model receiver, was tested for different combinations of angle between the receiver and wind. Wind angle was varied from 0 deg (wind parallel to receiver surface) to 90 deg (wind perpendicular to receiver surface) with average receiver surface temperatures 100 and 110°C. It is observed that at 90 deg. Angle heat loss was maximum. Heat loss measurement was carried for three combinations of wind skirt that has an angle of 15, 30 and 45 deg. It is observed that the heat loss was minimum for 30 deg wind skirt angle. For wind angle of 67.5 and 90 deg sharp rise in heat loss was noted at higher velocities.**

Keywords—Solar Energy; Solar Concentrators, Receives; Heat Loss

I. Introduction

Energy demand is continually increasing, with electricity being the energy of choice. The electricity production, however, does not come free. There is cost associated with the infrastructure for setting up new power production facilities and the rising cost and fast depleting natural resources such as oil, coal, and natural gas. One solution is to steer away from conventional methods and look for novel, alternative, renewable, energy resources, such as solar energy. The sun is an excellent source of radiant energy, and is the world's most abundant source of energy. It emits electromagnetic radiation with an average irradiance of 1353 W/m2 on the Earth's surface [1]. The solar radiation incident on the Earth's surface is comprised of two types of radiation beam and diffuse, ranging in the wavelengths from the ultraviolet to the infrared (300 to 200 μm), which is characterized by an average solar surface temperature of approximately 6000 K [2]. The amount of this solar energy that is intercepted is 5000 times greater than the sum of all other inputs terrestrial nuclear, geothermal and gravitational energies, and lunar gravitational energy [2]. To put this into perspective, if the energy produced by 25 acres of the surface of the sun were harvested, there would be enough energy to supply the current energy demand of the world. When dealing with solar energy, there are two basic choices. The first is photovoltaics, which is direct energy conversion that converts solar radiation to electricity. The second is solar thermal in which the solar radiation is used to provide heat to a thermodynamic system, thus creating mechanical energy that can be converted to electricity. In commercially available photovoltaic systems, efficiency is of the order of 10 to 15 percent, where in a solar thermal system, efficiency is as high as 30 percent are achievable [2].

The solar collector is the key element in a solar thermal energy system. The function of the collector is quite simple; it intercepts the incoming solar radiation and converts it into a usable form of energy that can be applied to meet a specific demand, such as generation of steam from water. Concentrating solar collectors are used to achieve high temperatures and accomplish this concentration of the solar radiation by reflecting or refracting the flux incident on the aperture area (reflective surface), Aa onto a smaller absorber (receiver) area, Ar. The receiver's surface area is smaller than that of the reflective surface capturing the energy, thus allowing for the same amount of radiation that would have been spread over a few square meters to be collected and concentrated over a much smaller area, allowing for higher temperatures to be obtained. These concentrating solar collectors have the advantage of higher concentration and are capable of much greater utilization of the solar intensity at off-noon hours than other types of solar concentrators. However, one of the major problems of using a dish type parabolic collector is that two-dimensional tracking is required. Most concentrating collectors can only concentrate the beam normal, otherwise the focal region becomes scattered and off focus, therefore requiring the concentrator to follow the sun throughout the day for efficient energy collection. For the parabolic concentrator, continuous tracking is needed; if oriented east-west, the concentrator requires an approximate 30 deg/day motion and if north-south, an approximate 15 deg/hr motion. This tracking also must accommodate a ± 23:5deg/yr declination excursion [1].

Fig. 1 shows a typical set up where all radiation falling on the solar concentrator is focused on the receiver. Except the exposed face of the receiver, rest of the part of the receiver is insulated. A working fluid like water or air or thermic fluid is circulated in the receiver. Heat is conducted inside the receiver from the exposed face and transferred to the working fluid. Part of the heat is lost from the receiver especially by way of convection losses to the air. Air movement around the receiver especially over the exposed face, causes this convective heat loss. As naturally the air velocity and direction changes from time to time the heat losses from the receiver also change. Electrical measurements are accurate and precise. This is a driving force to propose one such method and experimental set up where field condition can be simulated and corresponding heat losses can be recorded. When any solar concentrator, receiver system is in operation in the field knowing the operating conditions like operating temperatures, wind velocity, wind direction and ambient temperature it is possible to calculate heat losses.[3] The aim of the work is to develop an experimental set up that will be used to simulate the receiver heat loss in the laboratory. Based on this aim following objectives are derived

1. To develop receiver Plane type and an experimental setup for determination of heat loss from solar concentrator receiver.

2. To measure the conduction and radiation heat loss from the receiver and separate it out from the total loss to calculate convection loss

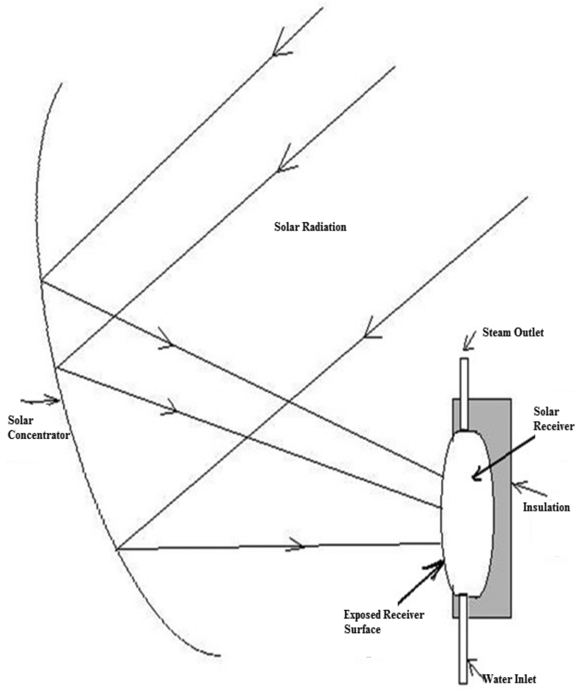


Fig. 1. Solar Concentrator

II. LITERATUR REVIEW

Investigations were made to identify the convective heat loss for receivers. Leibfried [4] reported a combined numerical and experimental study of natural convection in a side-facing open cavity. Significant variations in the local Nusselt numbers along the cavity surfaces were predicted, these variations being correlated to features of the flow field. Correlating equations for the experimentally determined surface-average and cavity-average Nusselt numbers were developed in terms of a Rayleigh number based on the height of the cavity, the Rayleigh number ranging between 3.5 x 106 and 1.2 x 109. The surface-average Nusselt number for the back wall was found to be in qualitative agreement with an existing empirical correlation for an isothermal vertical plate, but was over-predicted by the correlation. For the bottom plate, good quantitative agreement was obtained between the experimental results and an existing empirical correlation for an isothermal horizontal flat plate. Agreement between the data and the existing correlation was relatively poor for the top plate. The numerical predictions, which covered the Rayleigh number range from 103 to 107, were in good agreement with experimental data for the back plate, under-predicted the data for the bottom plate, and over-predicted the top-plate data. Clausing et al. [5] established a correlation based on the understanding of the physics of the convection heat loss associated with a large central cubical cavity. Laminar steady-state natural convection in a two-dimensional rectangular open cavity was investigated numerically [6]. Isotherms and streamline plots are obtained in a shallow open cavity with aspect ratio of 0.143 for Rayleigh numbers up to 106 using constant properties, by imposing approximate boundary conditions at the opening. This method has been tested and compared to cases where computations are carried out into an enlarged external domain. Results show that outgoing flow patterns and the heat transfer results are governed by strong characteristics of the heated cavity. These findings compare favourably with experimental results for Ra = 106

Five cavity geometries Cylindrical, Hetroconical, Spherical, Elliptical and Conical were investigated [7]. Results indicated that Variations in concentrator rim angle and cavity geometry cause large variations in the power profiles produced inside the cavity; thus, a desired power profile may be achieved without significantly reducing thermal efficiency. Skok et. al. [8] investigated numerical and experimental study of buoyancy driven flow in a side facing open cavity receivers. In this study, a two-dimensional numerical simulation was employed to predict the flow pattern in the cavity and both local and average Nusselt numbers for the surfaces of the cavity. Clausing et al. [5] developed a model to calculate the convection heat loss of a large cubical cavity based on the hypothesis that two factors governed the convection heat loss, the ability to transfer mass and energy across the aperture and the ability to heat air inside the cavity. An analytical method was developed based on the above assumption. It is concluded that the latter factor was of greatest importance. Based on the work by Clausing et al., Leibfried et. al. [4] developed a more generalized model that can be used for both downward and upward facing cavities with various geometries.

Prakash et. al. [9] studied five different types of models. The methodology employed in this analysis gives an instantaneous, steady-state system efficiency, Three days are considered, the winter solstice, the summer solstice and the equinox. The hour-by-hour solar input was estimated for site 400 N latitude. Elliptical cavity was most extensively analyzed. The thermal efficiencies of the four other cavity geometries were also examined. It was found that for the same cavity aperture and insulation thickness, cavity geometry has almost no effect on system efficiency. Stine and McDonald [11] proposed an extended correlation of the Nusselt number for a cylindrical shaped frustum receiver incorporating aperture size, surface temperature and receiver tilt angle.

Siangsukone et. al. [12] has presented work on modelling and simulation of the ANU 400 m2 Paraboloidal dish concentrator system with a direct steam generating cavity receiver and the steam line. The TRNSYS model predictions are compared with plant data measured with the ANU (Australian National University) dish. The SG3 system has been modelled using the TRNSYS program. Taumoefolau et. al. [13] has reported an experimental investigation based on an isothermal electrically heated model cavity receiver. The convection loss of the model receiver has been measured 0 deg to 90 deg Numerical analysis of the problem shows good agreement Paitoonsurikarn et. al [14] from ANU has reported a correlation model for predicting the natural convection loss from open-cavity solar receivers. Initial indications are that by incorporating and angle dependant length scale, the correlation is more accurate and general than those previously published. Paitoonsurikarn et. al. [15] had reported a new correlation based on the numerical simulation results of three different cavity geometries. Taumoefolau et. al. [13] investigated natural convection losses from cavity type receivers, an electrically heated model receiver, was tested at inclinations varying from -90 deg (cavity facing up) to +90 deg (cavity facing straight down), with test temperatures ranging from 45°Cto 65°C. Ratios of the aperture diameter to cavity diameter of 0.5, 0.6, 0.75, 0.85 and 1.0, were used. Numerical modelling of the convection losses from the cavity was carried out for positive angles with the commercial computational fluid dynamics software package, Fluent 6.0. Good agreement was found between the numerical flow patterns at the aperture region with the schlieren images and between measured and predicted values for heat loss. Kumar et. al. [16] presented a two dimensional model to estimate natural convection heat loss from modified cavity receiver (hemisphere with aperture plate) of fuzzy focal solar dish concentrator. Both insulation conditions and no insulation conditions are used for estimation of heat loss. The convection heat loss of the modified cavity receiver was estimated for an inclination of receiver from 0 deg (cavity aperture facing sideways) to 90 deg (cavity aperture facing down). From the experimental data points Nusselts number correlations were developed. The maximum convection heat loss occurs at 0 deg inclinations for both cases of the receiver, which is 63.0% with insulation and 42.9% without insulation of the total heat loss. The convection heat loss from the receiver decreases to a minimum value as its inclination increases to 90 deg. Prakash et al. [9] has reported experimental and numerical study of the steady state convection heat losses occurring from a downward facing cylindrical cavity receiver has been carried out. From all the data points Nusselt number correlations were proposed for the natural convection heat losses.

Previous discussion is limited to the natural convection and without wind situations. However, in the real applications wind effect is dominant for heat loss. Very few investigations are observed in windy situations Prakash et.al. [9] reported an experimental and numerical study of the steady state convective losses occurring from a downward facing cylindrical cavity receiver of length 0.5 m, internal diameter of 0.3 m and a wind skirt diameter of 0.5 m. The experiments are conducted for receiver inclination angles of 0 deg (sideways facing cavity), 30 deg, 45 deg, 60 deg and 90 deg (vertically downward facing receiver). The numerical study is performed using the Fluent CFD software. The experimental and the numerical convective loss estimations agree reasonably well with a maximum deviation of about 14%. Paitoonsrikaran et. al. [10] reported the studies of combined force and natural convection heat loss. Convection heat loss for three receivers with different dimensions was investigated. It was observed that the heat loss is actually reduced below the natural convection value by wind speeds up to about 7 m/s.

Many investigations are done numerically using different simulator programs for receiver heat loss prediction. Wind flow has the major effect on heat loss when the flow direction is parallel to the receiver plane or at different angle. All the concentrators are needed to be track during the operation and hence the position of the receiver is changing continually. In case of the Scheffler applications Scheffler's are provided with tracking mechanism while receivers are fixed. The angle between the wind flow and the receiver axis will then play an important role in the heat loss. In the present work a methodology is established to predict the heat loss characteristics of the receiver through electrical measurement which can be simulated to the field situations.

III. Experimental Setup and Methodology

Fig. 2 illustrates the test set up for heat loss measurements from the solar receiver. An electrical heating element is placed inside the solar receiver, to be tested. Power supply is given to this electrical heating element through a controller which can be triggered by a thermostat. An electrical watt meter is also placed in the circuit between the power supply and the electrical heating element. A thermostat is also placed inside the solar receiver. This thermostat can be set to different operating temperatures. As the temperature inside the solar receiver reaches a preset temperature, thermostat triggers the controller and power supply to the electrical heating element is cut. When the controller gets signal from the thermostat, it cuts off the power supply. If the temperature drops the preset value then electrical supply is restored. A fan is provided to blow air towards the receiver face to simulate a condition of natural wind and convective losses. As in figure fan blows air directly opposite to the receiver face. The direction is shown as. Fig. 3 illustrates that the position of fan can be varied at different preset angle. Accordingly the air blows over the exposed face of the receiver at a predetermined angle. Angle of air flow can be varied in vertical as well as horizontal plane.

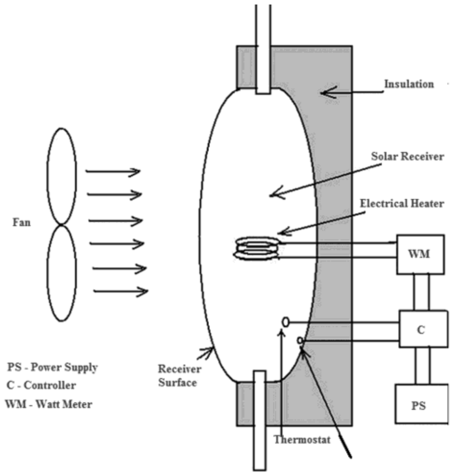


Fig. 2. Test Setup for Heat Loss Measurement from the Receiver

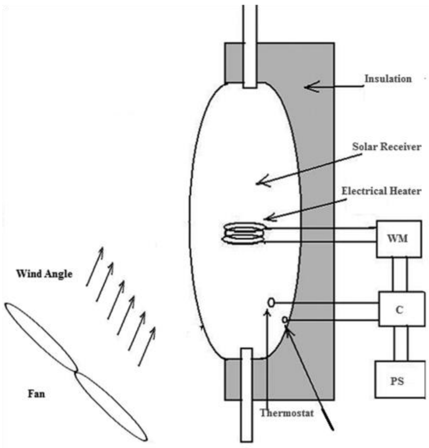


Fig. 3. Test Setup for Heat Loss Measurement from the Receiver with Fan at Angular Position

A. Experimental Methodology

An electrically heated experimental simulator of a solar receiver has been constructed to allow direct measurement of losses under laboratory conditions. The details of the arrangement in the laboratory are shown in Fig. 4 and electrical diagram is shown in Fig 5. Model receiver consists of a mild steel receiver with 0.5 m diameter. It has a mass of 26 kg and water capacity of 14 kg. It is provided with insulated electrical heater provided at the bottom pipe inlet as a source of heat input. Receiver surface has been painted with black (black board) paint. The entire structure is covered by a sheet metal casing and all internal spaces filled with wool insulation. The model receiver is attached to the left of a trolley by a welded plate. There are 7 K-type thermocouples that measure the receiver surface temperature, The temperature of the surface is controlled by a self tuned Eutech 808 PID temperature controller that regulates the power level to the heating coil. During operation, a time interval of approximately one hour is required for the system to reach steady state. Temperatures are logged once the steady state is achieved. Controller and cut-out turn ON and OFF the electrical current once the steady state is attained.

Fig. 4. Laboratory Arrangements for Heat Loss Measurement from the Receiver

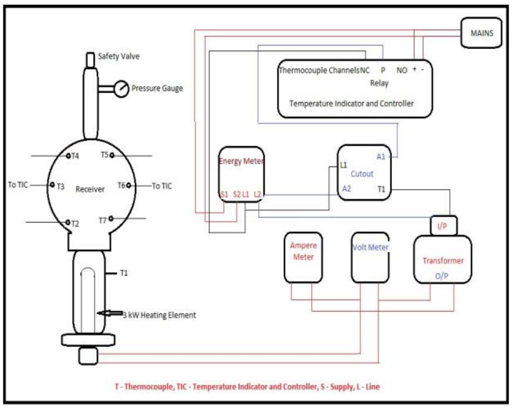


Fig. 5. Laboratory Arrangements for Heat Loss Measurement from the Receiver

Voltmeter and Current meter are used to measure the heat input to the receiver. Energy meter was used to measure the instantaneous kWh input. Auto transformer is used to vary the electrical input to measure the loss. Test is carried out in a closed room where outside wind velocities will not affect the testing and wind velocity remains a controlled parameter.

Receiver surface is provided with thermocouples as shown in Fig 5. Output of the thermocouple is supplied to the Eutech make 808 PID controller. Desired temperature or pressure is set on thermostat and power supply is put on. Initially temperature of the fluid inside the receiver increases and reaches to a preset value. At this point the thermocouple provides a signal to the controller and power supply to electrical heating element will be cut. Hereafter the system is in equilibrium. Once the equilibrium condition is reached, readings can be taken. After equilibrium is reached every time when electrical circuit is ON Voltage, Current, ON time and OFF time is recorded. Again as soon as the receiver surface is reached to the preset value controller will put OFF the circuit and again the OFF time is recorded. Every time before starting the readings, temperatures are recorded to calculate the average receiver surface temperature as an average of all. For calculation of total heat loss from the receiver surface at different wind velocities fan is ON once the equilibrium is reached. As soon as the fan is ON again the system is allowed to reach to its equilibrium. (i.e. ON and OFF of the circuit) Every time wind velocities are measured at the receiver surface at four different corners by hot wire anemometer and fan regulator is set till the average velocity is set to the required value. To ensure a constant speed of the fan it is provided with constant electrical regulated supply. With the increase in wind speed heat loss increases and that affects the ON and OFF time. For the sake of experimentation time limits setup is provided with transformer so that at high wind velocities current and voltage can increased to increase the heat input to the system.

Wind is allowed to direct over the receiver at different angles by changing the position of the receiver. Many such readings can be obtained for different combination of parameters like different operating and ambient temperatures, different wind velocities, different angle of approach of wind with reference to the exposed face of the receiver etc. Total Heat loss is calculated as

$$Q_{total} = V \times I \quad (1)$$

To determine the conduction heat loss, measurements of loss were made with a cover plug in the aperture as shown in Fig 6. It is assumed that conduction is the same for all inclination angles, with the thermal resistance associated with the boundary layer on the outside of the casing being considered negligible [13]. System is allowed to reach at the steady state by heat input to the receiver water. Once the steady state is achieved ON time and OFF time was recorded. To calculate radiation heat loss emissivity of black paint was measured from standard emissivity apparatus. Emissivity is measured as 0.91 for black paint (black board paint). This value of emissivity was also confirmed from Moghaddam [17]. Radiation heat loss is then measured by Boltzmann’s law. Force convection loss is calculated as per equation 1. Convection heat loss at different combinations of wind velocity and angle of wind with respect to receiver is then calculated.

Fan is installed with the duct directing wind towards the receiver. Wind velocities are set with different settings of step-less dimmer-stat. Angle of receiver and wind velocity are then set by turning the receiver at different angles as marked on the ground as shown Fig 7. Receiver surface temperatures are measured with calibrated K type thermocouple and calibrated temperature indicator and controller. Wind velocity was measured with hot wire anemometer. Velocity of wind was measured at four locations of the receiver surface and then considered as an average velocity on the receiver surface. ON time and OFF time are then recorded during the experimentation for calculations of the total heat loss and then convection loss is calculated after subtraction of the conduction and radiation heat loss.

It was expected that the provision of the wind skirt will reduce the heat loss from receiver surface. However, the wind-skirt angle is the governing factor for heat loss mechanism as it develops the wind eddies formed at receiver surface. Hence receiver was also tested for three different angle viz. 15 deg, 30 deg and 45 deg wind skirt angle. These angles are calculated by considering the shadow effect on the receiver surface after reflection of the beam radiation from Scheffler concentrator. Fig 8 represents the photographs of these types of wind skirt as developed.

Fig. 6. Measurement of Conduction Heat Loss from the Receiver

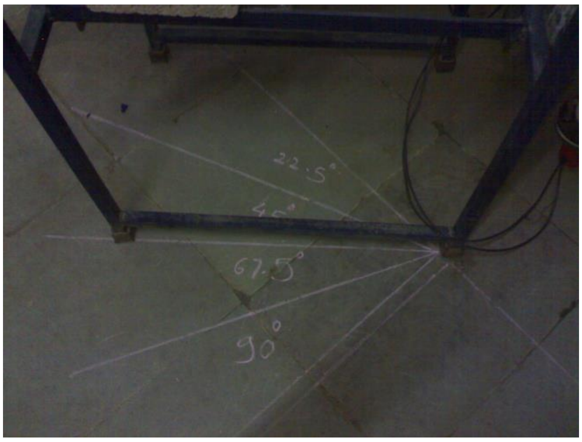


Fig. 7. Photograph for Angular Settings of the Receiver with respect to wind direction

Fig. 8. Photograph for three different wind skirts

B. Instrumentations

All the thermocouples used for experimentation are K type with a range of 0°C to 400°C. The accuracy of the measurement was ±1.5%. All the thermocouples were numbered to identify the calibration status. These are calibrated against the traceable national standard CC/ECL/0378/10-11 valid till July 14, 2012. Maximum deviation observed between UUC reading and the standard reading was ±0% with an uncertainty of the measurement ±1.3°C Controller type 808 PID was used for experimentation. It has a range of 0 to 200°C, Resolution of 0.1°C. The accuracy of the measurement was ±1.5%. All the channels for thermocouple were numbered and calibrated. These are calibrated against the traceable national standard CC/ECL/0378/10- 11 valid till July 14, 2012. Maximum deviation among all the channels UUC reading and the standard reading was ±0.25% of FSD, with an uncertainty of the measurement ±1.3

Thermal anemometer for velocity measurement Model AM4204 and Make - HTC was used for measurement of wind velocity. It was combination of hot wire and standard thermistor that delivers rapid and precise measurement even at low air velocity value. Sensor structure for air velocity is tiny glass bead thermistor and for temperature it was precision thermistor. The telescopic probe was 12 mm diameter and 940 mm maximum length. Air velocity measurements can carried out in m/s, kmph, knots. All the measurements were made in m/s with a range of 0.2 to 20 m/s, Resolution 0.1 m/s, and accuracy of reading. To vary the heat input to the electrical heater Current Transformer was used, Make ARGO, Model AS, V Range 0 to 270, I Range 0 to 20, Sr. No - 34409, Max Load 20 Amp. To measure the voltage and current voltmeter and Ampere meter was used, Volt Meter - AC, Range 0 to 260, Class1.0, Freq - 50 Hz, LC - 2 volt, IS 1248-68, Make - Automatic Electric Company Ltd. Sr. No - 11/86/2928/3, Ampere Meter - Make MECO-V, Range - 0 to 20 amp, LC - 0.2 amp, Sr. No 44787

Fig. 9. Photograph of Thermocouples and Temperature Indicator

Fig. 10. Photograph of Temperature Controller

Fig. 11. Photograph of Current Transformer, Voltmeter and Amperemeter

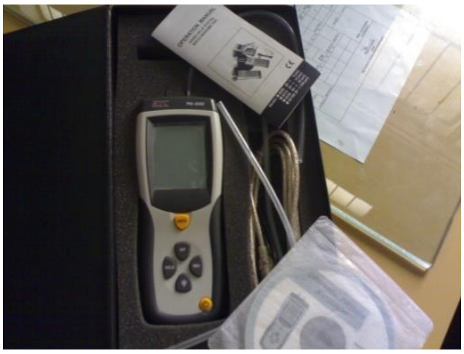


Fig. 12. Photograph of Hot Wire Annemometer

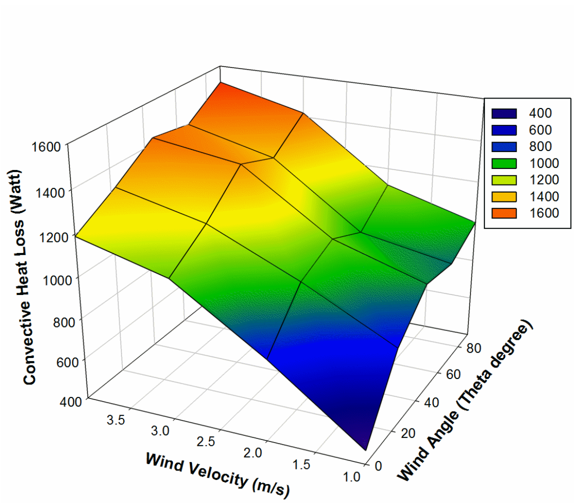


Fig. 13. Effect of wind velocity and wind angle on convective heat loss from receiver surface (no wind skirt)

IV. Test Result and Discussion

Experiments were conducted with calibrated instrumentation. Heat loss characterization was studied with laboratory method for two different cases, one without wind skirt and other with wind skirt. This topic discusses the results of two cases.

A. Heat Loss Characterization of Scheffler Receiver without Wind Skirt

Fig. 13 shows the effect on heat loss from the receiver surface for different values of wind velocity and wind angle at 110°C receiver surface temperature. Results show that heat loss from the receiver was increased with increased in velocity. With the increase in wind velocity from 1 m/s to 2 m/s loss rises by 10 to 15%. However, sharp rise was noted at higher velocities of 4 to 5 m/s. It may be due to the turbulence effect that breaks the stagnant layer of air at the receiver surface. With the increase in velocity of the air, inertia force increases that tend to increase in heat transfer coefficient and such effect is increased at rapid rate with the high velocities of the air. At 90 deg angle of wind (head on wind) large turbulence was observed that increases heat loss compared with the other angles. As the receiver was tilted to the 67.5 deg receiver heat loss is reduced. Same behaviour was observed for all the velocities of the wind. As the wind strikes the box it gets reflected and forms a stagnant zone near the receiver surface which reduces the loss. However, at 45 deg angle as wind blows over the surface of the receiver it follow the development of the continuous boundary layer over the surface and increases the convective zone that leads to increase in heat loss at these angle. With further increase in inclination receiver surface reflects the wind and only part of the receiver may be in contact with the wind velocity and stagnant layer tends to increase this leads to decrease in heat loss from the receiver surface Heat loss is minimum at 0 deg angle i.e. parallel to receiver.

From all the data points' correlation was developed using MINITAB software release 14. The developed correlation is [18]

$$Q_{total} = 47.86 V^{0.925} \theta^{0.0772} T_s^{0.632} \quad (2)$$

The R^2 value indicates that the predictors explain 88.2% of the variance in Heat Loss. The adjusted R^2 is 87.2%, which accounts for the number of predictors in the model. Both values indicate that the model fits the data well. The predicted R^2 value is 84.53%. Because the predicted R^2 value is close to the R^2 and adjusted R^2 values, the model does not appear to be over fit and has adequate predictive ability [19].

B. Heat Loss Characterization of Scheffler Receiver with Wind Skirt

Usually receivers are provided with wind skirts in order reduce the heat loss. With the provision of wind skirt part of the wind gets deflected from its edges and reduces the air movement near the receiver surface. This reduces the heat transfer coefficient and hence heat loss from the surface. However, the angle of wind skirt should be precisely selected to avoid the shading effect over the concentrator. To study the effect of this wind skirt angle over the heat loss three different arrangements were selected. The angle of these wind skirts was decided from the trigonometry and the dimensions of the concentrator as shown in the Fig. 14. Trials were conducted for the same wind speed with the same inclination and average receiver surface temperature.

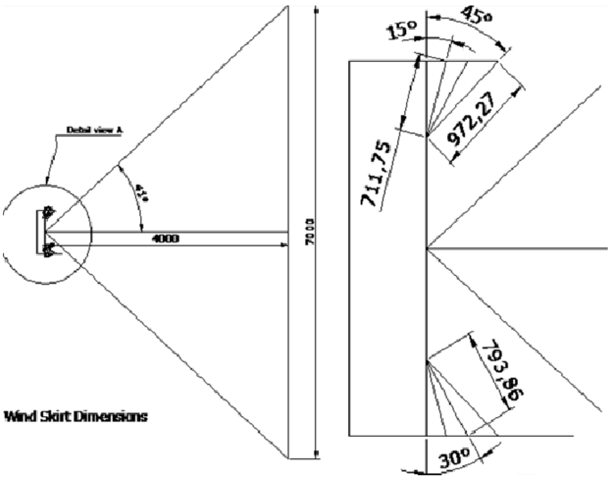


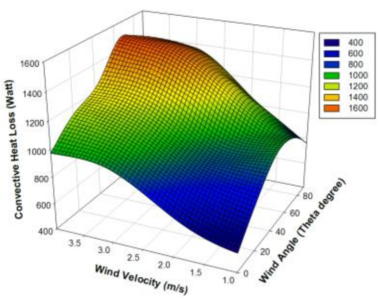
Fig. 14. Angles of Wind Skirt (Linear Dimesions are in mm)

Fig. 15 shows the effect of wind velocity, wind angle and wind skirt angle on convective heat loss from the receiver surface. Heat loss from the receiver was reduced as compared with no wind skirt. For 90 deg angle of wind flow (head on wind) it is observed that the heat loss is reduced by 23% for all wind skirt angles and it is observed that this heat loss was more or less same for all wind skirts and no any significant effect was observed. Air flows towards the skirt and it strikes at the receiver surface as well as the wind skirt surface. From the wind skirt air reflects back that may be the cause of decrease in heat loss as compared with no wind skirt. For the wind angle of 67.5 deg to 0 deg it is observed that heat loss from the receiver surface was lower for wind skirt angle of 30 deg for other two wind skirt angle. According to Clausing [5], two distinct zones are formed within the receiver as a result of the air temperatures and air flow patterns. The zone from which convective heat loss occurs is called the convective zone and the zone in which the convective losses are nullified is called the stagnation zone. Hence for 30 deg angle stagnation zone may be dominating compared with other two wind skirt. Compared with no wind skirt heat loss was reduced with wind skirt for all other angles except for 0 deg angle i.e. wind parallel to the receiver. With wind velocity of 2 m/s. at an angle 90 deg heat loss is maximum for wind skirt angle of 30 deg it indicates that stagnant zone was reduced due to the increase in velocity and formation of eddies at the receiver surface. However, for other angle heat loss again decrease for wind skirt angle of 30 deg since with the inclination stagnation zone increase. With wind velocity of 3 and 4 m/s it is observed that heat loss was minimum for wind skirt angle of 45 deg till the angle of receiver and wind varies from 90 deg to 45 deg It is also observed that the heat loss again minimum for the 30 deg wind skirt angle for wind angle of 45 deg to 0 deg This is due to the formation and breaking of the stagnant zone. For all wind skirt angles heat loss increase was approximately linear for wind flow angle of 67.5 deg. For 30 deg wind skirt angle variation in the heat loss for low wind velocity up to 2 m/s, heat loss at 45 and 67.5 deg wind angle was small. However, for wind velocity above 2 m/s heat loss is more for 67.5 deg wind flow angle. For wind skirt angle of 45 deg, heat loss variation for 90, 67.5 and 45 deg was very small for all wind velocities. For 30 and 45 deg wind skirt angle heat loss variation is small for 0 and 22.5 deg for all velocities but it increases sharply for all other wind angles.

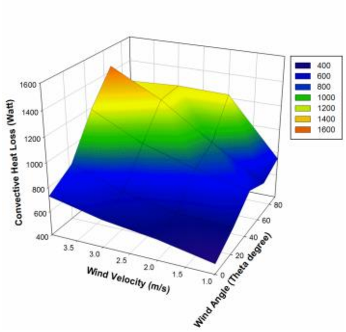
From all the data points' regression analysis was performed to obtain the Nusselt number correlation as a function of Reynold Number, Wind angle, Wind skirt angle, and non dimensional ratio (Slant length of the skirt to inner diameter of wind skirt). Correlation was developed using MINITAB software R14[18].

$$N_u = R_e^{0.32}\theta^{0.1}\left(\frac{1}{\beta}\right)^{0.5}\left(\frac{L}{D}\right)^{1.22} \quad (3)$$

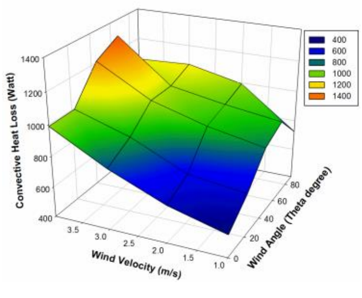
The R^2 value indicates that the predictors explain 82.6% of the variance in Heat Loss. The adjusted R^2 is 81.3%, which accounts for the number of predictors in the model. Both values indicate that the model fits the data well. The predicted R^2 value is 79.58%. Because the predicted R^2 value is close to the R^2 and adjusted R^2 values, the model do not appear to be over fit and has adequate predictive ability.[19] Uncertainty of heat loss measurement was $\pm$ 1.22% [20].



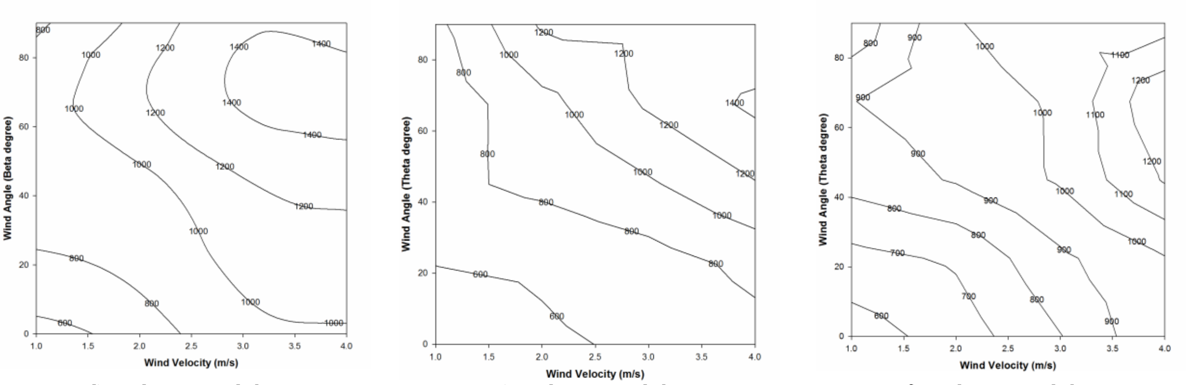
a) 15 degree wind skirt



b) 30 degree wind skirt



c) 45 degree wind skirt



d) 15 degree wind skirt e) 30 degree wind skirt f) 45 degree wind skirt

Fig. 15. Effect of Wind Velocity, Wind Angle and Wind Skirt Angle on Convective Heat Loss from Scheffler Receiver

With all the data points ANFIS (artificial neural network fuzzy inference system) prediction model was developed. Triangular membership function was used. Figure 16 and 17 shows ANFIS Structure and Training data and testing data used. Figure 18 and 19 represents scatter diagram for correlation prediction and ANFIS predicted. Error in ANFIS prediction is of the order of 11%

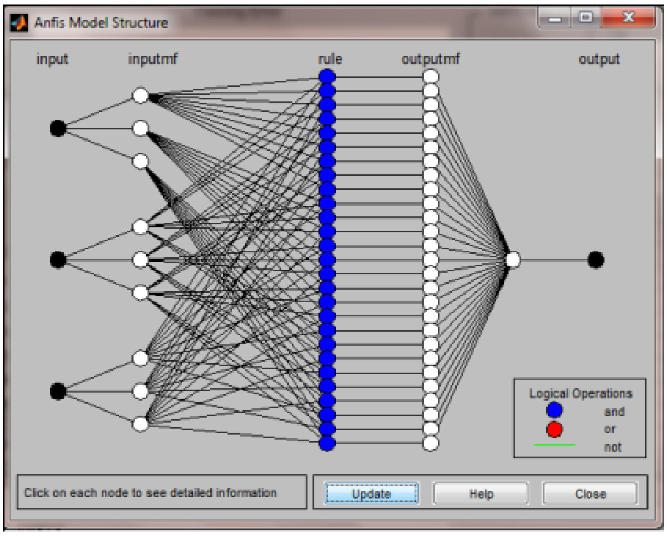


Fig. 16. ANFIS Structure

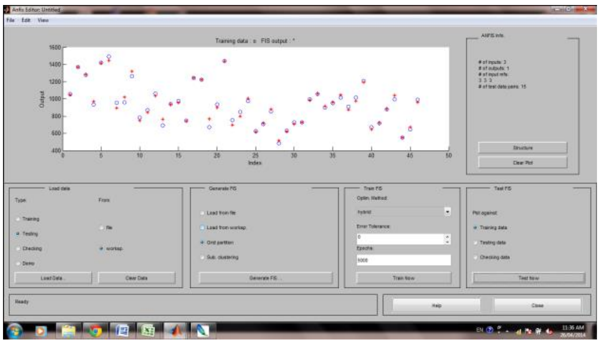


Fig. 17. Training Data and Testing Data

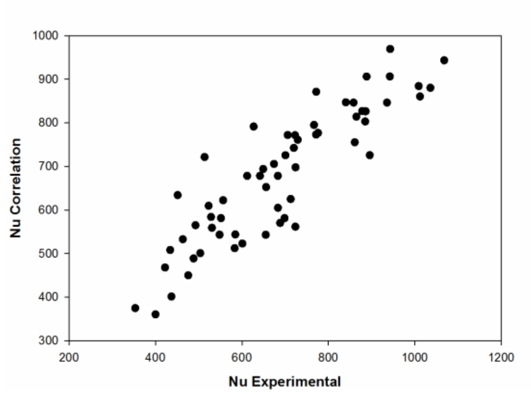


Fig. 18. Scatter Diagram for Correlation Prediction vs Experimental Results

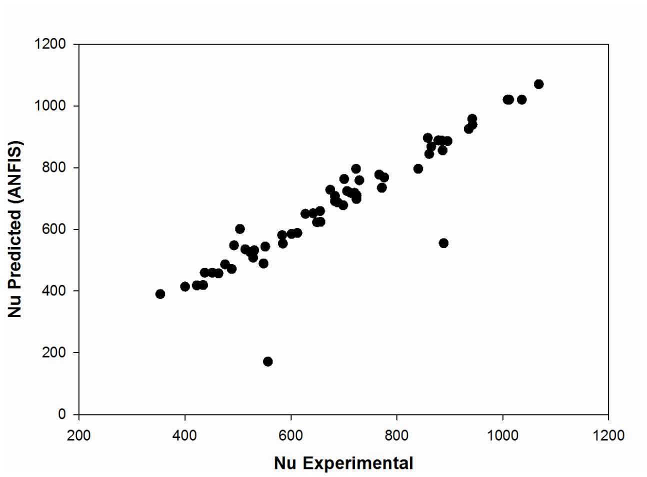


Fig. 19. Scatter Diagram for ANFIS Prediction vs Experimental Results

V. CONCLUSIONS

An experimental setup was developed to simulate the heat loss from the solar receiver. Spherical plane type receiver was used for heat loss measurement in boiling zone. Field test was conducted to compare the performance of plane and cavity type receiver. Experiments were conducted to measure experimental convective heat loss no-wind condition and for different combinations of wind velocity and receiver angle. Convective heat loss from the receiver was found to be the highest with the receiver facing wind at 90 deg and smaller for wind at 0 deg i.e. parallel to the receiver. With increase in wind velocity from 1 to 2 m/s loss rises by 10 to 15% and sharp rise was noted for the higher wind velocity. This may be due to the decrease in stagnant zone and increase in convective zone at the receiver surface. Test results with wind skirt shows that the heat loss was lower for 30 deg wind skirt angle for low velocity at all angles of the wind with respect to receiver. However, at higher velocities wind skirt angle 45 deg has a minimum heat loss for 90 to 45 deg angle of wind. For all wind skirts, the increase in velocity heat loss increases at wind flow angle of 67.5 deg. It is difficult to compare the results of the field test with the lab test. Lab test results were based on steady state measurements where as the field is at unsteady conditions. Also the wind velocity at field was not constant at the receiver surface and not a unidirectional. However, the receiver was tested at field with an average receiver surface temperature of 130°C. Wind velocity was constantly measured during the experimentation which is in the range of 4 - 5 m/s. At these field conditions heat loss was observed as 4.3 kW where as the lab test reported maximum heat loss at 4 m/s and average receiver surface temperature 110°C it is about 1.9 kW. This project was aim to develop the laboratory method that simulates the field condition so that the receivers can be tested without field setup. The project was also aim to develop the methodology that can be used in two phase heat loss characterization with electrical measurements only. Deviations are observed with field test and lab test. This is because the wind was directed with uniform velocity in lab test however; at field wind is turbulent. Similar methodology can be used to measure other different types of receiver. Methodology can be used to measure heat loss from complex shapes also.

References

[1] Sukhatme S. P., *Solar Energy*, Third Edition, TATA McGraw Hill Publication, ISBN (13): 978-0-07-026064-1, 71-107, (2000)

[2] Geyer, Michael, and Stine, William B., *Power from the Sun www.powerfromthesun.net* accessed on 7th March 2011. (2011)

[3] Chandak A. G., Somani S. K., Sardeshpande V., *Apparatus for heat loss characterization of solar ovens*, Indian Patent application no. 1607/MUM/2010 (2010)

[4] Leibfried U., Ortjohann J. Convective heat loss from upward and downward-facing cavity solar receivers: measurements and calculations. *ASME Journal of Solar Energy Engineering*, 117, 75-84 (1995)

[5] Clausing A. M. An Analysis of Convective Heat Looses from Cavity Solar Central Receivers. *Solar Energy*, 27, 295-300 (1981)

[6] Chan Y. L., Tien C. L., A numerical study of two dimensional, laminar natural convection in a shallow open cavity. *International Journal of Heat and Mass Transfer*, 28, 603-612 (1985)

[7] Harrist J. A, Lenz T G., Thermal Performance of Solar Concentrator/Cavity Receiver System. *Solar Energy*, 34 135-142 (1985)

[8] Skok H., Ramadhyani S., Schoenhals R. J. *International Journal of Heat and Fluid Flow*, 12, 36-45 (1991)

[9] Prakash M., Kedare S. B., Nayak J. K. Investigations on heat losses from a solar cavity receiver. *Solar Energy*, 83, 157-170 (2009)

[10] Paitoonsurikarn S, T. Taumoefolau, K. Lovegrove. Estimation of convection loss from paraboloidal dish cavity receivers. In: *Proceedings of 42nd Conference of the Australia and New Zealand Solar Energy Society (ANZSES)*, Perth, Australia. (2004)

[11] Stine W. B., McDonald C. G. Cavity Receiver convective heat loss. *Proceedings of the International Solar Energy Society (ISES)* Solar World Conference, Kobe, Japan (1989)

[12] Siangsukone. P. and Lovegrove K. Modeling of a steam based Paraboloidal Dish concentrator using the computer source code TRNSYS. *Proceedings of Solar 2002 - Australian and New Zealand Solar Energy Society* Paper 1 (2002)

[13] Taumoefolau T. and Lovegrove K. An Experimental Study of Natural Convection Heat Loss from a Solar Concentrator Cavity Receiver at Varying Orientation. *Proceedings of Solar 2002 - Australian and New Zealand, Solar Energy Society* Paper 1 (2002)

[14] Paitoonsurikarn S. , Lovegrove K., 2003. On the study of convection loss from open cavity receivers in solar paraboloidal dish applications. In: *Proceedings of 41st Conference of the Australia and New Zealand Solar Energy Society (ANZSES)*, Melbourne, Australia (2003)

[15] Paitoonsurikarn S. and Lovegrove K. A New Correlation for Predicting the Free Convection Loss from Solar Dish Concentrating Receivers. Clean Energy? – Can Do! - *Proceedings of Conference of the Australia and New Zealand Solar Energy Society ANZSES* (2006)

[16] Kumar N. S., Reddy K.S. Numerical investigation of natural convection heat loss in modified cavity receiver for fuzzy focal solar dish concentrator, *Solar Energy*, 81 846-855 (2007)

[17] Moghaddam S., Lawler John, McCaffery Collin, Kim Jungho. Heat Flux Based Emissivity Measurement, American Institute of Physics, Downloaded 15 April 2011 from *ip address 128.8.34.39. http://proceedings.aip.org/proceedings/cpcr.jsp* (2005)

[18] Minitab Inc. MINITAB Statistical Software, Release 14 for Windows, State College, Pennsylvania. MINITAB is a registered trademark of *Minitab Inc*. (2003)

[19] Chapra S. C., Canale R. P, *Numerical Methods for Engineers*. Tata Mc Graw Hill Publication, Third Edition, ISBN 0-07-042139-0, pp 533 - 537, (2000)

[20] Holman J. P. *Experimental Methods for Engineers*. McGraw hill - international edition, Fifth Edition, pp 40 - 49, ISBN 0 - 07 - 100501 - 3, (1989)