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EXPERIMENTAL INVESTIGATION OF THE THERMAL PERFORMANCE OF A PARABOLIC DISH CONCENTRATING SOLAR COLLECTOR

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ABSTRACT

Thermal performance of a point focus concentrating solar collector comprising a 2000 mm diameter, and a focal length of 665 mm symmetric parabolic dish concentrator covered with reflective aluminum tiles of 0.9 reflectivity and SiC honeycomb volumetric absorber, which use atmospheric air as heat transfer fluid is experimentally investigated. The absorber was tested for two different mass flow rates. This is an attempt to assess the potential of this collector as a component of solar cooker with heat storage, a prototype that has a potential to enable indirect and off-sun cooking. The prototype of a solar cooker under investigation is intended to be used in rural areas (in Mozambique) to satisfy the multiple domestic needs in thermal energy as part of a global effort to mitigate the consequences of one of the severe problems the world face today (desertification and deforestation), some of which are attributed to climate change. Thermal efficiency of the collector was estimated for the two mass flow rates. Preliminary results show that at the target temperature range the collector efficiency remained above 70 % and that the higher the mass flow rate, the lower the temperature of the air leaving the collector.

Key words: Energy; absorber; solar collector; Parabolic Dish; renewable sources; thermal performance; solar cooker.

INTRODUCTION

External energy sources are an important factor for human life and development. Sufficient access to quality energy is one of the indicators of human development.

Mozambique is ranked among the countries with lowest access of electricity per capita, with only 18 % of its population connected to national electricity grid [14]. Like many other African countries, it has the majority of its population living dispersed in rural areas and depending mainly on biomass as a source of thermal energy for their multiple domestic needs. Currently, however, the sources of biomass are becoming scarcer due to deforestation, changes in land use, desertification, etc. This reality forces people (in particular women and girls) to walk long distances in order to fetch firewood, thus pushing to the secondary plan other important activities for human development such as education. .

Luckily, the country enjoys the status of one of those possessing a considerable potential of renewable sources of energy. In particular, due to its geographical location the country has good solar radiation intensity. Therefore, solar radiation is a potential candidate to supply thermal energy to satisfy the multiple domestic needs. However, cooking is an energy intensive activity requiring medium-to-high temperatures, and thus demanding appropriate technology for harnessing and use in an efficient way.

The usefulness of solar radiation in different applications has long been recognized. But the low density at the Earth's surface makes harvesting of solar energy at high temperature difficult. In direct sun, however this is possible using solar concentrators.

Over a global growing concern, on the future of conventional energy resources, solar concentrating collectors became one of the major focuses of research endeavors mainly for large scale (industrial) applications, see [2-3] Currently, however there is also a growing concern over energy scarcity in small scale applications as the world witnesses the depletion of the major energy sources for people living in rural areas in developing countries. Thus, in some groups of interest small scale prototypes became a major focus of research aiming to assess its potential to benefit people in rural areas of developing countries, as a source of thermal energy for multiple domestic purposes (e.g. cooking, sterilization).

The potential impact of the adoption of solar cooking technologies in developing countries like Mozambique in general and in its rural areas in particular, include but not limited to more time available to women and girls to dedicate to educational activities, which clearly have a potential to bring more women empowerment, improve the quality of life and also of natural environment.

Solar cooking has witnessed improvements in both research and design and deployment. In fact, solar cookers of different configurations have been suggested and some have been applied. However, the more widespread solar cooker types allow for direct cooking in which case collection and cooking are performed simultaneously. Normally the cooking pot is placed onto focal point of the concentrator and the person performing cooking is subject to perform it under sun; facing all potential consequences arising from excessive exposition to solar radiation.

Solar cookers of the type mentioned above can only be useful while solar radiation is available, but become obsolete during off- sun periods (cloudy days and during evening). While availability of thermal energy is also important during night for cooking food or warming food already cooked during day-light, solar cookers of this type can no longer help. To overcome this limitation, a solar stove with heat storage, which has a potential to be used as cooking device, has been proposed. The prototype comprises different components: A collector subsystem (parabolic dish solar concentrator-PDSC and absorber), tracking mechanism and heat storage (HS).

The prototype being investigated has a potential to allow for indirect cooking in which case solar radiation can be collected and converted into thermal energy during day-light and stored in storage component for later use. This can be done through different configurations. For example, the absorber can be a component part of the storage in which case the storage is directly illuminated and heated and the heat is absorbed in a storage medium [12]- [13] or the absorber and heat storage are different sub-unities. It the latter case a heat carrier medium is required in order to transport thermal energy to the storage in charging mode and from the storage to the utilities during heat extraction ([6] [10] [11] [15])..

In many applications, PDSC appears coupled with Sterling engine for electricity production purposes ([8]-[20]) and some studies on application of PDSC that have thermal energy as an end product for cooking purposes used oil as heat transfer fluid ([9]-[21]). Meanwhile common applications of solar collectors using air as HTF are intended for space heating and crop drying [19]. Scarcely are found examples of successful applications of PDSC that use air as heat transfer fluid that focus on solar thermal energy use to meet cooking needs.

The performances of the components are interdependent in some way and together they determine the performance of the prototype. However, the performance of the collector subsystem is crucial and can affect the performance of the enter system. If the collector has poor performance, then the storage component becomes meaningless.

To assess its potential to be used as a component of a solar cooker intended to enable indirect cooking, in this paper, thermal performance of a PDSC coupled with Silicon Carbide (SiC) honeycomb as absorbing material and air as heat transfer fluid (HTF) is experimentally investigated. The collected heat is stored as sensible heat in a rock-bed HS. The rock-bed is a two phase system comprising a solid material and HTF. During charging mode the HTF enters the storage at high temperature and as it goes through the voids gives up heat to the rocks and emerges at the exit at low temperature. As a consequence the bed temperature rises. To enable the purpose of this paper, outdoor tests were performed at Norwegian University of Science and Technology-Department of Energy and Process Engineering (NTNU-DEPE). The present paper will not include thermal analysis on the storage, thus leaving the discussion to the next paper.

NOMENCLATURE

A_{ap}	[m^2]	Concentrator aperture area
A_{abs}	[m^2]	Absorber area
C	[..]	Geometric Concentration ratio
I_b	[W/m^2]	Beam radiation Intensity
η_o	[..]	Optical Efficiency
η_{th}	[..]	Thermal Efficiency
$\dot{Q}_u$	[W]	Usel heat rate
$\dot{Q}_l$	[W]	Heat loss rate
$\dot{Q}_{abs}$	[W]	Heat rate generated at the absorber surface
$\dot{m}$	[kg/s]	Mass flow rate
U_l	[W/m^{2}K]	Overall loss coefficient
C_P	[J/kgK]	Specific heat at constant Pressure
F	Indice	Refers to fluid
T_{out}	[K]	Temperature of the air leaving the collector
T_{av}	[K]	Average temperature at absorber surface
T_a	[K]	Ambient temperature
A	[m^2]	Cross-sectional area of the pipe
v_{av}	[m/s]	Average air velocity
$\rho(T)$	[kg/m^3]	Density as a function of Temperature
μ	[kg/m.s]	Dynamic viscosity of air
D	[m]	Hydraulic diameter
v	[m/s]	Air speed through channels

Subscripts

f	Fluid
th	Thermal
out	Leaving the system
av	Average
abs	Relative to absorber
a	Ambient
o	Optical

BACKGROUND AND GOVERNING EQUATIONS

Concentrating solar collectors are optical devices that concentrate low density solar radiation falling on their reflective surface onto a small area (absorber), thus turning the low energy density into high energy density. At the absorber surface, solar radiation is converted into thermal power and, due to high flux of solar radiation falling on it, high temperatures are produced and this effect can be used for different purposes including cooking.

Receivers are an integral part of any solar collector. In the context of PDSC applications, cavity receivers and volumetric receivers are the two types of receivers commonly used [1].

Witness of the relative importance of solar receivers, is the endeavor among researchers that have been focused on receiver modeling, not to name all, see for example [10]- [16] or through experiment ([4]- [5]-[6]).

Receivers for concentrating collectors should be designed in such a way to intercept as much reflected radiation from the concentrator as possible while at the same time ensuring that radiative heat losses from the receiver are minimized. Therefore, their size is determined by the compromise between intercepted radiation maximization, and heat loss from the receiver minimization to enable maximum power at sufficient temperatures for the intended application [16]. For high temperature applications, solar absorbers should have high temperature stability [8].

Consonant to these requirements, [4] performed experimental investigation of heat losses from a small cavity receiver, using simulated solar heat source and a Synthetic Schlieren technique for flow pattern visualization out of the cavity. In addition to this, numerical modelling of convective heat losses from the receiver through CFD analysis was carried out for positive angles.

In their turn [9] designed, fabricated, studied and compared the performance of three different receivers (Volumetric Flash-VF, Volumetric Box-VB and conical tube-CT) using a PDSC designed to charge the HS. The receivers were designed to use oil as HTF. One of their findings suggested that the CT was appropriate for the concentrator due to its high thermal efficiency compared to the others, for a given flow rate.

Using a different HTF from [9], [6] investigated the thermal performance of two similar flat solar absorbers made of different materials (stainless steel fiber wire mesh and SiC honeycomb) using air as HTF in a small scale PDSC. Their findings point out that HTF temperature and flow rates correlate negatively, while efficiency and flow rates correlate positively. In either case, the stainless steel absorber had a good performance as compared to honeycomb absorber.

Most recently, [7] investigated the thermal performance of a point-focus PDSC with cylindrical receiver intended for hot water or steam generation applications. The purpose was to find the optimal operating temperature and the impact of environmental conditions on heat loss mechanisms of the external receiver under windy conditions and in different facing directions of the blackened absorber. Their findings confirmed the well-known strong effects of absorber wall temperature, wind velocity and direction and ambient air temperature on total heat loss from the absorber.

The attainment of high temperatures is directly connected to the concentration ration of the collector. This notion relates the aperture area of a solar collector with the area of its absorber Concentrator aperture area A_{ap} is the area of the collector that intercepts solar radiation. The concentration ratio C is the ratio between aperture area of the concentrator and the absorber area, A_{abs}:

$$C = A_{ap} / A_{abs} \tag{1}$$

Not all the solar radiation incident onto the concentrator surface is intercepted and captured by the absorber. The optical efficiency η_o is the ratio of the energy absorbed by the absorber to the energy incident on the concentrator surface.

$$\eta_o = P_{abs} / (A_{ap} I_b) \tag{2}$$

Again, not all the energy absorbed at the absorber surface is converted into useful energy. The high energy density at the absorber surface gives a high temperature. Therefore, temperature gradients between absorber and ambient arise and become a driving force for heat loss; the absorber becomes hotter than the surrounding components and ambient. As such, heat losses may occur in different modes (conductive, convective and radiative).

Energy balance equation formulated at the absorber can be written as:

$$\dot{Q}_{abs} = \dot{Q}_u + \dot{Q}_l \tag{3}$$

$$\dot{Q}_u = \left(\dot{m} c_P\right)_f \left(T_{out} - T_{in}\right) \tag{4}$$

$$\dot{Q}_l = A_{abs} U_l \left(T_{av} - T_a\right) \tag{5}$$

Here U_l is the overall heat loss coefficient. Solving for $\dot{Q}_u$, the following expression is straight forward:

$$\dot{Q}_u = \dot{Q}_{abs} - \dot{Q}_l \Leftrightarrow \dot{Q}_u = \eta_o A_{ap} I_b - A_{abs} U_l \left(T_{av} - T_a\right) \tag{6}$$

To characterize thermal performance of a solar concentrator, the concept of thermal efficiency is used. This concept refers to the ratio between useful energy carried by the HTF and the energy falling onto concentrator aperture.

$$\eta_{th} = \frac{\dot{Q}_u}{A_{ap} I_b} = \frac{\left(\dot{m} c_P\right)_f \left(T_{out} - T_{in}\right)}{A_{ap} I_b} = \eta_o - \frac{A_{abs} U_l}{A_{ap} I_b}\left(T_{av} - T_a\right)$$

$$\eta_{th} = \eta_o - \frac{U_l}{\frac{A_{ap}}{A_{abs}} . I_b} . \left(T_{av} - T_a\right) \Rightarrow \eta_{th} = \eta_o - \frac{U_l}{C . I_b} . \left(T_{av} - T_a\right) \tag{7}$$

Assuming all reflected radiation reaches the absorber surface,

$$\dot{Q}_l = \eta_o A_{ap} I_b - \dot{Q}_u \Rightarrow U_l A_{abs} \left(T_{av} - T_a\right) = \eta_o A_{ap} I_b - \dot{Q}_u$$

This energy balance enables the computation of U_l as a function of absorber temperature:

$$U_l = \frac{\eta_o A_{ap} I_b - \dot{Q}_u}{A_{abs} \left(T_{av} - T_a\right)} \tag{8}$$

METHODOLOGY

For the test to take place, a SiC honeycomb absorber 105 x105x105 mm^3 from the Chinese Company Pingxiang Zhongying Packing Co. LTD was delivered at DEPE-NTNU.

A cubic receiver housing with truncated cone terminal was prepared in Department's Lab and the Sic was inserted within the housing, see Fig.1.In [3], advantages of SiC honeycomb absorber that favor its application in solar industry are outlined; not to name all, high temperature tolerance, thermal shock resistant and also mechanical strength are some of them.

The receiver was then mounted onto parabolic dish concentrator through a support structure that enabled it to have the lower surface to be located at the focal region of the concentrator, see Fig.2.

Then the receiver was connected to an air carrying pipe comprising stiff pipe which in turn was connected to the flexible pipe (made of 3 pieces of corrugated steel flexible pipe connected one after the other). The pipe was intended to carry the heated air from the receiver to the storage.

To monitor temperatures at the irradiated surface of the absorber and also temperature of the air leaving the absorber, 3 to 4 thermocouples were used. One at the absorber surface, 3 at different positions along the air carrying pipe of which the first was positioned 5 cm above the upper surface of the absorber.

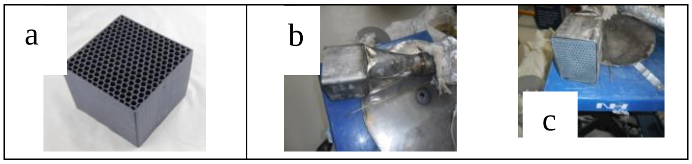


Figure 1 shows: a) SiC honeycomb absorber; b) Absorber housing and c) The receiver

Then absorber housing was covered with one layer of 5 mm Aerogel insulation, followed by a cover of two layers of reflective aluminum film on top. The geometric concentration ratio of the collector is 272.

Property	Values	Property	Values
Nr channels	17x14	Side wall thickness	1.3 mm
Channel dimension	7.5 mm	Specific surface area	366 m^2/m^3
Wall thickness	1.2 mm	Void (%)	57 %
Specific heat	750 J/kg.k	Thermal Conductivity	50-250 W/m.k

Table.1. SiC honeycomb properties

Figure 2 .Shows receiver mounted on the setup (left) and irradiated receiver during tests (right).

The forced convection is generated by a fan with speed controller-P.Lemmens air movement Company Type ESB3 Rev04-010009 connected to the HS outlet**.** The air speed is measured at fan outlet which has the same diameter as the air carrying pipe. Assuming no air leakage along the pipe and in steady state the air speed at the fan outlet can be used to calculate the air speed along the carrying pipe.

The average air mass flow rate is estimated through the expression: $\dot{m} = A v_{av} \rho_{av}(T)$ (9)

In this expression A is the cross sectional area of the pipe, v_{av}, the average air speed (air speeds were measured each 30 min, using a handheld W_M Digital Anemometer DA 4000 and from the data the average was calculated); $\rho_{av}(T)$ the average air density as a function of temperature (taken from tables of thermophysical properties of air).

The mentioned set of thermocouples was then connected to a data logger which in turn is interfaced to a PC through a labview program to log temperatures each second.

The incident beam flux was measured using a pyrheliometer at meteorological station located at the roof of Physics Department building located at the same Campus as the test location.

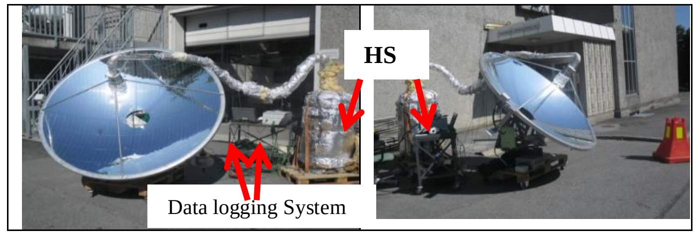


Fig.3. shows the setup during the experiment.

In the first experiment, run under average air mass flow rate of 4.4 g/s, only 3 thermocouples were used to monitor temperature of air along the pipe. The thermocouple located at the surface of the absorber had a cupper support to help fix it at the central honeycomb absorber channel. However, due to high flux density the metal was melt and the sensor did not give the readings. So, T_1 is the first thermocouple measuring the temperature of air leaving the collector.

In the second test, run under average air mass flow rate of 5.52 g/s, another thermocouple was positioned at the absorber surface. Therefore, T_1 represents reading from that thermocouple, while T_2 is the temperature of air leaving the collector given by the first of 3 thermocouples along the air carrying pipe.

The pressure drop is an important parameter for the collector performance as it is a source for an informed choice of pumping power. With Reynolds number estimated at 725.5, through the small channels of the absorber the flow regime is laminar and the pressure drop can be calculated as in [18] by :

$$\Delta p = (32.\mu.v.\Delta x)/D^2 \qquad (10)$$

The power required can be calculated by:

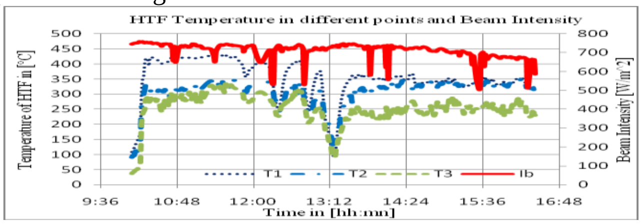
$$P = \dot{m}.\Delta p / \rho(T) \qquad (11)$$

RESULTS AND DISCUSSION

Figure.4 shows temperature of air leaving the collector at different points along the pipe and beam intensity at the first test.

As can be seen from fig.4, the air leaves the collector at temperatures above 350 °C during the experiment; with even high temperature between 10:19 and 11:34 probably due to high beam radiation intensity registered during this period. The sharp drops in air temperature encounter justification on weather conditions, shading of sun by clouds. This factor can be witnessed by the curve of beam radiation. Other factors are radiation loss due to imperfect focussing and increased convective heat losses. In most of the cases, the drop in air temperature is not simultaneous with drop in beam radiation intensity. Due to thermal inertia, when beam radiation drops due to cloud, the absorber is still hot and hot air is pumped in; only after some time it also drops. This effect is also observed on the efficiency curve in fig.6, which is characterized by sharp piques.

Figure 5 compares power input to the collector during the experiment and useful heat carried by air. From this chart, the effect of thermal inertia phenomena can also be noted, in which case useful heat seems to pique when power input is low. Generally, it can be seen that the air was able to carry out most of energy provided. This fact is attested by the efficiency curves in fig.6. Variation of the curves can also be attributed to changing wind speed and loss of radiation due to periodically imperfect focusing.



.Fig.4. Shows temperature of air leaving the collector at different points along the pipe and beam intensity at the first test.

In fig.7, behaviour of beam radiation intensity and temperatures of the air along the air carrying pipe (T_2 to T_4) during second test are compared. The experiment was run under an average air mass flow rate of 5.42 g/s. Thus, the effect of an increased air mass flow rate on the temperature of air leaving the collector can be noted here. This can be easily seen by the fact that although beam radiation intensity was slightly lower in the first experiment as compared to the same quantity in the second experiment, air temperature is lower in the second experiment.

The solar heat is absorbed on the receiver surface, pass into the material by conduction and is then transferred to the air flow. Thus, the temperature of the surface is considerably higher than the air exit temperature, dependent on the conductivity of the absorber material and the heat transfer process. A simplified model for heat transfer will be discussed. As the radiation distribution on the receiver was not monitored, constant radiation intensity on the top was assumed. The flow through the receiver is laminar by a clear margin. This means that the heat transfer coefficient h is independent of channel air speed and that the pressure drop is given by equation 10. The width between parallel hexagonal channel sides is 5.5 mm and Δx is length. Since μ varies approximately as $T^{2/3}$ in the actual region, v will be depressed in the hot, central region, and mass flow is reduced because of the 1/T temperature dependence. Temperature differences may thus be increased during the air heating process. The temperature differences will to a certain degree be reduced by sideways heat conduction in the material, but due to the hexagonal structure, sideways conductivity is about half of the longitudinal one.

Heat transfer SiC – air is calculated assuming the same flow and temperature in all channels. With T_s & T_a representing absolute temperatures in SiC & air; Q_s heat current in SiC, Q_{s0} front surface value, the equations governing the heat transfer with the assumptions mentioned above are given in [17]

(12) $Q_s = \Lambda . T_s'$; Λ is longitudinal heat conductivity of absorber material times its total cross section. The apostrophe indicate derivation with respect to distance x from receiver surface.

(13) $Q_s' = H.(T_s - T_a)$; $H = h.L$, h is heat transfer coefficient and L total inside wall length of channels.

(14) $Q_{s0} - Q_s = (T_a - T_{amb}).J$; T_{amb} is ambient temperature, J is air mass flow times its heat capacity. By derivations and substitutions these equations can be converted to an equation in the variable part of T_s: $Y'' + a.Y' - b.Y = 0$,

$$Y = T_s - T_{amb} - Q_{s0} / J \qquad a = H / J \qquad b = H / \Lambda$$

(15)

Assuming constant coefficients, the acceptable solution of this equation is $Y = Y_0 . \exp(-\alpha . x)$, $\alpha = \left(\sqrt{a^2 + 4b} + a\right) / 2$

(16)

A solution to the model [17] indicated in eqs. (12 to.16) is shown in Fig. 8. As discussed above, the model is based on an homogeneous heating on the top and the same air flow through all channels. The first condition is not satisfied giving also unequal flow, amplified by the fact that the viscosity of air increases with temperature, giving higher flow through the cooler parts. SiC thermal resistance is temperature dependent, and the curves in Fig. 8 are calculated using data for siliconized SiC given in [22]; 40 W/ K·m at 25 °C, 15 W/Km at 630 °C. Using standard models as given by e.g. [18]., to calculate the flow, a median value of α = 52.7 was found. The linear model (eq. 16) did not deviate very much from a numerical solution with variable parameters. A central surface temperature of 1000 °C is observed in the 2nd test. The difference from the model could easily be due to peaked irradiation giving a higher central flow than assumed in the model. Using a material with high thermal conductivity will reduce receiver surface temperature. We see that the depth of the receiver is suitable; it has wider channels than much of the conventional materials. The above discussion shows that the absorber should be as narrow as possible, still catching most of the incoming radiation. It should be made of a material with very high heat conductivity to allow transversal temperature equalization and avoid cool air and easy flow in the peripheral channels to mix with the hotter air from the inner ones. Good heat conductivity will also reduce the front temperature. A model should be made to optimize the receiver, both with respect to pumping power and heat loss.

The shape of the present receiver allows easy flow of cool air in the outer parts, and reduces mass flow of hot air in the central part, mixing at the output. This gives higher heat loss and increased pumping power for a given flow and temperature of air output.

It should be noted that for the experiments performed under the two different air mass flow rates, the transient behaviour of thermal efficiency indicate a fairly good performance in both cases with the average thermal efficiency around 70 %. Thus, the efficiencies are about the same, which enables the user to choose the relevant temperature optimal for the application by selecting the air flow from the heat conservation. This evidence suggests that, Honeycomb SiC absorber, despite the economic concerns around it as compared to some of its competitor materials, holds a potential to be used as an absorber for PDSC in small scale applications like solar cooker with HS.

The average loss coefficient was found to be around $51.22 \ W/m^2K$. The average pressure drop through the receiver was found to be around 330.2 Pa for the first test which implies a fan power of 4.416 W.

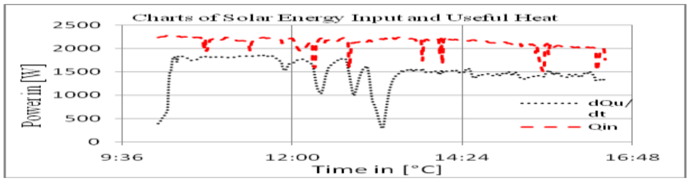


Fig.5. Shows transient behaviours of energy input to a concentrator and useful heat carried by HTF.

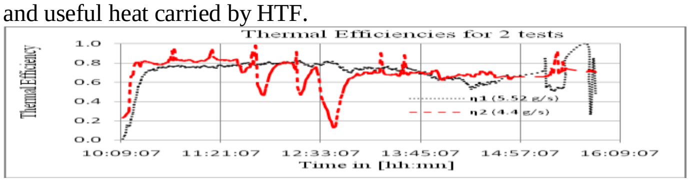


Fig.6. Shows thermal efficiencies behaviours for 2 experiments.

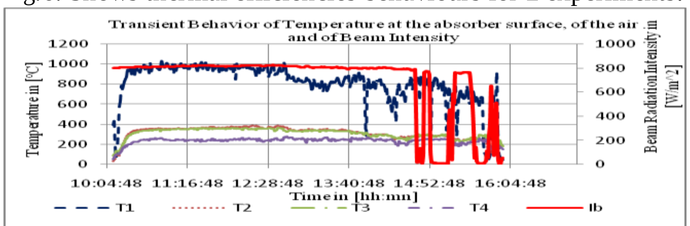


Fig.7. Temperature profiles and Beam radiation at 2nd test

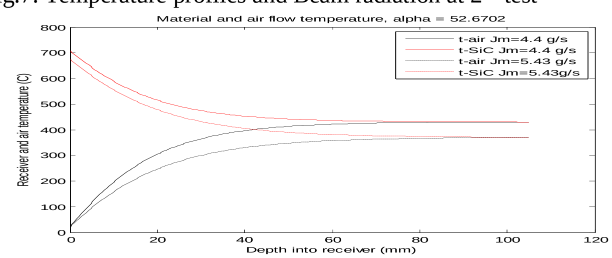


Fig.8 .Modelled temperature dependence in air and receiver material for the two test cases assuming homogeneous flow.

CONCLUSION

Thermal performance of a point focus concentrating solar collector comprising a 2000 mm diameter parabolic dish concentrator covered with reflective aluminum tiles of 0.9 reflectivity and a focal length of 665 mm and SIC honeycomb volumetric absorber, which use atmospheric air as HTF is experimentally investigated. The absorber was tested for two different mass flow rates in order to evaluate its potential as absorber material for solar cooker prototype with HS. The results indicate that the increase in flow rate leads to decrease in the temperature of HTF. Besides, the two flow rates gave a considerable good collector thermal efficiency of around 0.7. The results of this study, concerning both high temperatures and efficiency show that an air based solar cooker is possible. However, further tests are required to assure this possibility.

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