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Experimental Investigation of the Thermal Performance on a Solar Parabolic trough Collector in the Caribbean Region

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Abstract

The main objective of this paper is to present a new tool to design parabolic trough collectors (PTC), based on the theoretical heat transfer principles, solar radiation models and weather conditions at the Colombian Caribbean Region, connect equations to find PTC dimensions, build one PTC for specific process conditions and compare predictions of the model and experimental performance. Related to the findings of this study, experimental performance data differs from model predictions, real efficiencies are lower than theoretical results. Finally, the use of solar energy should be increased at commercial and residential buildings, it makes necessary to create tools for new solar powered systems using parabolic trough collector.

Keywords: Parabolic Trough Collector, Solar Energy, Model, thermal performance, experimental investigation.

INTRODUCTION

The generation of electric power has been a subject of great importance in the last years, since while the demographic increase in the last century was almost four times, the increase in the demand of energy has been of approximately ten times. However, this growth has had unfavorable consequences for the planet, because the main sources of energy in the world are fossil fuels, which involve emissions of gases that have significantly deteriorated environmental conditions especially in recent Years [1]; According to the International Energy Agency (IEA) during the period 1973 to 2013 carbon dioxide emissions due to the burning of fuel have increased by 16.675 million tons as can be seen in figure 1, this figure provides a Clear idea of the damage that the environment has suffered when knowing the detrimental effect of this gas on the ozone layer [2]. The existence of zones with measurable solar radiation levels in Latin American countries such as Brazil, Chile and Mexico allow us to consider the implementation of electric power generation systems from this clean and renewable source [1]. Solar energy is a renewable resource able to supply industrial thermal demand, mainly in those productive processes that use temperatures below 120 ° C, this resource can be exploited using technology such as parabolic solar collectors (PTC) [4], which Are artifacts basically constituted by a parabola formed by a reflecting surface, a receiver tube located in the focal line thereof, through which circulates a working fluid that receives the direct solar radiation of the reflecting zone, which allows to take advantage of the radiation Solar incident on thermal energy [5]. Plants such as Torresol Energy located in Spain generate 50MW using PTC connected in series [6]. Likewise, companies specializing in solar thermal power projects such as Sener, focus their designs on collectors such as SenerTrough to increase the performance of their plants [7], while SkyFuel, located in the United States, aims their designs as the SkyTrough, to the economy of the installations by the development of a reflective film called ReflecTechPLUS [8], these equipments are installed in series for the greater use of their capacities

On the other hand, some tests with parabolic solar collectors with the reflective surface manufactured in stainless steel concluded that due to the low optical performance of the material, direct steam generation could not be achieved [9]. It has also been determined that thermal losses decrease when the receiver tube is coated with a translucent material [10].

The advances achieved in recent years in energy generation from unconventional sources has encouraged the interest of many sectors to work in this type of projects and thus to take advantage of the resources that the country offers, among the most Highlights are wind power and solar energy in the Colombian Caribbean region. The global solar radiation conditions of up to 6.5 kWh/m^2 at a certain time of the year for the city of Barranquilla [14] boosted simulations to determine the energy production of a parabolic solar collector plant, resulting in a value Of kWh that can cover 50% of the annual energy demand in the city [15]. In some cases where solar tracking systems have been implemented, the experimentally obtained values of parabolic collectors subjected to radiations lower than those presented in the Colombian Caribbean Coast approximate 65% [16]. The solar resource of the city of Barranquilla and the current energy crisis that crosses the region, makes necessary an experimental evaluation of the performance of a parabolic solar collector, so the main contribution of this work will be the construction and evaluation of an economic PTC as an alternative to harness solar thermal energy in the city of Barranquilla.

PERFORMANCE MODEL

The design process of the equipment for the solar capture was carried out using a spreadsheet in which the corresponding energy balance was carried out and the data that came from the solar radiation estimation model described in chapter 1 were interrelated, With the thermodynamic conditions to which the working fluid was to be carried. The thermodynamic properties of the water were taken as the average at the inlet and outlet temperatures of the receiving tube. The geographic data that are entered into the model are clearly related to the estimation of the solar radiation that will be perceived by the collector.

PTC heat transfer equations.

The heat acquired by the fluid will realize a one-dimensional energy balance in the receiver tube, taking as a boundary the internal surface of the receiver that is in contact with the fluid and assuming a steady state.

$$Q_u = Q_r - Q_l, \tag{1}$$

where $Q_u$is the heat transmitted per unit time to the fluid, Q_r is the rate of heat captured by the reflecting surface and transmitted to the receiving tube and Q_L is the total value of the loss rate presented by convection, radiation and conduction.

The receiver tube is divided into an infinite number of segments of equal length, to analyze each segment will be considered a uniform and normal flow to the surface of the receiver tube. A one-dimensional analysis can be applied by applying the first law of thermodynamics for each segment

$$\frac{dE}{dt} = \dot{Q} - \dot{W} + \sum_{in} \dot{m}(h) - \sum_{out} \dot{m}(h), \tag{2}$$

where h is the enthalpy of the water and E is the energy of the systems.

In considering that the properties of the fluid depend on temperature, the amount of heat per unit of time that must be transferred to bring the working fluid to the required conditions is dependent on the following mathematical expression

$$Q_u = C_p \dot{m}(T_{out} - T_{in}), \tag{3}$$

where is the C_p specific heat of the water at the average inlet temperature T_{in} and outlet T_{out} of the receiving pipe at a mass flow m ̇. The amount of heat required is directly proportional to the difference in temperatures at the ends of the receiving tube. The heat flux per unit area from the surface of the receiver to the fluid with mean temperature T_m is defined by [28].

$$\dot{q}_s = h_f\,(T_s - T_m) = \frac{Q_{útil}}{A_s}, \tag{4}$$

where the heat transfer coefficient $h_f$defines the ratio of the heat flow per unit area from the surface of the receiver tube A s to the fluid and the difference between the average temperature of the fluid in a section T_m.

$$Nu = 0.023\, Re^{0.8} Pr^{0.4}. \tag{5}$$

In the above equation, dimensionless factors such as the Prandtl Pr number and the Reynolds Re number of the substance inside the tube are involved. The relationship between the forces of inertia and the viscous forces of the substance is determined from the following equation

$$Re = \frac{V_{avr} D_i}{v_f}. \tag{6}$$

The result of this relation indicates if the flow that is presented is turbulent or laminar, is considered laminar for values smaller than 2300 and turbulent for values greater than 10000. The flow is considered totally developed at the moment in which the length of the tube is much Greater than the length required by the developing fluid which is 10 times the internal diameter D_i of the receiving tube. The average velocity of the fluid ($V_{pavrrom}$) into the receiver tube is calculated from the continuity equation and the volumetric flow, the kinematic viscosity value of the working fluid v_f is taken at the average temperature between the outlet and The tube inlet

$$V_{avr} = \frac{4\dot{m}}{\rho \pi D_i^2}. \tag{7}$$

The relation between the convection heat transfer coefficient h_f and the Nu value is directly proportional, the thermal conductivity of the water k and the internal diameter D_i also influence this relation as follow

$$h_f = \frac{k}{D_i} Nu. \tag{8}$$

The average temperature on the inner surface of the tube allows a reference of the average maximum temperature that can reach the working fluid, this temperature can be calculated using

$$T_s = \frac{T_{in} + T_{out}}{2} + \frac{\dot{q}_s}{h_f}. \tag{9}$$

Heat losses in the receiver tube are related to the characteristics of the working fluid as well as to the characteristics of the tube itself, by the heat transfer mechanisms that are present, in addition to the difference between the surface of the tube. The heat rate assigned to the environment Q_L is defined as follow [17]

$$Q_L = U_L A_r (T_s - T_a), \tag{10}$$

where U_L is the total heat transfer coefficient between the tube and the environment obtained from the sum of the heat transfer coefficients for each of the present phenomena, A_r is the area of the receiver that is in contact with the outside , T_s is the average temperature of the outer surface of the receiver, which can be assumed to be identical to that of the inner surface by the properties of the material and the dimensions of the receiver tube, and the ambient temperature T_a this heat transfer coefficient is Calculated from the following expression [17]

$$U_L = h_w + h_r + U_c. \quad (11)$$

The coefficients corresponding to each phenomenon are calculated by analyzing that the conduction is a mechanism of heat transfer that is produced by direct contact between bodies [28], and that the only contacts that the receiver tube presents are its supports that are outside of the Section that was chosen as border is assumed that the losses by conduction are zero. The remaining mechanisms are present due to the interaction of the outer surface of the receiver with the surrounding fluid, in the case of convection is defined as a case of forced external convection because the air currents present in the environment collide with the outside of the receiver [28].

PTC Dimensions

The amount of light projected in the receiver tube largely defines the thermal performance of the equipment, so it is essential that the dimensions of the reflecting surface are related so that the entire solar image is intercepted according to the expression [17].

$$D_O = \frac{a \sin 0.267}{\sin \theta_r}, \quad (12)$$

where, D_O is the outer diameter of the receiver tube, this corresponds to the dimension of the tube whose internal diameter allows a turbulent regime to a mass flow, θ_r is the angle formed by the line from the apex to the center of the reflecting surface And the line from the focus to the highest point of the reflecting surface as shown in Figure 16, this parameter is known as rim angle (θ_r). The amplitude of the parabola (a) is defined by the area required to capture the amount of direct solar radiation (I_b), required to provide the necessary heat to give the proper temperature to the water and supply the heat losses (Q_r) taking into account a length of the manifold (L_c)

$$a = \frac{Q_r}{I_b \cdot L_c}. \quad (13)$$

After calculating the internal diameter D_i and the amplitude of the parabola, the parameter θ_r is defined as function of the amplitude of the parabola (a) and the focus (f) as follow

$$(16 \tan \theta_r)(f/a)^2 - 8\theta_r(f/a) - \tan \theta_r = 0. \quad (13)$$

RESULTS

Experimental analysis of the PTC designed with the model

Performance of the parabolic solar collector from January 21, 2017 to January 27, except for day 24, operation performance of the arrangement and a comparison will be made with the theoretical prediction of the equipment according to the model explained above. The results are divided into two stages, with different parameters modified during the tests.

The data of the environmental variables taken from the meteorological station of the group KAÍ of the University of the Atlantic were registered on January 21, 22 and 23 simultaneously with the data of Temperature and volumetric flow during the operation of the experimental assembly. This information allows to analyze the behavior of the collector during different environmental conditions and to know the maximum thermal performance that can be reached. Subsequently tests were performed on 25, 26 and 27 January, each day varying the volumetric flow of the collector to know the influence of this parameter on the performance of the equipment, also changed the slope of the collector (inclination of the designed collector Is 11°) to verify the variation of the yield due to this factor.

The figure 1 shows the comparison between the theoretical convection losses and the real values obtained during the operation of the tool.

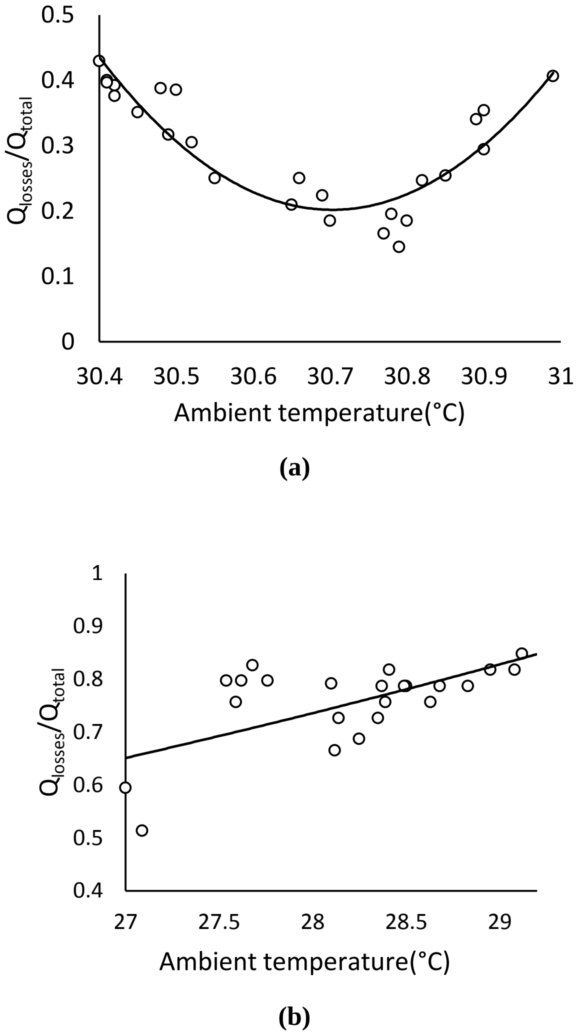
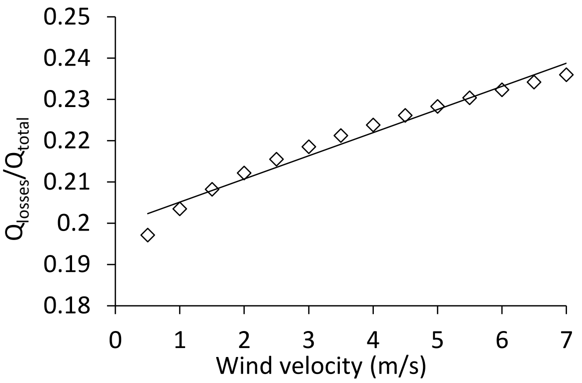
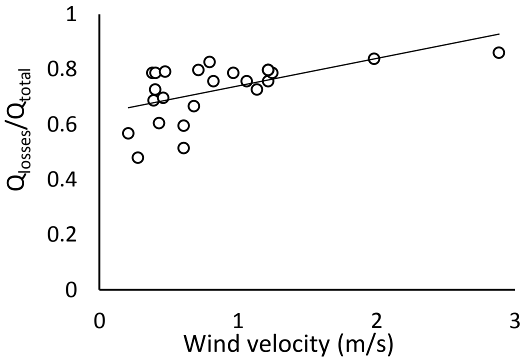


Figure 1. Convection losses due the ambient temperature, (a) Theoretical values, (b) Values obtained during the test

The shown in the figure 1 happens because the radiation model predicted an average of the ambient temperature around 30.7°C, close to the receptor tube temperature, meanwhile during the operation of the tool the receptor tube temperature was around 26,5°C. The next parameter to be compared is the wind velocity, the graph from the heat transfer analysis and the information obtained during the equipment operation are shown in the Figure 2.



(a)



(b)

Figure 2. Variation of the convection losses due the wind velocity, (a) Values obtained by the model, (b) Real thermal performance.

The model used to estimate the incident radiation in the place is the Transmisibility Model, these predicted amounts of radiation lower than the obtained during the operation of the system, these is shown in the Figure 3.

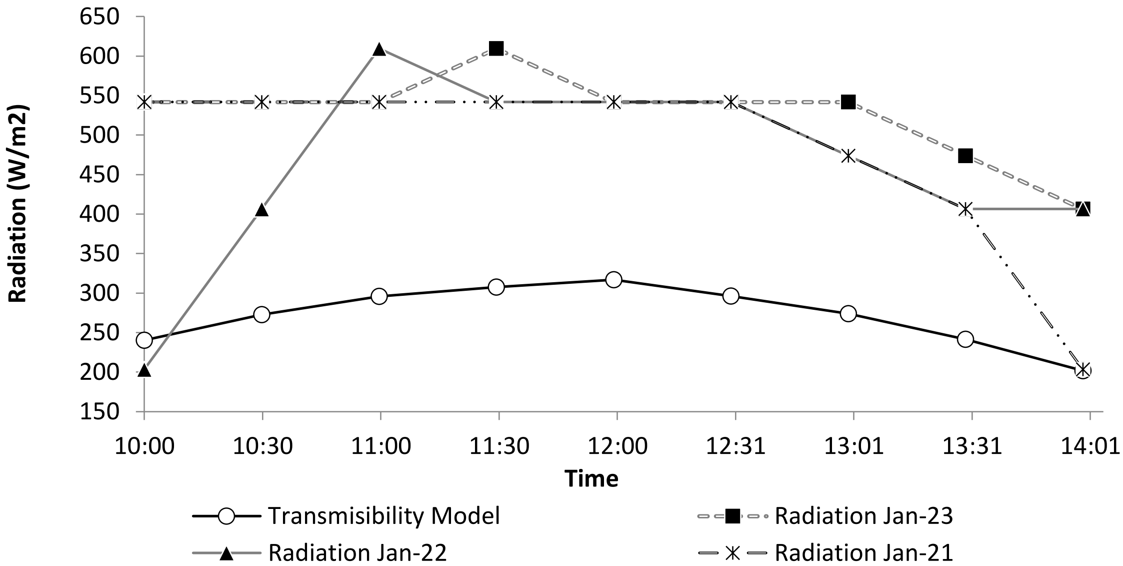


Figure 3. Comparison between the model radiation and incident radiation during the operation of the equipment.

As observed, the incident radiation values are higher than those of the radiation model, however this didn't allow to obtain values of performance higher than the expected ones in the model. This can be seen in Figure 4.

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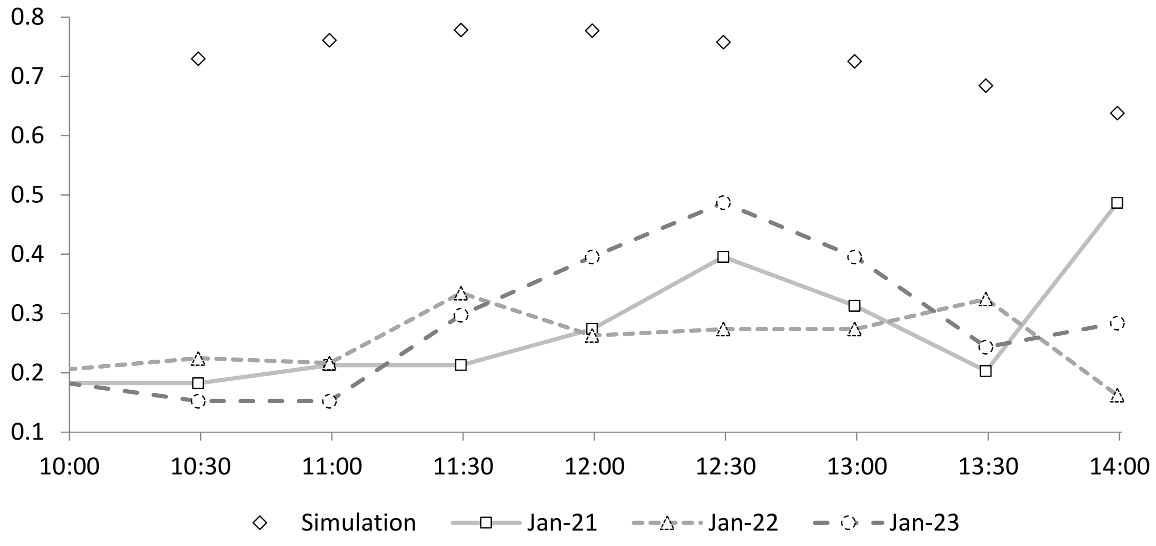

Figure 4. Comparison between the performance estimated by the model and the real performance obtained during the operation of the equipment.

Although the values of radiation were higher than expected, the performance achieved each day of operation of the equipment was lower, since the model raised a constant temperature increase of 5 ° C, obviously during the test it was different.

Figure 5 shows the increase in temperature throughout the day and the daily average.

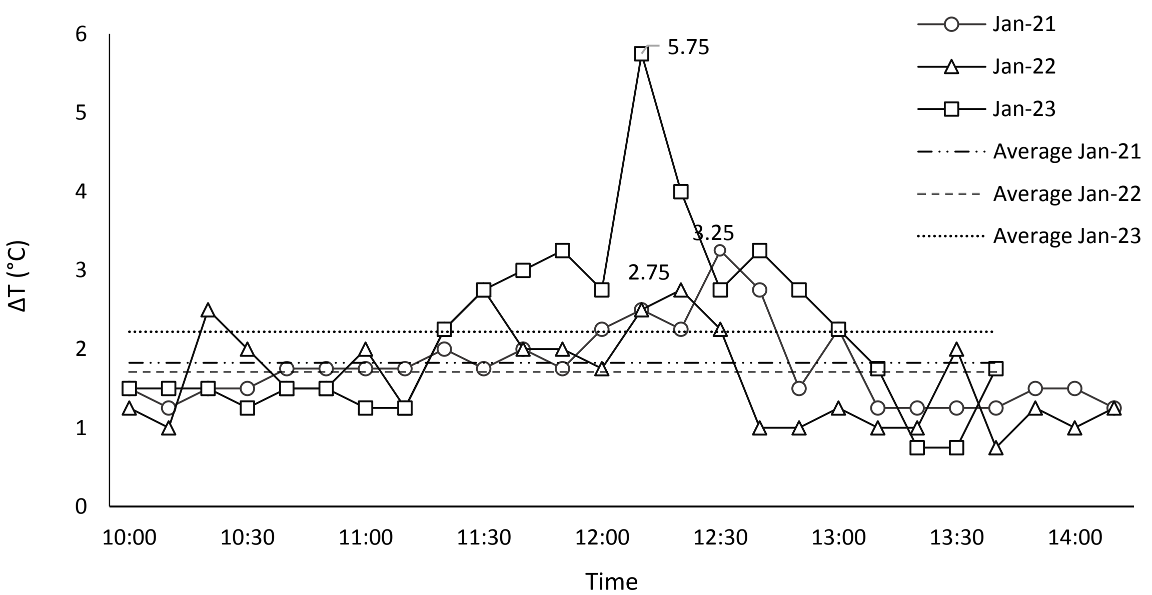

Figure 5. Temperature increase during the operation of the equipment.

For the second part of the test, the equipment was operating with variations in the volumetric flow. The model allowed to predict the values of performance for each volumetric flow, the graphs in the Figure 6 shows these values against a coefficient defined by the fraction between DT (the difference between the ambient temperature and the inlet temperature water to the collector) and I(incident radiation according the model radiation).

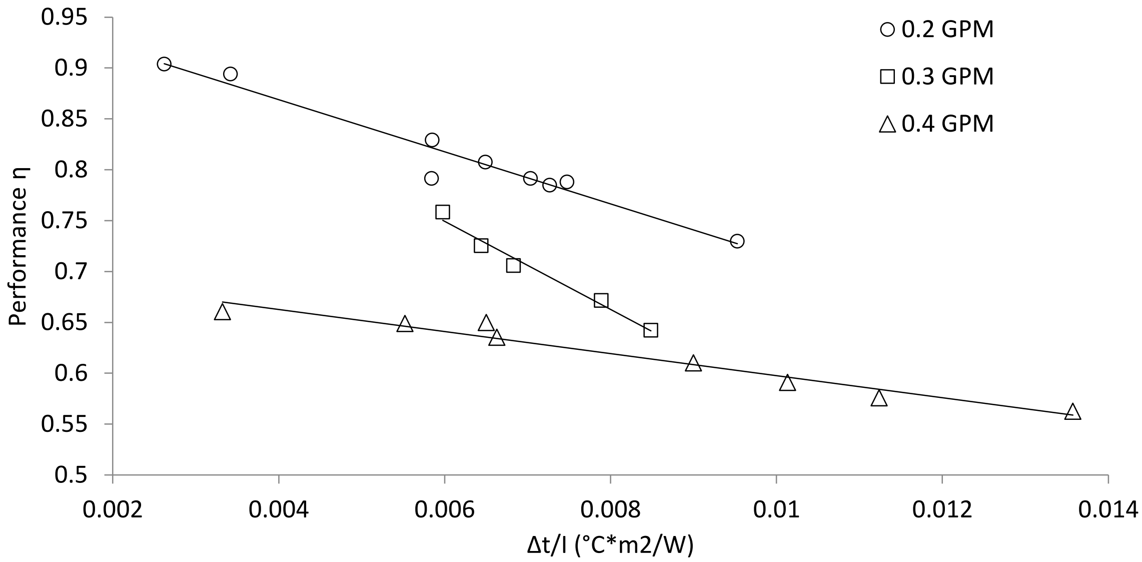

Figure 6. Collector's performance for each volumetric flow

The graph shown in the Figure 6 can be compared with the graphs obtained during the operation of the collector. The figure 7, 8 and 9 show the variation in collector's performance with the variation in the volumetric flow and the inclination defined previously (11°).

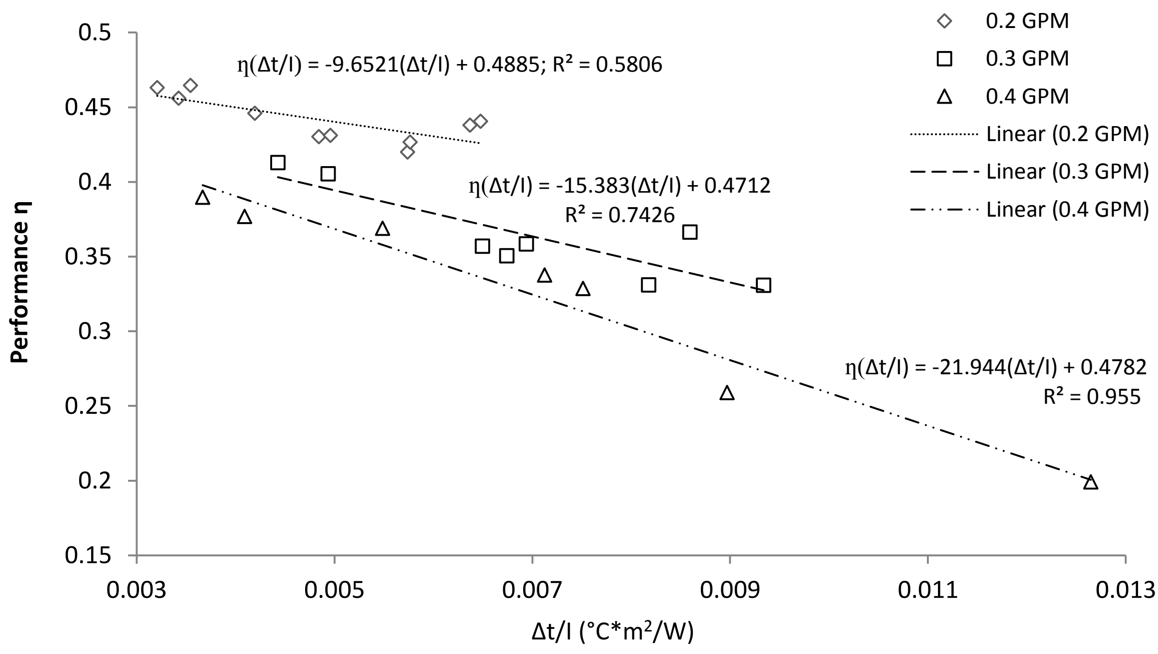

Figure 7. Collector's performance for different volumetric flows and an inclination of 17°

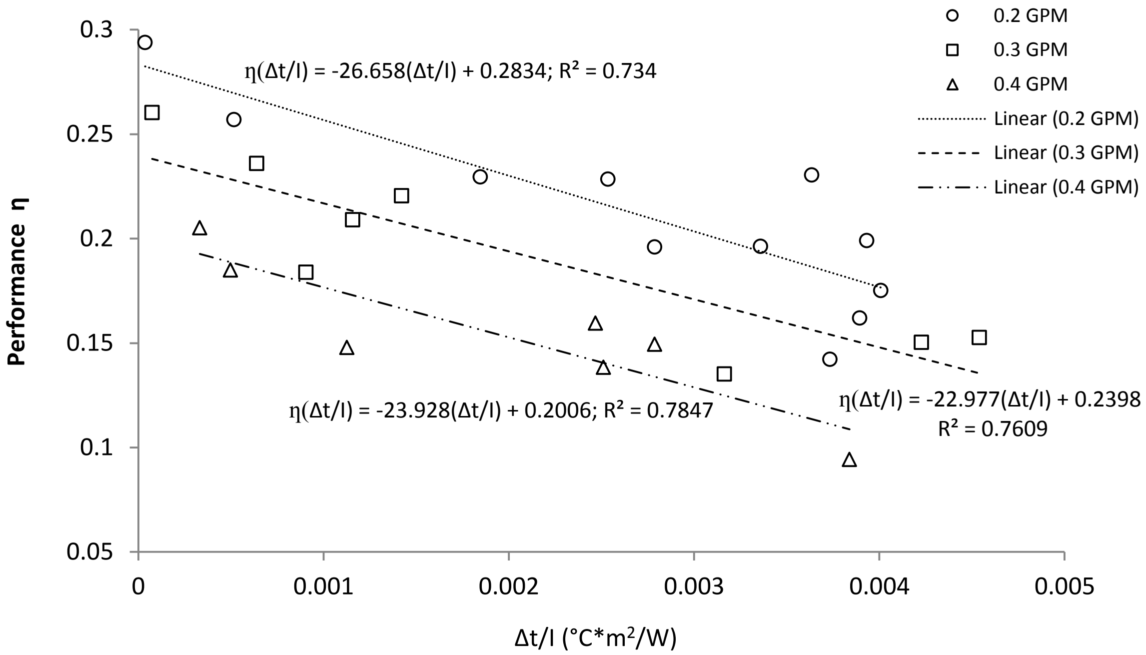

Figure 8. Collector's performance for different volumetric flows and an inclination of 5°

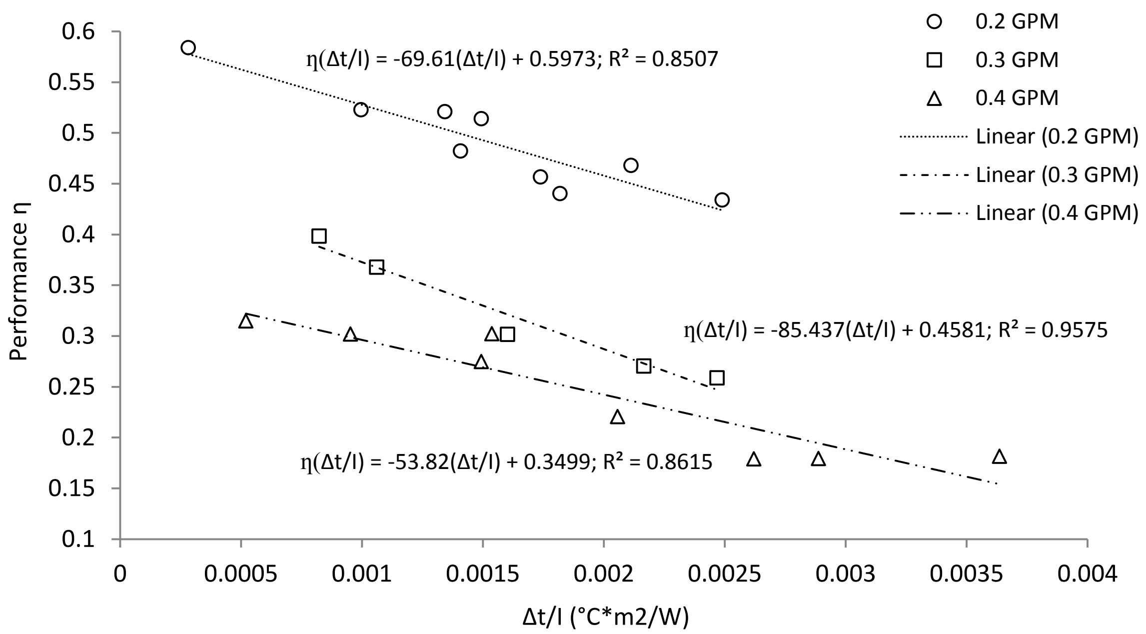

Figure 9. Collector's performance for different volumetric flows and an inclination of 11°

The values showed in the graphs 7,8 and 9 are lower than the predicted by the model, although, the graph 9 shows performance values higher than the graps 7 and 8, it proves this is the correct inclination, that is to say, the inclination that allows to reach the higher performance values (when the volumetric flow is 0.2 GPM). The figure 10 shows the influence of the inclination on the collector's performance.

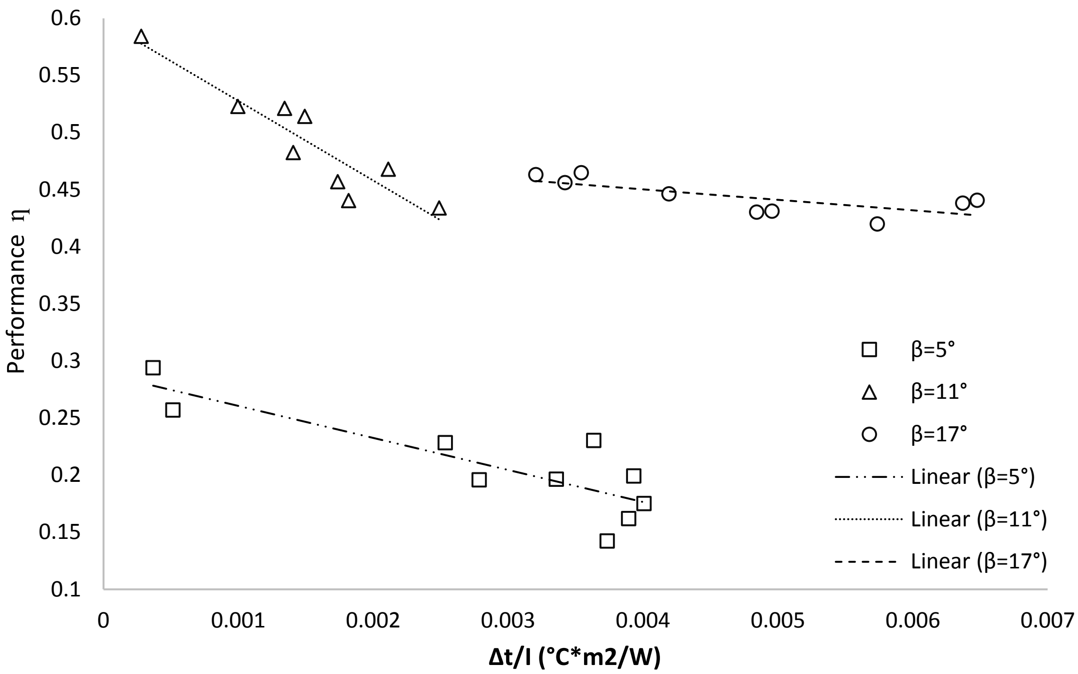

Figure 10. Collector's performance operating to 0.2 GPM in different inclinations.

CONCLUSIONS

The differences between theoretical results and experimental results are related with optical efficiency of the parabolic surface of the collector built, because it is compound by a group of flat mirrors, the model assumes that parabolic surface of PTC is always perpendicular to the sunlight beam.

The thermal losses are studied with respect to the ambient temperature. The heat transfer obtained is referred to the net external tube area in order to avoid confusion with the solar collector area. To study the effect of irradiance on collector, the energy balance was applied to calculate the mean temperature difference as function of the measurable fluid temperature. The experimental and theoretical result were presented for the efficiency based on mean solar collector fluid temperature and therefore is not require to correct for different fluids and flow regimes.

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