

Optimization of the Fins Dimensions for the Absorber with Fins on a Compact Thermal Solar Collector by Entropy Generation Criterion

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Abstract- This study presents the possibility of adding fins to plate absorbers to improve the performance of the flat plate collector. The objective of this theoretical work is to optimize the dimensions of the fins of the absorbers for a compact solar collector. The performance criterion used is the entropy generation, a global criterion followed by the realization constraint of the black body on the absorber. For that absorber behaves like a black body, the relation between the pitch “y” and the height “z” of fins on the absorber has been determined. The mathematical model of the entropy generation has been determined and simulated according to matlab software. The curves variation of the entropy generation has been drawn according to the pitch “y” of fins of the absorber, the pitch “p” and the height “h” of fins in the working fluid. The results show that the entropy generation is low if the pitch “y” is the lowest, the pitch “p” is raised possible and the height “h” is equal to 0.023 m. the difference between the temperature of the absorber and the one the glass must be low. More this the difference is low, more the optimal speed is low.

Keywords- Fins; solar collector; compact; entropy generation; height; pitch.

1. Introduction

The performance of thermal solar collectors depends on the energy taken by the working fluid. This energy depends on that stored in the absorber. The more the absorbers store the energy, the more effective it is. However, for an absorber to be ideal, it must behave as a black body, which is the most sought throughout the solar collector design. Indeed, at the level of flat plate collector, the performance improvement is sought, taking into account considerations such as the selectivity of the absorber [1], the inclination of the collector, insulation of low and side portions of the collector. Moreover much of studies were realized on the design of the flat plate collector with a view to improve their performance. Thus the effect to double or triple the glass on the output 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 reheated 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 [2]. 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 Chii-Dong Ho [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 collector is equipped with fins and operation is carried out with external recycle. Gerardo Diag [8] proposed the compact plate absorber with the rectangular and circular channels with a view to increase the thermal exchange performance between absorber and fluid. Suthuraman Ramasamy and al [9] 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 collector, 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 [10] 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 show an improved heat transfer with increase in depth of fin.

In this study, it was proposed to increase the surface of the absorber by fins and optimize the spacing and the height of the fins in order to obtain an absorber which behaves as a black body. This absorber contains not only fins plunged in water; but also find in the glass absorber space. This work consists to optimize the dimensions of these fins in such a way that the incident rays be captured so that the absorber behaves like a black body and that the heat transfer between the absorber and the working fluid is better. What generates the lowest degradations of energy. To place fins on the absorber in the glass-absorber space represent a new approach to increase the absorbed flow by the absorbers of solar collector. This optimization is done with the criterion of entropy generation.

2. Absorber of the Solar Collector Studied

For this study, the absorber of the solar collector is provided with right fins with rectangular section (Fig 1). 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.

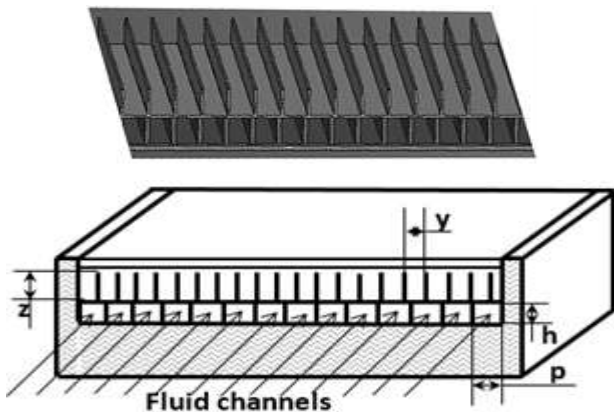


Fig. 1. Physical design: at the top-absorber with fluid flow channels and at the bottom-sectional collector.

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

3. Mathematical Models

3.1. Constraint of realisation of black body at absorber level

The maximum of solar energy is observed at thirteen o'clock at Cotonou [11]. The objective is to achieve a black body on the absorber to have it at its maximum energy. 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. We can express the incident power absorbed q_{abs} by the absorber after n reflections.

Before the 1st reflection:

$$q_{abs} = G\tau_v\alpha_{abs} \tag{1}$$

After the 1st reflection:

$$q_{abs} = G\tau_v\alpha_{abs} + G\tau_v\rho_{abs}\alpha_{abs} \tag{2}$$

After the 2nd reflection:

$$q_{abs} = G\tau_v\alpha_{abs} + G\tau_v\rho_{abs}\alpha_{abs} + G\tau_v\rho_{abs}^2\alpha_{abs} \tag{3}$$

After n reflection:

$$q_{abs} = G\tau_v\alpha_{abs}(1 + \rho_{abs} + \rho_{abs}^2 + \rho_{abs}^3 + \rho_{abs}^4 + \rho_{abs}^5 + \dots + \rho_{abs}^n) \tag{4}$$

Equation (4) can be rewritten by

$$q_{abs} = G\tau_v[\alpha_{abs}(1 + \sum_{n=1}^{\infty} \rho_{abs}^n)] \tag{5}$$

The product

$$[\alpha_{abs}(1 + \sum_{n=1}^{\infty} \rho_{abs}^n)] \tag{6}$$

tends to the value 1 after the 2nd reflection (n=2). Then

$$q_{abs} = G\tau_v \tag{7}$$

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

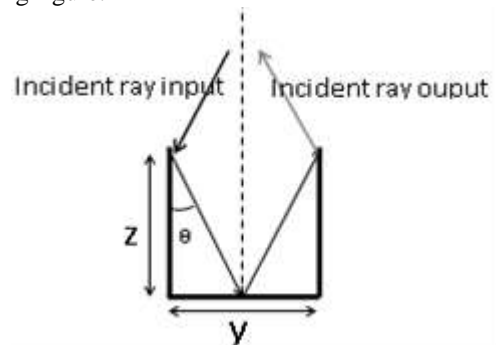


Fig. 2. Browsing the sunlight in the absorber.

Height z and the pitch y of the fins are bound by the following relationship where θ is equal to the zenith angle of the sun:

$$z = \frac{y}{2 \tan \theta} \quad (8)$$

3.2. Entropy generation

Entropy generation consists of a thermal contribution and mechanical contribution. It is deduced from the entropy balance between the heat source and the absorber on the other hand and between the absorber and the working fluid.

➤ Between the heat Source (the Sun) and the Absorber:

Entropy generation is expressed by:

$$S_{gen1} = \frac{q_L}{T_a} + \frac{q_u}{T_{abs}} - \frac{q_{abs}}{T_s} \quad (9)$$

Where $q_L = q_{abs} - q_u$ represents the loss power. Replacing with this new expression, entropy becomes:

$$S_{gen1} = \frac{1}{T_s} \left[q_{abs} \left(1 - \frac{T_a}{T_s} \right) - q_u \left(1 - \frac{T_a}{T_{abs}} \right) \right] \quad (10)$$

The useful power is determined by [12, 13]:

$$q_u = F' [q_{abs} - h_p (T_{mf} - T_a)] \quad (11)$$

F' is the collector efficiency factor. It express by [12]:

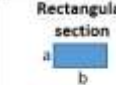
$$F' = \frac{h_{abs-f}}{h_{abs-f} + h_p} \quad (12)$$

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

$$h_{abs-f} = Nu_f \cdot \frac{\lambda_f}{D_h} \quad (13)$$

The Nusselt number is determined by table 1. For rectangular section, Nu_f and the friction factor f are determined by following laws regrouped in the following table [14, 15]:

Table 1. Nusselt number and friction factor for fully developed laminar flow in rectangular section [14, 15].

Rectangular section 	Nu Uniform heat flux	$f \cdot Re_{Dh}$
$\frac{b}{a} = 1$	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

$$T_{mf} = \frac{(T_{out} + T_{in})}{2} \quad (14)$$

The outlet temperature T_{out} is expressed by following relation [14, 15]:

$$\frac{T_{abs} - T_{out}}{T_{abs} - T_{in}} = \exp\left(\frac{-1}{\dot{m}_f C_p R_{tot}}\right) \quad (15)$$

Then

$$T_{out} = T_{abs} - (T_{abs} - T_{in}) \exp\left(\frac{-1}{\dot{m}_f C_p R_{tot}}\right) \quad (16)$$

With

$$\dot{m}_f = \rho_f \cdot V \cdot h \cdot p \quad (17)$$

$$R_{tot} = \frac{e_{abs}}{\lambda_{abs} \cdot p \cdot x} + \frac{1}{\eta_o \cdot h_{abs-f} \cdot A_t} \quad (18)$$

$$A_t = A_{fin} + A_b = (2 \cdot h + p) \cdot x \quad (19)$$

$$\eta_o = 1 - \frac{A_{fin}}{A_t} (1 - \eta_{fin}) \quad (20)$$

$$\eta_{fin} = \frac{\tanh(m \cdot h)}{m \cdot h} \quad (21)$$

$$m = \sqrt{\frac{h_{abs-f} \cdot P_{fin}}{\lambda_{abs} \cdot S_{fin}}} \quad (22)$$

$$P_{fin} = 2(e_{abs} + x) \quad (23)$$

$$S_{fin} = e_{abs} \cdot x \quad (24)$$

According to Petela [16],

$$T_* = \frac{3}{4} T_s = \frac{3}{4} (5770) K = 4330 K \quad (25)$$

Overall Coefficient of Heat loss h_p is the sum of the heat loss coefficients by the upper collector and the lower collector. It is determined by the expression [11]:

$$h_p = \frac{1}{\frac{1}{h_{c,abs-v} + h_{r,abs-v}} + \frac{1}{h_{c,v-a} + h_{r,v-a}}} + \frac{1}{\frac{e_{is}}{\lambda_{is}} + h_{wind}} \quad (26)$$

Differents terms $h_{c,abs-v}$, $h_{r,abs-v}$, $h_{c,v-a}$, $h_{r,v-a}$, h_{wind} in this expression are develop at the bottom.

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} are given by Mc Adams [13] where U_{wind} is the average speed of the wind.

$$h_{c,v-a} = h_{wind} = 5.7 + 3.8 U_{wind} \quad (27)$$

The convection coefficient between the glass and the absorber is obtained by:

$$h_{c,abs-v} = Nu \cdot \frac{\lambda_{air}}{y} \quad (28)$$

The Nusselt number Nu for an inclination i of the collector relative to the horizontal is given by [17]:

$$Nu = Nu_i = \frac{90-i}{90} Nu_0 + \frac{i}{90} Nu_{90} \quad (29)$$

Nu_{90} and Nu_0 are respectively the Nusselt number for $i = 90^\circ$ and for $i = 0^\circ$.

Nu_0 is given by the relationship of Jones and Smith [18]:

$$Nu_0 = \left[\left(\frac{1500}{Ra_y} \right)^2 + (0.081 Ra_y^{0.39})^{-2} \right]^{-\frac{1}{2}} \quad (30)$$

$$Ra_y = \frac{\varepsilon \cdot \beta \cdot (T_{abs} - T_v) \cdot y^3}{\nu \cdot \mathcal{D} \cdot x} \quad (31)$$

$$\mathcal{D} = \frac{\lambda_f}{\rho_a c_p} \quad (32)$$

Nu_{90} is given by the relationship of Bar-Cohen and Rohsenow [18]:

$$Nu_{90} = \left[\frac{576}{(Ra')^2} + \frac{2.873}{\sqrt{Ra'}} \right]^{-1/2} \quad (33)$$

$$Ra' = \frac{\varepsilon \cdot \beta \cdot (T_{abs} - T_v) \cdot y^4}{\nu \cdot \mathcal{D} \cdot x} \quad (34)$$

In this case, the distance x is equal to 1.

The radiation heat transfer coefficient between the glass and the atmosphere $h_{r,v-a}$ [16] and between the absorber and glass $h_{r,abs-v}$ are given by the equations:

$$h_{r,v-a} = \sigma \varepsilon_v (T_v^2 + T_a^2)(T_v + T_a) \quad (35)$$

$$h_{r,abs-v} = \sigma \frac{(T_{abs}^2 + T_v^2)(T_{abs} + T_v)}{\frac{1}{\varepsilon_{abs}} + \frac{1}{\varepsilon_v} - 1} \quad (36)$$

- Between Absorber and the working fluid :

In the case of forced convection and one-dimensional approach, entropy generation per unit length is determined by [19]:

$$S_{gen2}' = \frac{q'^2}{4T_i^2 \dot{m} c_p} \frac{D_h}{St} + \frac{2\dot{m}^3 f}{\rho^2 T_i D_h a^2} \quad (37)$$

Where q' is useful power per unit length. The useful power q_u is determined by (11). By introducing the Stanton

Number, the useful power q_u , entropy generation is calculated by:

$$S_{gen2} = \frac{\left[F' (q_{abs} - h_p (T_{abs} - T_a)) \right]^2 D_h \cdot Pr \cdot Re}{4x \cdot T_i^2 \dot{m} c_p \cdot Nu_f} + \frac{2\dot{m}^3 f \cdot x}{\rho_f^2 T_i D_h a^2} \quad (38)$$

The total entropy generation is the sum of entropies generations between the sun and the absorber and between the absorber and the working fluid.

$$S_{gen} = S_{gen1} + S_{gen2} \quad (39)$$

4. Hypothesis of Simulation

The variables that influence the performance are not only spacing and height but also other variables taken into account for the research done in the field. These variables are:

- The collector area S fixed at 1 m² (1m x 1m);
- The global insolation G taken at 800 W/m² (this variable is the average insolation of the hottest months at Abomey-Calavi in Benin) [11];
- The room temperature fixed at 30° C [20];
- 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]

5. Results and Discussions

Different mathematical models were programmed in Matlab. The different curves of variation are shown below.

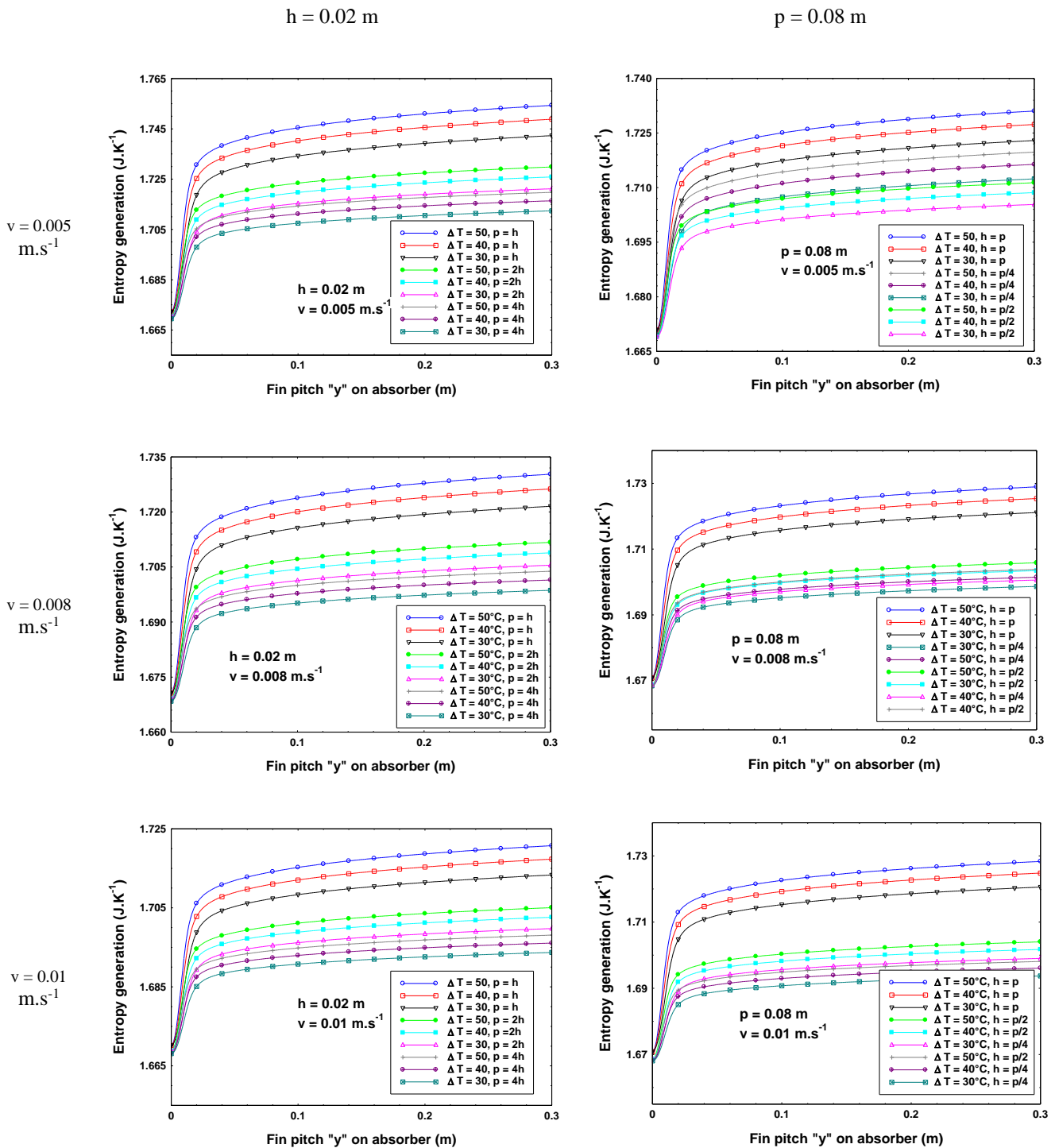


Fig. 3. Variation of entropy generation according to the pitch y on absorber.

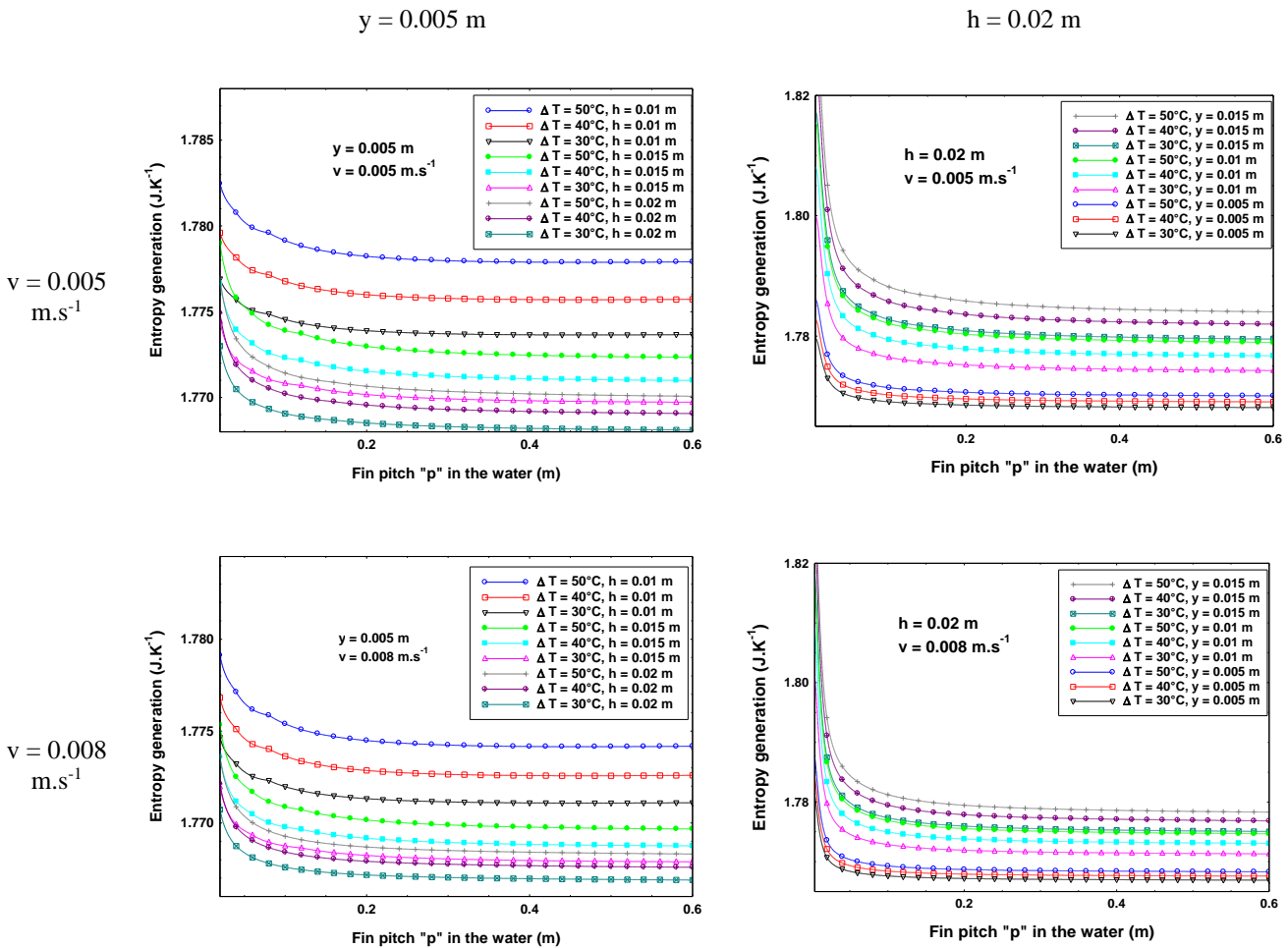
The effect of fin pitches at the absorber level on the entropy generation is shown on Fig 3. These curves are plotted for different values of the speed of water, the difference of temperature between the temperature of absorber and the one of glass, the height h and pitch p of fins in the water. Through these results we can notice that the entropy generation increases when the fin pitch "y" on the absorber increases. The lowest value of entropy generation is found when the pitch y tends to zero. These curves show that

the entropy increases fast for pitches "y" inferior to 0.04 m. But for pitches superior to 0.04 m, it increases lowly. The results show that the effect of fin pitches is negligible whether the pitches superior to 0.04 m. What can be justified by the fact that the sun's rays make more reflexions on the absorber before being sent again to the glass if the pitch "y" of fins on the absorber is low. This phenomenon increases the heat flow absorbed by the absorber and makes lower the heat losses due to the absorption. Furthermore, we observe a

great influence of the difference of temperature between the temperature of the absorber and the one of the glass on the entropy generation. More ΔT is low, more the entropy generation is low and more ΔT is raised, the entropy generation is raised. Then the entropy generation to $\Delta T=30^\circ\text{C}$ are inferior to those which are generated for $\Delta T=40^\circ\text{C}$ and those which are given by the last one are inferior to those which are generated for $\Delta T=50^\circ\text{C}$. When ΔT is raised, the heat lost of the absorber toward the glass is raised; that could give a raising entropy. Moreover, when we observe the Fig 3 for $h=0.02\text{ m}$ and for $p=0.08\text{ m}$; we notice that the speed of the working fluid has an effect on the entropy generation. When the speed is 0.01 m/s , the entropy

generation in the system is lower than when the speed is 0.008 m/s and this latter shows entropy that is lower than the one of speed 0.005 m/s . We will talk of 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 entropies generation by pitches $p=h$, $p=2h$ and $p=4h$ with $h=0.02\text{ m}$ are different. When the pitch is low the entropy is risen.

This report is confirmed by the Fig 4 and Fig 5 which respectively represents the effect of pitch p and the effect of the height h of fins in water on the entropy generation in the system.



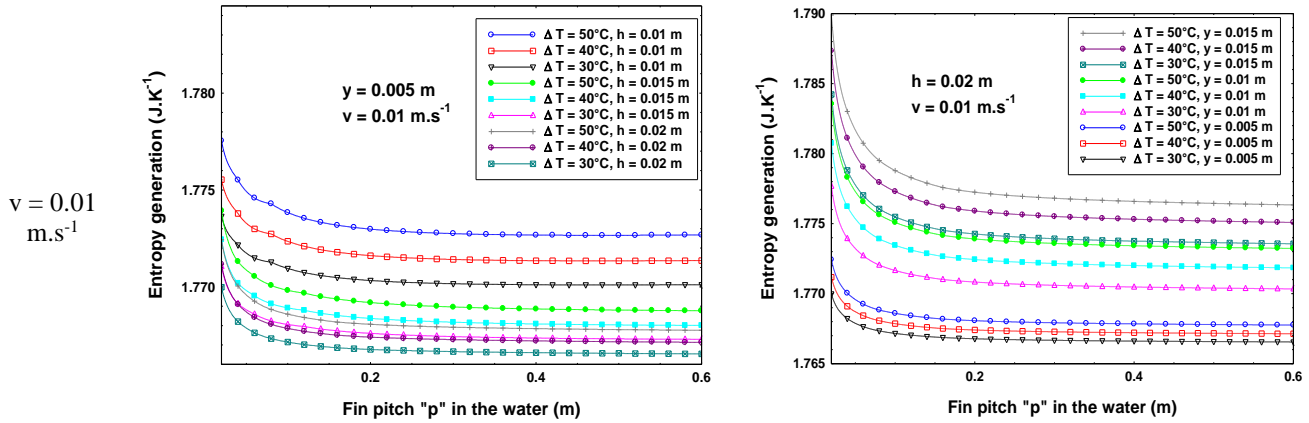
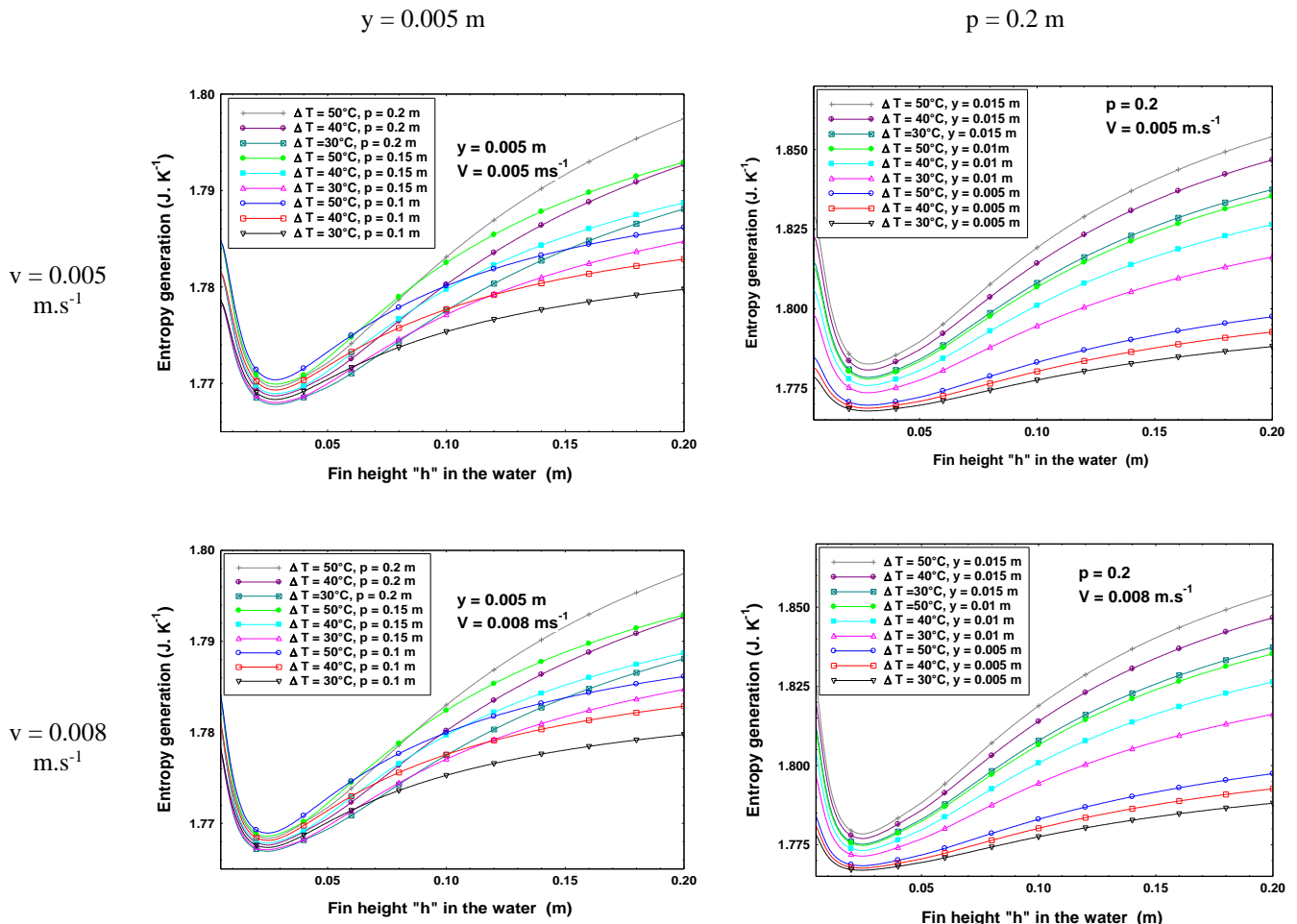


Fig. 4. Variation of entropy generation according to the pitch “p” in the water.

The Fig 4 reveals that for the low pitch “p”, the entropy generation is risen and for raised pitches, the entropy generation is low. In fact, when the height “h” and the velocity v are fixed, a low value of pitch “p” causes a friction higher than for a higher value of pitch “p”. So, about the low pitch “p”, the mechanical contribution in the entropy generation is more raised. When the pitch “p” is inferior to 0.1 m, the entropy generation rapidly decreases and for the

pitch whose the value is over 0.1 m, it decreases so slowly. From this result, we can say that the pitch “p” of fins in the water must be superior to 0.1 m and be certainly the greatest. Thus, a canal of water without fins is more considerable than the one with fins. It should also be noted that the effect of the difference of temperature ΔT on the entropy generation remains the same than the one in the case of the Fig 3. If ΔT is low, the entropy generation is also low



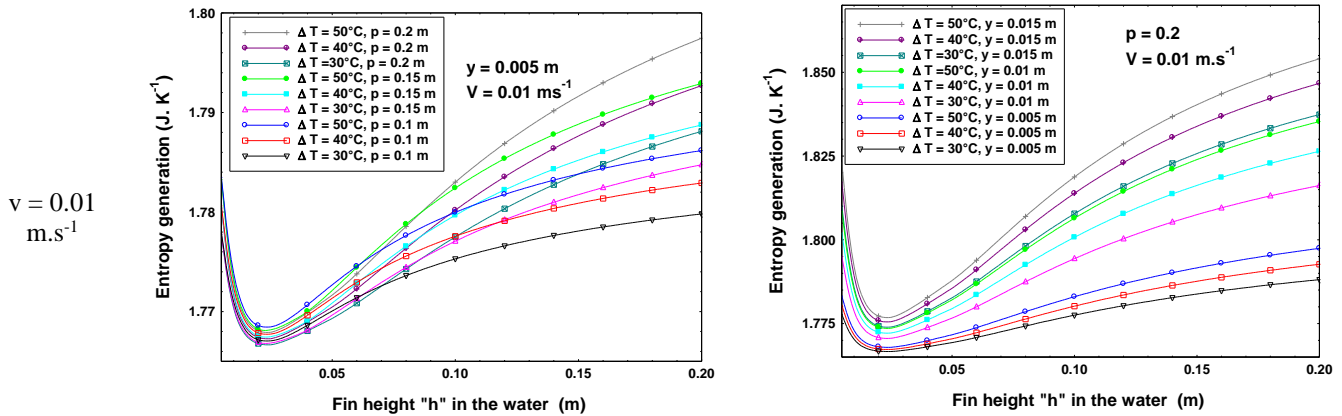


Fig. 5. Variation of entropy generation according to the height “h” in the water.

About the Fig 5, it is the evolution of entropy generation according to the height “h” of fins in the water. When the height is between 0 and 0.0235 m, the entropy decreases and reaches its minimal value of 1.767 J/K at h=0.0235 m. About the values of “h” superior to 0.0235 m, the entropy generation raises according to h. On these different curves, the minimal entropy is noticeable when the height “h” of fins in the water is 0.0235 m. The entropy generation is raised for the low values of the height “h” because of the raised value of the mechanical contribution in the entropy generation. This raised value of the mechanical contribution is due to a friction’s factor which is more raised when the height “h” is low. If the entropy generation is still raised for the values of the raised height “h”, it is due to a risen thermal contribution. In fact, for the values of the raised height “h”, the heat transfer coefficient between the absorber and the working fluid is low and consequently the useful flow is low. The figure 5 shows that the difference of temperature between the temperature of absorber and the one of the glass, keeps the same effects on the entropy generation than those in the case of the Fig 3 and Fig 4.

The Fig 6 presents the entropy generation according to the speed v of the water. For $\Delta T=50^{\circ}\text{C}$, when the speed is inferior to 0.016 m/s, the entropy decreases till to reach its minimum at the speed 0.016 m/s. When the speed is superior to 0.016 m/s, the entropy raises strongly according to the speed. The same paces are noticed for the differences of temperature $\Delta T=40^{\circ}\text{C}$ and $\Delta T=30^{\circ}\text{C}$ except that the minimum is reached at the speed 0.015 m/s for $\Delta T=40^{\circ}\text{C}$ and at the speed 0.014 m/s for $\Delta T=30^{\circ}\text{C}$. For the low speed of working fluid, the entropy generation is risen because of the raised thermal contribution. In the same way, for the raised speed of working fluid, the entropy generation is raised because of the risen mechanical contribution in the entropy generation. It should also be noted the minimum of entropy generation for $\Delta T=50^{\circ}\text{C}$ is higher than the one for $\Delta T=40^{\circ}\text{C}$ and this latter is also higher than the one of $\Delta T=30^{\circ}\text{C}$. This last remark confirms the effect of the difference of temperature between the temperature of absorber and the one of the glass noticed on the Fig 3, 4, 5, 6, 7 and 8. Then, one can say that the optimal speeds when $\Delta T=50^{\circ}\text{C}$, $\Delta T=40^{\circ}\text{C}$ and $\Delta T=30^{\circ}\text{C}$ are respectively of 0.016 m/s, 0.015 m/s and 0.014m/s.

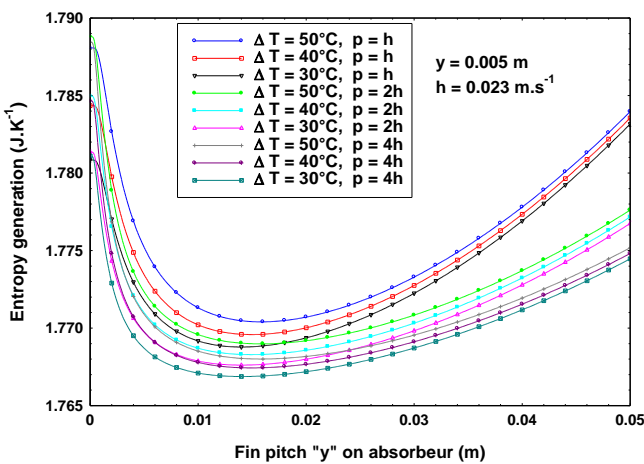


Fig. 5. Variation of entropy generation according the speed of water with h=0.023 m.

6. Conclusion

This work also led to understand that the entropy generation depends on the parameters of the flow channel of the thermal transfer fluid. These parameters in this study were the pitch p and the height h of the fins of the flow channel in the working fluid. The following conclusions can be drawn from the results of the present work:

- For a flat plate collector with fins on absorber, the fin pitch on the absorber must be lower than 0.04 m and to be weakest possible. When the fin pitch is higher than 0.04 m, the influence of the fin on the collector performance is negligible;
- The fin height on absorber must be chosen according to the fin pitch in order to realize a black body on the absorber;

➤ The fin pitch in the water must be higher than 0.1 m and to be largest possible;

➤ The optimal fin height in the water through this study is 0.023 m;

➤ The difference of temperature between the top absorber and the one of glass influence the entropy generation. More the difference is low more the entropy generation is low;

7. Nomenclature

A = Area (m²)

C_p = Specific heat at constant pressure kWh/kg° C

D_h = Hydraulic diameter (m)

e = Thickness (m)

G = global solar irradiance on any plane (W/m²)

g = Acceleration of gravity (N/kg)

Gr = Grashoff number

i = Inclination of the collector relative to the horizontal (°C)

h_{c,abs-v} = Convection coefficient between the absorber surface and the glass pane (W/m².K)

h_{abs-f} = Convection coefficient between absorber and the water (W/m².K)

h_p = Overall heat loss coefficient (W/m².K)

h_{r,v-a} = Radiation coefficient between the glass and the atmosphere (W/m².K)

h_{r,abs-v} = Radiation coefficient between absorber and the glass (W/m².K)

h_{c,v-a} = Convection coefficient between the glass and the atmosphere (W/m².K)

Pr = Prandtl number

q = Power (W)

Ra = Rayleigh number

Re = Reynolds number

S = Collector area (m²)/ entropy (J/K)
Fin section in the water (m²)

T = Temperature (K)

ΔT = Difference between the temperature of absorber and the one of glass (K)

U_{wind} = Wind speed (m/s)

x = Fin length on absorber (m)

y = Fin pitch on absorber (m)

p = Fin length in the water (m)

P_{fin} = Fin perimeter in the water (m)

z = Fin height on the absorber (m)

h = Fin height in the water (m)

V = Volume

β = Coefficient of expansion

ρ = Reflection coefficient

ρ_a = Air density (Kg/m³)

ρ_f = Fluid density

τ = Transmittance

α = Absorption coefficient

ε = Emissivity

λ = Thermal conductivity (W/m.K)

σ = Stefan-Boltzman constant (W / m²K⁴)

ℳ = Diffusivity (m²/s)

ν = Kinematic viscosity (m²/s)

Subscript

a = Ambiance

abs = Absorber

air = Air

f = Fluid

gen = Generation

in = Inlet

is = Insulator

m = Mean

out = Outlet

u = Useful

v = Glass

wind = Wind

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