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Modeling and Simulation of a Parabolic Trough Solar Concentrator

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Abstract: This work consisted of the mathematical modeling of a parabolic trough concentrator. To this end, a heat balance has been established for the different parts of the parabolic trough concentrator, which are the heat transfer fluid, the absorber and the glass. This allowed us to establish a system of equation whose resolution was done by the finite difference method. This digital resolution made it possible to obtain the temperatures of the different parts of our parabolic trough concentrator, namely, the heat transfer fluid, the absorber and the glass. The simulation of the heating process of the fluid is done in time steps of one hour, from six hours to eighteen hours. The results obtained show that the temperature difference between the inlet and the outlet of the solar collector is very large. A computer program has been developed to simulate the temperatures of the heat transfer fluid, the absorber tube and the glass as a function of time and space. These results were obtained for a typical day with regard to the variation of the temperatures of the heat transfer fluid, the absorber and the glass, as well as the powers and efficiency of the parabolic trough concentrator and various factors for the sake of improve the performance of our prototype.

Keywords: Modeling, Simulation, Parabolic Trough Concentrator, Heat Transfer Fluid, Temperature

1. Introduction

Sun, water, wind, wood and other plant products are all natural resources capable of generating energy thanks to technologies developed by humans. Their low impact on environment makes them energies of future in face of nuclear waste management problem and greenhouse gas emissions.

Solar energy is energy transmitted by sun in form of light and heat.

Solar thermal energy is transformation of solar radiation into thermal energy. This transformation can be either used directly (heating) or indirectly (production of water vapor to drive alternators and produce electrical energy).

By using heat transmitted by radiation rather than radiation itself, these modes of energy transformation set themselves apart from other forms of solar energy such as photovoltaic cells.

Two fundamental principles are used:

1) capture energy of sun's rays thanks to black body;
2) concentrate solar radiation at a point (solar oven).

This study objective is to model phenomenon of heat transfer in different compartments of parabolic trough solar concentrator. Here, temperature change as function of time is determined by establishing a heat balance that takes into account different heat exchanges, followed by digital model based on following steps [1]:

1) Solar energy reflected by concentrator falls on absorber;
2) Heat is recovered by heat transfer fluid and it heats up by circulating in absorber placed under a glazing which allows radiation to penetrate and minimizes the losses by infrared radiation by using greenhouse effect. This glazing also helps limit heat exchange with environment.

2. Description

It is presented as a module having a reflector of parabolic shape arranged cylindrically according to figure 1.

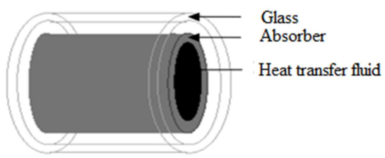

Figure 1. *Diagram of an absorber tube element.*

Heat exchanges here take place between three elements which are: Heat transfer fluid, absorber and glass, as shown in Figure 2.

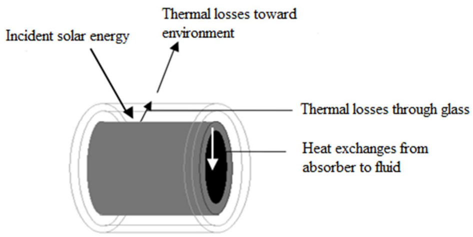

Figure 2. *Schematic of heat transfer between different parts of parabolic collector.*

3. Thermal Balance

3.1. Hypotheses

These assumptions are as follows:

1) Collector follows sun perfectly;
2) Flow of fluid in receiving tube is one-dimensional;
3) Diameters of tubes and surface of collector are constant;
4) Heat transfer by conduction in all elements of absorber is negligible;
5) Heat transfer fluid is incompressible;
6) Ambient temperature around parabolic trough is assumed to be uniform;
7) Effect of shadow of absorber tube on glass is negligible;
8) Solar flux at absorber is evenly distributed;
9) Glass is considered opaque to infrared radiation;
10) Temporal variations in thickness of absorber and glass are negligible.

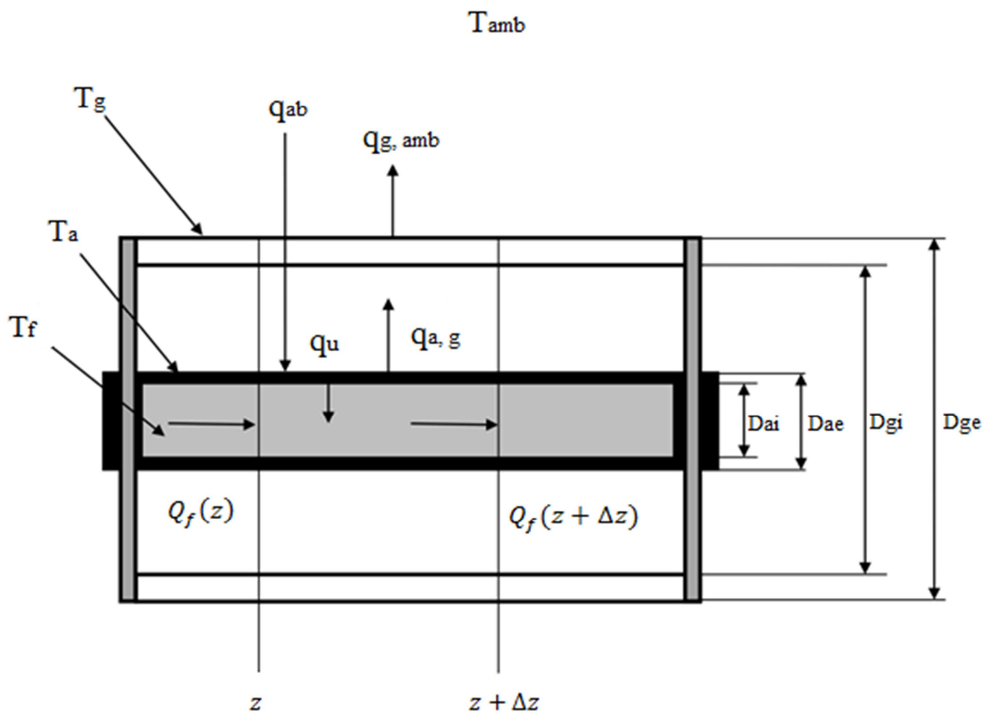

Figure 3. *Thermal balance of a surface element of parabolic trough concentrator.*

q_{ab}: heat quantity come from sun and receive by absorber.
q_u: useful quantity heat transferred to coolant by absorber.
qa, v: quantity of heat exchanged by convection between absorber tube and glass.
$q_{v,\, amb}$: heat quantity exchanged by convection between glass and ambient air.
$Q_f(x)$: heat flow of heat transfer fluid at abscissa x.
$Q_f(x + \Delta x)$: heat flow of heat transfer fluid at abscissa $x + \Delta x$.
Dai: internal diameter of absorber tube.
Dae: external diameter of absorber tube.
Dgi: inside diameter of glass.
Dge: external diameter of glass.
Tf: heat transfer fluid temperature.
Ta: absorber temperature.
Tg: glass temperature.
$\dot{V}$: volume flow.

3.2. Energy Balance in the Heat Transfer Fluid

Energy balance for heat transfer fluid circulating in absorber tube is expressed by the following relationship:

$$\frac{\partial}{\partial t}\left(\Delta Q_f(z,t)\right) = Q_f(z,t) - Q_f(z+\Delta z,t) + q_u.\Delta z(z,t) \quad (1)$$

Heat quantity recovered by fluid $\Delta Q_f(z,t)$ in interval Δz is given by:

$$\Delta Q_f(z,t) = \rho_f.C_f.S_{ai}.\Delta z.T_f(z,t) \quad (2)$$

where:
ρ_f: fluid density;
C_f: specific heat of fluid;
$S_{ai} = \pi.D_{ai}$: absorber internal exchange surface per unit length.
All thermo-physical properties of fluid depend on temperature T_f.
Heat quantity entering and leaving the element of length Δz is given by the following relations:

$$\begin{cases} Q_f(z,t)=\rho_f.C_f.\dot{V}.T_f(z,t) \\ Q_f(z+\Delta z,t)=\rho_f.C_f.\dot{V}.T_f(z+\Delta z,t) \end{cases} \quad (3)$$

Inserting equations (2) and (3) into equation (1) gives the following relation (4):

$$\rho_f.C_f.S_{ai}.\Delta z.\frac{\partial T_f}{\partial t}(z,t) = \rho_f.C_f.\dot{V}.T_f(z,t) - \rho_f.C_f.\dot{V}.T_f(z+\Delta z,t) + q_u(z,t).\Delta z \quad (4)$$

The partial derivative with respect to abscissa z is:

$$\frac{\partial T_f}{\partial z}(z,t) = \frac{T_f(z+\Delta z,t)-T_f(z,t)}{\Delta z} \quad (5)$$

Divide equation (4) by Δz and after substitution in equation (5), we get:

$$\rho_f.C_f.S_{ai}.\frac{\partial T_f}{\partial t}(z,t) = -\rho_f.C_f.\dot{V}.\frac{\partial T_f}{\partial z}(z,t) + q_u(z,t) \quad (6)$$

Initial conditions and boundary conditions of equation (6) are:

$$\begin{cases} T_f(0,t) = T_{f,inlet}(t) = T_{amb}(t) \\ T_f(z,0) = T_{f,initial}(z) = T_{amb}(0) \end{cases}$$

3.3. Energy Balance in the Absorber

The energy balance for absorber is given by the following relation:

$$\frac{\partial}{\partial t}\left(\Delta Q_a(z,t)\right) = \left(q_{ab}(t) - q_{a,g}(t) - q_u(t)\right).\Delta z \quad (7)$$

ΔQ_a: The amount of heat in absorber is expressed by:

$$\Delta Q_a(z,t) = \rho_a.C_a.S_{ae}.\Delta z.T_a(z,t) \quad (8)$$

where:
ρ_a: absorbeur density;
C_a: spécific heat of absorber;
$S_{ae} = \pi.D_{ae}$: absorber external exchange surface per unit length.
After substituting (8) in (7), we get the expression:

$$\rho_a.C_a.S_{ae}.\frac{\partial T_a}{\partial t}(z,t) = \left(q_{ab}(t) - q_{a,g}(t) - q_u(t)\right) \quad (9)$$

Initial condition of equation (9) is:

$$T_a(z,0) = T_{a,initial}(z) = T_{amb}(0)$$

3.4. Energy Balance in the Glass

The energy balance in the glass is given by:

$$\rho_g.C_g.S_g.\frac{\partial T_g}{\partial t}(z,t) = \left(q_{a,g}(t) - q_{g,amb}(t)\right) \quad (10)$$

where:
ρ_g: glass density;
C_g: specific heat of glass;
$S_g = \pi.D_g$: glass external exchange surface per unit length.
Initial condition of equation (10) is:

$$T_g(z,0) = T_{g,initial}(z) = T_{amb}(0)$$

4. Heat Exchanges

4.1. Heat Exchanges Between Absorber and Fluid

We have:

$$q_u = h_f.S_{ai}\left(T_a - T_f\right) \quad (11)$$

The convective exchange coefficient h_f essentially depends on fluid flow regime.
Its expression is:

$$h_f = \frac{Nu.k_f}{D_{ai}} \quad (12)$$

where:
k_f: thermal conductivity of fluid;
Nu: Nusselt number, given by Gnielinsky correlation [2]:

$$Nu = \frac{(\chi/8)(Re_f-1000)Pr_f}{1+12.7\sqrt{\chi/8}\left(Pr_f^{2/3}-1\right)} \quad (13)$$

χ: is coefficient of friction calculated from Petukhov relation [2]:

$$\chi = \left(0.0790ln(Re_f) - 1.64\right)^2 \quad (14)$$

Re_f: Reynolds number translated by the following relation [3]:

$$Re_f = \frac{4.\rho_f.\dot{V}}{\pi.\mu_f.D_{ai}} \quad (15)$$

with: μ_f: dynamic fluid viscosity;
Pr_f: number of Prandtl given by the following expression:

$$Pr_f = \frac{\vartheta_f}{\alpha_f} \quad (16)$$

$\vartheta_f = \frac{\mu_f}{\rho_f}$: kinematic viscosity; $\alpha_f = \frac{k_f}{\rho_f . C_f}$

1) For laminar flow ($Re_f < 2300$), Nusselt number is expressed by the following value [4]: Nu = 4.36;
2) For turbulent flow ($Re_f > 2300$), Nusselt number is calculated by Gnielinsky correlation [2].

4.2. Heat Exchanges Between Absorber and Glass

We have:

$$q_{a,g} = q_{a,g_{convection}} + q_{a,g_{radiation}} \tag{17}$$

Radiation exchange $q_{a,g_{radiation}}$ in annular space is given by Thorsten [3] :

$$q_{a,g_{radiation}} = \frac{\sigma . S_{ae}\left(T_a^4 - T_g^4\right)}{\frac{1}{\varepsilon_a} + \frac{1-\varepsilon_g}{\varepsilon_g}\left(\frac{D_{ae}}{D_{gi}}\right)} \tag{18}$$

where:

$\sigma = 5.670 \times 10^{-8}\ W.m^{-2}.K^{-4}$: Stefan Boltzmann's constant;

ε_a: absorber emissivity;

ε_g: glazing emissivity.

The internal heat exchange by convection in air space between absorber and glazing is given by Thorsten [3], knowing that this air space is mobile :

$$q_{a,g_{convection}} = \frac{2\pi . k_{eff,air}}{\ln\left(D_{gi}/D_{ae}\right)}\left(T_a - T_g\right) \tag{19}$$

Where, $k_{eff,air}$ is equivalent thermal conductivity of air, given by:

$$\frac{k_{eff,air}}{k_{air}} = 0.386\left(\frac{Pr_{air}}{0.861 + Pr_{air}}\right)^{1/4}\left(\frac{\left[\ln\left(D_{gi}/D_{ae}\right)\right]^4}{L^3\left(D_{ae}^{-3/5} + D_{gi}^{-3/5}\right)^3} Ra_L\right)^{1/4} \tag{20}$$

Where:

L: average thickness of annular layer located between absorber and glass, equal to:

$$L = 0.5\left(D_{gi} - D_{ae}\right) \tag{21}$$

Ra_L: Rayleigh number for air, defined as follows [3]:

$$Ra_L = \frac{g . \beta_{air}\left(T_a - T_g\right)L^3}{\alpha_{air} . \vartheta_{air}} \tag{22}$$

Where:

g: gravity intensity ($g = 9.81\ m.s^{-2}$).

β_{air}: thermal coefficient of volumetric expansion of air.

α_{air}: thermal diffusivity defined by:

$$\alpha_{air} = \frac{k_{air}}{\rho_{air} . C_{p_{air}}} \tag{23}$$

ρ_{air}, $C_{p_{air}}$, k_{air},: are respectively density, specific heat at constant pressure and thermal conductivity of air.

ϑ_{air}: kinematic viscosity of air, expressed by: $\vartheta_{air} = \frac{\mu_{air}}{\rho_{air}}$

Where:

μ_{air}: dynamic air viscosity.

If in absence of convection (trapped air space is immobile) between absorber and glass, the flow per unit length is written [5]:

$$q_{a,g_{conduction}} = \frac{2\pi . k_{air}}{\ln\left(D_{gi}/D_{ae}\right)}\left(T_a - T_g\right) \tag{24}$$

All thermodynamic properties of air in annulus ($k_{air}, Pr_{air}, \beta_{air}, \alpha_{air}, \vartheta_{air}, \rho_{air}, C_{p_{air}}, \mu_{air}$) depend on average temperature of annulus.

A semi-empirical equation is used for determination of average temperature of annular space for a cylindrical-shaped absorber [6]:

$$T_{average\ annular} = 320 + \left[(0.11 \times \varepsilon_a) + 0.57\right] \times \left(T_g - 320\right) \tag{25}$$

On other hand, average temperature in annular space, can be expressed by:

$$T_{average\ annular} = 0.5\left(T_g + T_a\right) \tag{26}$$

4.3. Heat Exchange Between Glass and Surrounding Environment

It's assumed that heat transfer between transparent envelope (glass) and environment is also due to exchange by convection and radiation.

$$q_{g,amb} = q_{g,amb_{convection}} + q_{g,amb_{radiation}} \tag{27}$$

The amount of convective heat from glazing to environment is expressed by:

$$q_{g,amb_{convection}} = h_V . S_{ge}\left(T_g - T_{amb}\right) \tag{28}$$

h_V: wind exchange coefficient, defined by the following expressions [6]:

$$\begin{cases} h_V = 5.7 + 3.8 V_V & 0 < V_V < 4\ m.s^{-1} \\ h_V = 7.3 V_V^{0.8} & 4\ m.s^{-1} < V_V < 40\ m.s^{-1} \end{cases} \tag{29}$$

V_V: wind speed $[m.s^{-1}]$.

The amount of heat radiating from glazing to environment can be expressed by the following relationship:

$$q_{v,amb_{radiation}} = \varepsilon_g . \sigma . S_{ge}\left(T_g^4 - T_{amb}^4\right) \tag{30}$$

5. Absorbed Energy

The thermal power emitted by sun and received by concentrator is [7]:

$$q_{ab} = S_e . R_d(\beta) \tag{31}$$

S_e: effective area of a reflector of the parabolic trough;

$R_d(\beta)$: sunlight or direct radiation.

Since the surface of absorber S_a receives the same power as S_e, then the concentration has the effect of increasing power per unit area at the level of S_a; but due to optical losses around transparent envelope (glass) and absorber, the power per unit length of absorber is:

$$q_{ab} = \tau_v . \alpha_{ab} . \rho_m . S_e . R_d(\beta) . k(\theta) \tag{32}$$

Where:
ρ_m: Mirror reflectance factor;
α_{ab}: Absorption coefficient of absorber tube;
τ_g: Coverage transmission coefficient (glass);
$k(\theta)$: Angle of incidence modify.

For cylindro-parabolic collectors, the following equation [8], gives coefficient k (θ) as a function of angle of incidence:

$$k(\theta) = 1 - b_0\left(\frac{1}{\cos(\theta)}\right) \quad (33)$$

Where:
b_0: coefficient of modified angle less than zero for a cylindro-parabolic concentrator.

6. Thermal Power Losses in a Parabolic Trough Concentrator

The coefficient of thermal losses U_L is expressed by [9]:

$$U_L = \left(\frac{1}{C_1[(T_a - T_{amb})/(1+\lambda)]^{0.25}} + \frac{D_{ai}}{D_{ae}} \times \frac{1}{h_V}\right)^{-1} + \left(\frac{\sigma(T_a^2 + T_{amb}^2)(T_a + T_{amb})}{[\varepsilon_a - 0.04(1-\varepsilon_a)(T_a/450)]^{-1} - [(D_{ai}/D_{ae})(1/\varepsilon_g) + \lambda/\varepsilon_g]}\right) \quad (34)$$

Where:
λ: wind loss coefficient, obtained by equation (35):

$$\lambda = \frac{D_{ai}}{D_{ai}^{1.4}}(0.61 + 1.3\varepsilon_a)h_g^{-0.9} exp[0.00325(T_a - 273)] \quad (35)$$

$$C_1 = \frac{1.45 + 0.96(\varepsilon_a - 0.5)^2}{D_{ai}(1/D_{ai}^{0.6} + D_{ae}^{0.6})^{1.25}} \quad (36)$$

$$h_V = 4V_V^{0.58} D_{ae}^{-0.42} \quad (37)$$

7. Numerical Solving

The finite difference method we use for our problem resolution, consists in approximating derivatives of equations obtained above, by means of Taylor expansions and is deduced directly from definition of derivative.

Discretization of One-Dimensional Equations (1D)

The space is discretized into N + 1 nodes of coordinates z_j (j varying from 0 to N) regularly spaced. Note Δz the space step. Time is discretized in constant step intervals Δt.

Let us note T_j^n the temperature at the node $z_j = j\Delta z$ and at time $t = n\Delta t$.

We use an explicit before order 1 scheme to evaluate temporal and spatial derivative:

$$\left(\frac{\partial T}{\partial t}\right)_j^n = \frac{T_j^n - T_j^{n-1}}{\Delta t} \quad (38)$$

$$\left(\frac{\partial T}{\partial z}\right)_j^n = \frac{T_j^n - T_{j-1}^n}{\Delta z} \quad (39)$$

For the fluid

Equation (6) gives:

$$\rho_f . C_f . S_{ai} . \frac{T_{f,j}^n - T_{f,j}^{n-1}}{\Delta t} = -\rho_f . C_f . \dot{V} . \frac{T_{f,j}^n - T_{f,j-1}^n}{\Delta z} + q_u(z,t)$$

By replacing $q_u(z,t)$ by its expression and by expressing $T_{f,j}^n$ depending on other parameters, we get:

$$\left(1 + \frac{\Delta t.\dot{V}}{\Delta z.S_{ai}} + \frac{\Delta t}{\rho_f(j)C_f(j)S_{ai}} h_f . S_{ai}\right) T_{f,j}^n + \frac{\Delta t}{\rho_f(j)C_f(j)S_{ai}} h_f . S_{ai} T_{a,j}^n = \frac{C_f(j-1)\rho_f(j-1)\Delta t.\dot{V}}{C_f(j)\rho_f(j)\Delta z.S_{ai}} T_{f,j-1}^n + T_{f,j}^{n-1} \quad (40)$$

For the absorber

Equation (9) gives: $\rho_a . C_a . S_{ae} . \frac{T_{a,j}^n - T_{a,j}^{n-1}}{\Delta t} = \left(q_{ab}(t) - q_{a,g}(t) - q_u(t)\right)$

By replacing $q_{ab}(t)$, $q_{a,g}(t)$ and $q_u(t)$ by their expressions, we get:

$$\frac{\Delta t.h_f.S_{ai}}{\rho_a C_a S_{ae}} T_{f,j}^n + \left(1 - \frac{\Delta t}{\rho_a C_a S_{ae}}\left(\frac{2\pi.k_{eff,air}}{\ln(D_{gi}/D_{ae})} + h_f . S_{ai} + \frac{\sigma.S_{ae}\left(\left(T_{a,j}^2\right)^n + \left(T_{g,j}^2\right)^n\right)\left(T_{a,j}^n + T_{g,j}^n\right)}{\frac{1}{\varepsilon_a} + \frac{1-\varepsilon_g}{\varepsilon_g}\left(\frac{D_{ae}}{D_{gi}}\right)}\right)\right) T_{a,j}^n +$$

$$\frac{\Delta t}{\rho_a C_a S_{ae}}\left(\frac{2\pi.k_{eff,air}}{\ln(D_{gi}/D_{ae})} + \frac{\sigma.S_{ae}\left(\left(T_{a,j}^2\right)^n + \left(T_{g,j}^2\right)^n\right)\left(T_{a,j}^n + T_{g,j}^n\right)}{\frac{1}{\varepsilon_a} + \frac{1-\varepsilon_g}{\varepsilon_g}\left(\frac{D_{ae}}{D_{gi}}\right)}\right) T_{g,j}^n = T_{a,j}^{n-1} + \frac{\Delta t}{\rho_a C_a S_{ae}} \tau_g . \alpha_{ab} . \rho_m . S_e . R_d(\beta) . k(\theta) \quad (41)$$

For the glass

Equation (10) gives: $\rho_g . C_g . S_g . \frac{T_{g,j}^n - T_{g,j}^{n-1}}{\Delta t} = \left(q_{a,g}(t) - q_{g,amb}(t)\right)$

By replacing $q_{a,g}(t)$ and $q_{g,amb}(t)$ by their expressions, we get:

$$-\frac{\Delta t}{\rho_g C_g S_g}\left(\frac{2\pi.k_{eff,air}}{\ln(D_{gi}/D_{ae})} + \frac{\sigma.S_{ae}\left(\left(T_{a,j}^2\right)^n + \left(T_{g,j}^2\right)^n\right)\left(T_{a,j}^n + T_{g,j}^n\right)}{\frac{1}{\varepsilon_a} + \frac{1-\varepsilon_g}{\varepsilon_g}\left(\frac{D_{ae}}{D_{gi}}\right)}\right) T_{a,j}^n + \left(1 + \frac{\Delta t}{\rho_g C_g S_g}\left(\frac{2\pi.k_{eff,air}}{\ln(D_{gi}/D_{ae})} + \frac{\sigma.S_{ae}\left(\left(T_{a,j}^2\right)^n + \left(T_{g,j}^2\right)^n\right)\left(T_{a,j}^n + T_{g,j}^n\right)}{\frac{1}{\varepsilon_a} + \frac{1-\varepsilon_g}{\varepsilon_g}\left(\frac{D_{ae}}{D_{gi}}\right)} + \right.\right.$$

$$h_V.S_{ge} + \varepsilon_g.\sigma.S_{ge}\left(\left(T_{g,j}^2\right)^n + \left(T_{amb,j}^2\right)^n\right)\left(T_{g,j}^n + T_{amb,j}^n\right)\Bigg)\Bigg) T_{g,j}^n =$$

$$T_{g,j}^{n-1} + \frac{\Delta t}{\rho_g C_g S_g}\left(h_V.S_{ge} + \varepsilon_g.\sigma.S_{ge}\left(\left(T_{g,j}^2\right)^n + \left(T_{amb,j}^2\right)^n\right)\left(T_{g,j}^n + T_{amb,j}^n\right)\right) T_{amb,j}^n \quad (42)$$

Equations 40, 41 and 42 form a system of equations with three unknowns to be solved.
It is also important to know that time step is one hour [10].
This system can be in the following matrix form:

$$\begin{bmatrix} b_1 & c_1 & 0 \\ a_2 & b_2 & c_2 \\ 0 & a_3 & b_3 \end{bmatrix}\begin{bmatrix} T_{f,j} \\ T_{a,j} \\ T_{v,j} \end{bmatrix}^n = \begin{bmatrix} d \\ e \\ g \end{bmatrix}$$

Where:

$$b_1 = 1 + \frac{\Delta t.\dot{V}}{\Delta z.S_{ai}} + \frac{\Delta t}{\rho_f(j)C_f(j)S_{ai}} h_f.S_{ai}$$

$$c_1 = \frac{\Delta t}{\rho_f(j)C_f(j)S_{ai}} h_f.S_{ai}$$

$$d = \frac{C_f(j-1)\rho_f(j-1)\Delta t.\dot{V}}{C_f(j)\rho_f(j)\Delta z.S_{ai}} T_{f,j-1}^n + T_{f,j}^{n-1}$$

$$a_2 = \frac{\Delta t.h_f.S_{ai}}{\rho_a C_a S_{ae}}$$

$$b_2 = 1 - \frac{\Delta t}{\rho_a C_a S_{ae}}\left(\frac{2\pi.k_{eff,air}}{\ln\left(D_{gi}/D_{ae}\right)} + h_f.S_{ai} + \frac{\sigma.S_{ae}\left(\left(T_{a,j}^2\right)^n + \left(T_{g,j}^2\right)^n\right)\left(T_{a,j}^n + T_{g,j}^n\right)}{\frac{1}{\varepsilon_a} + \frac{1-\varepsilon_g}{\varepsilon_g}\left(\frac{D_{ae}}{D_{gi}}\right)}\right)$$

$$c_2 = \frac{\Delta t}{\rho_a C_a S_{ae}}\left(\frac{2\pi.k_{eff,air}}{\ln\left(D_{gi}/D_{ae}\right)} + \frac{\sigma.S_{ae}\left(\left(T_{a,j}^2\right)^n + \left(T_{g,j}^2\right)^n\right)\left(T_{a,j}^n + T_{g,j}^n\right)}{\frac{1}{\varepsilon_a} + \frac{1-\varepsilon_g}{\varepsilon_g}\left(\frac{D_{ae}}{D_{gi}}\right)}\right)$$

$$e = T_{a,j}^{n-1} + \frac{\Delta t}{\rho_a C_a S_{ae}} \tau_g.\alpha_{ab}.\rho_m.S_e.R_d(\beta).k(\theta)$$

$$a_3 = -\frac{\Delta t}{\rho_g C_g S_g}\left(\frac{2\pi.k_{eff,air}}{\ln\left(D_{gi}/D_{ae}\right)} + \frac{\sigma.S_{ae}\left(\left(T_{a,j}^2\right)^n + \left(T_{g,j}^2\right)^n\right)\left(T_{a,j}^n + T_{g,j}^n\right)}{\frac{1}{\varepsilon_a} + \frac{1-\varepsilon_g}{\varepsilon_g}\left(\frac{D_{ae}}{D_{gi}}\right)}\right)$$

$$b_3 = 1 + \frac{\Delta t}{\rho_g C_g S_g}\left(\frac{2\pi.k_{eff,air}}{\ln\left(D_{gi}/D_{ae}\right)} + \frac{\sigma.S_{ae}\left(\left(T_{a,j}^2\right)^n + \left(T_{g,j}^2\right)^n\right)\left(T_{a,j}^n + T_{g,j}^n\right)}{\frac{1}{\varepsilon_a} + \frac{1-\varepsilon_g}{\varepsilon_g}\left(\frac{D_{ae}}{D_{gi}}\right)} + h_V.S_{ge} + \varepsilon_g.\sigma.S_{ge}\left(\left(T_{g,j}^2\right)^n + \left(T_{amb,j}^2\right)^n\right)\left(T_{g,j}^n + T_{amb,j}^n\right)\right)$$

$$g = T_{g,j}^{n-1} + \frac{\Delta t}{\rho_g C_g S_g}\left(h_V.S_{ge} + \varepsilon_g.\sigma.S_{ge}\left(\left(T_{g,j}^2\right)^n + \left(T_{amb,j}^2\right)^n\right)\left(T_{g,j}^n + T_{amb,j}^n\right)\right) T_{amb,j}^n$$

Which give: $[A][T] = [B]$

$$\text{Or } [A] = \begin{bmatrix} b_1 & c_1 & 0 \\ a_2 & b_2 & c_2 \\ 0 & a_3 & b_3 \end{bmatrix};\ [T] = \begin{bmatrix} T_{f,j} \\ T_{a,j} \\ T_{v,j} \end{bmatrix};\ [B] = \begin{bmatrix} d \\ e \\ g \end{bmatrix}$$

The matrix A of system being tridiagonal, we will use Gaussian pivot method for resolution of our problem. Computation is carried out in two stages: triangularization and determination of unknowns.

Which give:

$$T_{a,j}^{n} = \frac{b_1(eb_3 - c_2 g) - b_3 a_2 d}{b_1(b_2 b_3 - c_1 a_2) - b_2 c_2 a_3}$$

$$T_{f,j}^{n} = \frac{db_1(b_2 b_3 - c_1 a_2) - db_2 c_2 a_3 - c_1 b_1(eb_3 - c_2 g) + c_1 b_3 a_2 d}{b_1^2(b_2 b_3 - c_1 a_2) - b_1 b_2 c_2 a_3}$$

$$T_{g,j}^{n} = \frac{gb_1(b_2 b_3 - c_1 a_2) - gb_2 c_2 a_3 - a_3 b_1(eb_3 - c_2 g) + a_3 b_3 a_2 d}{b_3 b_1(b_2 b_3 - c_1 a_2) - b_3 b_2 c_2 a_3}$$

8. Parameters Used in Numerical Simulation

Table 1. *Geometric parameters.*

Geometric characteristics	Value
Internal diameter of absorber	1.25 cm
External diameter of absorber	1.4 cm
Internal diameter of glass	1.75 cm
External diameter of glass	2 cm
Effective mirror width	1 m
Number of collector, Ntotal	3
Length element, Δz	0.1 m
Tube length of each manifold, Ltube	5 m
Focal distance	0.235 m

Table 2. *Optical properties of materials.*

Optical properties of materials used	Value
Absorption coefficient of absorber tube, α	0.8
Glass transmitivity, τ_g	0.8
Mirror surface reflectance, ρ_m	0.85
Emissivity of the absorber tube (visible), ε_a	0.12
Glass tube emissivity, ε_g	0.9

A computer program written in Matlab language has been developed to simulate outlet temperature of heat transfer fluid, absorber and glass of cylindro-parabolic solar concentrator.

9. Result and Discussion

In this part, we present results of numerical simulation, as well as the effect of some geometric and climatic parameters [10] on evolution of outlet temperature Tf.

9.1. Evolution of Outlet Temperature

Figure 4 shows the variation in temperatures of fluid, absorber and glass at outlet of absorber tube as a function of time.

1) For a typical day of year, for a constant volume flow, outlet temperature of heat transfer fluid reaches its maximum value around 400 K recorded at true solar noon for day of March 21. It mainly depends on $q_{ab}(t)$, which is a function of optical and geometric parameters of concentrator and of direct radiation received by sensor.
2) Note that temperature of absorber (Ta) is almost close to temperature (Tf) of fluid, which can reach 405 K at true solar noon for day of March 21. This can be justified by its high absorption power for visible solar radiation and low emissivity for long-wavelength infrared radiation, which is provided by its coating. It is possible to conserve most of incident solar energy and lose very little heat by long-wavelength radiation when absorbent surface becomes hot.
3) The glazing temperature (Tv) is lower than (Ta) and (Tf) which can reach 375K at true solar noon for day of March 21, because internal face of glass absorbs infrared radiation, which undergoes an increase of temperature (Tv) (greenhouse effect). Therefore that of external face is lower, which is close to the ambient environment subjected mainly to the speed of wind, which creates a phenomenon of convection on glass outside.

9.2. Effects of Geometric Parameters on Fluid Temperature

9.2.1. Influence of Absorber Length

Figure 5 represents fluid temperature variation at the outlet as a function of length of absorber tube for three values of direct radiation (1000 W / m^2, 700 W / m^2 and 500 W / m^2). Note that increase in fluid temperature (Tf) is proportional to length and intensity of solar flow.

9.2.2. Effect of Reflector Width (Concentrator Opening)

To see the effect of width of mirror on fluid temperature (Tf), we have chosen three values, 2m, 1m and 0.5m, where we notice in figure 6, the maximum value of (Tf) corresponds to the width of 2m (570 K) for day of March 21.

9.3. Thermal Losses

Absorber is the site of thermal losses [9]. Figure 7 shows change in coefficient of heat losses as a function of average temperature of absorber for three values of emissivity ε_p: 0. 9, 0.5 and 0.2. Note that losses increase with increase of absorber average temperature. The emissivity of 0.2 could further reduce radiation losses by adopting selective surfaces. Eliminating air between transparent glass casing and absorber (creating a vacuum) could significantly reduce convection losses.

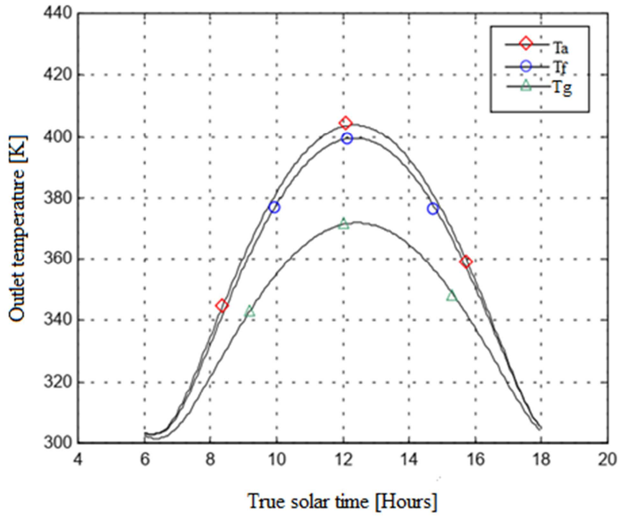

Figure 4. *Outlet temperature evolution of fluid (Tf), absorber (Ta) and glass (Tg) for March 21.*

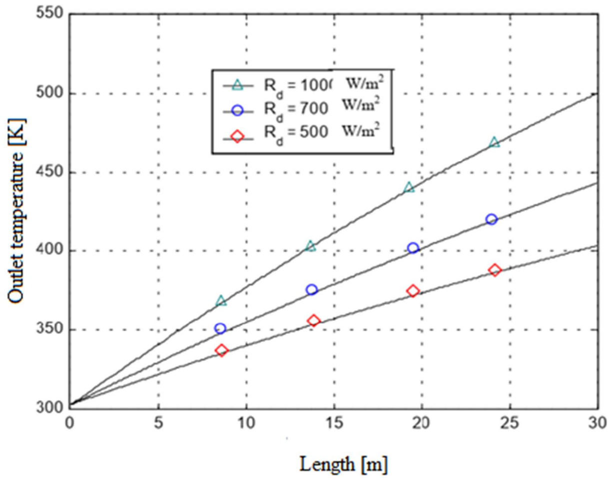

Figure 5. *Outlet temperature evolution as a function of length of absorber tube.*

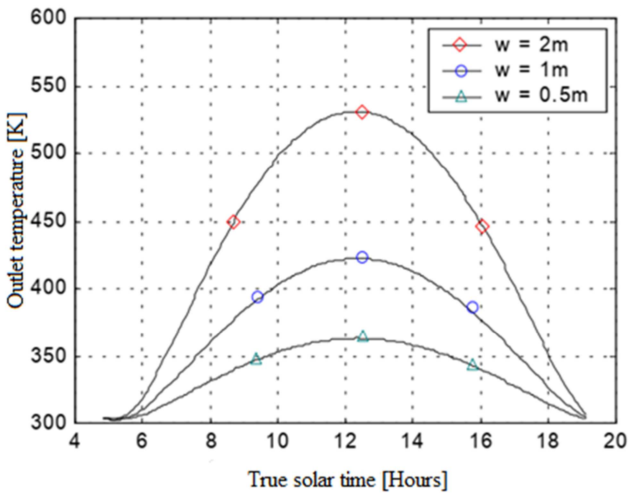

Figure 6. *Evolution of temperature Tf of fluid as a function of width of mirror.*

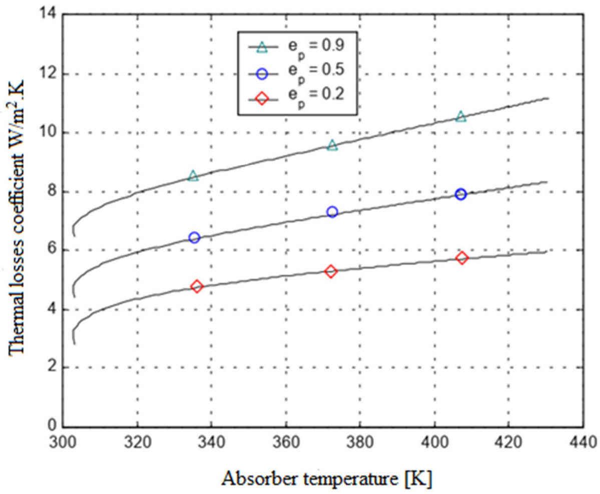

Figure 7. *Evolution of coefficient of thermal losses as a function of absorber temperature for different emissivities.*

10. Conclusion

This work proposes a numerical simulation of heating of heat transfer fluid which circulates in an absorber tube of a solar collector with a cylindro-parabolic type concentration effect.

From heat exchanges that take place, a mathematical model allowing monitoring of temperature of fluid, absorber and glass is established. The simulation of heating process of fluid is done in time steps of one hour, from 6 a.m. to 6 p.m. The results obtained show that temperature difference between input and output of sensor is very large.

A computer program has been developed to simulate temperatures of fluid, absorber tube and glass as a function of time and space.

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