Analysis of the energetic feasibility of cylindro-parabolic collectors integrated in solar towers in Adrar area.

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Abstract - In solar energy, the cylindro-parabolic collector plays a capital role in the production of the high temperatures, great than 250°C. Being very robust and with a lifetime relatively rather long, the cylindro-parabolic collectors are used to produce the electricity in the solar towers in the Adrar area, in Algerian Sahara, witch characterized by a high density of sunning 71.259 KWh/m^2.yr. In order to analyze its profitability, we proceeded to the mathematical modeling of the various elements of the cylindro-parabolic collector to lead to the advection equation by the recourse to the thermal balances in transient regime. A program, in language FORTRAN 90, was written to solve the system of algebraic equations system, deduced from the discretization method of the model equations. For one day typical and a volumetric flux of 0.6m^3/h, the simulation results highlighted a significant profit in temperature at the level of the refrigerant which could reach 200°C. Also, it was highlighted the impact of the sunning intensity and the collector length on the absorber temperature, T_1. The temperatures distributions in the absorber wall, T_2, and that of the envelope in glass, T_3, were calculated and we showed that its shape follows a Gaussian law whose the peak is reached in neighborhoods' noon. Such a temperature has a consequence on the vapor production, with a pressure to 18 bars, capable to convey the turbine intended for generation of electricity. In addition, the program enabled us to examine the influence of the rate flow of refrigerant on the diurnal distribution of its temperature at outlet of the absorber tube.

Key words: Solar energy, cylindro-parabolic collector, absorber, heat balance, finite differences.

Nomenclature

α	Absorptivity	[--]
τ	Transmissivity	[--]
ρ	Density of fluid	kg/m^3
λ	Thermal conductivity	W/m K
μ	Viscosity of fluid	Pa.s
β	Coefficient of volume expansion	K^{-1}
A	Cross section surface	m²
D	Diameter of collector	m
m	Statistical average	hr
σ^2	Statistical variance	hr^2
R_d	Global irradiation received	W/m^2
$\dot{Q}$	Heat flux exchanged	W
h	Coefficient of heat transfer	W/m^2 K
L	Length of the collector	m
Q_v	Refrigerant flow	m^3/s
V	Refrigerant flow rate	m/s
C_p	Specific heat of fluid	J/kg K
Nu	Nusselt number	[---]
Re	Reynolds number	[---]
Pr	Prandtl number	[---]
Re	Rayleigh number	[---]
Δz	Space width of integration step	m
Δt	Time Width of integration step	s
T	Temperature	K

Subscripts and superscripts

1	Refrigerant
2	Absorber wall
3	Enveloppe
ab	Absorbed, absorber.
Amb	Ambient
Ray	Radiation
Ext	Exterior
Int	Interior
Moy	Average
fs	Output refrigerant

I. Introduction

The energy constitutes an essential pillar for the development for the societies. Subject of scientific actualities, renewable energies (Solar, wind, biomass. etc.) represent an alternative which one will be able to consider with condition to develop of the thermal processes for their valorization. Although, the accumulated experience is considerable, this field has known a fulgurating technological development as well on the theoretical and experimental levels during three last decades. Indeed, in the thermodynamic field, its applications are numerous. As an example, the heating of water, the drying and the thermal centrals for the production of electricity are the scientific fields which contributed to this development via the integration of the solar collectors. The performances of the installation to which the collector belongs depend essentially on the useful energy that one always seeks to promote by:

- Increasing of the heat received by the absorber by elevation of the ratio of the absorptive radiation (coating of the absorber surface) or their concentration on a focal point.
- Reduction of the thermal losses towards the non receiving areas (needs a good thermal insulation) and forwards collector (between absorber and environment) .

The diversification of the use of this inexhaustible source led the scientific community to seek the geometrical forms in order to concentrate the incidental radiation. The form which lends itself is the parabolic shape. However, it is necessary to go gradually towards the hybrid systems in the medium term to flex this temperature increasing related organically to the fossil energy exploitation whose its corollary is the greenhouse gases. While being interested by the average temperatures [200°C, 500°C], the bibliographical analysis shows that the solar concentrators meet this aim rather well. However, we will limit our self to study the cylindro-parabolic collectors with the cheap refrigerants which could convey the captured energy at the absorber.

II. Solar radiation in Algeria

Algeria has a significant solar sink. From its climate, the maximum solar density in any point (clear sky, June) exceeds the 6 kW/m^2 and maximum annual received energy in Algeria is close to 2500 kW/m^2[1].

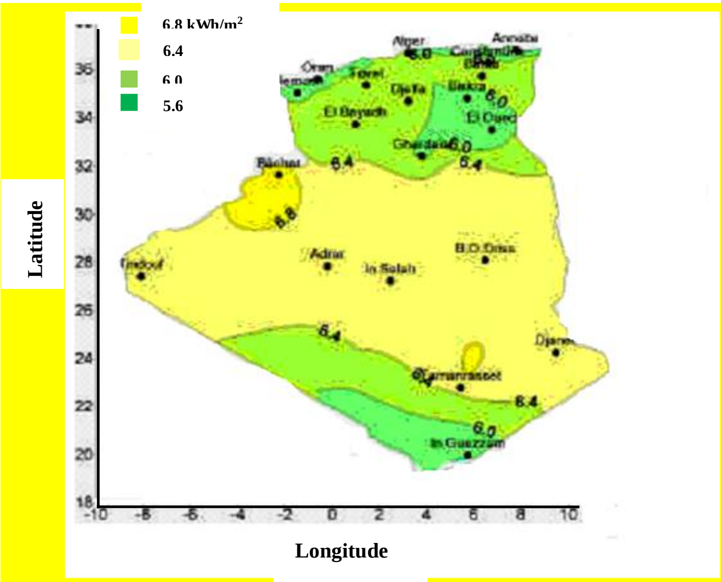

Fig. 1. The solar irradiation in various areas in Algeria.

The implantation of solar installations, dictated by the judicious choice of the preferred sites for a better collecting, is conditioned by the sweeping of all the territory (48 Provinces) which will enable us to select the sunniest cities for which the collector efficiency is optimal (figure 1). From its strategic importance, the area of Adrar (South-West of Algeria) is the favorable area for such projects. However, the problem of transport of energy produced could constitute a handicap which one must free oneself.

III. Definition of the Concentration Ratio

It is an indicator of the quantity of energy delivered per unit of area of a given collector. One distinguishes three definitions from the concentration [2]:

- Ideal concentration (Gauss)
- Geometrical concentration
- Optical concentration (energy)

The solar capturation can be carried out according to two manners:

III.1. Collecting without concentration

In this form of capturation, the solar radiation is transformed in heat at low temperature. The majority of the collectors are provided with a transparent cover transparent to the incidental solar radiation [3]. The flat plate collectors are covered with an anti-reflecting glass whose the role is to protect them from the environmental influences. Behind the glass tube, it is the absorber composed from the conduits crossed by a refrigerant which conveys the energy towards its place of storage or use [4]. Below, is placed the thermal isolation system in order to reduce the thermal losses. This equipment is integrated in a light trunk made up often of aluminum, synthetic matter or sheet steel. Its total weight per square meter of surface doesn't exceed 20 kilograms [5] (figure 2). In the case of a pointed concentrator with a flat absorber, the effective geometric concentration is equal to one.

The using temperature is generally less than 70 °C. Seldom, some of them, provided with a transparent cover of type «double glass", make it possible to reach operating temperatures close to 100 °C [3].

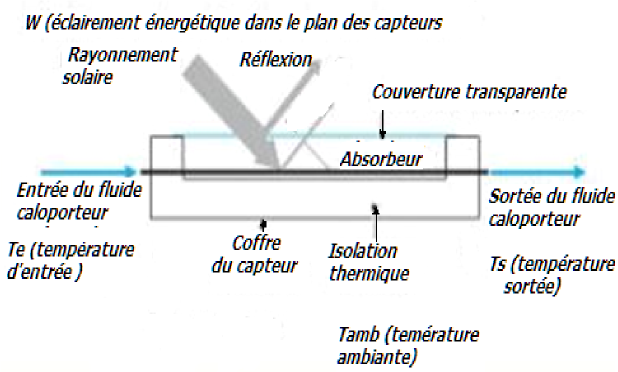

Fig. 2. Sketch of the plane collector with thermal interactions.

III.2. Collecting with concentration

In this form of collecting, several types of concentrators have been made to this day which we quote:

- The conical concentrator,
- The spherical concentrator,
- The cylindro-parabolic concentrator and,
- The heliostat field collector;
- The parabolic concentrator.

This third one, which will be the subject of this study, was used for a long time, because it ensures a high level of temperature and power [6]. Any luminous ray parallel with axis will be deviated by the reflective surface of the parabola towards the focal point F. The receiver is placed concentrically along this focal line. In order to optimize its performance, Balbir Singh et al. [7] have clearly showed that there must be equilibrium between the increasing thermal losses with the increasing aperture area. Figure 3 defines the various magnitudes characterizing the operating conditions of a collector with concentration.

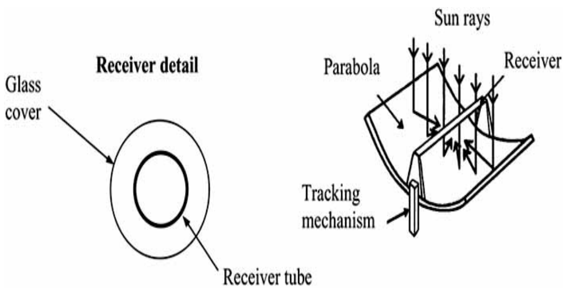

Fig. 3. Schematic diagram of a cylindro-parabolic collector.

For a standard CPC, we distinguish:

III.2.1. Reflector

With its parabolic shape, the reflector is an element of the collector, designed to transmit all the heat to the refrigerant which crosses the absorber (receiver). The more efficient would be a stainless plate with a polished face, make up from mirrors and protected by a plastic film. For that, the plate is constituted of metals having a high thermal conductivity (copper: 300W/m.K, aluminum: 200 W/m.K, steel: 60 W/m.K). Indeed, the increasing of its price depends entirely on the the thermal conductivity as well as the thickness. The mirrors of the reflectors, with a cylindro-parabolic shape, are easily constructed and less expensive. Comparatively to the parabolic collectors, their less low concentration ratios limit their exploitation to large fields of the chemical industry (Industry of cement, etc).

III.2.2. The Absorber

It is contained in a tube of glass, in vacuum. The absorber walls would practically be at the same temperature as with the periphery following a plain contact with the solar rays which cross the cover in glass (polycarbonate, the methacrylate or the tedlar) which plays the screen role to carry out the greenhouse effect. In order to increase the refrigerant temperature, one uses collectors thermally isolated. The reflectors focus is a copper tube (high thermal conductivity), painted in black, in order to improve the transfer of the radiative power to the refrigerant (water + antifreeze) to convey it with the storage tank.

IV. Modeling of the Cylidro-parabolic Collector

In order to study the thermal performance of the solar concentrator, the space-time profiles of the temperature are to be determined by the establishment of enthalpy balances on a differential element whose the length is dz. Taking into account its structure, the sketch of the collector exhibiting the thermal interactions between various parts is presented on the figure 4.

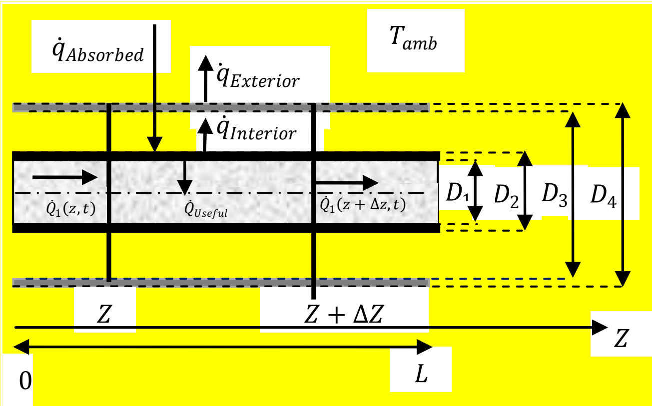

Fig. 4. Diagram of the absorber of a cylindro- parabolic collector.

Having emitted following simplifying assumptions:

- Negligible thermal losses
- Absence of phase change for the refrigerant.
- Constancy of the density and the specific heat,

the model is reduced to the balances on:

IV.1.The refrigerant

Generally, the refrigerants are essentially selected according to their thermal stability and their viscosity. Moreover, the diurnal variation of the sunning intensity imposes the taking in consideration the transient regime. Thus, the enthalpy balance is written:

$$\frac{\partial}{\partial t}\left(Q_1(z,t)\right) = \dot{Q}_1(z,t) - \dot{Q}_1(z+\Delta z,t) + \dot{q}_{usefl} * \Delta z \qquad (1)$$

where $q^{\cdot}_{usefl}$ is the useful power exchanged per unit of length of the tube between the refrigerant and the interior walls of the tube. The classical thermodynamics enables us to express the flow enthalpy by the following relation:

$$\dot{Q}_1(z,t) = \rho_1.c_{p_1}.\dot{Q}_v * \left(T_1(z,t) - T_{ref}\right) \qquad (2)$$

with, , A_1, the cross-section of the tube ($= \pi {D_1}^2/4$) and $T_{ref.}$ is the reference temperature , given by the conditions:

- Water: saturated liquid
- $T_{ref.} = 0°C$.

In addition, the power accumulated in the volume of control, $dv = A_1.\Delta z$, is expressed by:

$$\frac{\partial Q_1(z,t)}{\partial t} = \rho_1.c_{p_1}.A_1.\Delta z * \frac{\partial T_1}{\partial t} \qquad (3)$$

The combination of the equations (1), (2) and (3) makes it possible to establish the relation between the temperature of the refrigerant, , T_1, in any point in the collector and the energy exchanged via the absorber walls. Finally, the balance can be written:

$$\rho_1.c_{p_1}A_1\Delta z\frac{\partial T_1(z,t)}{\partial t} = \rho_1.c_{p_1}.\dot{Q}_v * [T_1(z,t) - T_1(z+\Delta z,t)] + q^{\cdot}_{usefl}(z,t).\Delta z. \qquad (4)$$

Introducing the flow rate, $\left(v = \dot{Q}_v \,/\, A_1\right)$ and taking into account the conditions initial and boundary conditions, the descriptive model of the profile of temperature is:

$$\begin{cases} \dfrac{\partial T_1(z,t)}{\partial t} + v * \dfrac{\partial T_1(z,t)}{\partial z} = \dfrac{q^{\cdot}_{usefl}(z,t)}{\rho_1.c_{p_1}A_1} & (5) \\ Z = 0,\ \forall t: \quad T_1(z,t) = T_f^e & (6) \\ t = 0,\ \forall z: \quad T_1(z,t) = T_{amb} & (7) \end{cases}$$

The equations (6) and (7) represent the conditions of inlet of the refrigerant and the initial condition to the absorber before the functioning start of the collector. In order to estimate the power received per unit of length, we introduce the transfer coefficient of heat, h_1 which is expressed by:

$$\dot{q}_{uusefl} = \pi D_1 h_1 (T_2 - T_1) \qquad (8)$$

By considering that the fluid undergoes an increasing of the temperature, and assuming that the flow is fully turbulent, the coefficient of heat transfer is expressed by the Dittus Boelter equation:

$$Nu_1 = \frac{h_1 * D_1}{\lambda_1} = 0.023 * Re_1^{0.8} * Pr_1^{0.4} \qquad (9)$$

with $\left(\frac{D_1+D_2}{2}\right) \cong D_1 \cong D_2$ and the properties of the fluid are estimated at average temperature of the film Re_1 and Pr_1 are Reynolds and Prandtl numbers, given by:

$$Re_1 = 4\frac{\rho_1 \dot{V}}{\pi D_1 \mu_1} \qquad (10)$$

$$Pr_1 = \frac{\mu_1 Cp_1}{\lambda_1} \qquad (11)$$

IV.2. The Absorber and the shell

The fluid, responsible for heat transfer between the wall of the absorber and the outer shell, is practically in a motionless state. It results a significant accumulation of energy from it in this elementary volume, $dV = \frac{\pi}{4}(D_3^2 - D_2^2).dz$. The analysis of heat exchange between the two walls leads to the writing of the heat balance:

$$\frac{\partial}{\partial t}\left(\Delta Q_2(z,t)\right) = \left(\dot{q}_{ab}(t) - \dot{q}_{int}(z,t) -\right)\Delta z \qquad (12)$$

Where $\dot{q}_{ab}(t)$ is the absorbed solar power which depends on the diurnal variation of the sunning intensity; whereas $\dot{q}_{int}(z,t)$ is the thermal power exchanged between the absorber and the outer shell. A similar analysis relating to the space confined between the envelope and the absorber pipe shows that:

$$\Delta Q_2(z,t) = \rho_2 Cp_2 A_2 \Delta z * (T_2(z,t) - T_{ref}) \qquad (13)$$

where A_2 is the surface of the annular section, $A_2 = \frac{\pi}{4}(D_3^2 - D_2^2)$. The equation of the balance is reduced to:

$$\begin{cases} \dfrac{\partial T_2(z,t)}{\partial t} = \dfrac{1}{\rho_2 Cp_2 A_2}\left(\dot{q}_{ab}(t) - \dot{q}_{int}(z,t) - \dot{q}_{usefll}(z,t)\right) & (14) \\ Z = 0, \forall t: \quad T_2(z,t) = T_f^e & (15) \\ t = 0, \forall z: \quad T_2(z,t) = T_{amb} & (16) \end{cases}$$

The equations (15) and (16) represent the imposed conditions taking into account the operating conditions of the collector. In the interval included between the pipe and envelope, we note the coexistence of the two heat transfer modes (free convection and radiation):

$$\dot{q}_{int} = \dot{q}_{int}^{conv} + \dot{q}_{int}^{ray} \quad (17)$$

For two cylinders whose surfaces are maintained at temperatures T_2 and T_3 constants, the linear density of exchange by free convection is:

$$\dot{q}_{int}^{conv} = \left(2\pi\lambda_{eff}/\ln\left(\frac{D_3}{D_2}\right)\right) * (T_2 - T_3) \quad (18)$$

The λ_{eff} value is given by Raithby and Hollands [8] whose the expression is:

$$\frac{\lambda_{eff}}{\lambda} = 0{,}386 * \left(\frac{Pr^{air}}{0{,}861 + Pr^{air}}\right)^{1/4} * \left(F_{cyl} * Ra_L\right)^{1/4} \quad (19)$$

Where F_{cyl} is the geometrical factor for concentric cylinders and Ra, the Rayleigh number whose the expressions are:

$$F_{cyl} = \frac{\left[\ln\left(\frac{D_3}{D_2}\right)\right]^4}{L^3\left(D_2^{-3/5} + D_3^{-3/5}\right)^5} \quad (20)$$

$$Ra_L = \frac{g.\beta_{air}.a_{air} * (T_2 - T_3)}{\mu_{air}\,\alpha_{air}} L^3 \quad (21)$$

with, $L = \frac{D_3 - D_2}{2}$, the characteristic length. The physical properties of the air are evaluated at the average temperature $T_{Ave} = \frac{T_2 + T_3}{2}$. If the space is occupied by air, which is supposed to obey to ideal gas law, the volume expansion coefficient is:

$$\beta_{air} \cong \frac{1}{T_{Ave}\,(K)} \quad (22)$$

Being at different temperatures, it creates a irradiative heat exchange between the two concentric cylinders, perfectly polished. The transferred power per unit of length is:

$$\dot{Q}_{int}^{ray} = \frac{\sigma\pi D_2 * (T_2^4 - T_3^4)}{\frac{1}{\varepsilon_{Abs}} + \left(\frac{1-\varepsilon_3}{\varepsilon_3}\right)\left(\frac{D_2}{D_3}\right)}. \quad (23)$$

Where ε_{Abs} and ε_3, are the emissivity of the absorber and that of the envelope glass. σ is the constant of Stefan-Boltzmann.

IV.3.The Shell and Environment

On the basis of diagram representing the collector functioning, the enthalpic balance can be expressed by:

$$\rho_3.Cp_3.A_3.\Delta z \frac{\partial T_3(z,t)}{\partial t} = \left(\dot{q}_{ab}(t) - \dot{q}_{ext}(z,t)\right)\Delta z \quad (24)$$

After simplification, this equation is reduced to:

$$\begin{cases} \frac{\partial T_3(z,t)}{\partial t} = \frac{1}{\rho_3.Cp_3.A_3}\left(\dot{q}_{ab}(t) - \dot{q}_{ext}(z,t)\right) \\ t = 0, \forall z: \quad T_3(z,t) = T_{amb} \end{cases} \quad (25)$$

where $A_3 = \frac{\pi}{4}(D_4^2 - D_3^2)$ whereas Cp_3 and ρ_3 are the physical properties of glass at the ambient temperature. In the developed model, the heat exchanged is the superposition of the two modes:

- Forced convection due to the circulation of the wind which the expression is:

$$\dot{q}_{ext}^{conv} = h_3 * \pi D_4 (T_3 - T_{Amb}) \quad (26)$$

With $D_3 \cong D_3 \cong \frac{D_3 + D_4}{2}$ and h_3 the coefficient of heat transfer by forced convection. In terms of adimensional numbers, he is written:

$$Nu_3 = \frac{h_3 D_4}{\lambda_{air}} = 0{,}3 + \frac{0{,}62 * Re_3^{1/2} * Pr_3^{1/3}}{\left[1 + (0{,}4/Pr_3)^{2/3}\right]^{1/4}} * \left[1 + \left(\frac{Re_3}{282000}\right)^{5/8}\right]^{4/5} \quad (27)$$

The Prandtl number and the physical properties of the air are estimated at the average temperature,

- Radiation: The contribution of the radiation to heat exchange from the envelope is written:

$$\dot{q}_{ext}^{ray} = \varepsilon_3 * \sigma * \pi D_4 (T_3^4 - T_{Amb}^4) \quad (28)$$

IV.4. Numerical Analysis of the Model.

The mathematical model is strongly nonlinear and its analytical resolution is quasi impossible. With this intention, the recourse to the finite differences method leads to an algebraic equations system. The discretization of the advection equation is written:

$$T_{1,i}^{P+1} = T_{1,i}^{P} - \frac{v * \Delta t}{\Delta z} * \left(T_{1,i}^{P} - T_{1,i-1}^{P}\right) + \frac{\dot{q}_{usefl}}{\rho_1 Cp_1 A_1} \quad (29)$$

It is clear that the calculation of T_i^{P+1} is a linear interpolation of $T_{1_{i-1}}^{P}$ and $T_{1_i}^{P}$. The analysis of the method stability is given by the condition CFL (Courant-Friedrichs-Levy) which is formulated by the relation:

$$C = \frac{v * \Delta t}{\Delta z} \ll 1 \quad (30)$$

For the calculation of temperatures, we elaborated a program in FORTRAN language. Taking account of the inherent factors with the geographical area and the typical day from the month considered, the program could be able to estimate the received power. The flowchart (figure 5) gives the principal steps for one typical day of June, at Adrar area, namely on June, 11th which would correspond to the 162 day of the year and without holding account weather factors (trouble, clouds etc). The results refer to a CPC length, $L = 30m$, with an average concentration ratio of 80 [9, 10].

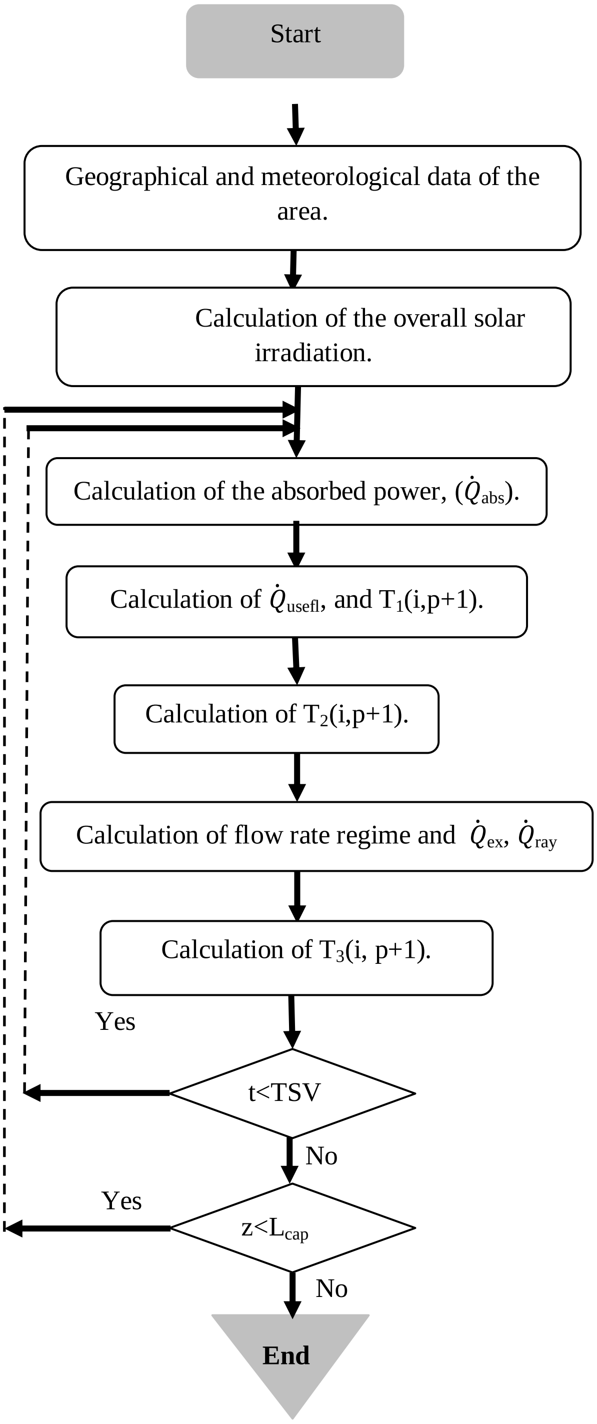

Fig. 5. Flowchart for calculation of the CPC.

V. Results and Discussion

V.1. Diurnal variation of the temperature of the output fluid

After having chosen geometrical size of the absorber:

$$D_1 = 0.066m; \ L=30m; \ Q_V = 0.6 \ m^3/sec.$$

and under the effect of a variable sunning along one day of June, with a real solar time close to 13hours, in the Adrar area, the results of simulation relating to $T_{fs}=T_1(z=L,T)$ are presented on figure 6. It is noted that $T_{fs}=T_1(z=L, T)$ follows a Gaussian law during the sunning period and T_{fs} reaches its low values with the sunrise at its sunset. The value maximum $(T_{fs})_{max}$ is close noon (midday T.U) and it be practically equal with 470.8K One notice that the CPC is relatively powerful so that it be exploited with some industrial and domestic uses for the heating. Beyond of sunset, the refrigerant temperature falls abruptly to keep a value of 345K what is insufficient, within thermal profitability, unless it is integrated a heat pump. That is the technique largely used in the solar towers for the generation of electricity by incorporation cycles thermodynamic of Stirling type or Brayton.

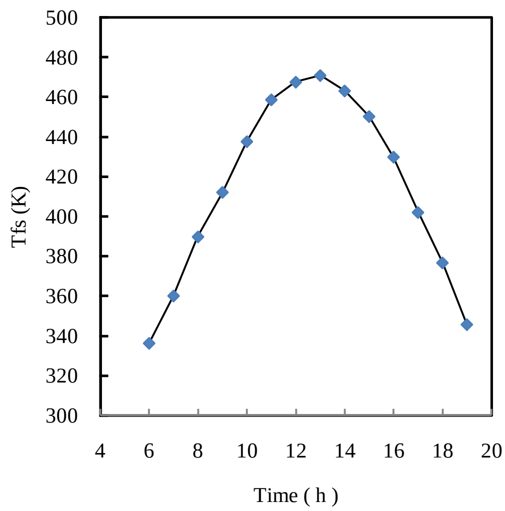

Fig. 6. Diurnal variation of the temperature of the output fluid from absorber pipe.

V.2. Diurnal variation of the temperature of the tube of the absorber

The mathematical analysis of the function $T_2=T_2$ (t) shows that the method of its integration is the Euler method of Euler, i.e.:

$$T_2(t+\Delta t) = T_2(t) + \Delta t * f_1(T_1, T_2, T_3, physical\ properties\ of\ fluids) \quad \ldots\ldots\ldots(31)$$

The results of simulation relating to $T_{ab}=T_2(t)$ are presented on figure 7. In an identical way, the shape of the function $T_{ab} = T_2(t)$ follows a law similar to that of $T_{fs} = T_1(t)$, i.e. that the two profiles reach their peaks at the same time. However, it is noticed that $T_2 > T_1$ during all the sunning period what the heat flow is oriented from the absorber wall towards the flowing refrigerant. It is reasonable to regard the heat transfer in a transient mode.

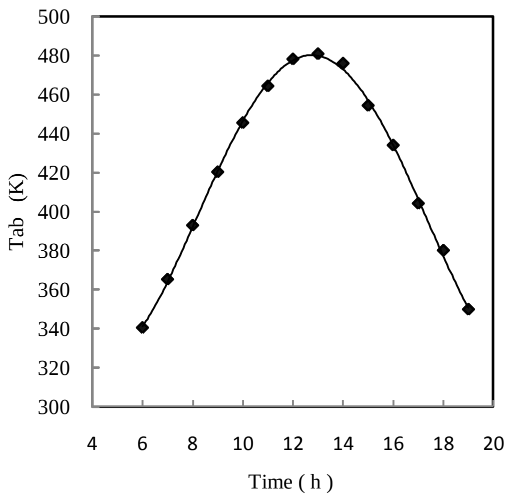

Fig. 7. Diurnal variation of the temperature of the absorber wall.

V.3. Diurnal variation of the temperature of the collector envelope

The mathematical analysis of the function $T_{env} = T_3(t)$ shows that it takes a form similar to that of $T_2 = T_2(t)$ and consequently, the method of its integration is the Euler method, announced in the preceding paragraph. Thus, we obtain:

$$T_3(t + \Delta t) = T_3(t) + \Delta t * f_2(T_2, T_3, T_{amb}, physical\ properties\ of\ fluids) \quad \ldots\ldots(32)$$

where f_2 is a function dependent on the variables T_2 T_3 and T_{amb} The results of simulation relating to $T_{env} = T_3(t)$ are presented on figure 8.

In an identical way, variation of the function $T_{env} = T_3(t)$ follows a Gaussian law similar to those of $T_1 = T_1(z = L, t)$ and $T_2 = T_2(t)$. However, we notice that during all the duration of sunning, the results of simulation are easily fitted by Gaussian curve whose equation is:

$$T = B * exp\left(-\frac{(t-m)^2}{2*\sigma^2}\right) \quad \ldots\ldots\ldots\ldots\ldots(33)$$

After the linearization, the values of the parameters (B, m, σ^2) are consigned through table I.

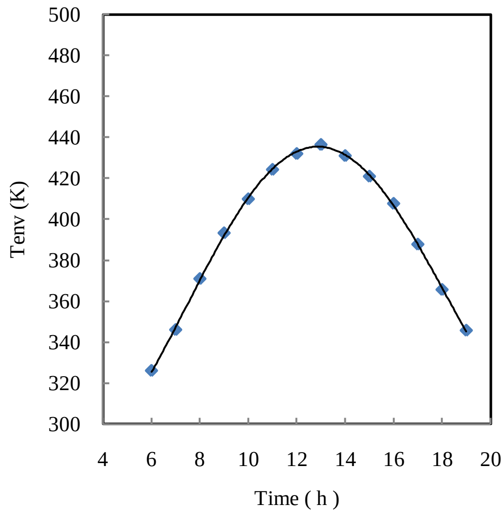

Fig. 8. Variation of the temperature of the collector envelope.

TABLE I
VALUES OF PARAMETERS OF THE GAUSSIAN MODEL FOR THE CURVES T_1, T_2 *and* T_3 .

	m(hour)	A	σ^2	Coefficient of standard deviation, R^2
$T_{fs}=T_1(z=L, t)$	12.5	463.87	61.728	0.9875
$T_{ab}=T_2(t)$	12.5	472.57	60.24	0.9867
$T_{env}=T_3(t)$	12.5	430.78	71.42	0.9923

V. 4. Influence of the length of the pipe absorber on the temperature of the output refrigerant

According to the results of simulation, one notices that the temperature of the refrigerant follows a linear law with the length of the pipe. The results are presented on figure 9. The slope of the curve $T = T(z)$ is:

$$\frac{\Delta T}{\Delta z} = 4.6123\left(\frac{K}{m}\right) \quad \ldots\ldots\ldots\ldots\ldots(34)$$

Such a result enables to predict the possibility of determining, a priori, the length necessary to reach the wished temperature of the refrigerant, on the outlet side of the absorber. The K.M.NG works show a linear relation between temperature and direct solar radiation [11]. Such results corroborate our analysis.

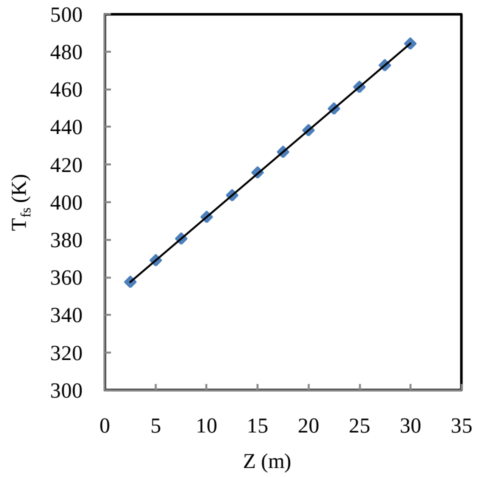

Fig. 9. Influence of the length of the pipe absorber on the temperature of the refrigerant

V. 5. Influence of the volumetric flow on the distribution of the refrigerant temperature at the output side of the absorber

During an interval of time given, the received quantity of energy is independent of the geometrical characteristics and the flow conditions in the collector. Any increasing of the volumetric flow involves an increase in the turbulence effects whose the corollary is an elevation in the intensity of the exchange. The results, presented through figure 10, confirm this assertion. However, for significant residence times, the low flows lead to temperatures which border the results obtained by the turbulence conditions of the flow. Most important of all is the reaching of the maximum temperatures in the temporal interval $[12h, 14h\ 30mn]$.

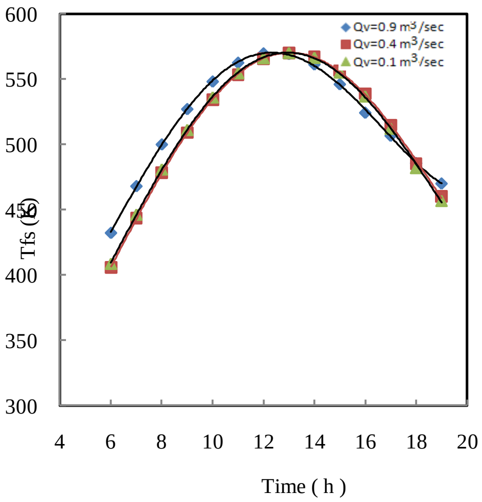

Fig. 10. Effect of the volume flow rate on the distribution of the diurnal temperature of the refrigerant at the output of the absorber

V. 6. Influence of the absorber length and sunning intensity on the temperature of the fluid

We can predict, without calculations that the refrigerant temperature rises in a quasi linear way as well with the length of the absorber pipe as with the sunning intensity. Their impact could be expressed by the ratios $\left(\frac{\Delta T}{\Delta z}\right)$ and $\left(\frac{\Delta T}{\Delta R_d}\right)$. Without the recourse to the simulation technique, it would be impossible to evaluate their exact values. To answer this question, we present the results of the influence of these amounts through the figure 11.

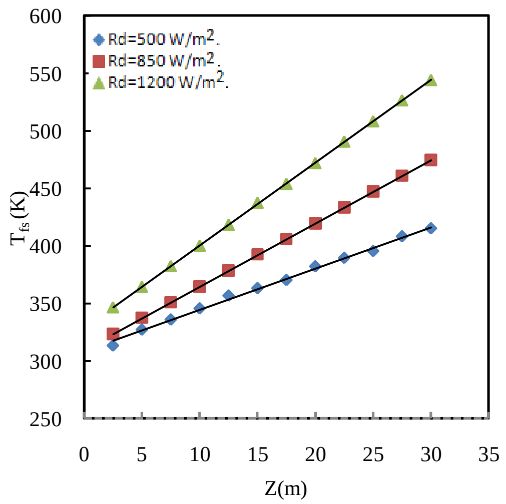

Fig.11. Influence of the absorber length and sunning intensity on the temperature of the fluid.

Their slopes vary with the intensity of sunning, whereas their respective values calculated from the figure 11 are consigned in table II.

TABLE II
IMPACT OF THE INTENSITY OF SUNNING AND LENGTH OF THE TUBE ON THE OUTPUT REFRIGERANT TEMPERATURE

Intensity of sunning (R_d (W/m^2))	500	850	1200
$\left(\frac{\Delta T}{\Delta z}\right)\left(\frac{K}{m}\right)$	3.5779	5.4973	7.2035

To give an overall outline of the interaction of amounts Z and R_d and their impact on the temperature, we consider to plot the curve of variation of de $\left(\frac{\Delta T}{\Delta z}\right) = f(R_d)$ (figure 12) where the data are relating to those given in table II.

The slope of the curve is:

$$\frac{\Delta(\Delta T/\Delta Z)}{\Delta R_d} = 0.0052\left(\frac{mK}{W}\right) \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots(35)$$

It is clear that the length and the sunning intensity contribute favorably to increase the temperature of the working fluid [12, 13]. Thus, for a wished temperature, one can adapt for the choice to these two parameters.

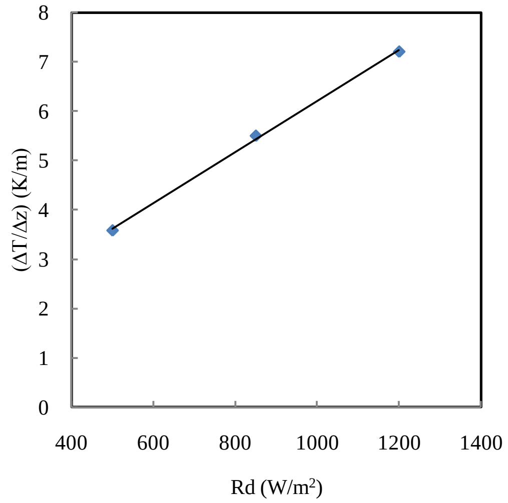

Fig.12. Influence of the sunning intensity on the increasing ratio of the refrigerant temperature.

VI. Conclusions

Solar energy is the source of energy most easily exploitable via solar collectors (thermal or photovoltaic). Thus one must develop this alternative to attenuate the exploitation of fossil energy with his corollary the greenhouse whose consequence effect is the planetary reheating. For geographical and industrial considerations, we select the Adrar area. The mathematical modeling of the operation of cylindro-parabolic collector in unsteady state led to the equation of advection who's the solution is facilitated by the finite differences method. Such a model is the theoretical transcription of the first principle of the thermodynamics applied for each part of the collector. One must mention that the temperatures, at various levels of the collector, follow a law Gaussian law. However, the temperature of the absorber, T_2 remains highest of all. It reaches values close to 480 K what has as a consequence a considerable profit in temperature of the refrigerant, ΔT_1=200 K. Such a rise is reflected by an increasing in pressure of about 18 bars, which leads to the production of a vapor capable to convey a turbine for the generation of the electricity intended to cover the needs for Adrar area. For significant requirements of hot water, the results show that the flow turbulence in the absorber is very positive. Such a situation is reflected by increase in maximum temperatures about 570K. Nevertheless, a similar result is obtained for low flow rates.

Otherwise, the absorber length and sunning intensity contribute to the rise in the temperatures for all the absorber sections. The superposition of these two amounts affects the temperature of the absorber with an increasing ratio . At this step, it is imperatively necessary to validate the model in order to apply it to real cases. Then, the development of such a collector, in real size, is the next stage to verify the model developed in this work.

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1