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Energy Balance and Thermal Evolution Analysis of Heat Transfer Fluids of Stirling Engine and Boiler at Focal Point of a Parabolic Solar Concentrator

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Abstract: A solar concentrator is a technology that converts direct solar radiation into heat. The parabolic concentrator is the best technology for producing electricity from solar energy, because of its high electrical efficiency, about 41%. This technique is the least mature due to the difficulties related to the storage of produced energy. This work concerns a system of two heat receivers, placed at the focus of the parabolic reflector. These two receivers are a boiler and a Stirling engine. The boiler is intended to heat a thermal fluid that will be stored while Stirling engine will produce electricity directly. We studied the thermal balance and the evolution of the temperature of thermal fluids of Stirling engine and the boiler installed at focal point of reflector. The concentrator used is a parabola with surface of 12.6 m². The simulations were carried out in the vicinity of direct radiation measured at 1 pm o'clock local time. The temperature recorded at the focal point varies from 30°C to 900°C for a duration of 80 seconds; when the direct radiation is about 900W/m². This temperature increases from 30°C to 1050°C, for an operating time of 120s. The average temperature of the three fluids in the receiver (permanent fluid in the boiler, heat transfer fluid to be stored, and thermal fluid of Stirling engine) increases from 30°C to over 400°C in less than 1500s. These thermal fluids at this temperature make it possible to operate turbine through the thermal storage system and Stirling engine, to produce electricity.

Keywords: Parabolic Concentrator, Dual Receiver, Heat Boiler, Stirling Engine

1. Introduction

Energy independence is crucial for the socio-economic development of a country. However, the current context where some energy sources such as fossil fuels are tending towards depletion and also the effect of greenhouse gases, pushes the scientific to consider solutions through research in the exploitation and control of other energy sources, in particular renewable energy sources. Among these sources of renewable energies the emphasis is more on solar energy sources.

The exploitation of the solar radiation for the production of electric energy is currently carried out by two different ways [1-3]: the first one, which is the most used, is the photovoltaic technique which allows the conversion of solar radiation directly into electricity; the second technique, which is the least known, is based on a first conversion of direct solar radiation into heat. Then this heat is converted into other forms of energy or stored.

This last technique consists in concentrating the direct radiation of the Sun via an optical system called concentrator to produce heat. There are two categories of concentrating solar systems: linear systems (parabolic trough concentrators and Fresnel concentrators) that concentrate solar radiation towards a linear focal point system (tower systems and parabolic concentrators) that concentrate towards a point [4].

In this work, we will focus on one of the techniques of spot concentration system, namely the concentration technique based on parabolic mirrors. This technique has the best performance and is adaptable to isolated sites [5, 6]. It is the least used because of the difficulty of storing heat and its high initial cost [7, 8].

The work is devoted to the study of a solar concentrator of parabolic type with double receiver. These two receivers are based on the use of two types of thermodynamic cycles: the Stirling cycle and the steam cycle. This last cycle will allow this system to have the possibility of producing electricity in case of solar intermittence.

The general objective of this work is to model a parabolic concentrator with direct electricity production and thermal storage.

As specific objectives we have:

1) to study the thermal balance of the boiler containing the permanent thermal fluid, the tube containing the heat transfer fluid and the cylinder containing the working gas of the Stirling engine;
2) to study the evolution of the temperature at the focal point of the concentrator;
3) to study the evolution of the temperature in the permanent fluid of the boiler, heat transfer fluid of the tube, conducting heat to the storage system and of the working gas of the Stirling engine.

2. The Energy Balance Dual Receiver Parabolic Concentrator

The system consists of the reflector, the boiler, the Stirling engine and the thermal fluids. We make an inventory of the thermal energy distribution from the input to the output of this system.

By application of the first principle of thermodynamics the heat balance of a system can be described by the following equation [9, 10]:

$$\left[\begin{bmatrix} heat\ flow \\ accumulated \end{bmatrix}\right] = \begin{bmatrix} incoming \\ flows \end{bmatrix} - \begin{bmatrix} out \\ flows \end{bmatrix}$$

Figure 1 shows the schematic model of the two receivers of the thermal energy concentrated by the reflective surface of the parabola. The heat concentrated on the surface of the receiver is transferred by conduction to the receiver and then to the thermal fluid boiler and the receiving tube of the Stirling engine. The heat is then transferred from the heat transfer fluid on the inner wall of the receiver to the permanent heat transfer fluid, which in turn transfers the heat it has stored to the coil tube by convection. Heat will be transferred by conduction from the outer wall in contact with the permanent fluid to the inner wall in contact with the heat transfer fluid. In the serpentine tube, the heat is transmitted by conduction from the inner wall of the tube to the heat transfer fluid. Once heated to a defined temperature, the heat transfer fluid is expelled to the thermal storage tank where it will be discharged by exchanging its heat with the materials in the thermal storage tank which will store the thermal power of the fluid [9]. At the same time, heat is transferred from the receiver to the working gas of the Stirling engine, which will produce mechanical work and the latter will produce electricity.

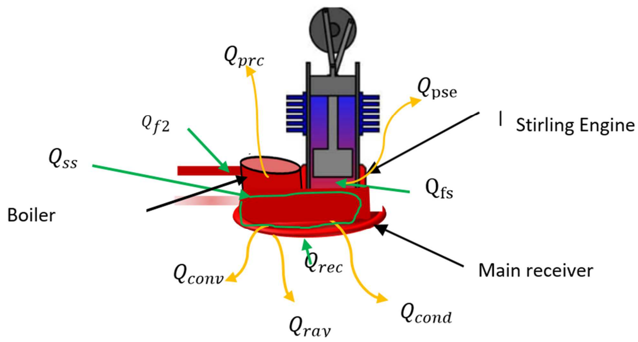

Figure 1. *Modèle schématique du récepteur.*

In this diagram:

Q_{rec}: amount of heat from the concentrator; Q_u: power amount of heat usable by both receivers; Q_{f2} amount of heat received heat transfer fluid; $Q_{s,\in}$: amount of heat received by the gas from the work of the Stirling engine; Q_{prc}: heat losses from the heat transfer between the permanent thermal fluid of the boiler and the heat transfer fluid of the coil tube; Q_{pse}: heat losses related to the transfer from the receiver to the Stirling engine tube and from the tube to the thermal fluid of the Stirling engine work;

Q_{conv} , Q_{cond} and Q_{ray} represents respectively the quantity of heat related to the transfer by radiation of the receiver, by convection of the air and by conduction within the receiver.

The quantity of heat Q_ss usable to the thermal fluid of the two sub-receivers is expressed by:

$$Q_{ss} = Q_{fms} + Q_{f2} \tag{1}$$

Q_{fms}: the thermal energy received by the working gas of the Stirling engine;

Q_{f2}: the thermal energy received by the heat transfer fluid which will be stored.

The energy balance of the global system will then be:

$$Q_{ss} = Q_{rec} - Q_{PG} \tag{2}$$

Q_{rec}: the thermal energy produced by the concentration system at the receiver;

Q_{PG}: amount of heat lost by the overall system expressed as:

$$Q_{PG} = Q_{P1} + Q_{P2} \tag{3}$$

Q_{P2} : thermal power lost during heat transfer to two receiving systems expressed in (4).

$$Q_{P2} = Q_{pse} + Q_{Prc} \tag{4}$$

In this work we have neglected the convection losses at the receiver and the heat exchange between the absorber of the Stirling engine and the boiler.

2.1. Receiver Energy Balance - Stirling Engine

Some of the thermal energy received by the main receiver is transferred by conduction to the Stirling engine receiver. This heat is transferred by convection from the Stirling receiver to the thermal working fluid of the engine. The thermal energy received by this heat transfer fluid of the Stirling engine (gas) $Q_{ms,in}$ during the isochore heating, makes this gas expand, which will create an isothermal expansion of the piston which will in turn make the engine work. The thermal power received by this gas is obtained by applying the first principle of thermodynamics [11]. The amount of heat received by the engine is equal to the sum of the amount of heat of the system during the isochore heating phase and the isothermal expansion phase. During the isothermal expansion, the amount of heat received by the system is equal to the work recovered during this same phase [12-16].

The amount of heat during the isothermal expansion phase is the amount of heat received by the engine thermal fluid expressed as (5):

$$Q_{fms} = Q_{det} = \int_{v_m}^{v_M} \frac{mRT_{réc}}{v} dv = mRT_{réc} ln\left(\frac{v_M}{v_m}\right) \quad (5)$$

T_{rec} is the temperature of the receiver.

During the phase of isochore heating of any gas the amount of heat is expressed by (6) [14, 17]:

$$Q_{ch} = mc_v\left(T_{réc} - T_{fse}\right) \quad (6)$$

c_v: molar capacity of the gas considered for constant volume heating.

The amount of heat received by the Stirling engine $Q_{ms,in}$ is expressed by the equation (7):

$$Q_{ms,in=}mc_v\left(T_{réc} - T_{fse}\right) \quad (7)$$

T_{se} is the initial temperature of the Stirling engine.

The amount of heat received by the gas will be:

$$Q_{fms} = Q_{ms,in} - Q_{cd_{r-rs}} - Q_{cv_{rs-f}} \quad (8)$$

$Q_{cd_{r-rs}}$: amount of heat lost during the transfer by conduction from the external wall of the Stirling engine receiver to the internal wall of this receiver;

$Q_{cv_{rs-f}}$: amount of heat lost during heat transfer from the inner wall of the Stirling engine receiver to the engine thermal fluid.

$$Q_{cd_{r-rs}} = \frac{\lambda_{rse-rsi} A_{rs}}{e_{rse-rsi}} (T_{réc} - T_{se}) \quad (9)$$

$$Q_{p_{f1-t}} = h_{fse-t} A_{rs} (T_{réc} - T_{se}) \quad (10)$$

$e_{rse-rsi}$: Stirling engine receiver thickness; $\lambda_{rse-rsi}$: Stirling engine conductivity; A_{rs}: the surface of the Stirling engine receiver; h_{fse-t}: the transfer coefficient by convection from the engine receiver to the engine thermal fluid (gas).

2.2. Energy Balance Received by Boiler

The boiler coupled to the Stirling engine, receives part of the thermal power concentrated at the focal point of the reflector. This boiler has the shape of a cube which contains a permanent thermal fluid with a large thermal storage capacity. This thermal fluid of the boiler once heated will transfer the heat to a serpentine tube. The serpentine tube, bathed in the permanent thermal fluid of the boiler, will in turn transfer the heat to the heat transfer fluid. The heat transfer fluid then conducts the stored thermal energy to the thermal storage tank. After discharge, the thermal fluid cools down and returns to the boiler to be reheated. This cycle continues throughout the day if there is sufficient sunlight.

2.2.1. Boiler Energy Balance - The Permanent Fluid

Figure 3 shows the cross-section of the schematic model of the boiler.

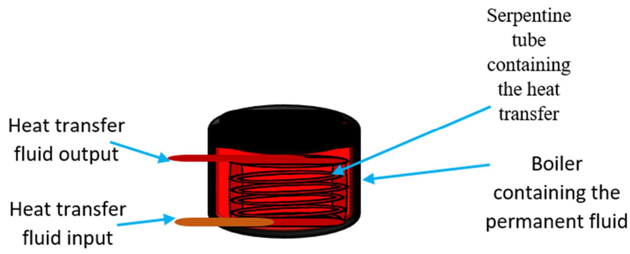

Figure 2. *Section of the boiler.*

A part of the thermal power concentrated on the surface of the receiver is transmitted by conduction to the inner wall of the boiler. Then it is transferred to the permanent fluid by convection from the inner wall.

The amount of heat Q_{f1} received by the permanent thermal fluid in the boiler is given by the equation (11):

$$Q_{f1} = Q_{rc} - Q_{p_{c-f1}} \quad (11)$$

Q_{rc}: amount of heat received by the boiler;

$Q_{p_{c-f1}}$: losses related to the transfer between the boiler and the permanent fluid.

We will consider that the transfer losses by convection in this case, which amounts to express $Q_{p_{c-f1}}$ by (12) [18]:

$$Q_{p_{c-f1}} = h_{cf1} A_c\left(T_{réc} - T_{f1}\right) \quad (12)$$

A_c: boiler surface.

The thermal energy Q_{rc} per unit volume received by the boiler is expressed as:

$$Q_{rc} = \rho_c c_{prc} (T_{réc} - T_{amb}) \quad (13)$$

ρ_{cf1} is the density and c_{prc} is the specific heat.

The power stored by the permanent thermal fluid in the boiler is defined by (14) [19]:

$$Q_{f1} = \rho_{cf1} c_{prf1} \frac{dT}{dt} \quad (14)$$

ρ_{cf1} is the density of the permanent fluid $cprf$ is the specific heat of the permanent fluid.

The expression of the heat balance of the permanent thermal fluid of the boiler becomes [3]:

$$\rho_{cf1} c_{prf1} \frac{dT}{dt} = \rho_{cf1} c_{prc} (T_{réc} - T_{amb}) - h_{cf1} A_c (T_{réc} - T_{f1}) \quad (15)$$

2.2.2. Energy Balance Permanent Fluid - Heat Transfer Fluid of the Serpentine Tube

The permanent fluid of the boiler will transfer the heat it has stored to the heat transfer fluid contained in the serpentine tube. The transfer will be done by a first convection between the permanent fluid and the external wall of the tube, by conduction through the wall of the tube then by a second convection between the internal wall of the tube and the heat transfer fluid. Once heated to a defined temperature, this heat transfer fluid is expelled to the thermal storage tank.

The amount of heat received by the heat transfer fluid contained in the serpentine tube Q_f2 is expressed by:

$$Q_{f2} = \dot{m}_{cf2} c_{prf2} \frac{dT}{dt} = Q_{f1} - Q_{cv_{f1-t}} - Q_{cd_t} - Q_{cv_{t-f2}} \quad (16)$$

The amount of heat from the boiler is expressed as:

$$Q_{rc} = Q_{f2} + Q_{p_{c-f1}} + Q_{cv_{f1-t}} + Q_{cd_t} + Q_{cv_{t-f2}} \quad (17)$$

With $\dot{m}_{cf2}$ is the mass flow rate of heat transfer fluid 2, c_{prf2} is the specific heat of fluid 2, $Q_{cd_{f1-t}}$ the energy lost in transfer by conduction within the serpentine tube $Q_{cv_{f1-t}}$ energy lost in transfer between the permanent fluid and the outer wall of the serpentine tube, and $Q_{cv_{t-f2}}$ is the power lost in transfer between the inner wall of the serpentine tube and the heat-transfer fluid with definite stays. The $Q_{cv_{f1-t}}$ and $Q_{cv_{t-f2}}$ are expressed by:

$$Q_{cd_t} = \frac{2\pi L}{\ln\left(\frac{R_2}{R_1}\right)} \left(T_{f1} - T_{f2}\right) \quad (18)$$

$$Q_{p_{f1-t}} = h_{f1-t} A_t \left(T_{f1} - T_t\right) \quad (19)$$

$$Q_{p_{t-f2}} = h_{t-f} A_t \left(T_t - T_{f2}\right) \quad (20)$$

λ_t is the conductivity of the tube, e_t is the thickness of the tube, A_t is the surface area of the serpentine tube, h_{f1-t} is the convective transfer coefficient of the permanent fluid tube, and h_{t-f} is the convective transfer coefficient of the heat transfer fluid tube with defined residence time [20, 21].

3. Materials and Method

The study focuses on the thermal balance and modeling of the evolution of the temperature of the receiver and that of the thermal fluids of a parabolic concentrator with double receiver at the focal point.

The simulation was developed in python language in the Jupiter notebook environment.

Direct radiation measurements were made on the roof of the Physics Department of the Faculty of Science and Technology [22].

The simulation is performed with a parabolic concentrator with an opening diameter of 4 meters (m). The corresponding surface is 12.56 m².

The characteristics of the receiver, the boiler and the thermal fluids used are shown in the table 1.

We used the same fluid for the boiler and the coil tube: Jarytherm® DBT oil. This oil was used because of its stability and thermal performance [23] table 1.

The type of Stirling engine adaptable to our work is the β-type free piston engine as a prototype of the LMEE of the BATIMAC project using helium as the working gas [17]. This motor has an electrical power output of 3kW, a hot head temperature of 400°C and a cold head temperature of 40°C.

Table 1. *Material characteristics and thermal fluid.*

P Parameters	Chaleur spécifique	Density	Dynamic viscosity	Thermal conductivity	Convective transfer coefficient	Mass flow	Initial temperature	Working temperature
Aluminum (main) receiver [24, 25]	à 600°C 1101,86 J/Kg.k, standards 900 J/Kg.K	2.7g/cm^3		226 W/m.K,			30°C	.600°C
boiler								
30°C								
400°C [26] 2 liter capacity								
Permanent thermal fluid [27, 28]	chaleur spécifique à 350°C à pour 2,64 J/kg.K	799 kg/m^3	0.923 kg/m.h	0.101 W/m.K,	Re=11759.20; Pr=0.4020; Nu=173.309; hcvf1=7525 W/m².K	0.001 m^3/s	30°C	350°C
Heat transfer fluid [27, 28]		799 kg/m^3	0.923 kg/m.h	0.101 W/m.K,	Re=11759.20; Pr=0.4020; Nu=173.309; hcvf1=7525 W/m².K	0.001 m^3/s	30°C	350°C
Stirling engine								
Fluid Stirling engine (gas) [29]	cv=3132 J/kg.K à 200bar	0,178.10^{-3} g/cm^3 à 20°C	0,0001863 poise	0.30121 W/m.K à 500°C	70 kg/m²/K		30°C	400°C [17] Voulue 600°C

3.1. Modeling the Evolution of the Temperatures of the Receiver and the Thermal Fluids of the System

The schematic model of the receiver a system were presented by the figures 1 (II). In this part we will establish the mathematical models of the evolution of the temperatures of the thermal fluids.

The amount of heat that the main receiver can absorb is calculated from:

$$Q_{rec} = m_{rec}.Cp_{rec}(T_{rec} - T_{amb}) \tag{21}$$

The amount of heat recovered by the boiler's permanent fluid is expressed as:

$$Q_{rc} = \dot{m}_{f1}Cp_{f1}(T_{réc} - T_{f1}) \tag{22}$$

The amount of heat recovered by the Stirling engine is expressed as:

$$Q_{ms,in} = \dot{m}c_v(T_{réc} - T_{fse}) \tag{23}$$

The main receiver receives the concentrated solar flux, part of it is used by this receiver and part is lost during the transfer. According to the thermodynamic principle, the heat balance as a function of the useful power is expressed [30] by (24):

$$Q_{u1} = m_{rec}.Cp_{rec}(T_{rec} - T_{amb}) \tag{24}$$

the amount of usable heat at the main receiver is expressed as:

$$Q_{u1} = I_{bn} \times A_{app}(\rho\gamma\alpha\tau) - Q_{P1} \tag{25}$$

With Q_{P1} all the powers related to transfer losses by conduction, by convection and by radiation.
the usable power becomes:

$$Q_{u1} = I_{bn} \times A_{app}\rho\gamma\alpha_{(S_r)t}\tau - h_{cv_{cd}}(T_{réc} - T_{amb}) - (\varepsilon_{tot})\sigma A_{rec}(T_{réc}^4 - T_{ciel}^4) \tag{26}$$

With $h_{cv_{cd}}$ expressed in (26):

$$h_{cv_{cd}} = \left(\left(h_{cv_{r\text{-}air}}A_{rec}\right) + \left(\frac{\lambda_{r-ms}\frac{A_{rms}}{A_{rec}}}{e_{r-ms}} + \frac{\lambda_{r-rc}\frac{A_{rrc}}{A_{rec}}}{e_{r-rc}}\right)\right) \tag{27}$$

The heat balance at the receiver leads to the differential equation expressed in (28):

$$\frac{dT_{rec}(t)}{dt} = \frac{I_{bn} \times A_{app}(\rho\gamma\alpha\tau)}{m_{rec}.Cp_{rec}} - \frac{Q_{P1}}{m_{rec}.Cp_{rec}} \tag{28}$$

From this expression we can establish the model of the evolution of the temperature of the receiver as a function of time.

$$T_{rec}(t) = T_{amb} + \frac{I_{bn} \times A_{app}(\rho\gamma\alpha\tau) - Q_{P1}}{m_R.Cp_R}t \tag{29}$$

The heat transfer between the inner side of the boiler and the permanent fluid is done by convection [31, 3, 28]. The heat balance at this level leads to the differential equation expressed by:

$$\frac{dT_{f1}(t)}{dt} + \frac{h_{cvf1}S_c}{\rho_{cf1}c_{prf1}}T_{f1}(t) = \frac{h_{cvf1}S_c}{\rho_{cf1}c_{prf1}}T_{réc} \tag{30}$$

3.2. Temperature Evolution Functions of the Internal Heat Transfer Fluids of the Two Receivers at the Focus of the Concentrator

From this differential equation we can establish the model of evolution of the temperature of the fluid as a function of time expressed by:

$$T_{f1}(t) = Ce^{\frac{-h_{cvf1}S_c}{\rho_{cf1}c_{prf1}}t} + T_{réc} \tag{31}$$

After the integration of this equation, by setting as a boundary condition that at the initial time the temperature of the fluid is equal to the ambient temperature, we have.

$$T_{f1}(t) = (T_{amb} - T_{réc})e^{\frac{-h_{cvf1}S_c}{\rho_{cf1}c_{prf1}}t} + T_{réc} \tag{32}$$

The heat transfer to the heat transfer fluid is done through two convections and one conduction [25]. Here we neglect the first convection within the permanent fluid and the conduction within the coil tube. The heat balance for the heat transfer fluid leads to the differential equation expressed by:

$$\frac{dT_{f2}(t)}{dt} + \frac{h_{t-f2}A_t}{\rho_{cf2}c_{prf2}}T_{f2}(t) = \frac{h_{t-f2}A_t}{\rho_{cf2}c_{prf2}}T_{f1} \tag{33}$$

The integration of this equation leads to:

$$T_{f2}(t) = De^{\frac{-h_{t-f2}A_t}{\rho_{cf2}c_{prf2}}t} + T_{f1} \tag{34}$$

With D the integration constant.

By setting as a boundary condition that at the initial time, the temperature of the heat transfer fluid is equal to the ambient temperature, the evolution model of the temperature of the heat transfer fluid as a function of the temperature of the permanent fluid is:

$$T_{f2}(t) = (T_{amb} - T_{f1})e^{\frac{-h_{t-f2}A_t}{\rho_{cf2}c_{prf2}}t} + T_{f1} \tag{35}$$

The thermal power that the helium, the working gas of the Stirling engine can receive is, by applying the principle of thermodynamics [32]:

$$mc_v\frac{dT_{fms}(t)}{dt} = h_{fms}A_{ms}(T_{réc} - T_{fms}) \tag{36}$$

This relation leads to the following differential equation:

$$\frac{dT_{fms}(t)}{dt} + \frac{h_{fms}A_{ms}}{mc_v}T_{fms}(t) = \frac{h_{fms}A_{ms}}{mc_v}T_{réc} \tag{37}$$

The solution of this differential equation taking as initial condition $T_{fms} = Tamb$

$$T_{fms}(t) = (T_{fms}(t) - T_{réc})e^{\frac{-h_{fms}A_{ms}}{mc_v}t} + T_{réc} \tag{38}$$

4. Results and Discussion

In this section, we present the results obtained. The curves representing the temperatures are the results of the transient state of each thermal fluid.

Figure 3 shows the variation in direct radiation measured at the study site with a solar tracking system at 02_11_21.

This figure shows that the direct radiation for this day varies from 200W at 8am to 800 around 9am and reaches a peak with a value of 1050W around 1pm and decreases to a value of 600W around 5pm.

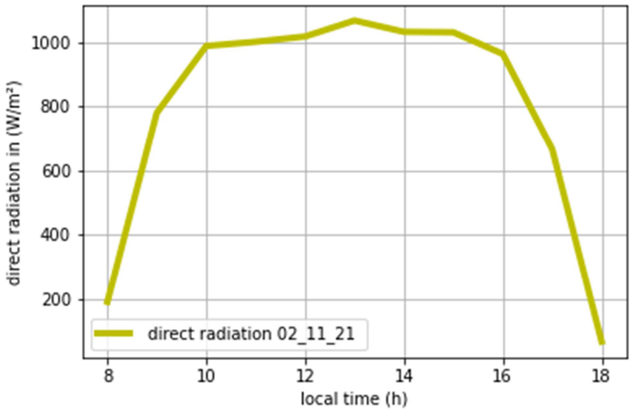

Figure 3. *Direct radiation measured for the day of 02_11_21.*

Figure 4 shows the evolution of the thermal power concentrated by the reflector surface as a function of the direct radiation received during the day.

The observation of this figure indicates that the power received exceeds 9000W between 10am and 3pm. We notice that the thermal power is much more important around 1pm with a peak of production of 10000W.

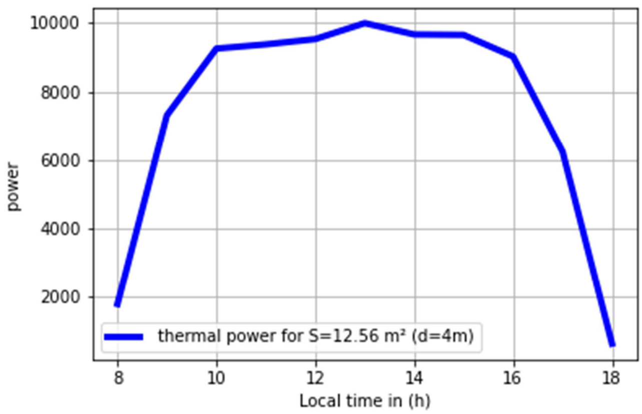

Figure 4. *Solar power and thermal power according to the direct radiation received for a diameter of 4m.*

Figure 5 shows the evolution of the temperature at the focal point of the reflector during 80 seconds (s) for the concentration of direct solar radiation in figure 1.

The analysis of this figure shows that in 80s, the temperature reached 640°C at around 8 am while the maximum temperature of 850°C is reached at 12pm. It decreases to 580°C around 4 pm.

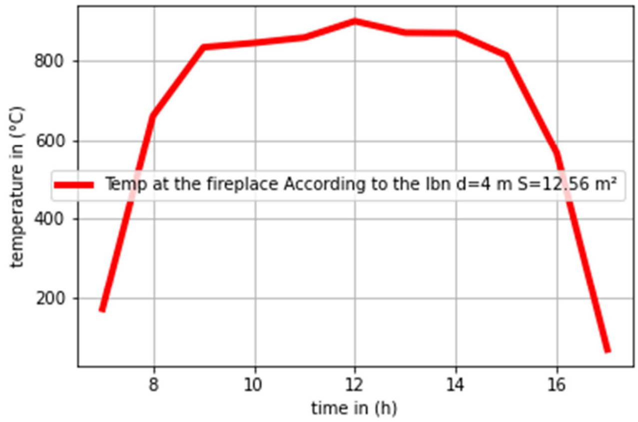

Figure 5. *The evolution of the temperature at the surface of the main receiver.*

Figure 6 shows the evolution of the temperature at the focus for a parabolic reflector of direct radiation fixed at 900W/m² in 80s.

The observation of this figure allows to affirm that the temperature produced varies according to the surface of this reflector. We notice that in 80s we can produce a temperature of 350°C with 6m², 500°C with 8 m², 650°C with 12.56 m² at the focal point of the concentrator.

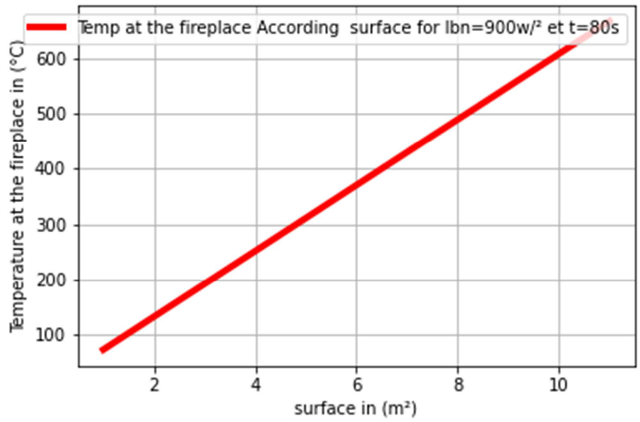

Figure 6. *Variation of the temperature according to the surface of the concentrator for a direct radiation of 900W/m² during a time of 80 seconds (s).*

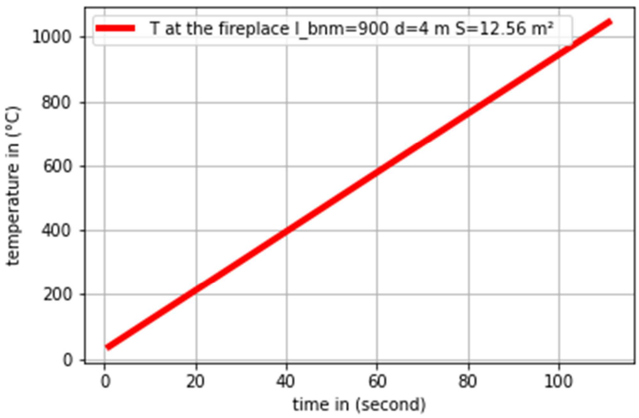

Figure 7. *Variation of the temperature at the focus of the concentrator of a surface of 12.56 m² a direct radiation fixed at 900W/m².*

Figure 7 shows the evolution of the temperature produced by a concentrator of a surface of 12.56 m² with a direct radiation fixed at 900 W/m² in a concentration time of 0s to 120 seconds (s).

The analysis of this figure confirms that with this surface the temperature at the focal point can reach 1050°C in 120s.

Figure 8 shows the temperature variation of the thermal fluid in the boiler when the temperature of the receiver is set to 600°C. The initial temperature of the fluid is 30°C.

The temperature of the permanent fluid goes from 30°C to 190°C in 500s, to 300°C in 1000s and reaches 500°C in 3000s. We notice that here the temperature reached by the fluid never reached the temperature of the receiver; this is due to the losses in the heat transfers.

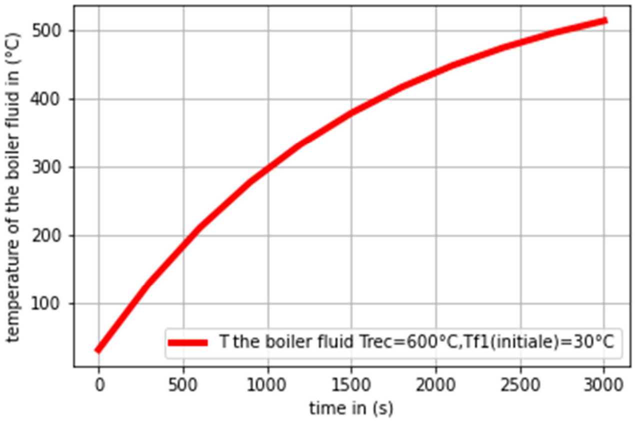

Figure 8. *Variation of the temperature of the thermal fluid of the boiler for a concentration time of 0 to 3000s Trec=600°C.*

Figure 9 shows the temperature variation of the heat transfer fluid contained in the serpentine tube which is itself immersed in the thermal fluid of the boiler.

The observation of this curve shows that this fluid reaches a temperature of 320°C in 1500s then 500°C in 3000s.

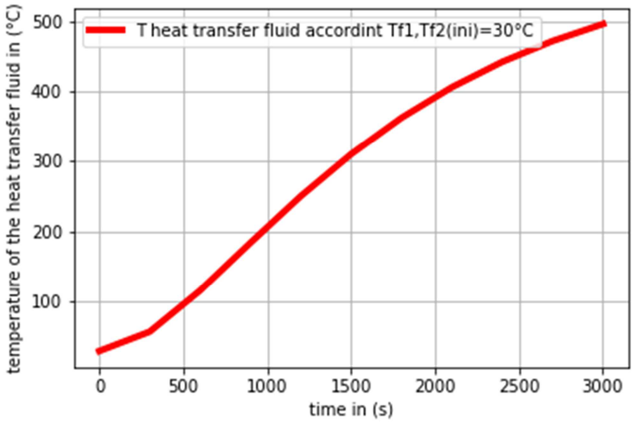

Figure 9. *Variation of the temperature of the heat transfer fluid in the coil tube as a function of the heating time.*

Figure 10 shows the variation of the temperature of the permanent fluid in the boiler and of the heat transfer fluid in the coil tube.

We note that there is a difference between the temperature of the permanent fluid in the boiler and the temperature of the heat transfer fluid in the coil tube. The temperature of the permanent fluid is always higher than that of the fluid contained in the serpentine tube.

This is due to heat transfer losses by convection and conduction from the boiler's permanent fluid to the coil tube and the coil tube's heat transfer fluid.

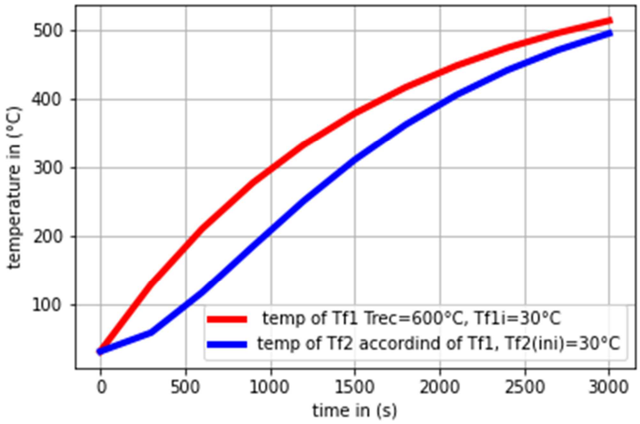

Figure 10. *Variation of the temperature of the thermal fluid of the boiler and the coil tube.*

Figure 11 is the curve of the variation of the temperature of the thermal fluid (helium gas) of work of the Stirling engine for a temperature at the hearth of 600°C and an initial temperature of 30°C.

The analysis of this figure shows that the temperature of this gas reaches 320°C at about 1500s and continues to grow until reaching 450°C at about 3000s.

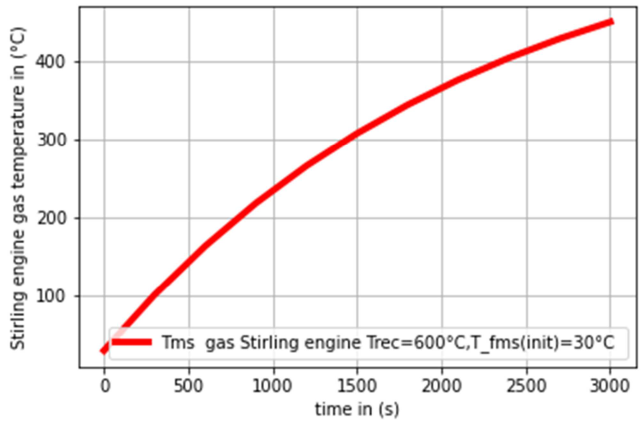

Figure 11. *Variation of the temperature of the working gas (helium) of the Stirling engine.*

Figure 12 shows the temperature evolution of the three thermal fluids of the receiving devices at the focus of the reflector as a function of the heating times.

The observation of this figure makes it possible to see that the temperature of the permanent fluid of the boiler grows reaches 300°C in 1000s then reaches 514°C in 3000s. The thermal fluid of the Stirling engine grows to 240°C in 1000s and reaches 450°C in 3000s. As for the heat transfer fluid of the serpentine tube, we notice a delay compared to the other two at the beginning. It reaches 200°C in 1000s, increases until about 1500s, reaches 300°C and exceeds the temperature of the thermal fluid of the Stirling engine to reach 500°C in 3000s.

This difference in temperature is due to heat transfer losses and to the categories of solid and fluid materials that receive the heat.

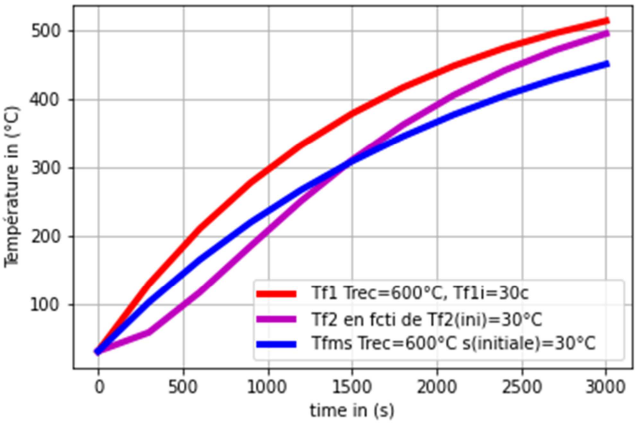

Figure 12. *Curve of the variations of the 3 thermal fluids according to the times of the heating.*

This study shows that it is possible to heat two receivers at the focal point of a parabolic concentrator. The previous results prove that with a parabolic concentrator of 4m diameter of opening, we can bring to more than 400°C the thermal fluids of the two receivers in 2250s. That is the thermal energy sufficient to operate these two receivers at normal thermal power.

5. Conclusion

In this work, we first studied and modeled the heat balances of the different substances (fluid and solid) of the two receivers at the focus of a parabolic concentrator. Then we performed the simulation of these theoretical models:

We noticed that with a parabolic concentrator of 12.56 m² for a direct radiation measured during a day the thermal power produced at the focal point had a peak with 10000W around 13:00. We have seen again in 80s with a radiation received at 13:00 we can produce at the focus of this reflector a temperature of 900°C.

We have highlighted the influence of the reflector surface on the production of thermal energy when direct radiation is fixed at 900W/m². We have also seen that for a concentration time of 80s the temperature varies very strongly.

It appears from this study that with a surface of 12.56 m² during a concentration time of 0 to 120s the temperature increase from 30°C to 1050°C.

For thermal fluids, we have fixed the temperature of the receiver at 600°C. In order to observe, the evolution of the temperature of the different fluids as a function of time while fixing the initial temperature of 30°C.

Thus the permanent fluid in the boiler has increased from 30°C to 350°C in 1500s to 514°C in 3000s. The heat transfer fluid of the serpentine tube which receives the temperature of the permanent fluid increased from 30°C to 320°C in 1500s and reaches 500°C in 3000s. The working gas of the Stirling engine has also increased from 30°C to 320°C in 1500s and reaches 450°C in 3000s. This engine reaches the temperature necessary to produce electricity in less than 1600s.

This study highlights the thermal production of a parabolic concentrator with two receivers at the boiler and Stirling engine. It proves the possibility of coupling a solar installation based on a parabolic concentrator directly producing electricity and heat production for thermal storage.

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