Effect of Height Difference between Solar Thermal Collector & Storage Tank on the Entransy Destroyed

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Abstract— The difference in level between the top of the solar collector and the bottom of the storage tank is one of the parameters that can be used to improve the performance of a solar collector that operates by thermosiphon. The present research studies the influence of this factor on the criterion of entransy destroyed. The results of our analyzes showed that the distance between the top of the solar collector and the bottom of the storage tank recommended for a better performance is 1.1 m according to the entransy destroyed criterion.

Key words: Solar Collector; Thermosiphon; Storage Tank; Level; Entransy Destroyed

I. Introduction

Renewable energy source widely used worldwide is solar thermal energy. In its basic form, a solar thermal collector is designed to intercept solar radiation, absorb that radiation to convert it into heat energy, and then deliver that heat to a heat transfer fluid. Solar water heating systems have reached technical maturity and can be used in most climate conditions as they harness the solar energy to useful heating. Thermosiphon solar water heating systems are the simplest and most widely used solar energy collection and utilization devices. Water in the riser tubes get heated and flows to the storage tank due to density difference. Distributions temperatures in the collector and the flow of water in the riser tubes and connecting pipe of a solar water heating system is necessary to understand the performance of a thermosiphon solar water heating systems. Distributions temperatures and the water flow depend on the thermosiphon head, which is due to the change in density of water caused by water temperature rise in the solar collector, the collector slope, the type and dimensions of storage tank, the height difference between solar thermal collector and storage tank. The influence of these parameters on the performance of the thermosiphon solar collector has been the subject of some studies. The effect of wind speed, number of glazing cover, ambient temperature, gap spacing between absorber plate and the glazing cover, tilt angle and the emissivity of the absorber plate one the overall heat-loss coefficient of a flat-plat collector was studied in [1]. In [2], the author has presented an analysis of water circulation rate in natural circulation systems and compared the theoretical and experimental inlet and outlet water temperatures of the collector. Other researchers was studied the performance of the solar water heater system with a horizontally arranged storage tank [3-4]. To see the influence of tube type on the performance of a solar water heater system, it was realized an experimental study using steel and polypropylene tubes in [5]. In [6], it was evaluated the variation of thermal efficiency according to the height between the top of the solar collector and the bottom of the storage flask for the overall heat loss coefficient fixed. In their model study of the naturally circulating solar water heater system, the authors [7] varied between -30 cm and 60 cm the height between the top of the sensor and the bottom of the flask storage. They found that the impact of the variation of this height is very small on the out temperature but significant on the speed of the flow of water.

The difference in height between the top of the solar collector and the bottom of the storage tank influences the working of the solar water heater system [6,7]. The purpose of this study is to determine the optimal height for best performance of the solar water heater running in natural circulation by considering the operating parameters of the system at each moment of the day. The absorber of the solar collector used is with fins in the space between the absorber and the glass. In the case of this absorber, the fins pitch recommended is 0.005 m [8,9]. The water circulates directly in contact with the absorber and the height of the water channel is 0.005 m. The section of the solar collector used is shown below.

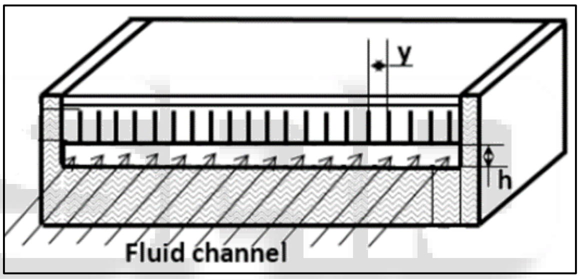

Fig. 1: Section of Solar Collector Used

II. Thermal balance of the system

The thermal balance will be at the solar collector, the storage tank and the water pipes between the solar collector and the storage tank.

A. Thermal Balance at the Solar Collector

The main elements on which the balance will be made at the solar collector are the glass cover, the absorber and the water. To make the thermal balance at these elements, the first principle of the thermodynamics is used. The expression of this principle, by considering each element as a system, is the following:

$$\frac{dU}{dt} = Q_{ent} - Q_{sor} + Q_{in} \qquad (2.1)$$

With:

$\frac{dU}{dt}$: The variation of the internal energy of the system;

Q_{ent} : The heat flux entering of the system;

Q_{sor} : The heat flow out of the system;

Q_{in} : The heat flow generated inside the system

Thermal balance at the glass: The overall balance on the glass cover gives:

$$m_v c_v \frac{dT_v}{dt} = S \cdot \alpha_v \cdot G + S \cdot H_1 - S \cdot H_2 \qquad (2.2)$$

With:

$S \cdot \alpha_v \cdot G$: The solar radiation absorbed by the glass;

$H_1 = (h_{c,ab.v} + h_{r,ab.v})(T_{ab} - T_v)$: The heat exchanged by convection and radiation between the absorber and the glass;

$H_2 = (h_{c,v.a} + h_{r,v.a})(T_v - T_a)$: The heat exchanged by convection and radiation between the glass and the ambience.

Thermal balance at the absorber: The absorber is the element that receives most of the incident solar radiation. Part of this radiation is exchanged with the glass, another part is lost from the bottom of the collector through the thermal insulation and most of it is exchanged with the heat transfer fluid. The balance at the level of the absorber gives:

$$m_{ab} c_{ab} \frac{dT_{ab}}{dt} = S \cdot \alpha_v \cdot \alpha_{ab} \cdot G - S \cdot H_1 - S \cdot H_3 - S \cdot H_4 \qquad (2.3)$$

With:

$S \cdot \alpha_v \cdot \alpha_{ab} \cdot G$: The solar radiation received by the absorber;

$H_3 = h_{ab.f}(T_{ab} - T_f)$: The heat exchanged between the absorber and the heat transfer fluid;

$$H_4 = \left(\frac{1}{\frac{e_{is}}{\lambda_{is}} + \frac{1}{h_{wind}}} \right)(T_{ab} - T_a)$$

The heat lost from the bottom of the collector.

Thermal balance at the heat transfer fluid: Considering that the temperature and the heat transfer fluid speed vary only along the y axis, the heat balance on the heat transfer fluid gives:

$$m_f c_f A \cdot dx \frac{dT_f}{dt} = dS \cdot H_3 - \dot{m}_f \cdot c_f \cdot dT_f \qquad (2.4)$$

A is the section of the water canal

$$A = l \cdot h \qquad (2.5)$$

$$S = 2 \cdot l \cdot x \qquad (2.6)$$

$$dS = 2 \cdot l \cdot dx \qquad (2.7)$$

With l and x respectively the width and height of the water channel. Replacing A by its expression and dividing the equation by dx, we obtain:

$$m_f \cdot c_f \cdot l \cdot h \cdot \frac{dT_f}{dt} = 2 \cdot l \cdot H_3 - \dot{m}_f \cdot c_f \cdot \frac{dT_f}{dx} \qquad (2.8)$$

Recapitulation of balance equations of the solar collector: Different equations reflecting the thermal balances show relations between different parameters that compose them. The system of equations below summarizes these balance equations:

$$\begin{cases} m_v c_v \frac{dT_v}{dt} = S \cdot \alpha_v \cdot G + S \cdot H_1 - S \cdot H_2 \\ m_{ab} c_{ab} \frac{dT_{ab}}{dt} = S \cdot \alpha_v \cdot \alpha_{ab} \cdot G - S \cdot H_1 - S \cdot H_3 - S \cdot H_4 \\ m_f \cdot c_f \cdot l \cdot h \cdot \frac{dT_f}{dt} = 2 \cdot l \cdot H_3 - \dot{m}_f \cdot c_f \cdot \frac{dT_f}{dx} \end{cases} \qquad (2.9)$$

In practice, the enthalpy variations according the time of the collector components are weak, that is to say the terms in $m \cdot c \cdot \frac{dT}{dt}$ can be neglected [10].

Equation system then reduces to:

$$\begin{cases} S \cdot \alpha_v \cdot G + S \cdot H_1 - S \cdot H_2 = 0 \\ S \cdot \alpha_v \cdot \alpha_{ab} \cdot G - S \cdot H_1 - S \cdot H_3 - S \cdot H_4 = 0 \\ 2 \cdot l \cdot H_3 - \dot{m}_f \cdot c_f \cdot \frac{dT_f}{dx} = 0 \end{cases} \qquad (2.10)$$

The last equation of the (2.10) is a differential equation whose solution (after resolution) is given by:

$$\frac{T_{ab} - T_f}{T_{ab} - T_0} = \exp\left(-\frac{2 \cdot l \cdot h_{ab.f}}{\dot{m}_f \cdot c_f} x \right) \qquad (2.11)$$

To obtain the outlet temperature of the hot water, x takes the value of L = 1 m. The system of equations to solve at the level of the solar collector is then written:

$$\begin{cases} S \cdot \alpha_v \cdot G + S \cdot H_1 - S \cdot H_2 = 0 \\ S \cdot \alpha_v \cdot \alpha_{ab} \cdot G - S \cdot H_1 - S \cdot H_3 - S \cdot H_4 = 0 \\ \frac{T_{ab} - T_f}{T_{ab} - T_0} - \exp\left(-\frac{2 \cdot l \cdot h_{ab.f}}{\dot{m}_f \cdot c_f} L \right) = 0 \end{cases} \qquad (2.12)$$

The convection coefficient between the glass and the absorber $h_{c,ab.v}$ depends on the configuration of the fins of the absorber. The characteristic dimension is equal to y.

$$h_{c,ab.v} = Nu \frac{\lambda_{air}}{y} \qquad (2.13)$$

The Nusselt number Nu for an inclination i of the collector relevant to the horizontal is given by [8]:

$$Nu = Nu_i = \frac{90 - i}{90}.Nu_0 + \frac{i}{90}.Nu_{90} \qquad (2.14)$$

Nu_{90} and Nu_0 are respectively the Nusselt number for i = 90° and for i = 0°.

Nu_0 is given by the relationship of Jones and Smith [11]:

$$Nu_0 = \left[\left(\frac{1500}{Ra_y} \right)^2 + \left(0.081 Ra_y^{0.39} \right)^{-2} \right]^{-\frac{1}{2}} \qquad (2.15)$$

$$Ra_y = \frac{g.\beta.(T_{abs} - T_v) y^3}{\nu.\Lambda} \qquad (2.16)$$

Nu_{90} is given by the relationship of Bar-Cohen and Rohsenow [12]:

$$Nu_{90} = \left[\frac{576}{(Ra')^2} + \frac{2.873}{\sqrt{Ra'}} \right]^{-\frac{1}{2}} \qquad (2.17)$$

$$Ra' = \frac{g.\beta.(T_{ab} - T_v) y^4}{\nu.\Lambda.x} \qquad (2.18)$$

$$\Lambda = \frac{\lambda_{air}}{\rho_{air} cp_{air}} \qquad (2.19)$$

The coefficient of heat transfer by radiation between the glass and the ambience $h_{r,v.a}$ and the one between the absorber and the glass $h_{r,ab.v}$ are calculated by the following relationships [13]:

$$h_{r,v-a} = \sigma \varepsilon_v \left(T_v^2 + T_a^2 \right)(T_v + T_a) \qquad (2.20)$$

$$h_{r,abs-v} = \sigma \frac{\left(T_{abs}^2 - T_v^2\right)\left(T_{abs} + T_v\right)}{\frac{1}{\varepsilon_{abs}} + \frac{1}{\varepsilon_v} - 1} \quad (2.21)$$

The convective heat transfer coefficient between the glass and the ambient air and between the bottom of the collector and the ambient air can be determined by the relation of Mac Adams:

$$h_{c,v-a} = h_{wind} = 5.7 + 3.8U_{wind} \quad (2.22)$$

The coefficient of heat transfer by convection between the heat transfer fluid (water) and the absorber is calculated using the Nusselt number determined in [14]:

$$Nu(L) = \left[(h/L)Ra_h^*\right]^{1/2} . \frac{1 - \exp\left[-2.84 + \left(\left((h/L)Ra_h^*\right)^{1/2}\right)^{-2/3}\right]}{\sqrt{24}} \quad (2.23)$$

$$Ra_h^* = \frac{g.\beta.\left(T_{ab} - T_f\right)h^4}{\nu.\Lambda_f.x} \quad (2.24)$$

$$\Lambda_f = \frac{\lambda_f}{\rho \cdot c_f} \quad (2.25)$$

$$h_{c,ab.f} = Nu(L)\frac{\lambda_f}{h} \quad (2.26)$$

The mass flow $\dot{m}_f$ is determined by:

$$\dot{m}_f = \rho_f \cdot A \cdot U_f \quad (2.27)$$

In natural convection, the fluid velocity cannot be imposed as in forced convection. It is governed by the dilation effect of the water. By considering that the flow is established in the collector, in the hot water pipe and in the cold water pipe and that the velocity in the storage tank is zero, the Bernoulli equation is applied to the different sections of heat transfer fluid.

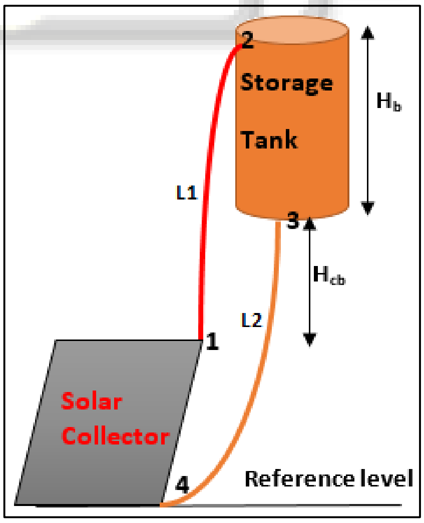

Fig. 2: Diagram representing the Thermosiphon Solar Water Heater System

By neglecting the singular losses of load on a rectilinear stretch 1 -2, we have:

$$P_1^* = P_2^* + 4C_f \frac{L}{D}\rho\frac{U_f^2}{2} \quad (2.28)$$

$$P^* = P + \rho g Z \quad (2.29)$$

C_f is the average coefficient of friction, P is the pressure at altitude Z, U_f is the velocity of the water. By applying this relation successively to the different sections, we have:

$$P_1 + \rho g Z_1 = P_2 + \rho g Z_2 + 4C_f \frac{L_1}{D_1}\rho\frac{U_f^2}{2} \quad (2.30)$$

$$P_2 + \rho_m g Z_2 = P_3 + \rho_m g Z_3 \quad (2.31)$$

$$P_3 + \rho_0 g Z_3 = P_4 + \rho_0 g Z_4 + 4C_f \frac{L_3}{D_3}\rho_0\frac{U_f^2}{2} \quad (2.32)$$

$$P_4 + \rho_m g Z_4 = P_1 + \rho_m g Z_1 + 4C_f \frac{L_4}{D_4}\rho\frac{U_f^2}{2} \quad (2.33)$$

ρ_m is the average density in the sensor and in the storage tank because in these two elements the temperature of the fluid is not constant. In the hot and cold water pipe, the fluid temperature is assumed to be constant because these pipes are well insulated. The pipes are chosen so that the hydraulic diameter D1 = D2 = D4 = D_h. lengths of the lines L_1 and L_2 are taken equal. By doing the successive sum and member to member of the equations, we have after reduction:

$$gZ_{12}(\rho_0 - \rho) = \frac{4U_f^2 C_f}{2D_h}\left[(\rho_0 + \rho)L_1 + \rho_m L_4\right] \quad (2.34)$$

$$\rho_m \approx \frac{\rho_0 + \rho}{2} \Rightarrow \rho_0 + \rho \approx 2\rho_m \quad (2.35)$$

$$\rho_0 - \rho = \rho_0 \beta\left(T_f - T_0\right) \quad (2.36)$$

$$C_f = \frac{16\upsilon}{U_f D_h} \quad (2.37)$$

$$Z_{12} = H_{cb} + H_b \quad (2.38)$$

$$L_1 = Z_{12} + 0.5 \quad (2.39)$$

The speed is then determined by the following expression:

$$U_f = \frac{g.\beta.D_h^2\left(H_{cb} + H_b\right)}{32.\upsilon.\left(2H_{cb} + 2H_b + L_4 + 1\right)}\left(T_f - T_0\right) \quad (2.40)$$

$$D_h = \frac{2l \cdot h}{l + h} \quad (2.41)$$

The length L_4 =L.

B. Thermal balance at the Hot Water Pipe

The hot water, at the outlet of the solar collector is conveyed by a circular pipe to the storage tank which is separated from the solar collector. This water experiences heat loss on its path between the collector and the storage tank. To reduce these heat losses, the pipe has been thermally insulated. In our case, the internal and the external diameter of hot water pipe connecting the collector to the storage tank are respectively 10 mm and 12 mm.

The water in the pipe and the inside face of the pipe being supposed to be at the same temperature, the thermal balance on the hot water gives:

$$\dot{m}_f \cdot c_f \cdot dT_{tu} = -dh_{tu.a}\left(T_{tu} - T_a\right) \quad (2.42)$$

$dh_{tu.a}$ is the overall coefficient of heat loss between the pipe water and the environment for an elementary length dl_{tu} of the pipe. It is the reverse of the overall thermal resistance between water and the environment. This resistance is expressed by:

$$R_{tu.a} = \frac{1}{dl_{tu}} R \tag{2.43}$$

$$R = \frac{\ln\left(\frac{r_{tu}^{ex}}{r_{tu}^{in}}\right)}{2\pi\lambda_{tu}} + \frac{\ln\left(\frac{r_{tu}^{ex} + e_{is.tu}}{r_{tu}^{ex}}\right)}{2\pi\lambda_{is.tu}} + \frac{1}{2\pi h_{wind}\left(r_{tu}^{ex} + e_{is.tu}\right)} \tag{2.44}$$

$$dh_{tu.a} = \frac{1}{R_{tu.a}} \tag{2.45}$$

Replacing the expression of $dh_{tu.a}$ in the previous relation, we have after arrangement:

$$\frac{T_{tu} - T_a}{T_{out.ab} - T_a} = \exp\left(\frac{-L_1}{R \cdot \dot{m}_f \cdot c_f}\right) \tag{2.46}$$

L_1 is the distance traveled by the hot water in the pipe. The temperature T_{tu} corresponding to L_1 equal to the temperature of the water inlet of the storage tank T_{eb}.

C. Thermal Balance at the Water of Storage Tank

The hot water that comes into the storage tank from the top spends sometimes in the latter before ending up to be ready to go back into the solar collector where it will be heated again. During its passage through the storage tank, the hot water loses some of its energy through the walls of the storage tank to the environment. This lost energy decomposes in two, namely the energy lost by the lateral surface of the tank and that lost by the two bases. The latter is negligible in front of the 1st for the following reasons:

- Its exchange surface is weak; which makes its value lower in front of the 1st
- The practical conditions of installation of the storage tank such as the installation of the balloon on a support, the surface of the base which is tangent to the direction of the wind, make even lower its value.

Considering that the variation of the temperature in the storage tank is linear and according to the height of the tank, the thermal balance of the hot water between the inlet and the outlet of the storage tank gives:

$$\dot{m}_f \cdot c_f \cdot dT_b = -d\phi_b \tag{2.47}$$

$d\phi_b$ is the elemental heat exchanged with the outside on a height dh_b and can be expressed by:

$$d\phi_b = \frac{T_b - T_a}{R_{b.a}} dh_b \tag{2.48}$$

$$R_{b.a} = a + b + c \tag{2.49}$$

$$a = \frac{\ln\left(\frac{r_b + e_b}{r_b}\right)}{2\pi\lambda_b} \tag{2.50}$$

$$b = \frac{\ln\left(\frac{r_b + e_b + e_{is.tu}}{r_b + e_b}\right)}{2\pi\lambda_{is.b}} \tag{2.51}$$

$$c = \frac{1}{2\pi h_{wind}\left(r_b + e_b + e_{is.tu}\right)} \tag{5.52}$$

The resolution of (2.47) leads to the following relation:

$$\frac{T_b - T_a}{T_{in.b} - T_a} = \exp\left(\frac{-h_b}{R_{b.a} \cdot \dot{m}_f \cdot c_f}\right) \tag{2.53}$$

To maintain the stratification within the storage tank, it is recommended a tank height to tank diameter ratio greater than two [15-16]. For this study the volume of the storage tank is 90 liters for the 1 m^2 of the collector. The height of the storage tank is 0.9 m and the diameter 0.36 m.

D. Thermal Balance at the Cold Water Pipe

The water that has been heated, which is now at the bottom of the storage tank returns to the solar collector to be warmed as long as it has the ability to remove energy from the absorber. But before going to the collector, its passage in the cold water pipe connecting the bottom of the storage tank to the solar collector causes it to lose some calories and still drop its temperature. The mathematical model explaining the evolution of the temperature in this pipe is the same in the case of the hot water pipe that connects the solar collector to the top of the storage tank. The two pipes being placed under the same conditions, the heat exchange coefficients are the same. It should be noted that the length of the cold pipe and that of the hot pipe are not the same. The evolution of the temperature is expressed by:

$$\frac{T_{tu'} - T_a}{T_{out.b} - T_a} = \exp\left(\frac{-L_1}{R \cdot \dot{m}_f \cdot c_f}\right) \tag{2.54}$$

III. Entransy Balance on the Solar Collector

The performance of a solar thermal collector is conditioned by the following principal aspects:

- The power of the solar radiation captured by the absorbing surface exposed to the sun;
- The heat flux ceded to the heat transfer fluid;
- Heat losses by convection-conduction-radiation between the absorber and the external environment and those by conduction through the absorber;
- The pressure losses of the friction between the heat transfer fluid and the absorber.

Like an energy or entropy balance, the entransy balance takes into account these different aspects. There are two types of entransy namely the entransy of heat transfers and the entransy of loads losses. As we are in natural convection, the entransy of loads losses is neglected.

This balance on the absorber is expressed by the following relation [17]:

$$q_{ab}.T_{ab} - h_p\left(T_{ab} - T_a\right)T_{ab} - \nabla.\left(e_{ab}.Q.T_{ab}\right) + e_{ab}Q.\nabla T_{ab} = 0 \tag{3.1}$$

Where Q is the heat flux density expressed as:

$$Q = -\lambda_{ab}.\nabla T_{ab} \tag{3.2}$$

T_{ab} and T_a are respectively absorber and ambient temperatures. q_{ab} is the power absorbed by the absorber. e_{ab} and λ_{ab} are respectively the thickness and the thermal conductivity of the absorber, h_p is the overall Coefficient of Heat loss. In (3.1), the first term represents the entransy contribution by the solar flux, the second term represents the entransy produced by heat losses between the absorber and the exterior environment, the third term is the entransy transferred to the working fluid by convection, and the fourth term represents the entransy produced by conduction through the absorber.

By integrating over the whole domain, the Gauss formula gives:

$$\iint_{\Omega}\left(q_{ab}.T_{ab}-h_p\left(T_{ab}-T_a\right)T_{ab}-e_{ab}.\lambda_{ab}.\nabla T_{ab}.\nabla T_{ab}\right)dS = \oint_{\Gamma}\left(e_{ab}Q.\nabla T_{ab}\right).ndS \qquad (3.3)$$

This équation can be writen again by:

$$\iint_{\Omega}\left(q_{ab}.T_{ab}\right)dS-\oint_{\Gamma}\left(e_{ab}Q.\nabla T_{ab}\right).ndS = \iint_{\Omega}\left(h_p\left(T_{ab}-T_a\right)T_{ab}+e_{ab}.\lambda_{ab}.\nabla T_{ab}.\nabla T_{ab}\right)dS \qquad (3.4)$$

The first term to the left of (3.4) is the entransy provided by the solar flux on the absorber and the second term on the left is the entransy ceded to the heat transfer fluid. The term to the right is the entransy destroyed by the heat losses in the collector.

So that the collector performance is high, you must minimize the entransy destruction [18, 19]. With (3.4), the entransy destruction E_{Td} can express.

$$E_{Td} = \iint_{\Omega}\left(h_p\left(T_{ab}-T_a\right)T_{ab}+e_{ab}.\lambda_{ab}.\nabla T_{ab}.\nabla T_{ab}\right)dS \qquad (3.5)$$

or

$$E_{Td} = \iint_{\Omega}\left(q_{ab}.T_{ab}\right)dS-\oint_{\Gamma}\left(e_{ab}Q.\nabla T_{ab}\right)n.dS \qquad (3.6)$$

Entransy ceded to the working fluid E_f can be expressed by [4, 20]:

$$E_f = \oint_{\Gamma}\left(e_{ab}Q.\nabla T_{ab}\right).ndS = \frac{1}{2}\dot{m}_f.c_f\left(T_{out}^2-T_{in}^2\right) \qquad (3.7)$$

The entransy brought by the solar flux E_s, after integration gives:

$$E_s = q_{ab}.S.T_{ab} \qquad (3.8)$$

In summary:

$$E_{Td} = q_{ab}.S.T_{ab}-\frac{1}{2}q_u.\left(T_{out}+T_{in}\right) \qquad (3.9)$$

Equation (3.9) represents the entransy destruction by heat loss. q_{ab} is determined by considering the effect of the fins on the absorption capacity of the absorber. Indeed, these fins which are placed on the absorber gives it the ability to behave like a black body; Because the incident rays which reach the absorber undergo several reflections before coming out. The power q_{ab} after n reflections can be determined by [8]:

$$q_{ab} = G.\tau_v\left[\alpha_{ab}.\left(1+\sum_{n=1}^{\infty}\rho_{ab}^n\right)\right] \qquad (3.10)$$

G is the global solar radiation, τ_v is the glass transmittance. α_{ab} and ρ_{ab} are respectively the absorber absorption and reflection coefficients. After two reflections, q_{ab} become:

$$q_{ab} = G.\tau_v \qquad (3.11)$$

The number of thermal entransy destruction E_{Td}^* is the ratio of the thermal entransy destruction by the total entransy in the collector, that is to say the entransy brought by the solar radiation.

$$E_{Td}^* = \frac{E_{Td}}{E_s} \qquad (3.12)$$

IV. METHOD & MEANS OF RESOLUTION

The different results presented come from a written program and implemented in the Matlab version 2010a environment. The method used to solve the equation system giving the temperatures of the glass, the absorber and the outlet of the water is the Newton-Raphson method. The time step used is 15 minutes. This step corresponds to the time step used by the online PVgis software by which the sunlight data was collected. Values from this program are carried in the Excel 2013 software where the different curves have been plotted.

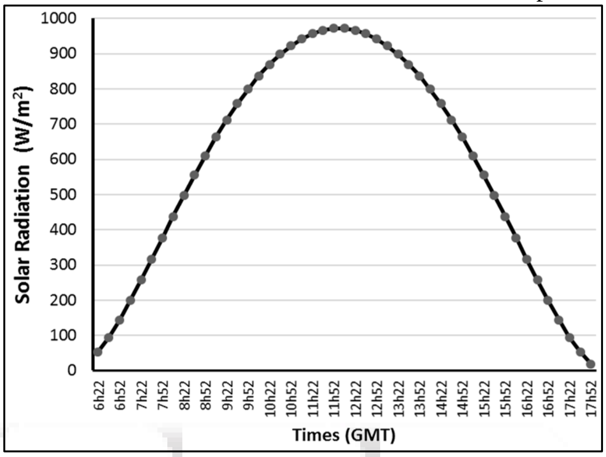

Fig. 3: Solar radiation on Abomey-Calavi

The radiation G considered is that of the month of September. Its evolution according to GMT time is shown in Fig.3.

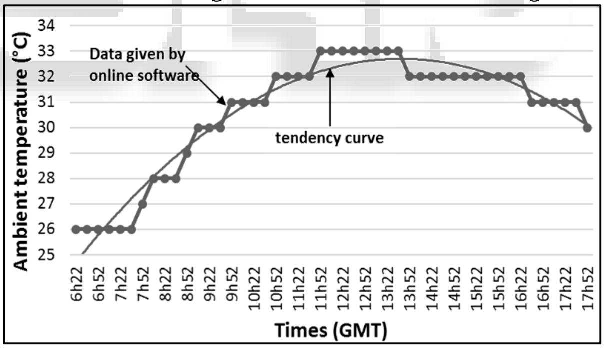

Fig. 4: Ambient temperature in Abomey-Calavi

The ambient temperature also varies during the day and is shown in Fig.4.

V. FIXED PARAMETERS DURING THE SIMULATION

The fixed parameters during the simulation of the results are:

- The capture area S fixed at 1 m^2;
- The thickness of the absorber equals to 0.0015 m;
- The insulation used is glass wool;
- The thickness of the insulation is 0.1 m;
- The wind speed equals to 2 m / s;
- The absorption coefficient of the absorber set at 0.9;
- The reflection coefficient of the absorber set at 0.1
- The reflection coefficient of the glass fixed at 0.05
- The absorption coefficient of the glass is 0.05
- The transmittance of the glass fixed at 0.9

VI. VALIDATION OF RESULTS

In most of the work done on solar water heaters, the rate of thermal efficiency as a function of $\frac{h_p(T_{ab}-T_a)}{G}$ is often plotted.

In order to see the conformity of the results obtained in this work, the variation of the thermal efficiency of the sensor brought back to a conventional flat plat solar collector as a function of $\frac{h_p(T_{ab}-T_a)}{G}$ is represented.

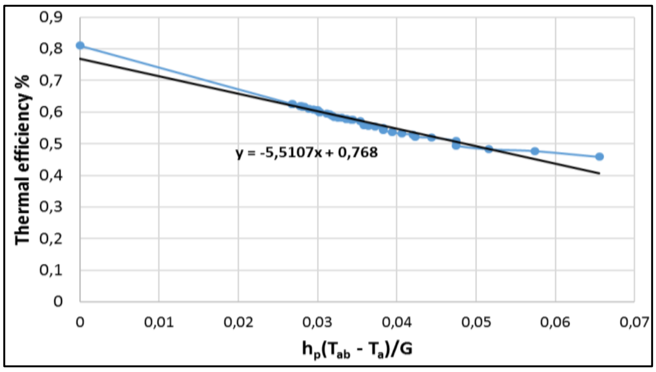

Fig. 5: Evolution of the Thermal Efficiency According to $\frac{h_p(T_{ab}-T_a)}{G}$

According to the results of this evolution, the regression line of the variation of the thermal efficiency as a function of $\frac{h_p(T_{ab}-T_a)}{G}$ has for equation:

$$\eta = -5.5107\frac{h_p(T_{ab}-T_a)}{G} + 0.768 \quad (4.1)$$

This result is confirmed by those found experimentally in [21] and [22].

VII. RESULTS, ANALYZES & INTERPRETATIOS

Curves showing the evolution of entransy destroyed in the solar collector are plotted.

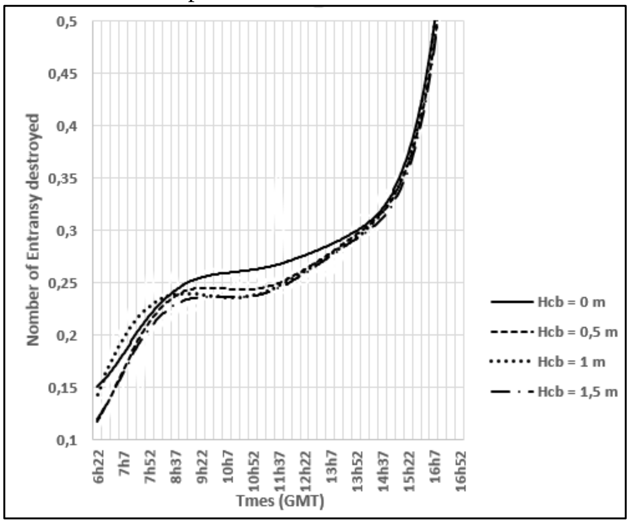

Fig. 6: Evolution of the Number of Entransy Destroyed According to the times

Figure.6 shows the evolution of entransy destroyed at each moment of operation of the solar thermal collector during the day. The curves show that the entransy destroyed varies during the day. It is weak at the beginning of the day and increases from morning to evening. This growth is not uniform. It is strong between 6:22 a.m. and 8:52 a.m. and weak between 8:52 a.m. and 2:52 p.m.. Beyond 2:52 p.m. she believes very strongly. This

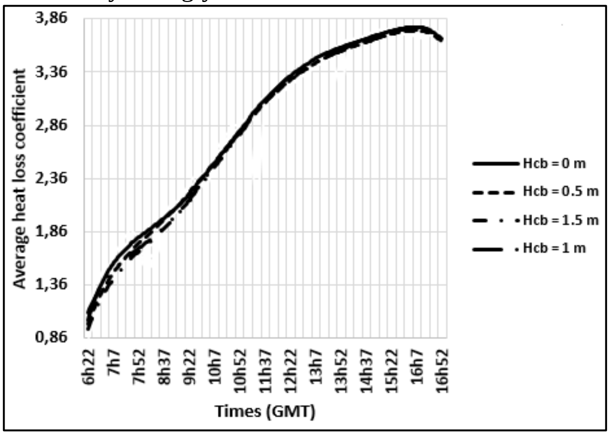

Fig. 7: Evolution of Average Coefficient of Heat Loss According To The Times

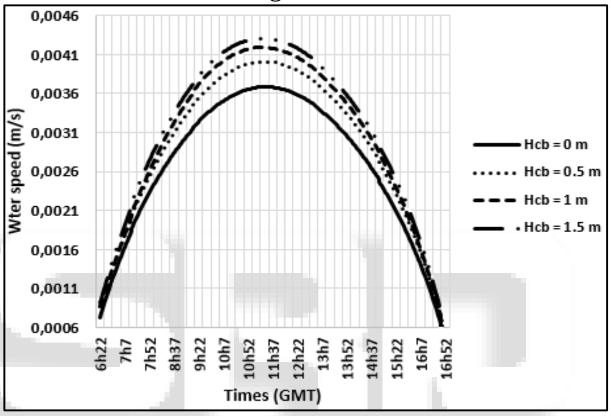

Fig. 8: Evolution of Water Speed According To The Times

Trend of the destroyed entransy curve can be explained by the appearance of the overall coefficient of heat loss during the day shown in Fig.7. This figure shows that heat loss is low and as the day progresses, heat loss increases. It can also be noted that unlike the case of the heat transfer fluid velocity (Fig.8), which is strongly influenced by the solar radiation reaching the solar collector, the number of entransy destroyed is little influenced. This can be seen by comparing the curves of Fig.3 and Fig.6. The number of entransy destroyed is then much more influenced by the operating conditions and heat exchange between the different parts of the solar collector as in the case of the overall coefficient of heat loss. Moreover, it is generally found by the Fig.6, Fig.7 and Fig.8 that when the height between the high level of the solar collector and the bottom of the storage tank is low, the number of entransy destroyed is high, the overall coefficient of heat loss is high and the speed of the water is low. This observation is visible in the interval of

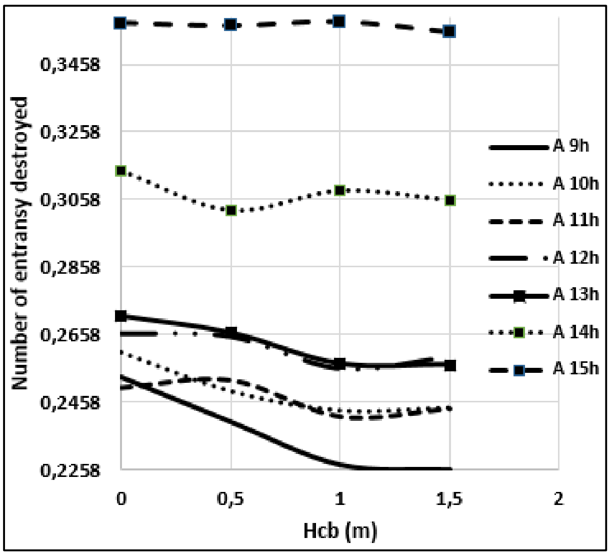

Fig. 9: Evolution of the number of Entransy destroyed according to Hcb

9 a.m. to 3 p.m., interval in which the solar flux is high (Fig.6). The curves in Fig.9 give more details on the influence of the height between the high level of the solar collector and the bottom of the storage tank on the number of entransy destroyed. Through these curves, it can be seen that the entransy destroyed is high when the bottom of the storage tank is at the same altitude as the high level of the solar collector. When this altitude increases, the entransy destroyed generally decreases to reach its minimum value: this is the case of the entransy curves destroyed at 9 a.m., 10 a.m., 11 a.m., 12 o'clock and 1 p.m. when the minimum value of the entransy destroyed is obtained for a height between the high level of the solar collector and the bottom of the storage tank equals to 1.1 m. for 2 p.m. and 3 p.m. curves, the destroyed entransy has two minimum values: one at 0.5 m and the other at 1.5 m.

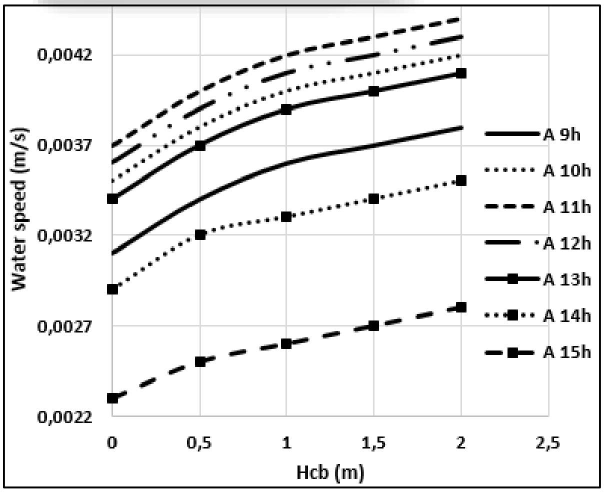

Fig. 10: Evolution of the water speed according to Hcb

The Fig.10 shows the influence of the height between the high level of the solar collector and the bottom of the storage tank on the speed of the water in the sensor. These curves reveal that the speed increases when this height increases. This growth in speed versus height between the top level of the sensor and the bottom of the storage tank is uniform for certain times of the day: this is the case for the 9 a.m., 10 a.m., 11 a.m., 12 o'clock and 1 p.m. speed curves. It is nonuniform at other times of the day: 2 p.m. and 3 p.m. speed curves are rapid growth for height values between 0m and 0.5m and become less fast beyond.

VIII. Conclusion

The theoretical study of the entransy destruction as for the height between the high level of the solar thermal collector and the bottom of the storage tank in a solar water heater system in a day, made the purpose of this study. The results obtained show that the height between the high level of the solar collector and the bottom of the storage tank has a great influence on the performance of the solar water heater system, precisely on the entransy destroyed, which is a loss of energy transfer ability. To minimize the loss of energy transfer ability caused in the thermosiphon solar water heater system through the position of the storage tank relevant to the high level of the solar collector, it is necessary that the high level of the solar collector and the bottom level of the storage tank be separated with height of 1.1 m and that the storage tank in height relevant to the solar collector. We notice that in the solar water heater system, the entransy destroyed is less influenced by the solar radiation coming towards solar collector.

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