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Numerical Study of a Cylindro-Parabolic Cooker "Blazing Tube"

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Abstract

The objective of this work is to numerically determine the thermal performance of the parabolic cylinder cooker commonly "blazing tube". These performances were determined by establishing heat balances at the different levels of the system. The equations obtained have been discretized; simplifying assumptions have been made to facilitate their resolution. We adopted Gauss Seidel's method using MATLAB software to solve these equations. The temperatures of the coolant, the glass and the absorber were determined as a function of time and along the tube. The thermal efficiency was also determined. It emerged that the different temperatures evolve linearly as a function of the length of the tube. Yield and temperatures depend on the amount of sunshine.

Keywords

Solar Cooker, Heat Transfer Fluid, Temperature, Yield, Sunshine

1. Introduction

The foods consumed by men are mostly cooked beforehand. The source of energy necessary for this cooking is generally fossil fuel. This increasingly rare, perishable and expensive source of energy is also polluting. It should be noted above all that in African countries in general, the most widely used source of energy is wood. These reasons lead researchers to turn to renewable energies, including solar. The literature shows that several solar cookers have been developed and studied. Box type cookers are the most common. This type of cooker consists of an insulated box with a transparent glass cover [1]. Several authors have taken an interest in the study of these cookers [2] [3] [4] [5]. Studies have shown their effectiveness and limits, so they added improvement elements such as reflectors, fins and even storage systems. So-called solar panel cookers have also been the subject of research by several researchers [6] [7] [8] [9]. These cookers are made up of panels that can be made of cardboard or any other material (even recycled). These panels are foldable and covered with reflective materials. The particularity of these cookers is that they are easy to build, very inexpensive and transportable. Unfortunately they are ineffective. However, improvement work has been carried out by these authors. Another type of cooker is the parabolic solar cooker. It is a parabolic shaped cooker with a reflective surface. The rays are reflected at the focus where the pot is placed. Studies carried out by various authors [10]-[15] show that thanks to the concentration of the sun's rays, these cookers perform well. A major inconvenience is that the entire device remains under the sun's rays, indisposing the user. This is why I refer to them as direct solar cooker. In contrast to these cookers, there may be indirect cookers. These cookers are the subject of this study, more specifically the parabolic cylinder solar cooker commonly called "blazing tube". A numerical study on MATLAB is made to determine the performance of this type of cooker in order to allow possible improvements.

2. Materials and Methods

2.1. Description of the Device

The device on which the simulation study will be carried out is the parabolic cylindrical solar cooker called "blazing tube", an indirect solar cooker. It is made up of:

- an aluminum reflector,
- a transparent pyrex glass tube under vacuum,
- a copper absorber tube covered with black paint containing a heat transfer fluid which is vegetable oil,
- a hearth or cooking box where the container (kitchen utensil) is located.

It is also equipped with wheels to facilitate its mobility.

The solar cooker concentrates the solar rays through the reflector on the absorber tube filled with oil (linear concentration). The heated oil rises by thermosiphon to the cooking hearth where the pot is located [16].

Figure 1 presents ours device.

2.2. Heat Balances at the Level of the Various Components

Before performing the heat balances, we made some simplifying assumptions.

Hypotheses

- the heat transfer fluid is incompressible;
- the ambient temperature Ta around the receiver is uniform;
- the solar flux at the level of the absorber is uniformly distributed;
- a transient regime for each heat transfer medium;

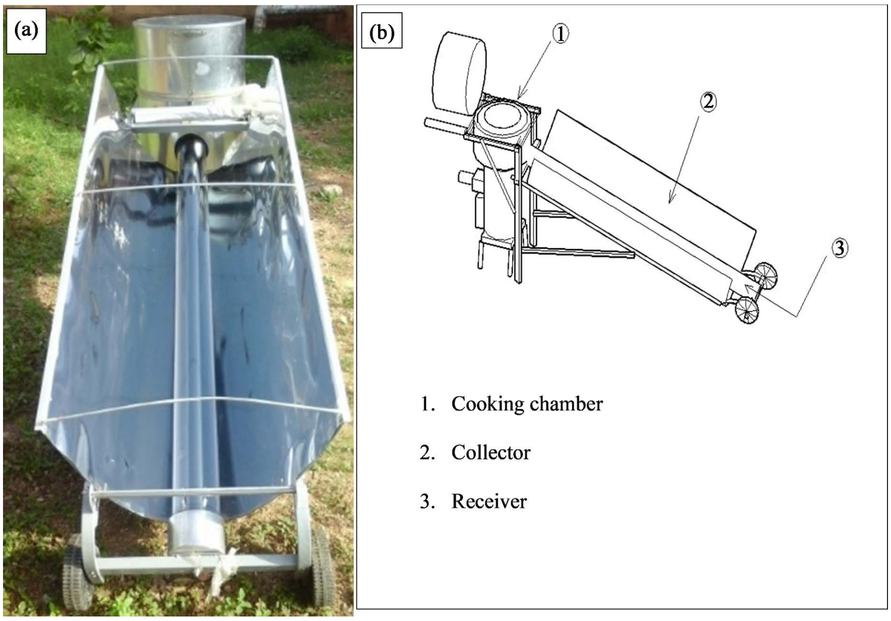

Figure 1. Cylindrical-parabolic cooker ((a) Front view, (b) Side image).

➢ One-dimensional heat transfer.

2.2.1. Heat Balance at the Glass

The heat transfer is done according to the three modes of transfer that is to say the convection, the conduction and the radiation. During this heat exchange we observe gains but also losses.

The glass exchanges heat with the external environment but also with the absorber.

- Thermal balance of exchanges at the level of the external wall of the glass:

$$m_v \frac{C_{pv} \mathrm{d}T_{v,e}}{2\mathrm{d}t} = h_{cond,ve-vi} S_v \left(T_{vi} - T_{ve}\right) - h_{conv,ve-a} S_v \left(T_{ve} - T_a\right) - h_{ray,ve-c} S_v \left(T_{ve} - T_c\right) + P_v \tag{1}$$

- Thermal balance of exchanges at the level of the internal wall of the glass:

$$m_v \frac{C_{p,v} \mathrm{d}T_{v,i}}{2\mathrm{d}t} = h_{ray,abs-vi} S_{ext,abs} \left(T_{abs} - T_{vi}\right) + h_{conv,vi-abs} S_{ext,abs} \left(T_{abs} - T_{vi}\right) - h_{cond,vi-ve} S_v \left(T_{vi} - T_{ve}\right) \tag{2}$$

2.2.2. Heat Balance of the Absorber

- Thermal balance of the external wall of the absorber:

$$m_{abs} \frac{C_{p,abs} \mathrm{d}T_{abs,ext}}{2\mathrm{d}t} = \varphi_{cond,abs.e-abs.i} - \varphi_{ray,abs-vi} - \varphi_{conv,abs-vi} + P_{abs} \tag{3}$$

- Thermal balance of the internal wall of the absorber:

$$m_{abs} \frac{C_{p,abs} \mathrm{d}T_{abs}}{2\mathrm{d}t} = -\varphi_{conv,abs-f} - \varphi_{cond,abs.i-abs.e} \tag{4}$$

- Overall thermal balance at the level of the absorber:

$$m_{abs}\frac{C_{p,abs}\mathrm{d}T_{abs}}{\mathrm{d}t}=h_{ray,abs-vi}S_{abs}\left(T_{vi}-T_{abs}\right)-h_{conv,abs-vi}S_{abs}\left(T_{abs}-T_{vi}\right)-h_{conv,abs-f}S_{abs}\left(T_{abs}-T_{f}\right)+P_{abs} \quad (5)$$

2.2.3. Overall Heat Balance of the Heat Transfer Fluid Overall Heat Balance of the Heat Transfer Fluid

$$m_f\frac{C_{p,f}\mathrm{d}T_f}{\mathrm{d}t}=h_{conv,f-abs}S_{abs}\left(T_{fp}-T_f\right)+\dot{m}C_{p,f}\frac{\mathrm{d}T_f}{\mathrm{d}z} \quad (6)$$

Conditions to the limits:

$$T_f\left(j,0\right)=T_{amb}$$

$$T_f\left(0,n\right)=T_{fe}$$

2.3. Evaluation of Heat Exchange Coefficients

- convection between the heat transfer fluid and the absorber

Heat transfer between the absorber tube and the heat transfer fluid (oil in our case) takes place by natural convection with a regime that can be laminar or turbulent.

In natural convection we have:

$$\overline{N_u}=0.86\times\left[Ra\sin\left(\theta\right)\right]^{0.25} \quad [17] \quad (7)$$

$$\text{with} \quad Ra=\frac{g\beta D_h\Delta T}{\alpha\nu} \quad (8)$$

$$\beta:\frac{1}{T_f}$$

$$T_f=\frac{T_p+T_\infty}{2}$$

$$h_{f-abs}=\frac{N_u\lambda_f}{D_h} \quad (9)$$

- Convection between the absorber and the glass

Heat transfer between the absorber tube and the glass (where air is trapped between them) takes place by natural convection. The heat transfer coefficient is determined using the following relationships: [18]

$$h_{convabs-vi}=\frac{2\pi k_{eff}}{\ln\left(\frac{D_{int,v}}{D_{ext,abs}}\right)} \quad (10)$$

where

$$\frac{k_{eff}}{k_f}=0.386\left(\frac{Pr_{air}}{0.861+Pr_{air}}\right)^{1/4}\times Rac^{1/4} \quad (11)$$

and

$$Rac = \frac{\ln\left(\frac{D_{int,v}}{D_{ext,abs}}\right)^4}{L^3\left(D_{ext,abs}^{1} + D_{int,v}^{-3/5}\right)^5} \times Ra \tag{12}$$

$$L = \frac{D_{int,v} - D_{ext,abs}}{2}$$

The Rayleigh number is defined as follows:

$$Ra = \frac{g\beta_{air}\rho_{air}L^3\left(T_r - T_0\right)}{\alpha_{air}\mu_{air}} \tag{13}$$

- Convection between the glass and the ambient

The heat exchange between the glass and the surrounding environment takes place by convection.

It can be determined by several linear relationships depending on the wind speed:

Hottel and Woertz relationship

$$h_{conv,ve} = 5.67 + 3.86Vv \quad [19] \tag{14}$$

Watmuff *et al.* relationship

$$h_{conv,ve} = 2.8 + 3.3Vv \quad [20] \tag{15}$$

For our work we have opted for the relation of Hottel and Woertz which is more rigorous and more used.

- Radiation between the glass and the celestial vault

$$h_{ray,ve} = \sigma\varepsilon\left(T_v + T_c\right)\left(T_v^2 + T_c^2\right) \tag{16}$$

- Radiation between the absorber and the glass

$$h_{ray,abs-vi} = \frac{\sigma}{\frac{1}{\varepsilon_{abs}} + \frac{1-\varepsilon_v}{\varepsilon_v}\left(\frac{D_{ext,abs}}{D_{int,v}}\right)} \tag{17}$$

- Conduction between the glass walls

$$h_{cond,ve-vi} = \frac{\lambda_v}{\ln\left(\frac{D_{ext,v}}{D_{int,v}}\right)} \tag{18}$$

- Conduction between the walls of the absorber

$$h_{cond,abs.i-abs.e} = \frac{\lambda_{abs}}{\ln\left(\frac{D_{ext,abs}}{D_{int,abs}}\right)} \tag{19}$$

2.4. Determination of Powers

- Power received by the collector

$$P_c = S_c I \tag{20}$$

- Power absorbed by the glass

$$P_v = I\alpha_v \rho_c C_g S_v \tag{21}$$

- Power absorbed by the absorber

$$P_{ab} = I\alpha_{ab}\tau_v \rho_c C_g S_{abs} \tag{22}$$

2.5. Thermal Efficiency

Le rendement thermique est le rapport entre la puissance thermique gagnée par le fluide caloporteur et la puissance solaire reçue par le capteur. Il est donné par la relation suivante:

The thermal efficiency is the ratio between the thermal power gained by the heat transfer fluid and the solar power received by the collector. It is given by the following relationship:

$$\eta = \frac{\dot{m}_f c_{pf}\left(T_{s,f} - T_{e,f}\right)}{I\left(A_c \times C_g + \frac{1}{2}S_{abs}\right)} \tag{23}$$

2.6. Calculation of Solar Radiation

The global radiation RG is expressed by the relationship:

$$RG = RDIR + RDIF \tag{24}$$

$RDIF$ is diffuse radiation

$$RDIF = \frac{1}{2}\left(1+\cos\left(\frac{pi}{180}\beta\right)\right)2SRDIFH + \left(1-\frac{1}{2}\cos\left(\frac{pi}{180}\beta\right)SA\left(I_0 \sin\left(\frac{pi}{180}h\right)\right)\right) + RDIFH$$

$RDIFH$ is diffuse radiation on a horizontal surface

$$RDIFH = I_0 C\left(\sin\left(\frac{pi}{180}h\right)\right)\left(0.271 - 0.2939\exp\left(-B\Big/\sin\left(\frac{pi}{180}h\right)\right)\right)$$

$RDIR$ is direct radiation

$$RDIR = I_0 CA\exp\left(-B\Big/\sin\left(\frac{pi}{180}h\right)\right)\left(\cos\left(\frac{pi}{180}I\right)\right)$$

3. Results

The equations were discretized by the finite difference method. They are then solved by the Gauss Seidel method in a computer program using MATLAB software. This program made it possible to obtain the curves below.

Figure 2 and Figure 3 represent respectively the temporal evolution of direct sunshine and global sunshine by simulation.

It can be seen that the overall sunshine increases (8 a.m. to 12 p.m.) then decreases as a function of time. The curve is "bell-shaped". It reaches its maximum around 12 noon with a value of approximately 920 W/m^2. Direct sunlight reaches a maximum value of approximately 860 W/m^2.

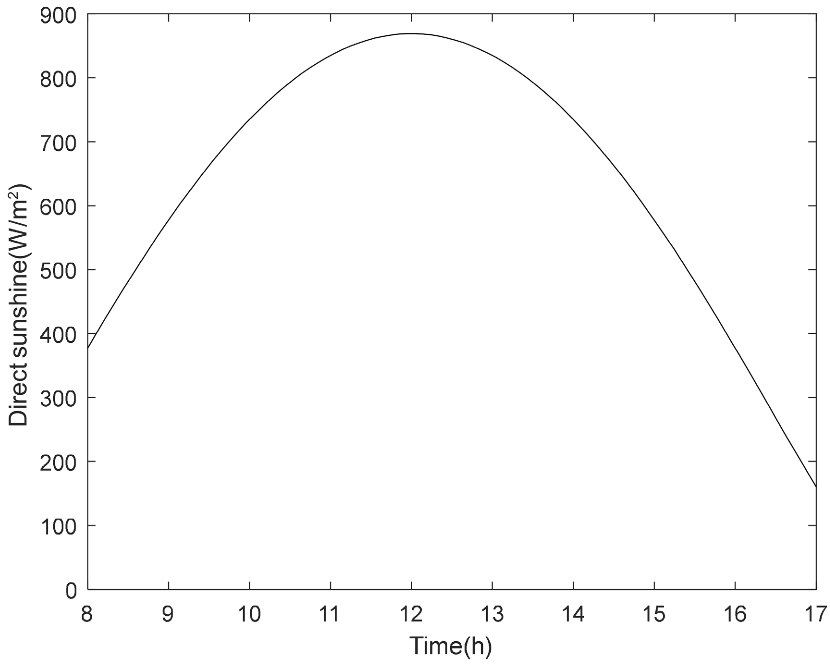

Figure 2. Temporal evolution of direct sunshine.

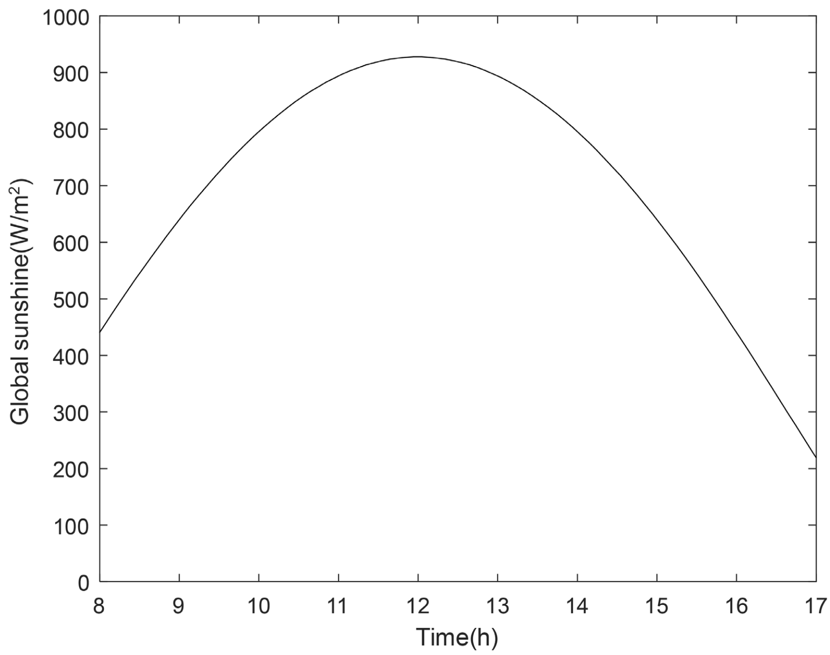

Figure 3. Temporal evolution of global sunshine.

Figure 4 presents the variations of the different temperatures (temperature of the absorber, temperature of the fluid, temperature of the glass) as a function of the length of the tube for $t = 13$ h (t represents the time).

It can be seen that all the curves have the same shape and that the temperatures evolve in an increasing way according to the length of the tube. We also note that the temperature of the absorber is higher than that of the fluid which is higher than that of the glass. We find the same profiles in the simulation results of Chekirou *et al.* [18]. This shows that our results are satisfactory. The maximum temperature of the fluid, *i.e.* the outlet temperature, is around 145˚C.

Figure 5 shows the evolution over time of the temperature of the absorber (red), the heat transfer fluid (black) and the glass (blue).

Temperatures change during the day or time. This means that the temperatures

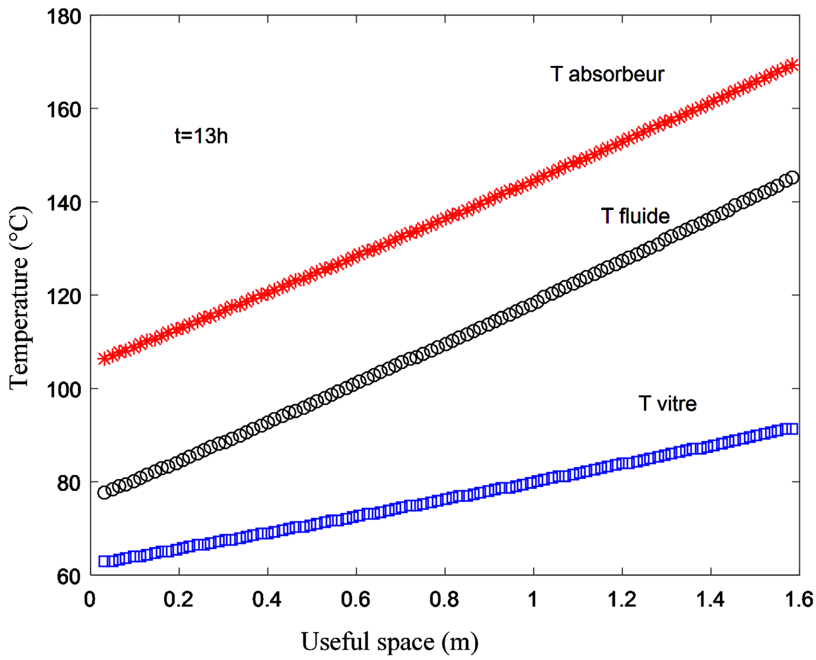

Figure 4. Evolution of the temperature of the fluid (black), of the absorber (red) and of the glass (blue) according to the length of the tube.

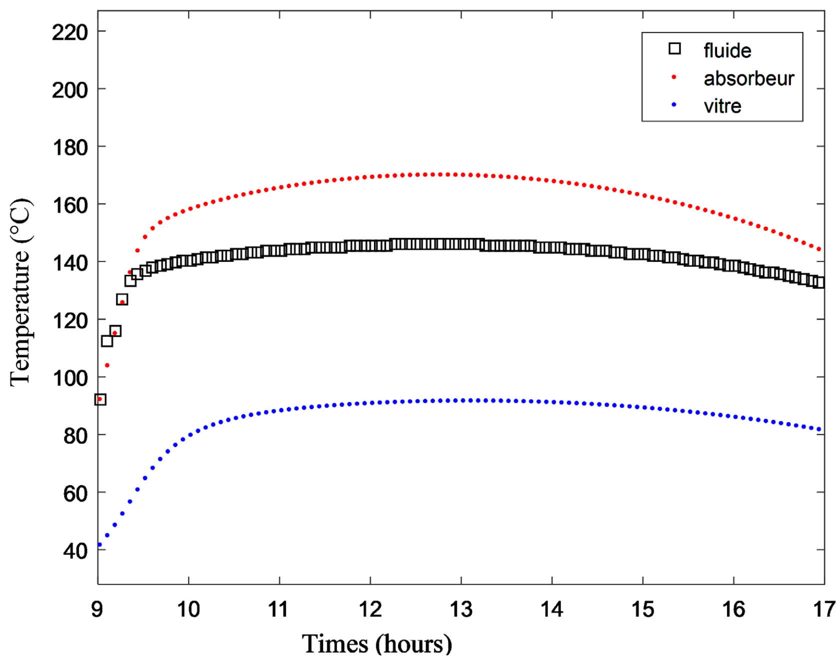

Figure 5. The evolution over time of the temperature of the absorber (red), the heat transfer fluid (black) and the glass (blue).

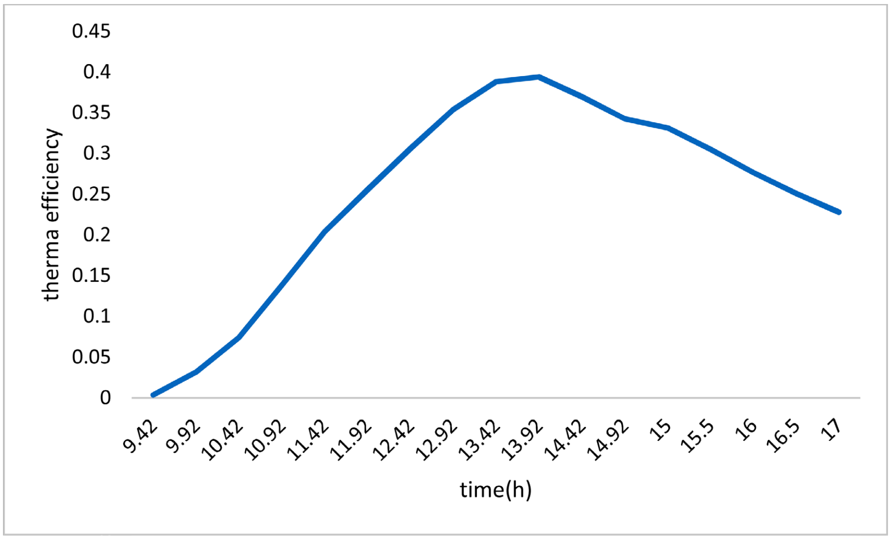

Figure 6. Evolution of thermal efficiency.

are a function of the sunshine. During the day the temperature of the absorber is the highest and that of the glass the lowest throughout the day, the temperature of the fluid is then between the two. The same profiles are obtained in the work of Chekirou *et al.* [18], Boukhchana *et al.* [21].

Figure 6 shows the evolution of thermal efficiency as a function of time.

The thermal efficiency evolves in an increasing way until its maximum (0.40 to 0.33) around 1 p.m. The thermal efficiency curve has a bell shape, like that of sunshine. The efficiency depends on the solar power.

4. Conclusion

An algorithm developed in this study allowed us to determine the performance of the cylindrical parabolic solar cooker. Thus, it was possible to evaluate the temperatures of the various components of the system and of the heat transfer fluid. There is an increasing evolution of these temperatures along the length of the tube. Also the sunshine directly influences the evolution of the fluid temperature as well as on all the components.

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Conflicts of Interest

The authors declare no conflicts of interest regarding the publication of this paper.

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Nomenclature

Symbols	Designation	Unity
A	Atmospheric haze factor	
A_c	useful surface of the collector	m^2
B	Atmospheric haze factor	
C	correction factor	
C_g	geometric concentration -	-
C_p	specific heat	J/kg.K
D	Diameter	m
D_h	hydraulic diameter	m
f	collector efficiency factor	-
Gr	Grashoff number	-
h	height of the sun	(˚) degré
h_{cond}	heat exchange coefficient by conduction	$W/m^2·K$
h_{conv}	heat exchange coefficient by convection	$W/m^2·K$
h_{ray}	radiation heat exchange coefficient	$W/m^2·K$
I	solar radiation	W/m^2
I_0	Solar constant	W/m^2
K	overall heat exchange coefficient	$W/m^2·K$
K_{eff}	effective thermal conductivity of fluid	W/mK
l	collector length	m
m	mass	kg
$\dot{m}_f$	mass flow of the heat transfer fluid	kg/s
N_u	Nusselt number	-
P	Power	W
Pr	Prandtl number	-
Ra	Rayleigh number	-
Rac	simplified Rayleigh number	
S	concentrator opening area	m^2
T	temperature	K
V	wind speed	m/s

Greek letters	Designation	Unity
α	Absorption coefficient	-
ε	Emissivity	-
η	Thermal efficiency	-
λ	Thermal conductivity	W/m.K
ρ	Reflection coefficient	-
σ	Stephan-Boltzman constant	$W/m^2·K^4$
τ	Transmission coefficient	-
φ	Amount of heat exchanged	W

Indices	**Designation**
ab	absorbed
abs	absorbed
air	air
c	collector
e	External or input
ext	Outside
f	fluid
i	internal
int	Interior
r	receiver
s	exit
v	glass