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Comparative study of the energy performance of two thermosiphon solar collectors with different water channels: channel in round tubes and channel under flat plate

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***Abstract*— This theoretical study focuses on the comparison of the performance of two solar collectors with the same characteristics except for the different water circulation channels. One of the channels is copper tube of 10 mm diameter and the other is rectangular with 5 mm height. The method used to resolve the system of equations from thermal balances is that of Newton-Raphson. The code of our simulations has been validated. The results of our simulations show that the water temperature and thermal efficiency are high in the collector with rectangular channel while the global coefficient of heat loss and the number of entransy destroyed are higher in the collector equipped with round tube conduit.**

***Keywords*— temperature, absorber, thermal efficiency, entransy destroyed, heat loss**

I. INTRODUCTION

To obtain a good performance of flat plat solar collectors, selective surfaces and surfaces with less easy geometries are used. This impacts the cost of solar water heaters. The cost, performance and technology of solar collector are three concepts that should be the focus of researchers on the field. These notions must be treated with care in order to encourage people to use solar water heaters. Some authors have conducted research in this direction. Among others Zilan Ruken, concerned about the cost and thermal efficiency of flat plat solar collectors, proposed in this research for the Master, a galvanized iron absorber [1]. Aghilas BRAHIMI [2] studied the influence, on the thermal efficiency, of the internal and external parameters of the solar collector. this is also the case with F. Sahnoune et al [3] who studied the influence of external parameters on the thermal efficiency of a solar collector prototype. Gerardo Diag carried out an experimental study of the characterization of aluminum [4]. To increase the performance of the collector by acting on the working fluid channel, other authors suggested a double passage of the working fluid in a modified geometry channel [5-7].

These various researches make important contributions to improving the performance of solar water collectors. In the present work, a simple channel is suggested to allow water to have more contact with the absorber. Thus, the water will flow directly under the absorber unlike other configurations where it circulates in round tubes spaced between them. The energy performance of this proposed collector is then compared to that of the cylindrical water tube solar collector. The two collectors, which all operate as a thermosyphon, have the same characteristics except that the copper tube water channel has a diameter of 10 mm while in the other case, the height of the rectangular channel is 5 mm.

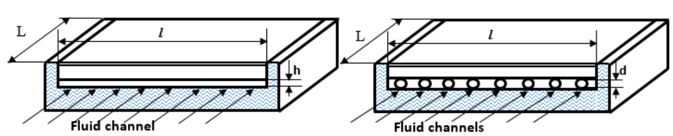

Fig. 1. Sections of solars collectors used; top: rectangular channel – bottom: round tube conduit

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II. THERMAL BALANCES

The thermal balance is made on the different subsystems of solar water heater.

A. Thermal analysis on the Solar water heater

The solar collector consists essentially of the glass, the absorber and the heat transfer fluid. The first principle of thermodynamics is used for the thermal balance on these different elements. Expressions used in these balances are identical for both solar collector are the same except the one expressing the exchange between the absorber and the heat transfer fluid because the both heat transfer fluid channels do not have the same configurations. The expression of the first principle of thermodynamics on each subsystem is:

$$\frac{dE}{dt} = \phi_{ent} - \phi_{sor} + \phi_{in} \tag{1}$$

With E is the internal energy ;

ϕ_{ent} : inflow heat flux ;

ϕ_{sor} : outflow heat flux;

ϕ_{in} : The heat flow generated within the system

1) Glass thermal balance : We can express this balance by:

$$m_v c_v \frac{dT_v}{dt} = A \cdot \alpha_v \cdot G + A \cdot K_1 - A \cdot K_2 \tag{2}$$

With:

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

$K_1 = \left(k_{c,ab.v} + k_{r,ab.v}\right)\left(T_{ab} - T_v\right)$: This is the convection and conduction heat exchanged between the absorber and the glass;

$K_2 = \left(k_{c,v.a} + k_{r,v.a}\right)\left(T_v - T_a\right)$: This is the convection and radiation heat exchanged between the glass and the ambience.

2) Absorber thermal balance at the: We can express this balance by :

$$m_{ab} c_{ab} \frac{dT_{ab}}{dt} = A \cdot \alpha_v \cdot \alpha_{ab} \cdot G - A \cdot K_1 - A \cdot K_3 - A \cdot K_4 \tag{3}$$

For the solar collector of which channel is copper tube, the thermal balance on the absorber gives:

$$m_{ab} c_{ab} \frac{dT_{ab}}{dt} = A \cdot \alpha_v \cdot \alpha_{ab} \cdot G - A \cdot K_1 + -\pi \cdot d \cdot n \cdot L \cdot K_3 - A \cdot K_4 \tag{4}$$

With $A \cdot \alpha_v \cdot \alpha_{ab} \cdot G$ the solar radiation absorbed by the absorber;

$K_3 = k_{ab.f}\left(T_{ab} - T_f\right)$ the heat exchanged between the absorber and the heat transfer fluid;

$K_4 = \left(\dfrac{1}{\dfrac{e_{is}}{\lambda_{is}} + \dfrac{1}{k_{wind}}}\right)\left(T_{ab} - T_a\right)$ the heat lost from the bottom of the collector.

Note that T_{ab} is the average temperature of the absorber.

1) Thermal balance at the heat transfer fluid: Considering that the speed and the temperature of the water vary only in the longitudinal direction y, the heat balance on the water gives:

$$\rho_f c_f S \cdot dL \frac{dT_f}{dt} = dA_f \cdot K_3 - \dot{m}_f \cdot c_f \cdot dT_f \tag{5}$$

S is the water channel section

$$S = l \cdot h \tag{6}$$

$$A_f = 2 \cdot l \cdot L \tag{7}$$

$$dA_f = 2 \cdot l \cdot dL \tag{8}$$

For the solar collector of which channel is copper tube,

$$S = n \cdot \frac{\pi \cdot d^2}{4} \tag{9}$$

$$A_f = \pi \cdot d \cdot n \cdot L \tag{10}$$

$$dA_f = \pi \cdot d \cdot n \cdot dL \tag{11}$$

2) Sommary of balance equations: Equations (12) translate the relationships between the different parameters of the different elements of the solar collector*:*

$$\begin{cases} m_v c_v \dfrac{dT_v}{dt} = A \cdot \alpha_v \cdot G + A \cdot K_1 - A \cdot K_2 \\ m_{ab} c_{ab} \dfrac{dT_{ab}}{dt} = A \cdot \alpha_v \cdot \alpha_{ab} \cdot G - A \cdot K_1 - S \cdot K_3 - S \cdot K_4 \\ \rho_f \cdot c_f \cdot S \cdot dL \cdot \dfrac{dT_f}{dt} = dA_f \cdot K_3 - \dot{m}_f \cdot c_f \cdot dT_f \end{cases} \tag{12}$$

Given that the enthalpy of the collector components varies very weakly according the time we neglect the terms in $m \cdot c \cdot \frac{dT}{dt}$ [8].

Equation system (12) become :

$$\begin{cases} A \cdot \alpha_v \cdot G + A \cdot K_1 - A \cdot K_2 = 0 \\ A \cdot \alpha_v \cdot \alpha_{ab} \cdot G - A \cdot K_1 - S \cdot K_3 - S \cdot K_4 = 0 \\ dA_f \cdot K_3 - \dot{m}_f \cdot c_f \cdot dT_f = 0 \end{cases} \tag{13}$$

The solution of the last equation of the (13), after resolution, is as follows:

$$\frac{T_{ab} - T_f}{T_{ab} - T_0} = \exp\left(-\frac{2 \cdot l \cdot k_{ab.f}}{\dot{m}_f \cdot c_f} L\right) \tag{14}$$

For the solar collector of which channel is copper tube,

$$\frac{T_{ab} - T_f}{T_{ab} - T_0} = \exp\left(-\frac{\pi d \cdot n \cdot k_{ab.f}}{\dot{m}_f \cdot c_f} L\right) \tag{15}$$

the length of the solar collector is L = 1 m. Equations system to be solved is then written:

$$\begin{cases} A\cdot\alpha_v\cdot G + A\cdot K_1 - A\cdot K_2 = 0 \\ A\cdot\alpha_v\cdot\alpha_{ab}\cdot G - A\cdot K_1 - A\cdot K_3 - A\cdot K_4 = 0 \\ \dfrac{T_{ab}-T_{out.f}}{T_{ab}-T_0} - \exp\left(-\dfrac{2\cdot l\cdot k_{ab.f}}{\dot{m}_f\cdot c_f}L\right) = 0 \end{cases} \quad (16)$$

Or
$$\begin{cases} A\cdot\alpha_v\cdot G + A\cdot K_1 - A\cdot K_2 = 0 \\ A\cdot\alpha_v\cdot\alpha_{ab}\cdot G - A\cdot K_1 - A\cdot K_3 - A\cdot K_4 = 0 \\ \dfrac{T_{ab}-T_{out.f}}{T_{ab}-T_0} - \exp\left(-\dfrac{\pi d\cdot n\cdot k_{ab.f}}{\dot{m}_f\cdot c_f}L\right) = 0 \end{cases} \quad (17)$$

for the solar collector with tube channel.

The heat transfer coefficient by convection between the absorber and the glass, $k_{c,ab.v}$, is a function of the characteristic dimension y and of the configuration of the absorber.

$$k_{c,ab.v} = Nu\frac{\lambda_a}{y} \quad (18)$$

With *Nu* the Nusselt number. for an inclination i from the horizontal, *Nu* is given by Droptein relationship [9]:

$$Nu = Nu_i = 0.060 - 0.017\left(\frac{i}{90}\right)Gr_a^{0.33} \quad (19)$$

$$Gr_a = \frac{g.\beta.(T_{ab}-T_v)\,y^3}{\nu_a^2} \quad (20)$$

The heat transfer coefficient by radiation between the ambience and the glass $k_{r,v.a}$ and the one between the glass and the absorber $k_{r,ab.v}$ are expressed by the following relationships [10]:

$$k_{r,v.a} = \sigma\varepsilon_v\left(T_v^2+T_a^2\right)\left(T_v+T_a\right) \quad (21)$$

$$k_{r,ab.v} = \sigma\frac{\left(T_{ab}^2-T_v^2\right)\left(T_{ab}+T_v\right)}{\dfrac{1}{\varepsilon_{ab}}+\dfrac{1}{\varepsilon_v}-1} \quad (22)$$

The heat transfer coefficient by convection between the ambient air and the glass and between the ambient air and the bottom of the collector can be calculated by the Hottel and Woertz relation [9]:

$$k_{c,v-a} = 5.67 + 3.86U_a \quad (23)$$

The heat transfer coefficient by convection between the absorber and the heat transfer fluid $k_{ab.f}$ is determined using the Nusselt number given by:

- Bar-Cohen and Rohsenow [11] for the solar collector with rectangular channel:

$$Nu_h = \left[\frac{144}{\left(Ra_h\,h/L\right)^2} + \frac{2,87}{\left(Ra_h\,h/L\right)^{1/2}}\right]^{-1/2} \quad (24)$$

$$Ra_h = \frac{\beta g\cos\left(\dfrac{\pi}{2}-i\right)\left(T_{ab}-T_{in.f}\right)h^3}{\nu.\Lambda_f} \quad (25)$$

$$\Lambda_f = \frac{\lambda_f}{\rho\cdot c_f} \quad (26)$$

$$k_{ab.f} = Nu_h\frac{\lambda_f}{h} \quad (27)$$

- For the solar collector of which channel is copper tube [12]:

$$Nu = 0,59Ra^{1/4} \quad (28)$$

$$Ra = \frac{\beta g\cos\left(\dfrac{\pi}{2}-i\right)\left(T_{ab}-T_{in.f}\right)d^3}{\nu.\Lambda_f} \quad (29)$$

$$k_{ab.f} = Nu\frac{\lambda_f}{d} \quad (30)$$

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

$$\dot{m}_f = \rho_f\cdot A\cdot U_f \quad (31)$$

For thermosiphon solar collector, the speed of the fluid cannot be imposed because it is generated by the expansion effects of the water. When the flow is established in the collector, in the hot water and the cold water pipes and considering that the water speed in the storage tank is zero, the Bernoulli equation is applied to the different sections of water.

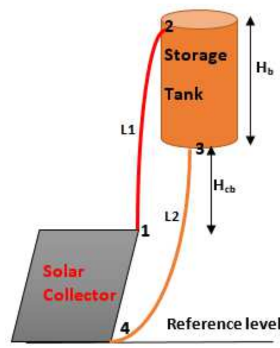

Fig. 2. Thermosiphon solar collector representation

Considering that the singular head losses on section 1 -2 are negligible, the speed is then expressed by [13]:

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

$$D_h = \frac{2l\cdot h}{l+h} \quad (33)$$

The length L_4 =L.

The temperatures T_f and T_0 are those obtained respectively at the inlet (point 4) and at the outlet (point 1) of the sensor. During heating in a day, a quantity of water considered can pass through the collector several times. This operating principle shows that the mass of water at the outlet of the collector (point 1) passes through the hot water pipe, the storage tank and the cold water pipe to still end up at the inlet of the collector (point 4) to be reheated. Thus, the states of the water in these different elements influence the state of the water at the input of the collector (point 4). The thermal balances on these different elements (hot water pipe, storage tank and cold water pipe) enable to determine the temperatures at points 2, 3 and 4 of the system [13]

$$\frac{T_2 - T_a}{T_1 - T_a} = \exp\left(\frac{-L_1}{R \cdot \dot{m}_f \cdot c_f}\right) \quad (34)$$

with

$$R = \frac{\ln\left(\frac{r_{tu}^{ex}}{r_{tu}^{in}}\right)}{2\pi\lambda_{tu}} + \frac{\ln\left(\frac{r_{tu}^{ex} + e_{is.tu}}{r_{tu}^{ex}}\right)}{2\pi\lambda_{is.tu}} + \frac{1}{2\pi h_{wind}\left(r_{tu}^{ex} + e_{is.tu}\right)} \quad (35)$$

$$\frac{T_3 - T_a}{T_2 - T_a} = \exp\left(\frac{-h_b}{R_{b.a} \cdot \dot{m}_f \cdot c_f}\right) \quad (36)$$

$$R_{b.a} = a + b + c \quad (37)$$

$$a = \frac{\ln\left(\frac{r_b + e_b}{r_b}\right)}{2\pi\lambda_b} \quad (38)$$

$$b = \frac{\ln\left(\frac{r_b + e_b + e_{is.tu}}{r_b + e_b}\right)}{2\pi\lambda_{is.b}} \quad (39)$$

$$c = \frac{1}{2\pi h_{wind}\left(r_b + e_b + e_{is.tu}\right)} \quad (40)$$

To not degrade the performance of the solar water heater, good stratification must be maintained in the storage tank. For this, it is recommended that the height be greater than double the diameter for cylindrical shaped balloons [14-15]. For the present work, the collector surface is 1 m^2 and the storage tank capacity is 90 liters. The height and diameter of the storage tank are 0.9 m and 0.36 m respectively.

$$\frac{T_4 - T_a}{T_3 - T_a} = \exp\left(\frac{-L_2}{R \cdot \dot{m}_f \cdot c_f}\right) \quad (41)$$

III. Method And Means of Resolution

To solve the equations systems, the Newton-Raphson method was used. Matlab version 2010a is used for programming and implementation with a time step of 15 minutes. The values of the results obtained from the Matlab software are transported to the Excel version 2013 for the plots of the curves.

The changes in solar radiation and the ambient temperature of Abomey-Calavi for the month of September are shown in Figures 3 and 4 respectively.

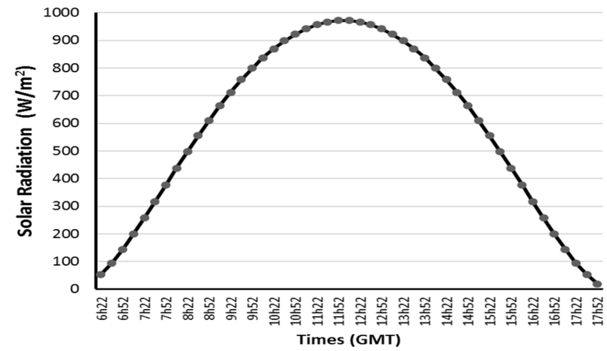

Fig. 3. Solar radiation on Abomey-Calavi.

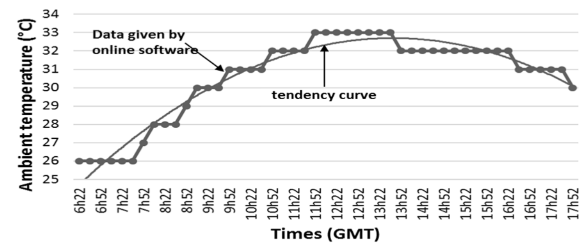

Fig. 4. Ambient temperature in Abomey-Calavi.

IV. Simulation Constants

For the simulation some parameters are fixed :

The absorber thickness is 0.0015 m and the insulation (glass wool) thickness is 0.1 m. The wind speed is 2 m/s. The absorber absorption and reflection coefficients are 0.9 and 0,1 respectively. The glass reflection coefficient and transmittance are 0.05 and 0,9 respectively.

V. Validation of Results

To validate our results, we compared them with those obtained in the literature and mainly those obtained by P. M. E. Coffi and al [16] and A. A. Karaghouli and W. E. Alnaser [17] who plotted the evolution of thermal efficiency as a function of $\frac{(T_{ab} - T_a)}{G}$.

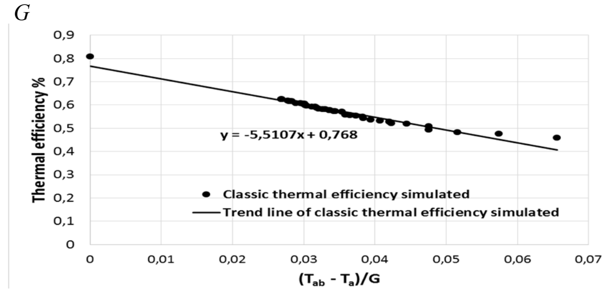

Fig. 5. Evolution of thermal efficiency as a function of $\frac{(T_{ab} - T_a)}{G}$.

The curve of the thermal efficiency as a function of $\frac{(T_{ab} - T_a)}{G}$ plotted within the context of this work for the collector with cylindrical duct (Fig.5) shows that the slope which represents the overall coefficient of heat loss is -5.5107. This value of hp, compared with those found by P. M. E. Coffi and al [16], A. A. Karaghouli and W. E. Alnaser [17] proves a conformity of the results. Indeed by P. M. E. Coffi and al, A. A. Karaghouli and W. E. Alnaser found respectively -5.4749 and -5.45 as values of hp.

VI. Results, Analyzes And Interpretations

The curves of the Fig. 6, 7 and 8 present respectively the evolutions of the temperatures of the glass, the absorber and the water outlet at the collector for the two types of solar collector.

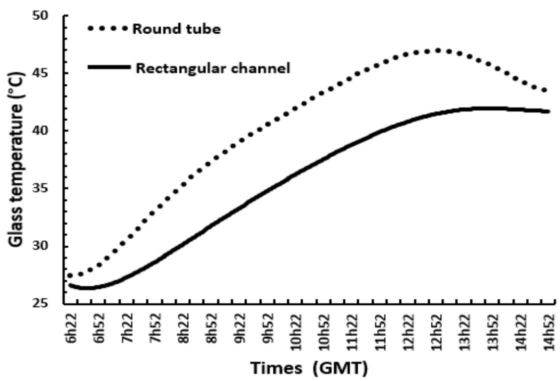

Fig. 6. Evolutions of the glass temperatures.

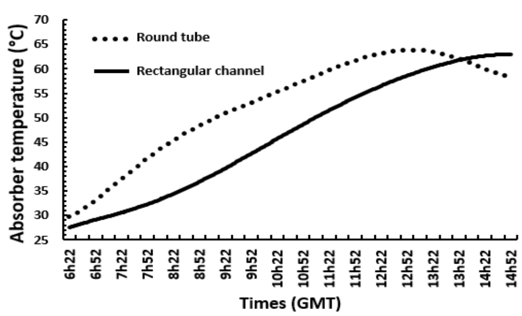

Fig. 7. Evolutions of the absorber temperatures.

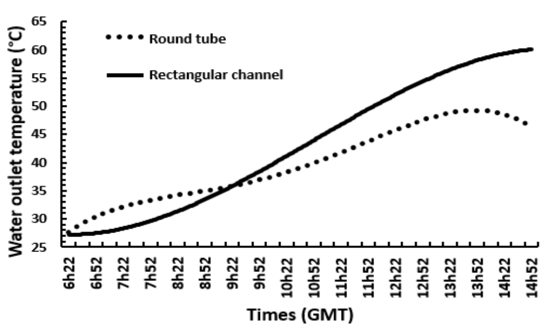

Fig. 8. Evolutions of the water outlet temperatures.

It can easily be seen that the collector with round tube has higher glass and absorber temperatures than those of the collector with rectangular channel. But for the water outlet temperature, it is higher for the collector with rectangular channel. Evolutions of these different temperatures reveal that the amount of heat received by the water is higher at the collector with rectangular channel than at the collector with round tube. This is explained by the fact that at the level of the collector with rectangular channel, the exchange is done directly between the water and the absorber and with much more exchange surface whereas in the case of the other collector the exchange surface is reduced and also the exchange between the water and the absorber is via the round tube. Note also that in the morning until 9:30, the water outlet temperature of the collector with round tube is higher than that of the collector with rectangular channel. We would better understand what happens when we analyze the curves of the Fig. 9 presenting the evolutions of the water mass flow rates at the two types of solar collector.

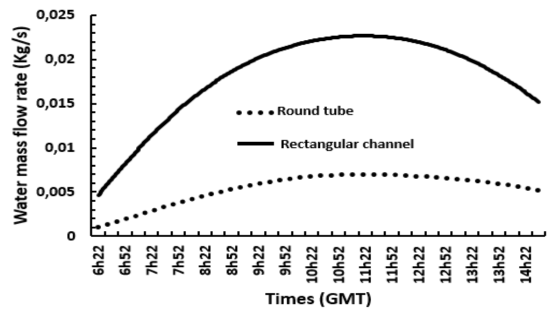

Fig. 9. Evolutions of the water mass flow rates.

The mass flow rate is higher for the collector with rectangular channel. This means that the quantity of water heated per unit of time at this collector is higher, hence its temperature is lower. Since both systems have the same collector surfaces and same storage tank capacites, the water cycle is faster at the collector with rectangular channel. So the number of cycles of water in this collector will be higher and the water will eventually be heated up. After 9:30 am, the water outlet temperature at the collector is higher at the collector with rectangular channel.

The phenomenon observed at the different temperatures results in a great loss of heat at the collector with round tube. Which is confirmed by the curves of the Fig. 10.

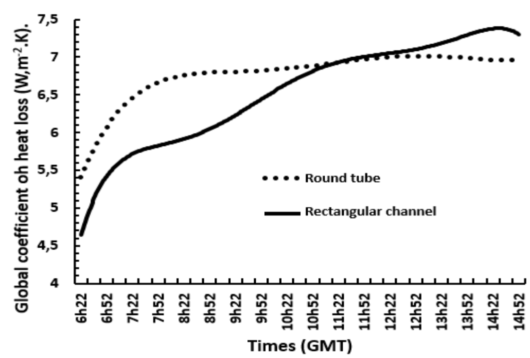

Fig. 10. Global coefficient of heat loss.

At these curves showing the evolution of the global coefficient of heat loss of the two solar collectors, we note that the collector with round tube generates more loss. In the afternoon, it is found that the global coefficient of heat loss at the collector with rectangular channel is higher. This last observation can be explained by the fact that in the afternoon, the water inlet temperature at the collector with rectangular channel is higher. However, for solar collectors, when the heat transfer fluid inlet temperature is high, the thermal efficiency falls, in other words, the losses are greater. However, this fall in efficiency does not make the collector with rectangular channel less efficient because the thermal efficiency of a solar collector is not only a function of the global coefficient of heat loss but also a function of the temperature of the absorber. The thermal efficiency curves shown in Fig. 11 confirm that the performance of the collector with channel is higher than that of the collector with round tube.

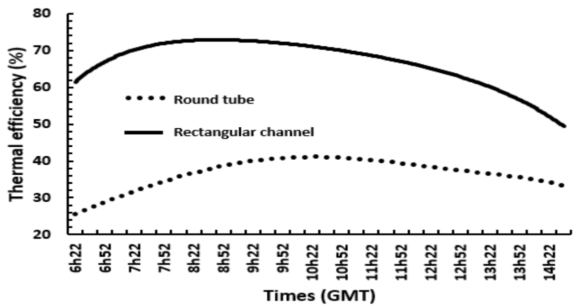

Fig. 11. Thermal efficiency.

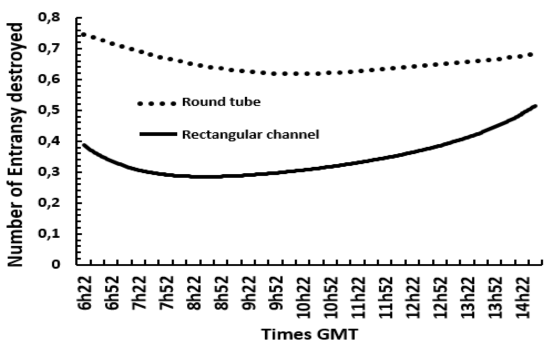

Fig. 12. Evolutions of the entransy destroyed.

On the Fig. 12 it is presented for the two types of solar collector the evolutions of the entransy destroyed which is the loss of the transfer energy ability. It is found that the collector with round tube has a higher number of entransy destroyed all day. It should also be noted that at the level of the collector with rectangular channel, the number of entransy destroyed is higher in the afternoon break than in the morning. This is also supported by a high water inlet temperature at collector with rectangular channel in the afternoon.

VII. Conclusion

The study of the behavior of these two types of solar collector on a typical sunny day enable to assess the daily performance of each of them. Through this theoretical study we see that a solar collector whose water circulates directly in contact with the absorber has a high performance compared to the conventional solar collector in which the coolant channel is a tube. The temperature and the water quantity heated by the rectangular channel is significantly higher than that of the round tube. The useful energy is then higher in the case of the rectangular channel. Thus, the efficiency is higher and the destroyed entransy is lower. But it is necessary to underline the big problem which will be faced for the realization of flat plat solar collectors whose working fluid will circulate directly in contact with the absorber. This problem is that waterproofness in the solar collector.

References

[1] R. Zilan, Optimization of the Geometry and Material of Solar Water Heater. Master, The middle east Technical University, 2001.

[2] A. BRAHIMI, Etude de performances d'un capteur solaire plan à eau. Mémoire de Master 2, Université de lorraine, 2016

[3] F. Sahnoune, M. Belhamel and M. Zelmat, "Etude comparative des performances thermiques d'un prototype de chauffe eau solaire pour deux sites algériens," Revue des Energies Renouvelables Vol. 14, no. 3, pp. 481 – 486, 2011

[4] G. Diag, "DRAFT: Experimental Characterization of Aluminum-Based Minichannel Solar Water Heater," Proceeding of the ASME 2013 Summer Heat Transfer Conference, Minneapolis, USA, copyrigth, pp. 14 – 19, July 2013.

[5] C. D. Ho, R. C. Wang, and T. C. Chen, "Double-Pass Flat-Plate Solar Air Heaters with External Recycle," presented at the Proceedings of European Congress of Chemical Engineering (ECCE-6), Copenhagen, pp. 1–9, 2007.

[6] C. D. Ho and T.-C. Chen, "Collector efficiency of double-pass sheet-and-tube solar water heaters with internal fins attached, 淡江理工學刊, vol. 10, no. 4, pp. 323-334, 2007.

[7] H.-M. Yeh and C.-D. Ho, "Collector Efficiency in Downward-Type Double-Pass Solar Air Heaters with Attached Fins and Operated by External Recycle," Energies, vol. 5, no. 8, pp. 2692–2707, Jul. 2012.

[8] J.A. Duffie and W.A. Beckman, Solar Engineering of Thermal Processes, 4th ed., John Wiley and Sons, 2006.

[9] M. Daguenet, Les Séchoirs solaires: théorie et pratique, Paris: UNESCO, 1985.

[10] K. Soteris, Solar Energy Engineering: Processes and Systems, 1st ed., Elsevier, 2009.

[11] A. Bar-cohen and W.M Rohsenow, "Thermal optimal arrays of cards and fins in natural convection," IEEE Transations on Components, Hybrids, and Manufacturing Technology, vol. CHMT-6, pp. 154-158, June 1983.

[12] J.-L. Battaglia and A. Kusiak, Introduction aux transferts thermiques Cours et exercices corrigés. Paris: Dunod, 2010

[13] M. Anjorin, W. C. Adihou, C. Awanto, A. C. Houngan and E. Sanya, "Effect of Height Difference between Solar Thermal Collector & Storage Tank on the Entransy Destroyed," International Journal for Scientific Research & Development, vol. 6, Issue 07, pp. 720-727, 2018.

[14] K. Hariharan, K. Badrinarayana, S.S. Murthy and M.V.K. Murthy, "Temperature stratification in hot-water storage tanks," Energy, vol. 16, pp. 977-982, 1991.

[15] A.J.N. Khalifa, A.T. Mustafa and F.A Khammas, "Experimental study of temperature stratification in a thermal storage tank in the static mode for different aspect ratios," ARPN Journal of Engineering and Applied Scieces, vol. 6, pp. 53-60, February 2011.

[16] P. M. E. Koffi, H. Y. Andoh, P. Gbaha, S. Touré, and G. Ado, "Theoretical and experimental study of solar water heater with internal exchanger using thermosiphon system," Energy Convers. Manag., vol. 49, no. 8, pp. 2279–2290, Aug. 2008.

[17] A. . Karaghouli and W. . Alnaser, "Experimental study on thermosyphon solar water heater in Bahrain," Renew. Energy, vol. 24, no. 3–4, pp. 389–396, Nov. 2001.