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EXPERIMENTAL EVALUATION OF HEAT LOSS FROM A MULTI-TUBE TRAPEZOIDAL CAVITY RECEIVER FOR A LINEAR FRESNEL SOLAR CONCENTRATOR CONSIDERING NATURAL AND FORCED CONVECTION CONDITIONS

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Abstract. *The minimization of heat losses if of primary importance upon designing a LFC's absorber. Additionally, the heat losses may be influenced by several parameters such as cavity geometry, materials, coating, temperature and ambient conditions. In this work, an indoors multi-tube trapezoidal cavity setup was experimentally studied. Heat losses were evaluated for natural and forced convection conditions. Also, the presence of a window glass, to minimize convection heat losses, was studied. The results showed a significant difference in heat losses considering the window glass and also that heat losses increases along wind speed.*

Keywords: *Solar Energy, Linear Fresnel Concentrator, Experimental Tests.*

1. INTRODUCTION

The industrial process sector uses more energy than any other sector in the planet. Just by itself, it is responsible for about 54% of the world's total delivered energy, (EIA, 2016). Most of this energy is consumed either in electrical or thermal energy forms. While electrical energy is used for air conditioning and operating motors, thermal energy is used in process heating applications, such as drying, cooking, bleaching, etc. (Sharma et al, 2017). Although the contribution of solar thermal heating to the industrial sector is still small, it is expected to increase in the following years. Moreover, the cost for collectors has decreased 50% in the past 20 years (IEA, 2017).

The linear Fresnel solar concentration can be used to attain fluid at medium temperatures (100-250ºC) (Singh et al, 2010; Iparraguirre et al, 2016). It is eminent that process heating is demanded for a wide variety of temperature ranges, varying along with different industrial processes. However, the majority of processes in areas such as pulp and paper, food processing, chemicals and pharmaceutical, automobile and other include medium temperature processes (Sharma et al, 2017).

A linear Fresnel solar concentrator prototype was developed as part of the Heliotermica project at the LEPTEN laboratory. The project includes the conception, thermal optimization and construction of an outdoors functional linear Fresnel workbench.

The absorber is probably the most crucial element on the LFC and should be thought carefully. A proper evaluation of the thermal mechanisms occurring in the LFC's absorber includes the entrant radiative heat, coming from the sun and entering the cavity's aperture and also the heat lost from several mechanisms in the absorber.

Additionally, the outdoors' setup heat loss evaluation is vulnerable to several environmental parameters such as temperature, solar irradiation, wind conditions, etc. Thus, in order to predict heat losses on the LFC under real conditions, tests performed indoors tend to be more reliable. A shorter replication of the actual outdoors absorber was built and simpler tests were performed in a controlled environment (indoors).

This study aims to evaluate the thermal losses occurring on an indoors multi-tube trapezoidal cavity absorber for natural and forced convection conditions. Tests were also performed with and without a glass window placed in the bottom of the cavity. The objective of the window glass is to create a greenhouse effect, reducing convective heat losses to environment. The overall heat loss coefficients were obtained for each configuration and plotted against the temperature difference with the environment. Also, a tendency curve for heat losses versus wind speed was outlined. The experimental setup and experimental procedures are described herein.

2. EXPERIMENTAL PROCEDURE

2.1 Experimental setup description and overall heat loss coefficient

The experimental setup consisted of a multi-tube trapezoidal cavity absorber module installed on a controlled indoors environment. For instance, no solar radiation is incident over the cavity. Additionally, the heat loss tests were performed by heating the tubes from inside with a constant power, and when the system reached steady state condition the data was collected.

A schematic of the experimental setup is shown by Figure 1. It is consisted of the multi-tube trapezoidal cavity absorber module, a varivolt transformer, T and K type thermocouples, AC watt transducer and an Agilent data reduction system. Each tube inside the absorber held a 96 Ω (±2%) cylindrical thermal resistance as a hating source. Also, a 55 W capacity, axial fan was used to flow wind along with the absorber. The varivolt transformer input was 220 Vac and the output ranged between 0-250 Vac. It could supply a maximum power of 10 kW.

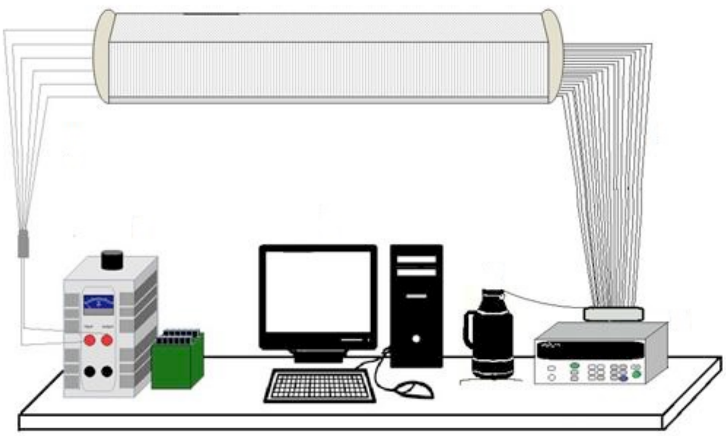

Figure 1: Schematic of the Experimental Setup

The absorber module was kept suspended with a 1.5 m distance between the aperture and surrounding objects, to avoid radiative interactions between the module and environment. This module of the receiver is only shorter lengthwise and all other dimensions are the same as those used in the linear Fresnel prototype. The cavity is built using two folded aluminum sheet of 2 mm thickness. The spacing between the aluminum sheets is filled with rock wool. Six stainless steel tubes, each with an outer diameter of 25.4 mm, were used as the absorber elements and they were fixed inside the trapezoidal cavity. Tests were performed with and without a glass window on the cavity's aperture. The presence of this glass creates a greenhouse effect. In this case, heat is coming from the top part of the cavity's mouth - where the tubes are allocated - and the heat is lost by radiation from tubes to the glass and cavity walls, and also by diffusion occurring on the trapped air molecules inside the cavity. Another considerable heat loss mechanism occurring on the cavity is conduction, which takes place along the aluminum sheets and through the rock wool. The heat reaching the outer part of the cavity is lost by radiation and convection to the environment (Dey, 2004). The setup's operational temperature is designed to be up to 250 ºC.

The cavity receiver is demonstrated by Figure 2(a). Also, Figure 2(b) shows the instrumentation profile, which was allocated at the middle section of the absorber. K type thermocouples were attached to the bottom surface of each tube and another at the top part of tube 3. T type thermocouples are used to measure the environment air, window glass, rock wool and the aluminum wall temperatures. In total, 17 temperature points were measured in this experimental setup.

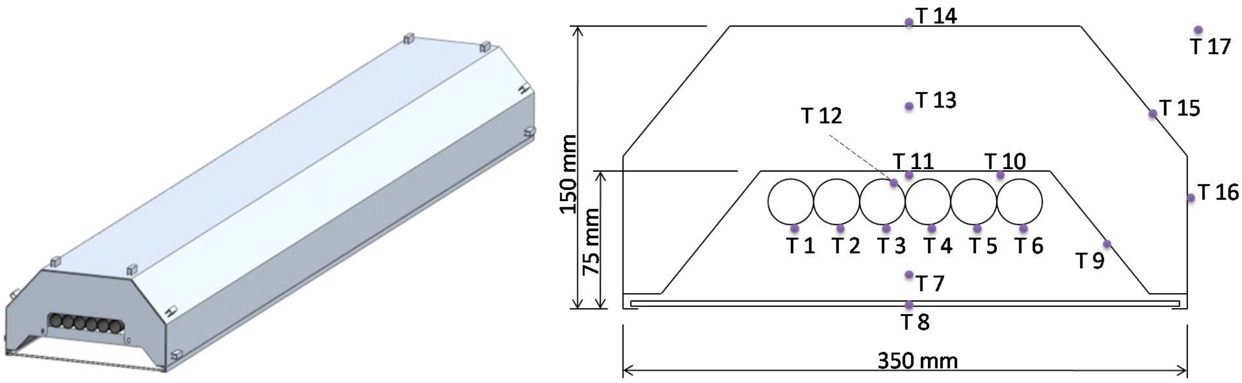

Figure 2: (a) Trapezoidal cavity receiver setup, (b) Instrumentation profile and cavity geometry

2.2 Experimental overall heat loss coefficient (UL)

In this work, the overall heat loss from the absorber was assumed to be equal to the total power dissipated by the resistances W_{ele} which can be written:

$$Q_{loss} = W_{ele} \tag{1}$$

Under steady state conditions the heat loss from the tubes Q_{loss} equals the rate in which heat is transferred from the absorber to the environment Q_{abs}. That can be written as:

$$Q_{loss} = Q_{abs} \tag{2}$$

The average absorber temperature, Tabs, is calculated considering the average of the six tube's surface temperatures,

$$T_{abs} = \frac{T_1 + T_2 + T_3 + T_4 + T_5 + T_6}{6} \tag{3}$$

Also, the rate at which heat is transferred from the absorber, in terms of the overall heat transfer coefficient, can be written as:

$$Q_{abs} = U_{abs} A_{abs} (T_{abs} - T_{env}) \tag{4}$$

Considering equations (1), (2) and (4), it is possible to write:

$$U_{abs} = Q_{abs} / A_{abs} (T_{abs} - T_{env}) \tag{5}$$

With exception of the extremities, the absorber is considered to be longitudinally isothermal. Also, the area, A_{abs}, of the tubes were considered as if all tubes merged in a single rectangular tube with edges 25.4 mm and 152.4 mm.

The overall heat loss coefficients were deduced for absorber temperatures of about 100 ºC, 125 ºC, 150 ºC, 175 ºC, 200 ºC and 225ºC for any of the test conditions. Tests were performed for natural convection and forced convection. Forced convection tests considered 1.9 m/s and 2.9 m/s wind speeds. The considered painting configurations were naked or non-painted tubes and fully painted tubes, using a multi-tube LFC specific selective painting on the last case. The emissivity of the polished stainless steel is 0.17 and the selected selective painting's emissivity is in between 0.3 and 0.6, depending on the painting configuration.

The experimental uncertainties were based on the standard uncertainties of T and K type thermocouples, which are 1 °C or 0.75 % and 2.2 °C or 0.75 %, whatever is greater, respectively for T and K type thermocouples. The uncertainty for the electric power transducer is 0.5 %. The maximum combined uncertainties for the overall heat loss coefficient were 2.2 % for the experimental data.

3. RESULTS AND DISCUSSIONS

The experimental setup was submitted to three different convective conditions, as mentioned above. For each of them, the overall heat loss coefficients, UL, were plotted against temperature difference between the absorber and environment. A polynomial best-fit curve was obtained from the experimented points and the curve can be used to predict heat losses on intermediary temperatures. Generally speaking, and as it was expected, the heat losses increased along with the absorber's temperature, with wind speed and with the increasing tube's surface emissivity.

Figure 3 displays the overall heat loss coefficients for non-painted and full painted tubes considering a window glass and also for full painted tubes without the window glass. It can be observed that the selective painting contributes to an increased heat loss for all experimented temperatures. The high emissivity of the painting also contributes to increase the radiative losses, as compared to non-painted tubes. At 180 °C temperature difference, the overall heat loss coefficients were of 7.76 W/m²K and 7.84 W/m²K for non painted and full painted tubes respectively, which represents a 1.03 % increase. Also, it can be seen that the presence of that window glass represents a reduction in heat losses. That can be explained by the fact that is reduces convective losses to environment and because the glass is opaque to medium temperatures radiative wavelengths, it also retains the tube's radiation reaching the cavity's aperture. For tests considering painted tubes and no window glass, the overall heat loss coefficient at 180 °C was 9.83 W/m²K. Therefore, removing the window glass represented a 25.4 % increase on the overall heat loss.

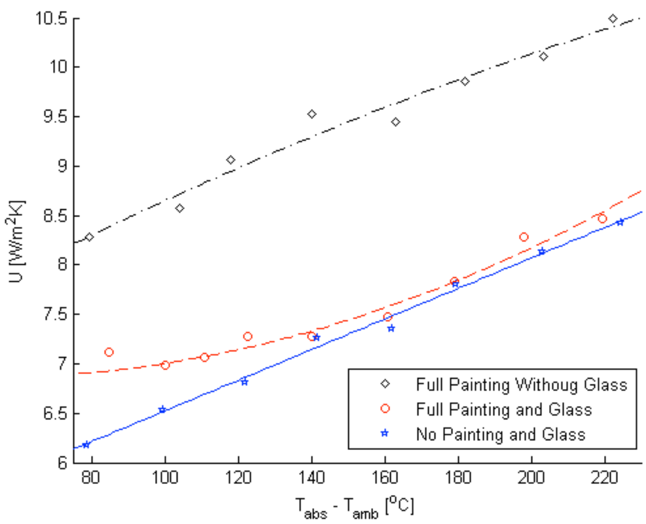

Figure 3: Overall heat loss coefficient comparison on tests considering different painting configurations and window glass conditions, for natural convection

The effects of forced convection over the heat losses are shown by Fig. 4 and Fig. 5. Figure 4 displays the overall heat loss coefficients for different temperatures considering full painted tubes and a 1.9 m/s wind speed. The overall heat loss coefficient at 180 °C temperature difference, for example, is of 9.66 W/m²K. As compared to the same painting condition in natural convection, the heat loss increased 23.2 % for a 1.9 m/s wind speed forced convection.

Similarly, Fig. 5 shows the overall heat loss coefficients for full painted tubes with and without a window glass considering a 2.9 m/s wind speed. The overall heat loss coefficients at 180 °C temperature difference is of 10.24 W/m²K and 23.28 W/m²K for tests with and without the glass window, respectively. For means of comparison, for tests considering the window glass, these results represented a 31.1 % and 6.00 % difference with natural convection and 1.9 m/s wind speed tests, respectively. Also, if we compare the two tests with 2.9 m/s, they differ with a 127 % increase when we take the window glass off.

The complete range of overall heat loss coefficients in between 100 °C and 225 °C tubes' average temperature are shown in

Table 1. For tests without the window glass the values of overall heat loss coefficients ranged between 8.31-10.17 W/m²K for natural convection and 22.33-24.23 W/m²K for 2.9 m/s wind speed forced convection. On the other hand if the window glass is considered, the overall heat loss coefficients ranged between 6.21-8.15 W/m²K for natural convection with non painted tubes and between 7.13-8.25 W/m² for full painted tubes. If 1.9 m/s wind speed is

longitudinally incident over the cavity, the values were between 8.18-10.13 W/m²K. Now for 2.9 m/s wind speed, the range corresponds to 9.50-11.25 W/m²K.

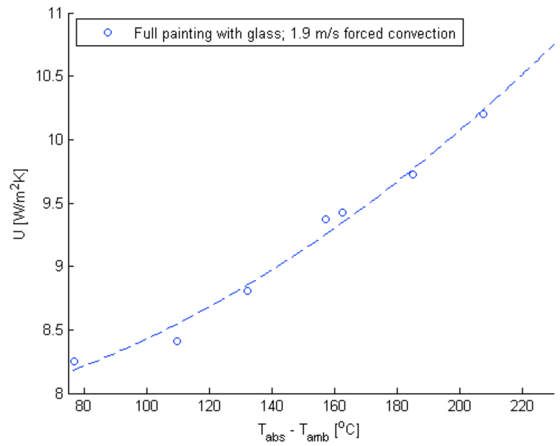

Figure 4: Overall heat loss coefficients for full painted tubes considering 1.9 m/s wind speed

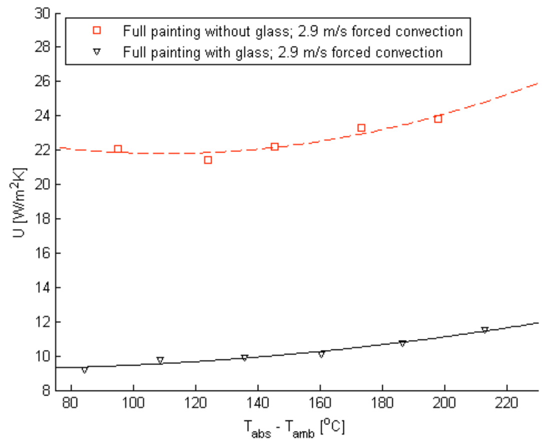

Figure 5: Overall heat loss coefficients for full painted tubes considering 2.9 m/s wind speed, with and without a window glass

Table 1: Overall heat loss coefficients for different painting and convective configurations

Overall Heat Loss Coefficient [W/m^{2}K]					
		Non Window Glass		Window Glass	
		No Painting	Full Painting	No Painting	Full Painting
Natural Convection		-	8.31-10.17	6.21-8.15	7.13-8.25
Forced Convection	1.9 m/s	-	-	-	8.18-10.13
	2.9 m/s	-	22.33-24.23	-	9.50-11.25

Comparing tests considering full painting of the tubes and a window glass placed on the cavity's mouth, which is probably the most likely real application, the difference in heat losses was in the range of 33.2 % - 36.4 % for the experimental tests. Additionally, Fig. 6 illustrates how the overall heat loss coefficient varies along with the increasing wind speed at a 180 °C temperature difference.

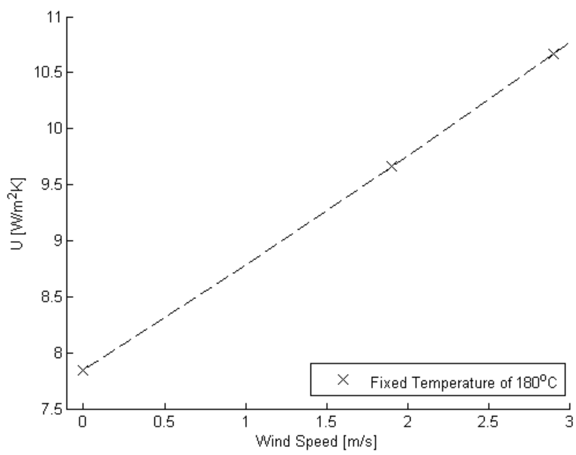

Figure 6: Overall heat loss coefficient versus wind velocity at 180 °C temperature difference between average tube temperature and environment

4. CONCLUSIONS

Although radiation is the dominant heat loss mechanism on medium temperature absorbers (Saxena et al, 2016), the experiments have shown that convection still plays an important role on the overall heat loss account and it must not be neglected. The overall heat loss coefficient has increased 33.2 % from natural convection to 2.9 m/s wind speed forced convection. Also, the overall heat loss coefficient was found varying almost linearly along with wind speed for the experimented temperatures.

Additionally, the presence of the window glass is of vital importance for non-evacuated absorbers. The window glass has reduced the heat loss in 14.2 % for natural convection condition and in 57.3 % on 2.9 m/s wind speed forced convection.

5. ACKNOWNLEDGMENTS

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6. REFERENCES

Dey, C.J., 2004. "Heat transfer aspects of an elevated linear absorber". *Solar Energy, 76(1), pp.243-249.*

EIA (U.S. Energy Information Administration). "International energy Outlook 2016". Ch. 7, Industrial sector energy consumption <http://www.eia.gov/forecasts/ieo/>. [Accessed Apr.2016]; 2016.

IEA (International Energy Agency). "World energy Outlook 2014". < https://www.iea.org/ >. [Accessed Sep. 2017]

Iparraguirre, I., Huidobro, A., Fernández-García, A., Valenzuela, L., Horta, P., Sallaberry, F., Osório, T. and Sanz, A., 2016. "Solar thermal collectors for medium temperature applications: a comprehensive review and updated database". *Energy Procedia, 91, pp.64-71.*

Saxena, A., Jhamaria, N., Singh, S. and Sahoo, S.S., 2016. "Numerical analysis of convective and radiative heat losses from trapezoidal cavity receiver in LFR systems". *Solar Energy, 137, pp.308-316.*

Sharma, A.K., Sharma, C., Mullick, S.C. and Kandpal, T.C., 2017. "Solar industrial process heating: A review." *Renewable and Sustainable Energy Reviews, 78, pp.124-137.*

Singh, P.L., Sarviya, R.M. and Bhagoria, J.L., 2010. "Heat loss study of trapezoidal cavity absorbers for linear solar concentrating collector" *Energy Conversion and Management, 51(2), pp.329-337.*

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