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Use the thermal efficiency criterion to optimize the fins dimensions for the absorber with fins on a compact solar collector.

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Abstract

The present work concerns the optimization of the fins dimensions for the absorber with fins on a compact solar collector. For this optimization, the criterion of thermal efficiency was used, accompanied by the condition of production of the black body on the absorber. This allowed to determine the optimum pitch and height of the fins on the absorber and in the heat transfer fluid, which is water. Based on our calculations 1 m^2 of collector surface, the mathematical models were determined. The curves showing the evolution of the thermal efficiency according the fins dimensions were plotted with the Easyplot software after a simulation in Matlab. The results show that the pitch y must be as small as possible, the pitch p must be as large as possible and the height h = 0.007 m.

Keywords: *Fins; compact solar collector; thermal efficiency; height; pitch.*

1. Introduction

The use of energy sources, which are responsible of green-house gases emissions, is one of the main causes of environmental degradation. Thus, these energy sources must be gradually replaced by the clean and renewable energy sources. Solar energy, is a renewable source of energy; and particularly solar thermal energy, one form of solar energy, is the subject of much scientific research over the past two decades. The system of solar water heaters is one of the solar thermal applications increasingly used in health centers, hotels, industries, sports centers, etc. These systems transform the radiant energy of the sun into heat energy absorbed by water flow through the collector. The efficiency of solar water heaters essentially depends on the performance of the solar collectors [1].

Different technologies of solar collectors used in solar water heaters are classified into three categories named flat plate solar collectors, evacuated tube solar collectors and parabolic concentrators. This study focused only on the flat plate solar collectors.

Moreover much of studies were realized on the design of the flat plate collector with a view to improve their performance. Thus, the effect of the double glass and the triple glass on the flat plate solar collector efficiency was studied [2-4]. Chii-Dong Ho and al [5] realized a theoretical study of the efficiency of a solar collector provided the rectangular fins placing inside the circular water tubes which are placed under the absorber. The tubes function in binomials. Part of the water heated in a first tube is by-pass to be re-heated in the second tube. It is then mixed with cold water which enter in the first tube. This system increase the outlet temperature of the heated water. In the same year, they had studied experimentally and theoretically the thermal performance for the double-pass flat-plate solar air heater with external recycle. The influences of recycle ratio and absorbing plate location on the heat-transfer efficiency enhancement as well as on the power consumption increasing have been also delineated [4]. Y. Raja Sekhar and al [6] evaluate the heat loss coefficients in solar flat plate collector. The absorber of this collector is compact with cylindrical channels. It is partly cylindrical because of the shape of the half cylinder shifted compared to the plane part of the absorber. Ho-Ming Yet and al [7] have investigated theoretically the collector efficiency in a downward-type double-pass external-recycle solar air heater with fins attached on the absorbing plate. They concluded that considerable improvement in collector efficiency is obtainable if the col-lector is equipped with fins and operation is carried out with external recycle. The work of Chong and al [8] on solar water heater using stationary V-trough collector has shown very promising result in overall thermal performance of the solar water heater. Gerardo Diag [9] proposed the compact plate absorber with the rectangular and circular channels in order to increase the thermal exchange performance be-tween absorber and fluid. Suthuraman Ramasamy and al [10] have experimentally studied the thermal performance of a solar collector provided with rectangular and circular fins placing inside the circular water tubes. In this solar col-lector, the water tubes are placed above the absorber. For the rectangular fins, the height and the thickness are respectively 5 mm and 2.5 mm. For the circular fins, the diameter is 2.5 mm. R. Sivakumar and al [11] used the Computational Fluid Dynamics (CFD) studies to simulate a transient heat transfer in an integrated collector-storage type flat plate solar water heater without and with fins, dimples and V-grooves in absorber surface. CFD studies showed an improved heat transfer with increase in depth of fin.

The particularity of the solar collectors studied in this work is on the level of the shape of the absorber. This absorber is provided with right fins with rectangular section on its internal face (face wet in water) and also on its external face (Fig 1).

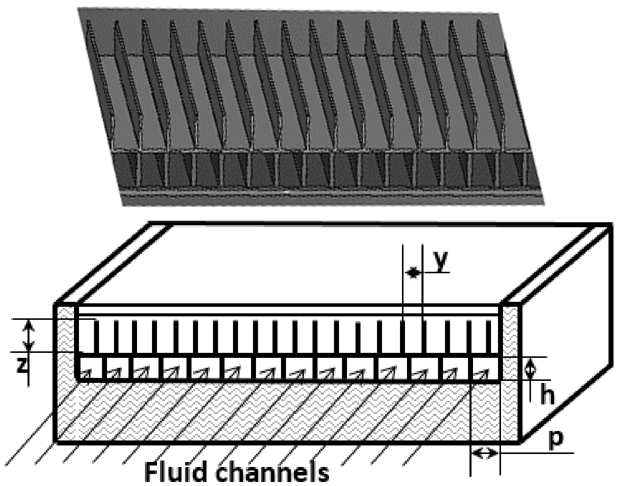

Fig. 1 Physical design: at the top-absorber with fluid flow channels and at the bottom-sectional collector.

This absorber contains not only fins immersed in the water, but also fins in the glass-absorber space. This work consists to optimize the dimensions of these fins in such a way that the incident rays are trapped so that the absorber behaves like a black body and the heat is better transferred to the heat transfer fluid. What generates the lowest the degradation of energy. To Place fins on the absorber in the glass-absorber space is a new approach to increasing the absorption of solar water heater absorbers.

The working fluid is the water that circulates in contact with the rear side of the absorber. The parameters to be optimized are the pitch (y) and the height (z) on the absorber and the pitch (p) and the height (h) in the water.

Mathematical models describing the different kinds of heat transfer involved were developed as well as mathematical models of the efficiency and the constraint of realization of the black body.

2. Thermal efficiency

The thermal efficiency is the ratio of useful flow by solar flow available on the solar collector. The useful flow is the net flow collected by the water. This flow is according to the parameters such as the transmission factor of the glass, the absorption factor of the absorber, the overall coefficient of heat loss, the collector efficiency factor. The thermal efficiency η is calculated by the following relation [12-15].

$$\eta = \frac{q_u}{S \times G} \tag{1}$$

Where G is the global solar irradiance on any plane (W/m^2) and S the Collector area (m^2). The useful power q_u is calculated by:

$$q_u = F'\lfloor q_{abs} - h_p (T_{mf} - T_a) \rfloor \tag{2}$$

q_{abs} is the solar intensity absorbed by the absorber and h_p the Overall heat loss coefficient (W/m^2.K). T_{mf} and T_a are respectively the mean temperature of the water and the ambient temperature. F' is the collector efficiency factor. It is express by:

$$F' = \frac{h_{abs-f}}{h_{abs-f} - h_p} \tag{3}$$

The convection heat transfer coefficient h_{abs-f} between absorber and the working fluid is given by:

$$h_{abs-f} = Nu_f \cdot \frac{\lambda_f}{D_h} \tag{4}$$

λ_f is the water Thermal conductivity and D_h the Hydraulic diameter. The Nusselt number is determined by table 1. For rectangular section, Nu_f and friction factor f are determined by following laws regrouped in table 1. [16-17]:

Table 1: Nusselt number and friction factor for fully developed laminar flow in rectangular section

Rectangular section b a	**Nu (uniform heat flux)**	***f.Re$_{Dh}$***
$\frac{b}{a}=1$	3.61	57
$\frac{b}{a}=1.43$	3.73	59
$\frac{b}{a}=2$	4.12	62
$\frac{b}{a}=3$	4.79	69
$\frac{b}{a}=4$	5.33	73
$\frac{b}{a}=8$	6.49	82
$\frac{b}{a}=\infty$	8.23	96

$$T_{mf}=\frac{T_{out}+T_{in}}{2} \tag{5}$$

Were T_{out} and T_{in} are the outlet and inlet temperature.

Useful power can be write again according to the specific heat at constant pressure C_p and the mass flow rate $\dot{m}_f$ of the water in following form:

$$q_u=\dot{m}_f.c_p(T_{out}-T_{in}) \tag{6}$$

Considering equations (2), (5) and (6), the mean temperature T_{mf} is determined by the expression below:

$$T_{mf}=\frac{q_{abs}.F'+F'.h_p.T_a+2.\dot{m}_f.c_p.T_{in}}{2.\dot{m}_f.c_p+F'.h_p} \tag{7}$$

Overall Coefficient of Heat loss h_p in the equation (2) is the sum of the heat loss coefficients by the upper collector and the lower collector. It is determined by the expression [18]:

$$h_p=\frac{1}{\frac{1}{h_{c,abs-v}+h_{r,abs-v}}+\frac{1}{h_{c,v-a}+h_{r,v-a}}}+\frac{1}{\frac{e_{is}}{\lambda_{is}}+\frac{1}{h_{wind}}} \tag{8}$$

The convection heat transfer coefficient between the glass and the atmosphere $h_{c,v-a}$ and the convection coefficient between the bottom of the collector and the atmosphere h_{wind} is given by Mc Adams [14] where U_{wind} is average speed of the wind.

$$h_{c,v-a}=h_{wind}=5.7+3.8U_{wind} \tag{9}$$

The convection coefficient between the glass and the absorber $h_{c,abs-v}$ is obtained by:

$$h_{c,abs-v}=Nu.\frac{\lambda_{air}}{y} \tag{10}$$

The Nusselt number Nu for an inclination i of the collector relative to the horizontal is given by [19]:

$$Nu=Nu_i=\frac{90-i}{90}.Nu_0+\frac{i}{90}.Nu_{90} \tag{11}$$

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 [20]:

$$Nu_0=\left[\left(\frac{1500}{Ra_y}\right)^2+\left(0.081Ra_y^{0.39}\right)^{-2}\right]^{-\frac{1}{2}} \tag{12}$$

$$Ra_y=\frac{g.\beta.(T_{abs}-T_v)\,y^3}{\nu.\Lambda} \tag{13}$$

Nu_{90} is given by the relationship of Bar-Cohen and Rohsenow [20]

$$Nu_{90}=\left[\frac{576}{(Ra')^2}+\frac{2.873}{\sqrt{Ra'}}\right]^{-\frac{1}{2}} \tag{14}$$

$$Ra'=\frac{g.\beta.(T_{abs}-T_v)\,y^4}{\nu.\Lambda.x} \tag{15}$$

The distance x is equal to 1. T_{abs} and T_v are respectively the temperature of the absorber and the glass. Λ is the Diffusivity, ν the Kinematic viscosity and β the Coefficient of expansion of the water.

Considering the emissivity coefficients ε_{abs} of the absorber and ε_v of the glass, the radiation heat transfer coefficient between the glass and the atmosphere h_(r,v-a) and between the absorber and glass $h_{r,abs-v}$ are given by the equations [18], [21-22]:

$$h_{r,abs-v}=\sigma\frac{\left(T_{abs}^2-T_v^2\right)\left(T_{abs}+T_v\right)}{\frac{1}{\varepsilon_{abs}}+\frac{1}{\varepsilon_v}-1} \tag{16}$$

$$h_{r,v-a}=\sigma\varepsilon_v\left(T_v^2+T_a^2\right)\left(T_v+T_a\right) \tag{17}$$

ε_{abs} and ε_v are respectively the absorber and the glass emissivity. σ is the Planck constant.

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3. Realization condition of black body on absorber

To absorb the totality of radiant energy, the receptor body should behave as a black body. In the solar water heater systems, to have an absorber which has the properties of a black body constitutes an advantageous factor in the performance of the solar collectors. The objective is to achieve a black body on the absorber. For this, it is necessary that the incident rays once reaching the absorber, make several reflections before exiting towards the glass. This will be possible if the height and pitch of the fins on the absorber are well defined. The incident power absorbed q_{abs} by the absorber can be expressed after n reflections according the transmission coefficient of the glass τ_v, the absorption and the reflection coefficients α_{abs} and ρ_{abs} [23].

Before the 1st reflection

$$q_{abs} = G.\tau_v.\alpha_{abs} \quad (18)$$

After the 1st reflection

$$q_{abs} = G.\tau_v.\alpha_{abs} + G.\tau_v.\rho_{abs}.\alpha_{abs} \quad (19)$$

After the 2nd reflection

$$q_{abs} = G.\tau_v.\alpha_{abs} + G.\tau_v.\rho_{abs}.\alpha_{abs} + G.\tau_v.\rho_{abs}^2.\alpha_{abs} \quad (20)$$

After n reflection

$$q_{abs} = G.\tau_v.\alpha_{abs}\begin{pmatrix} 1+\rho_{abs}+\rho_{abs}^2+\rho_{abs}^3+\rho_{abs}^4+ \\ \rho_{abs}^5+...+\rho_{abs}^n \end{pmatrix} \quad (21)$$

Equation (21) can be rewritten by

$$q_{abs} = G.\tau_v\left[\alpha_{abs}\left(1+\sum_{n=1}^{\infty}\rho_{abs}^n\right)\right] \quad (22)$$

The product $\left[\alpha_{abs}\left(1+\sum_{n=1}^{\infty}\rho_{abs}^n\right)\right]$ tends to the value 1 after the 2nd reflection (n=2).

Then

$$q_{abs} = G.\tau_v \quad (23)$$

The path of a sunbeam reached to the absorber is showed on the following figure.

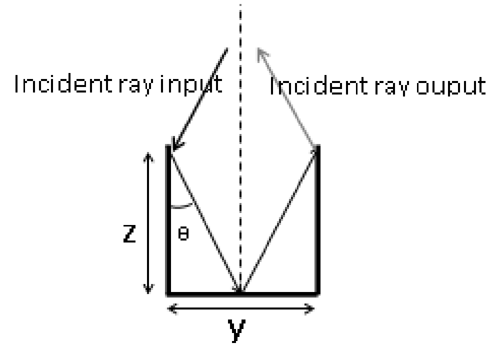

Fig. 2 Browsing the sunlight in the absorber.

The height z and the pitch y of the fins are bound by the following relationship:

$$z = \frac{y}{2\tan\theta} \quad (24)$$

Where θ is equal to the zenith angle of the sun.

4. Hypothesis of simulation

The variables that influence the performance of the flat plate solar collectors are not only spacing and height but also other variables. These variables are:

- The collector area S fixed at 1 m^2 (1m x 1m);
- The global insolation G taken at 800 W/m^2. This variable is the average insolation of the hottest months at Abomey-Calavi in Benin [24];
- The room temperature fixed at 30° C [25];
- The thickness of the absorber equal to 0.0015 m;
- The used thermal insulator is glass wool and this thickness is 0.1 m;
- Wind speed equal to 2 m/s;
- $\alpha_{abs} = 0.9$; $\alpha_v = 0.9$; $\rho_{abs} = 0.1$; $\rho_v = 0.05$; $\tau_v = 0.9$;
- Inclination i is fixed at 15° [12]

All these additional variables were considered while conducting the field work.

5. Results and discussions

The different mathematical models have been programmed in matlab. The curve of entropy generation have been drawn according to the pitch “y” of fins on the absorber, the pitch “p” and the height “h” of fins in working fluid for different values of the working fluid speed. To draw the curve of entropy generation according to one of these three parameters, that means “y”, “p” or “h”, we fix the two others. For instance, to draw the curves of entropy generation according to pitch “p”, we have firstly fixed “h”

to 0.02 m and we consider three values for the pitch “p” (p = h, p = 2h, p = 4h). Secondly, “p” is fixed to 0.08 m and we consider three values of “h” (h =p, h = 1/2p, h = 1/4p). For each value of “p” in the first case or each value of “h” in the second case, we consider three values which are the difference ΔT between the temperature of the absorber and the one of the glass (ΔT = 20°C, ΔT = 30°C, ΔT = 40°C).
The different curves of variation are shown below.
Major headings are to be column centered in a bold font without underline. They need be numbered. "2. Headings and Footnotes" at the top of this paragraph is a major heading.

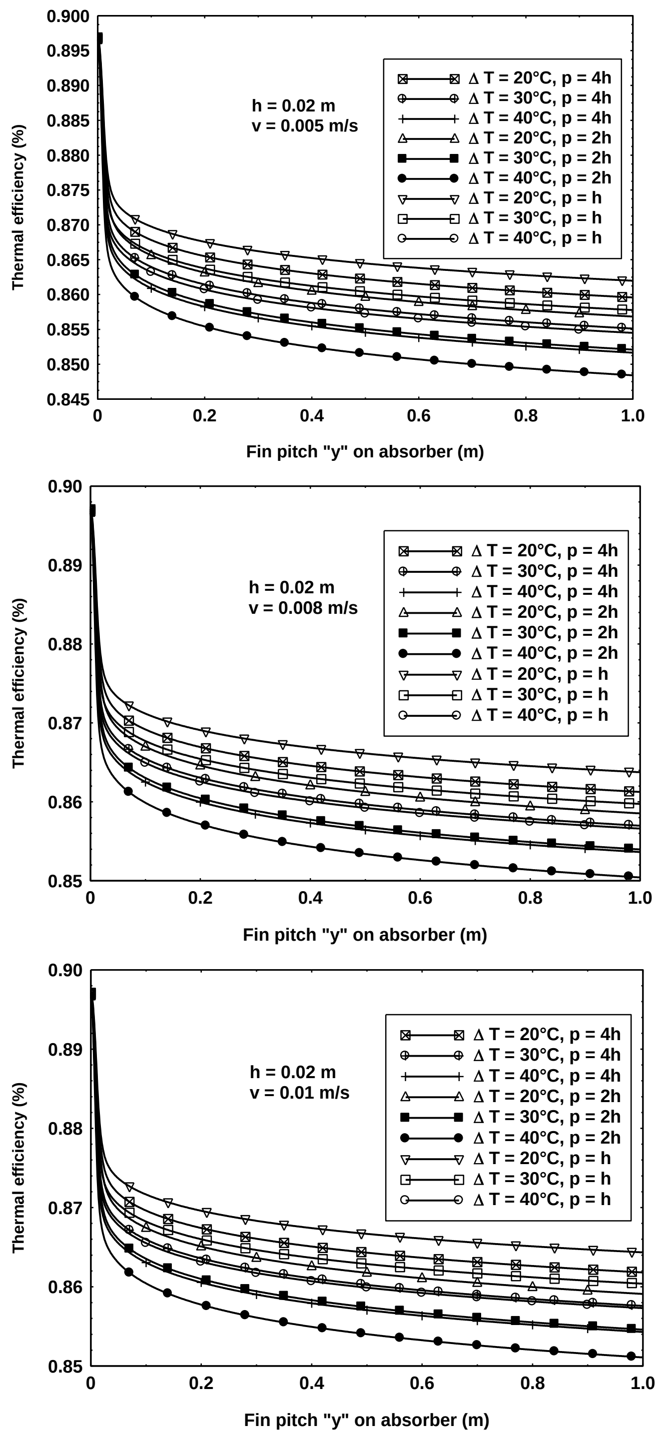

Fig. 3 Variation of thermal efficiency according to the pitch y on absorber for h=0.02 m with different values of water speeds

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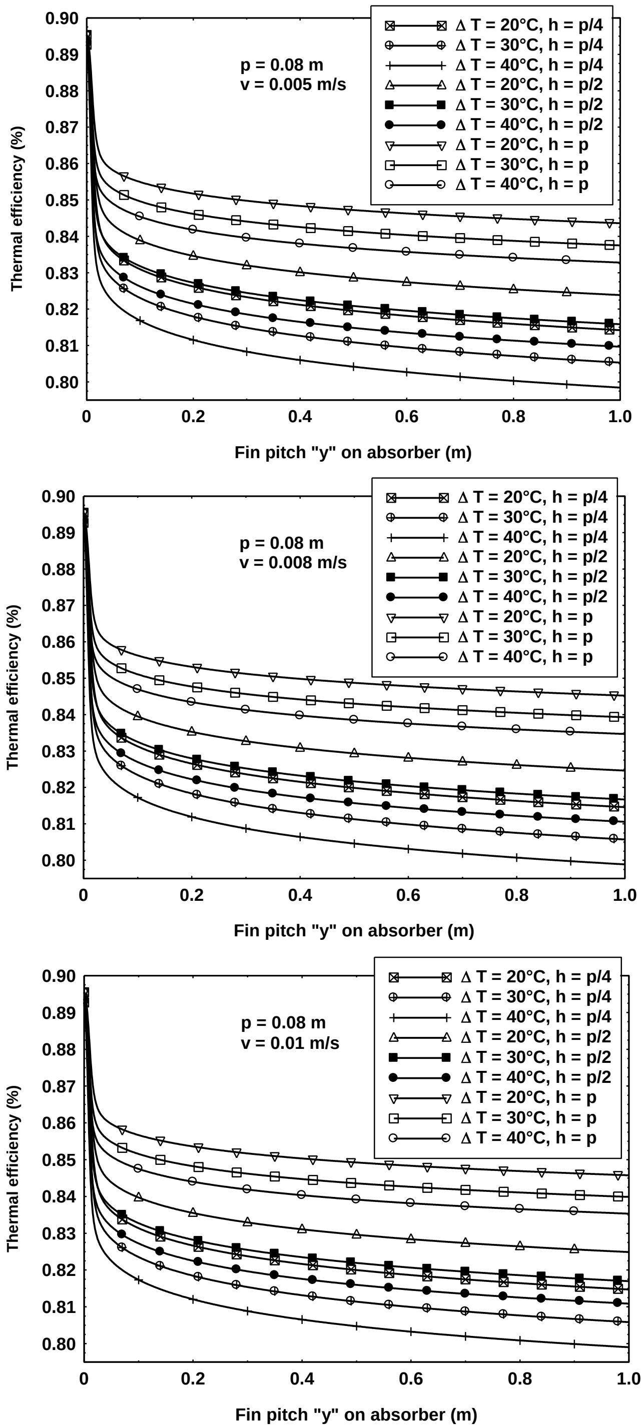

Fig. 4 Variation of thermal efficiency according to the pitch y on absorber for p=0.08 m with different values of water speeds

The variation of thermal efficiency according to the pitch y on absorber is shown on Fig 3 and 4. These curves were plotted for different values of the speed of water, the difference of temperature between the temperature of absorber and the one of glass (ΔT), the height h and pitch p of fins in the water. These results reveal that the thermal efficiency decreases when the fin pitch y on the absorber increases. The greatest value of thermal efficiency is found when the pitch y tends to zero. These curves show that the thermal efficiency decreases fast for pitches y inferior to (lower than) 0.04 m; while it decrease slowly with the pitches greater than 0.04 m. The results show that the effect of fin pitches is negligible whether the pitches superior to 0.04 m. Further-more, the difference of temperature between the temperature of the absorber and the one of the glass has greatly influenced the thermal efficiency. More ΔT is low, high is the thermal efficiency and more ΔT is high, lower is the thermal efficiency. Then the thermal efficiency to ΔT=20°C is higher than that found for ΔT=30°C, and that which is given by the last one is lower than that found for ΔT=40°C. When ΔT is high, heat loss of by the absorber toward the glass is high too; that could give a low thermal efficiency. Moreover, In Fig 3 and 4, for h=0.02 m and for p=0.08 m; we notice that the speed of the working fluid has an effect on the thermal efficiency. When the speed is 0.01 m/s, the thermal efficiency of the system is higher than when the speed is 0.008 m/s. This last shows thermal efficiency that is higher than the one of speed 0.005 m/s. We will discuss this aspect in the following of our analysis. Another remark concerns the pitch p and the height h of fins in the working fluid (water). The thermal efficiency for the pitches p=h, p=2h and p=4h with h=0.02 m are different from each other. When the pitch p is low the thermal efficiency is also low.

The Fig 5 and 6 reveal that for the pitch p between 0 and 0.05 m, the thermal efficiency decreases and reaches its minimal value at p=0.05 m. when p is superior to 0.05 m, thermal efficiency raises according to p. The minimal values obtained at p = 0.05 m and the maximal values obtained at p = 1 m vary according to the pitch y of fins on the absorber, the height h of fins in water, the speed of water and the difference between the temperature of the absorber and the one of the glass (ΔT). From this result, we can say that the pitch p of fins in the water must be as larger as possible. Thus, a canal of water without fins is more considerable than the one with fins. It should be noted also that the effect of the difference of temperature ΔT on the thermal efficiency remains the same than the one in the case of the Fig 3 and 4. If ΔT is low, the thermal efficiency is high.

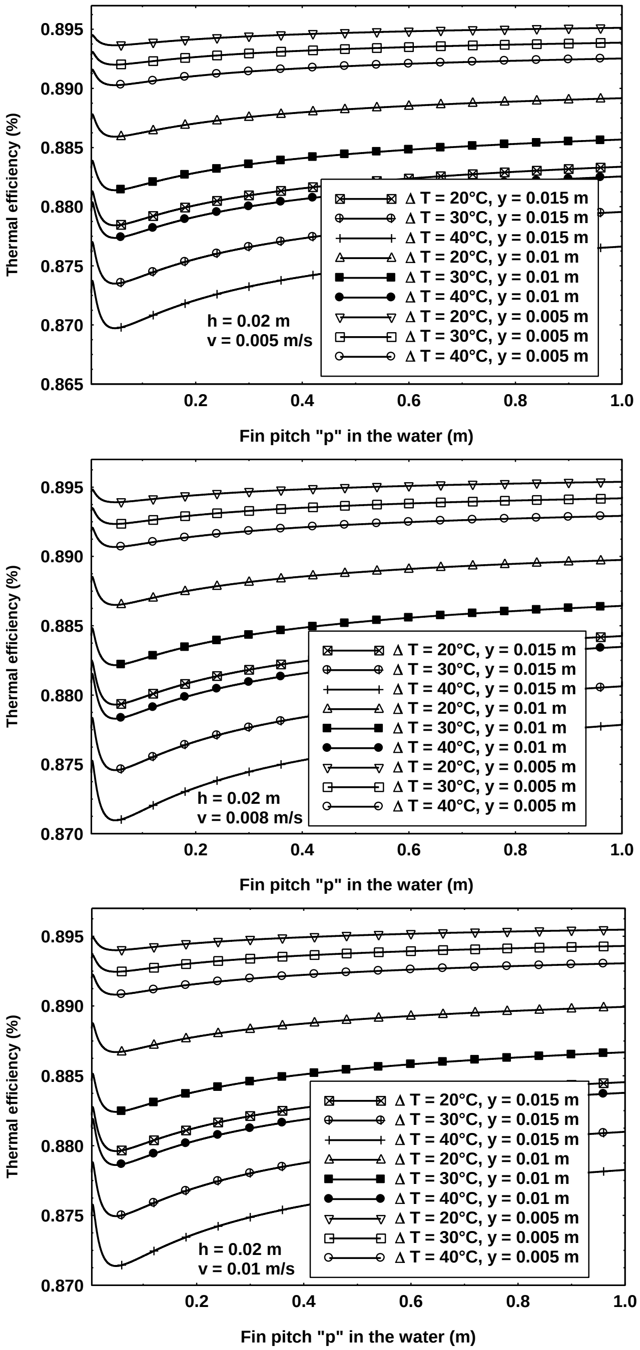

Fig. 5 Variation of thermal efficiency according to the pitch p in the water for h=0.02 m with different values of water speeds

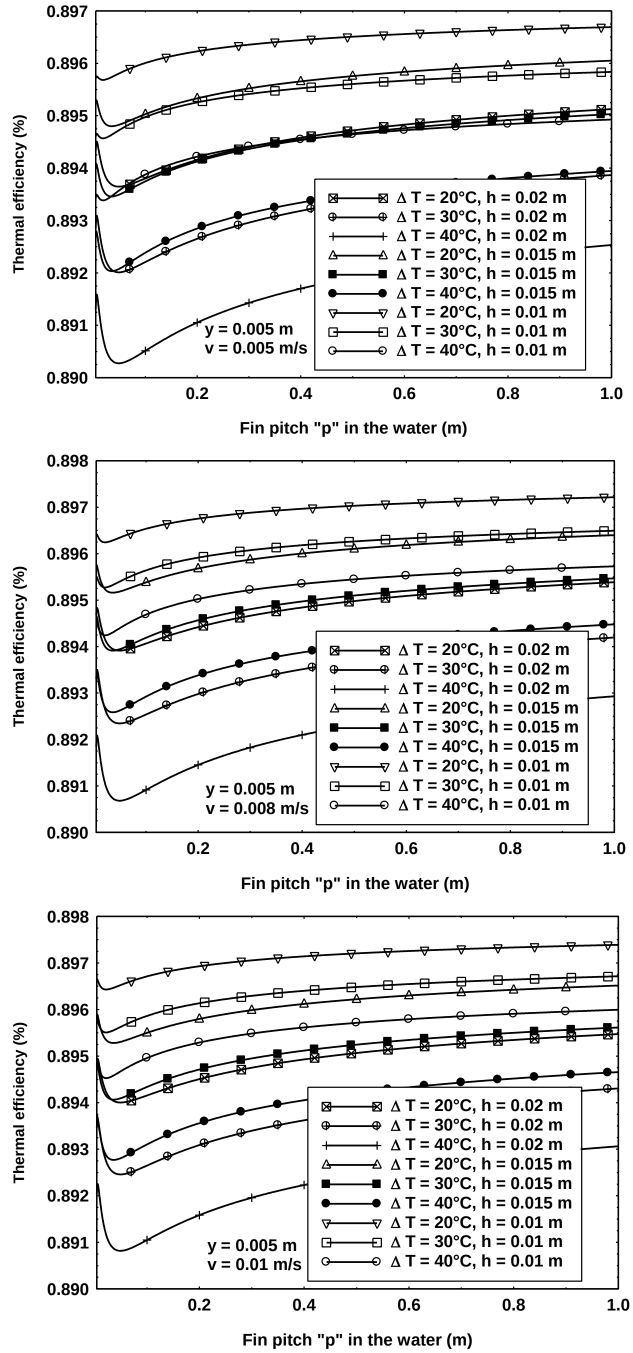

Fig. 6 Variation of thermal efficiency according to the pitch p in the water for y=0.005 m with different values of water speeds

The Fig 7 and 8 below present the evolution of thermal efficiency according to the height h of fins in the water.

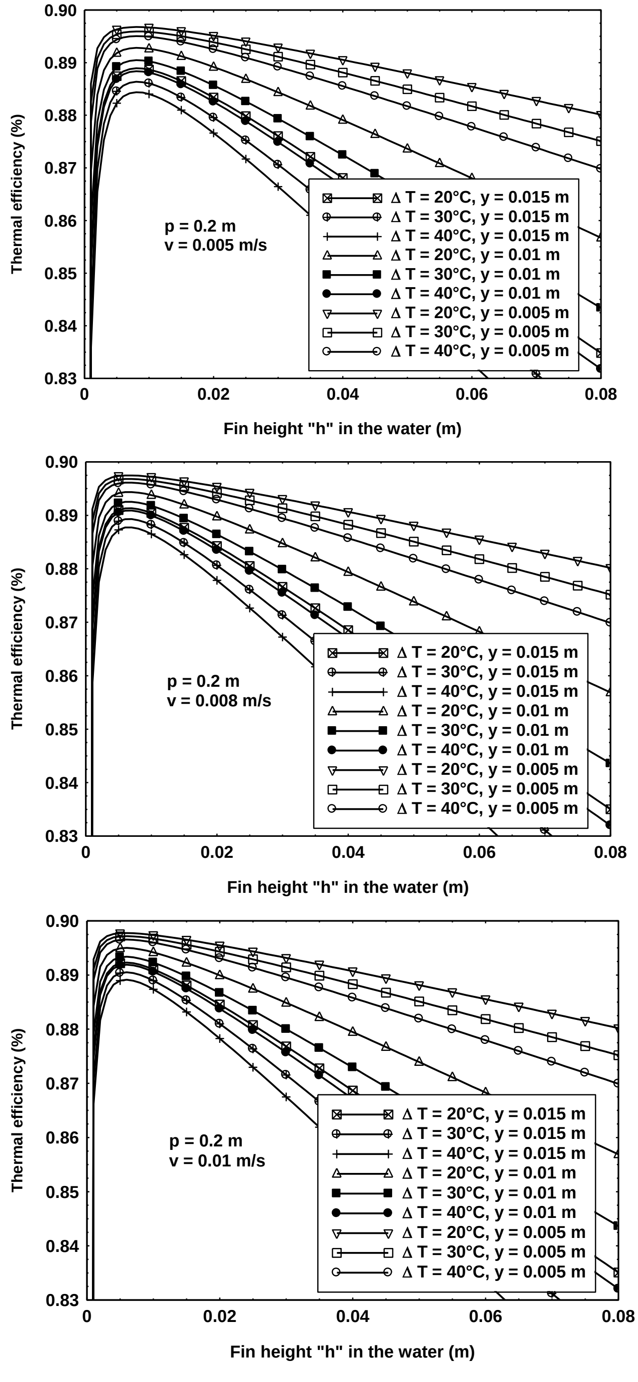

Fig. 7 Variation of thermal efficiency according to the height h in the water for p=0.2 m with different values of water speeds

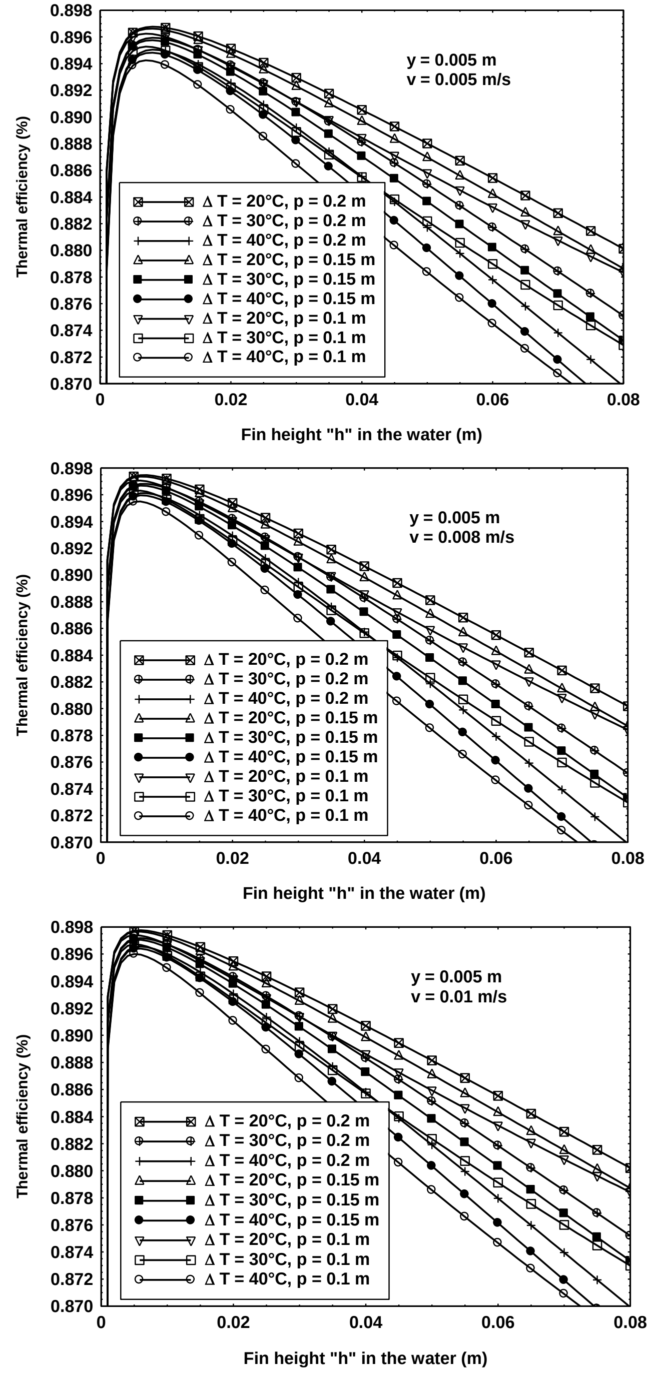

Fig. 8 Variation of thermal efficiency according to the height h in the water for y=0.005 m with different values of water speeds

When the height is between 0 and 0.007 m, the thermal efficiency increases and reaches its maximal value at h=0.007 m. When the height h is greater than 0.007 m, the thermal efficiency decreases according to h. On these different curves, the maximal value of thermal efficiency is noticeable when the height h of fins in the water is 0.007 m. The Fig 7 and 8 shows that the difference of temperature between the temperature of absorber and the one of the glass keeps the same effects on the thermal efficiency than those in the case of the Fig 3, 4, 5 and 6. These different figures show that the influence of the water speed is not negligible. Fig 9 at the bottom shows this influence.

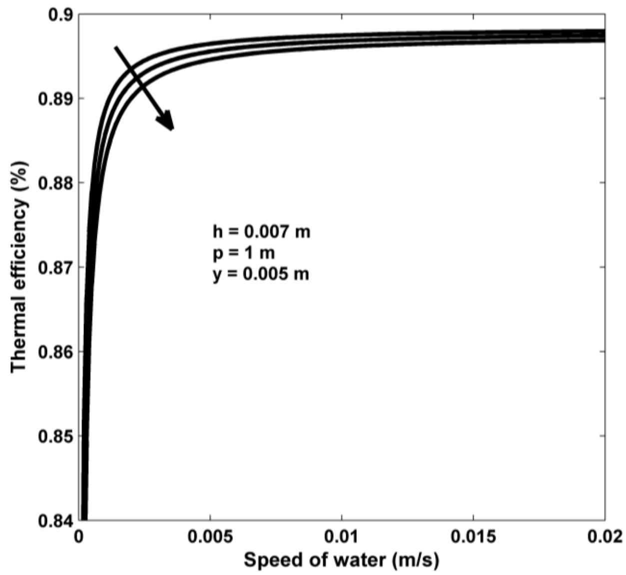

Fig. 9 Variation of thermal efficiency according the speed of water.

In Fig. 9, the variation of the efficiency according to the speed of the water is presented. These curves show that for speeds v inferior to 0.004 m/s, the efficiency increases fast from 0 to 0.896, from 0 to 0.895 and from 0 to 0.8938 respectively for the differences $\Delta T = 20\ °C$, $\Delta T = 30\ °C$ and $\Delta T = 40\ °C$. But for speeds superior to 0.004 m/s, it increases slowly. The results show that the effect of the speed is negligible if the speeds are superior to 0.004 m/s. Also, the speed of the water has a great effect on the outlet temperature and the mass of heated water. This is shown in Fig 10 below.

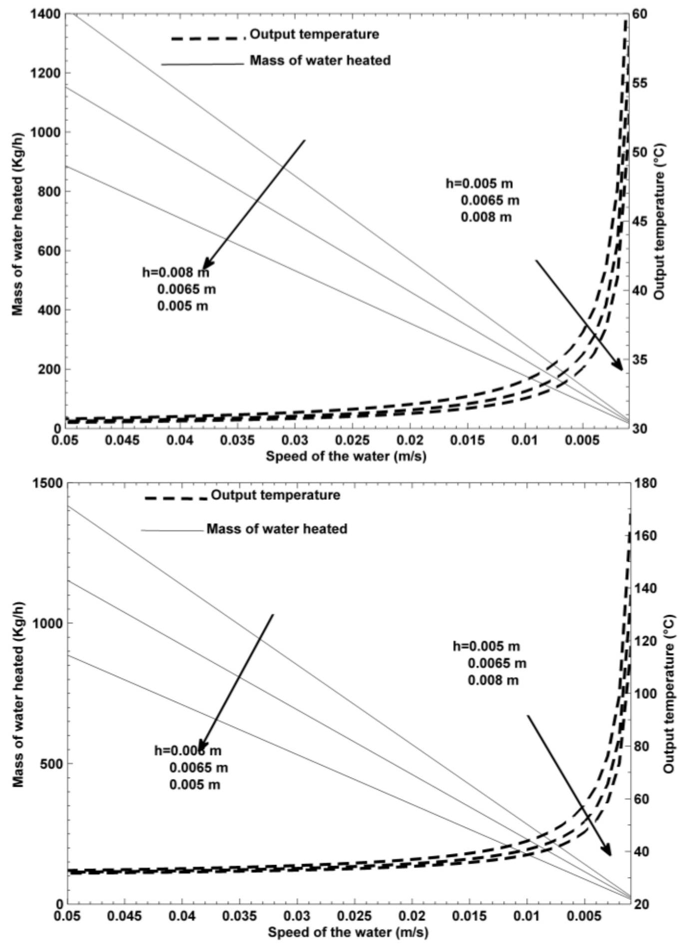

Fig. 10 Variation of the outlet temperature and the mass of water heated in one hour according to the speed of the water. Top figure: absorber length is 1 m. Bottom figure: absorber length is 2 m.

This figure shows that the outlet temperature of the water is high when the velocity of the water is low. In this case the mass of heated water is low. For high water speed, the outlet temperature of the water is low while the mass of heated water is high. This is justified by the fact that a large quantity of water has a high thermal capacity, which makes the variation of the temperature small while a small quantity of water has a small thermal capacity and the variation of its temperature is relatively high. It is also noted that the length of the collector influences the outlet temperature of the water. For a length of 2 m, the output temperatures are higher than those for a collector length of 1 m. This could be explained by the time which the water spends in the col-lector before exiting. For the length of 2 m, the water spends more time in the collector than for the length of 1 m.

6. Conclusion

This work contributes to improving knowledge in the field of solar thermal collectors. The aim of this work was to optimize the height and pitch of the fins on the absorber as well as the height and pitch of the fins in the heat transfer fluid for a compact solar collector with plane absorber by the performance criterion. The analysis of the different results shows that in order to obtain a better efficiency, the pitch y of the fins on the absorber must be less than 0.04 m and be as small as possible. The height of the fins on the absorber must be determined according to the pitch and the zenithal angle of the sun. Moreover, the optimum channel height of the heat transfer fluid is 0.007 m and the use of the fins in the fluid weakens the efficiency. Furthermore, the efficiency is weakened when the difference between the absorber temperature and the glass one is high. The influence of the fluid velocity on the yield is not negligible. If the speed is low, the efficiency is low too and water temperature is high; while at high speeds the efficiency is high too and the water temperature is low.

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