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Numerical Modelling of a Parabolic Trough Concentrator for the Design of a Thermodynamic Solar Power Plant

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Abstract: The Sahel region of Burkina, whose capital is Dori, is one of the regions of the country whose electricity coverage rate is very low, which happens to be a paradox because it has enormous solar deposits which are sources of energies. This is the essence of our study of a 50MW cylindrical-parabolic concentrator solar power plant in Dori. It should be noted that for the production of electricity through solar, several technologies exist, the choice of one over the other is made on technical and economic criteria. Essentially there are photovoltaic solar power plants, parabolic trough solar power plants, combined cycle solar power plants and tower power plants. The cylindrical-parabolic concentrator (CCP) solar power plant is chosen because it is more promising and its technology produces not only electricity but also sanitary hot water which is essential for this part of the country where it is very cold at certain period of the year. In this work, attention is paid to the numerical simulation of solar to thermal energy conversion using a parabolic trough concentrator for the design of a solar power plant in Dori. Therminol vp-1 is used as heat transfer fluid. The mathematical equations governing the operating principle of our PTC are described and solved using the finite difference method. The numerical results obtained indicate a thermal efficiency of 63.38% for our concentrator and an overall efficiency of 22.10% for the solar power plant. These results show real possibilities for electricity production in this region of the country.

Keywords: Parabolic Solar Concentrator, Therminol vp-1, Performance

1. Introduction

Access to energy is an essential component of economic, social and political development. It allows for the improvement of educational and health conditions that promote individual development and contributes to the development of economic activity. The production, distribution and management of energy resources are also key to the preservation of the environment and form an integral part of the solution to pursuing sustainable development objectives. 70% of the continent's electricity is generated by thermal power plants (40% coal and 30% gas) and 25% by hydro power plants [1]. With 14% of national electricity coverage in 2018, Burkina Faso is one of the least electrified countries in West Africa. [2]. Utilisation of the energy produced requires an adequate transmission network (high voltage) and efficient distribution services which still account for more than 80% of the cost of electricity. The low population density combined with a predominance of rural population are major constraints as they make the development of electricity infrastructures very expensive and limit economies of scale. It must also be noted that the population is increasingly resorting to generator sets for the production of electricity, even though this has serious negative consequences. In fact, the supply of fossil fuel for the generator has a strong impact on the environment and its price represents a particularly heavy burden for households. Considering these facts, it seems necessary, on the one hand, to decentralise production as close as possible to the users in order to limit the cost of the kWh produced and, on the other hand, to implement means of production that limit the use of fossil fuels. The aim is to promote innovative local solutions in order to provide the population (in rural areas) with sustainable, affordable, reliable and efficient energy. However, Burkina Faso's solar energy potential is significant, accounting for 60% of the country's total renewable energy potential [3]. The Research Institute in Applied Science and Technology and the National Metrological Direction evaluated the average sunshine at 5.5 kWh/m²/day for 3000 to 3500 hours per year [4, 5]. This potential solar resource can be used to develop both photovoltaic and thermal solar energy. Investments in this technology in Burkina Faso include the installation of a number of solar photovoltaic power plants such as the solar power plant in Zagoutouli (33 MWp), Ziga (1.1 MWp) and Nagréongo (30 MWp). There are no installations in CSP technology, specifically parabolic trough plants, which are suitable and profitable for the country. The parabolic trough power plants require high temperatures of 380°C to 1200°C to produce electricity and this is due to the mirrors having an optical efficiency of between 60% and 83% depending on the sizing [6-8]. The solar power plant is able to produce superheated steam thanks to parabolic trough concentrators in which the temperature can reach up to 1500°C [9-11]. Parabolic trough solar power plants are one of the most promising technologies to replace non-renewable energies such as gas, coal and nuclear power plants, for the production of electricity for various activities [12, 13].

This is the context in which this work on modelling parabolic trough concentrators is being carried out. The objective is to validate the developed model and then to determine the performance of the parabolic trough concentrator in the climatic conditions of Burkina Faso, to determine the evolution of the output temperature of the heat transfer fluids passing along the absorber and the thermal efficiency according to the input parameters for any time and day of the year in order to design a 50 MW solar power plant in Dori in the Sahel region.

2. Description of the Power Plant

The figure 1 shows the configuration of the parabolic trough solar power plant. This plant consists of two main elements: the solar field, which uses a series of long parabolic trough concentrators, and the heat transfer system (power block), which is an electrical generation system (Turbine + Generator). The concentrators used for the plant are of the LS-3 Type. This type of concentrator has a total length of L=99 m and a width of l= 5.76 m. The internal and external diameters of the glass are Di=0.065 m and De=0.07 m respectively, while those of the absorber are Di=0.045 m and De=0.04 m. The optical properties are uniform over the entire reflective surface. A solar tracking mechanism is implemented in accordance with the sinh of the sun. The values of these absorption and reflection coefficients are $\rho_A = 0.94$ and $\rho_m = 0.95$ respectively. The glass transmitivity is 0.77 and the absorber emissivity is estimated to be α=0.19. [14]

Our plant is a 50 MW high concentration solar thermodynamic power plant based on the Hirn cycle with reheat. Thus, the power block is determined by sizing a turbine type SST-300 (Siemens Steam Turbine) with a nominal power of 50 MW and a maximum pressure of 120 bar. The turbine inlet temperature is estimated at 520°C with a maximum expansion of 0.6 bar [15]. For the operation of the power plant, a combination of high and low pressure turbine is required. The P_e is estimated at 100 bar with a Te=380°C at the HPT inlet. The P_e is estimated to be 20 bar with a Te=380°C at the LPT inlet with a 0.6 bar expansion. The respective efficiencies of HPT and LPT are 85 and 88%. The pump and the electric generator have an efficiency of 70 and 97% respectively. The heat loss from the steam generator is estimated at 2%.

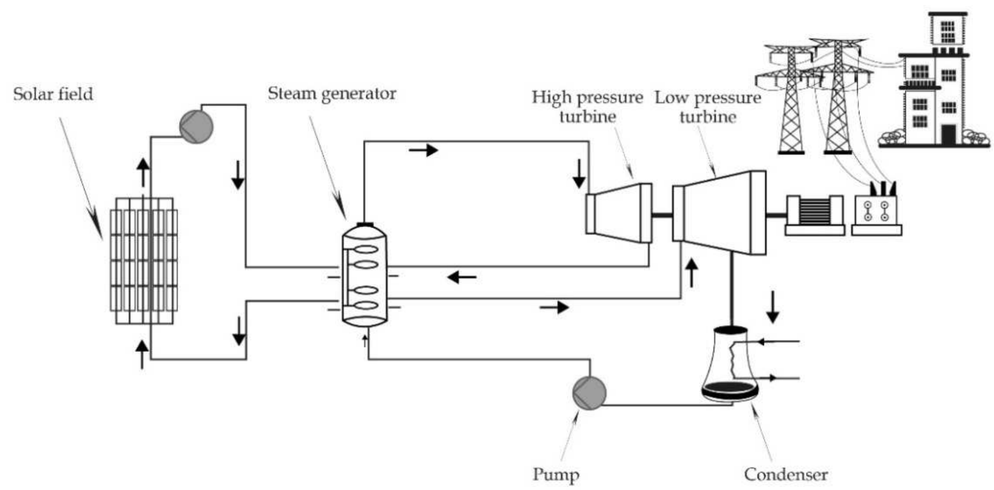

Figure 1. *Model of the solar thermodynamic plant studied.*

The power plant consists of long parabolic trough mirrors, oriented on the north-south axis and equipped with a sun tracking system. The sun's rays are concentrated on a horizontal tube, in which a heat-transfer fluid flows and transports the heat to the heat exchangers. The temperature of the fluid can reach up to 380°C. This energy is transferred to a water circuit where the elevated temperature of the steam drives the turbines to produce electricity.

3. Power Plant Dimensioning

It is necessary to size such a power plant in order to determine the different components and their thermodynamic properties for on-site implementation. Therefore, the area of the solar field and the number of LS-3 concentrators at the Dori site are estimated. The sizing shows 2016 concentrators covering an area of 110ha. The solar field is composed of 1 008 loops, spread over two areas, the North and South fields. Each surface will contain 504 loops of four (4) modules, distributed in two (2) rows. They will be aligned on the north-south line, and on a single axis of solar tracking, from east to west. The thermodynamic components such as pressure, temperature, entropy and enthalpy of the different elements of the power block were calculated using the steam entropy diagram (T-S). The power of these components and the amount of heat generated by the steam generator were determined using mathematical expressions found in the literature. These results are presented in Table 1.

Table 1. *The different characteristics of the power bloc and the solar field.*

Elements / Parameters	Turbine		Condenser	Pump	Steam generator		Solar field
	HP	LP			Oil	vapor	
Inlet pressure	100	20	0.6	0.6	-	-	-
Outlet pressure	20	0.6	0.6	100	-	-	-
Inlet temperature (°C)	380	380	90	90	30	-	-
Outlet temperature (°C)	224	90	90	90.30	-	-	390
Inlet enthalpy (kJ/ kg)	3050	3200	2535.6	380	-	-	-
Outlet enthalpy (kJ/ kg)	2837.5	2535.6	380	394.2	-	-	-
Inlet entropy	6.12	7.02	7.2	1.1	-	-	-
Outlet entropy	6.4	7.2	1.1	1.14	-	-	-
Amount of heat (kj/kg)			2155.6	-	3078.66	3018,3	3078.66
Total power (MW)	55.5		123.77	0.832	176. 77		-
Mass flow rate (kg/s)	57.42		57.42	57.42	-		1
Work (kj/kg)	967.5		-	14.2	-		-
Active area (ha)							110
Thermal efficiency of the facility (%)	31.38						
Overall efficiency (%)	22.10						

4. Mathematical Formulation

A section of the absorber is shown in figure 2. It consists of glass and a fluid flowing inside. The principle is to reflect the sun's rays onto a mirror, which are reflected back to the absorber. The absorber heats up and transfers the maximum amount of heat to the heat transfer fluid circulating along the absorber. To reduce heat loss, the absorber is placed in a glass tube. In order to write the mathematical equations governing the operating principle of the solar thermodynamic plant, the following simplifying assumptions were made: (i) the heat transfer fluid is incompressible; (ii) the parabola shape is symmetrical; (iii) the glass is considered opaque to infrared radiation; (iii) the fluid flow is one-dimensional.

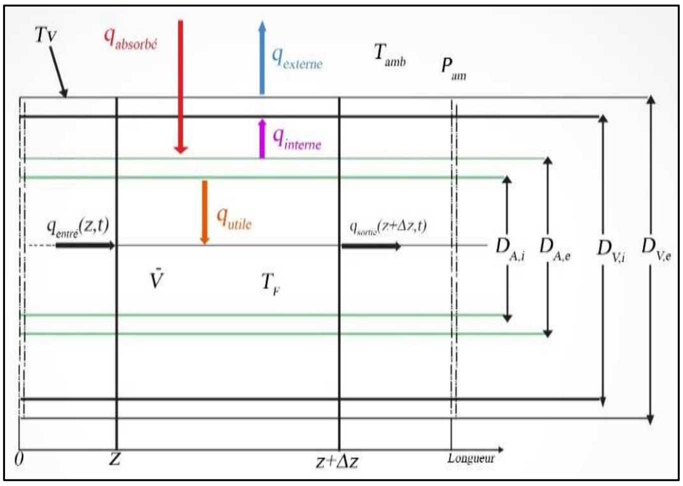

Figure 2. *Absorber section.*

One section of the absorber is modelled by writing the heat balance equations for the heat transfer fluid, the absorber and the glass. This will lead to partial differential equations for the temperature of the heat transfer fluid, the absorber and the glass. The distance along the collector is indicated by z. The equations for the heat change of the fluid over time t on an element of length Δz at position z and those for the absorber and glass are:

$$\rho_F . C_F . A_{A,interne} \frac{\partial T_F(z,t)}{\partial t} = -\rho_F . C_F . \dot{V} . \frac{\partial T_F(z,t)}{\partial z} + q_{utile}(z,t) \tag{1}$$

$$\rho_A . C_A . A_A . \frac{\partial T_A(z,t)}{\partial t} = q_{absorbé}(t) - q_{interne}(z,t) - q_{utile}(z,t) \tag{2}$$

$$\rho_v . C_v . A_v . \frac{\partial T_v(z,t)}{\partial t} = q_{interne}(z,t) - q_{externe}(z,t) \tag{3}$$

Thus the following initial and boundary conditions for the fluid, the absorber and the glass are given:

$$T_F(0,t) = T_{F,entré}(t) = T_{ambiant}(t),\ T_F(Z,0) = T_{F,initial}(z) = T_{ambiant}(0)$$

$$T_A(z,0) = T_{A,initial}(z) = T_{ambiant}(0),\ T_v(z,0) = T_{v,initial}(z) = T_{ambiant}(0)$$

5. Equations Discretisation

The discretized form of the equations (1-3):

$$\frac{dT_{f.i}(t)}{dt} = -\frac{\dot{V}}{A_{A,int}.\Delta z} . T_{f.i}(t) + \frac{\rho_f(T_{f.i-1}).C_{f.i}(T_{f,i-1})\dot{V}}{\rho_f(T_{f.i}).C_{f.i}(T_{f,i}).A_{A,int}.\Delta z} . T_{f,i-1}(t) + \frac{q_{utile}}{\rho_f(T_{f.i}).C_{f.i}(T_{f,i}).A_{A,int}.\Delta z} . \left(T_{A,i}(t), T_{f,i}(t)\right) \tag{4}$$

$$\frac{d(T_A(z,t)}{dt} = \frac{1}{\rho_A.C_A.A_A.} \left[q_{absorbé}(t) - q_{int}\left(T_{A,i}(t), T_{V,i}(t)\right) - q_{utile}\left(T_{A,i}(t), T_{f,i}(t)\right)\right] \tag{5}$$

$$\frac{dT_v(z,t)}{dt} = \frac{1}{\rho_v.C_v.A_v} \left[q_{int}(T_{A,i}(t), T_{v,i}(t)) - q_{ext}\left(T_A(t), T_{v,i}(t)\right)\right] \tag{6}$$

The discretization of equations 4-6 allowed us to have a three row three column matrix which will be solved in the Matlab 2020 software with the initial conditions. This matrix is:

$$\begin{cases} \dfrac{dT_{f.i}(t)}{dt} = -\dfrac{\dot{V}}{A_{A,int}.\Delta z} . T_{f.i}(t) + \dfrac{\rho_f(T_{f.i-1}).C_{f.i}(T_{f,i-1})\dot{V}}{\rho_f(T_{f.i}).C_{f.i}(T_{f,i}).A_{A,int}.\Delta z} . T_{f,i-1}(t) \\ \qquad + \dfrac{q_{utile}}{\rho_f(T_{f.i}).C_{f.i}(T_{f,i}).A_{A,int}.\Delta z} . \left(T_{A,i}(t), T_{f,i}(t)\right) \\ \dfrac{d(T_{A,i}(t)}{dt} = \dfrac{1}{\rho_A.C_A.A_A.} \left[\begin{array}{c} q_{absorbé}(t) - \dfrac{2\pi k_{eff.air}}{\ln\left(\frac{D_{V,i}}{D_{V,e}}\right)}\left(T_{A,i}(t) - T_{V,i}(t)\right) - \left(\dfrac{\sigma.A_{A,ext}.\left(T_{A,i}^4(t) - T_{V,i}^4(t)\right)}{\frac{1}{\varepsilon_A} + \frac{1-\varepsilon_v}{\varepsilon_v}\left(\frac{D_{A,e}}{D_{v,i}}\right)}\right) \\ -q_{utile}\left(T_{A,i}(t), T_{f,i}(t)\right) \end{array}\right] \\ \dfrac{\partial T_{v,i}(t)}{dt} = \dfrac{1}{\rho_v.C_v.A_v}\left[\dfrac{2\pi k_{eff.air}}{\ln\left(\frac{D_{V,i}}{D_{V,e}}\right)}\left(T_{A,i}(t) - T_{V,i}(t)\right) + \left(\dfrac{\sigma.A_{A,ext}.\left(T_{A,i}^4(t) - T_{V,i}^4(t)\right)}{\frac{1}{\varepsilon_A} + \frac{1-\varepsilon_v}{\varepsilon_v}\left(\frac{D_{A,e}}{D_{v,i}}\right)}\right)\right] \\ -\dfrac{1}{\rho_v.C_v.A_v}\left[h_V . A_{V.ext}\left(T_{V,i}(t) - T_{amb}(t)\right) + \varepsilon_V . \sigma . A_{v.ext}\left(T_{V,i}^4(t) - T_{amb}^4(t)\right)\right] \end{cases}$$

6. Numerical Method of Resolution

Equations (1-3) are discretised using the Finite Difference Method (FDM). The discretisation method consists in transforming these equations into an algebraic system. These algebraic equations describe the energy balances of the fluid, the absorber and the glass. The resulting algebraic equations are expressed as matrices and solved online using the Gauss-Seidel method. A numerical code was developed to simulate the outlet temperature of the heat transfer fluid flowing through the absorber, the absorber temperature and the glass temperature. The numerical code was developed using Matlab2020 software.

7. Results and Discussions

The results of the numerical code developed to simulate Tf, Tab and Tve are presented. The results are presented in the form of figures. In order to validate our results, The numerical results were compared with a similar parabolic trough concentrator studied by Mokhtar GHODBANE [6]. The curves in figure 3 show a good agreement with those obtained by the author Mokhtar GHODBANE. The estimated difference is 4.14%.

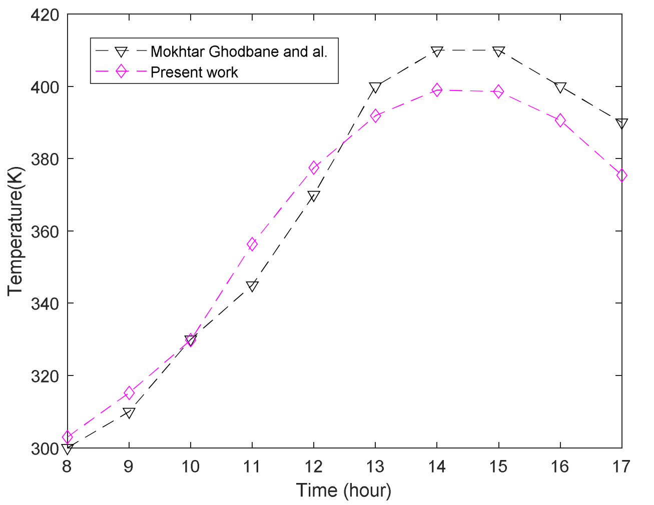

Figure 3. *Temporal evolution of the heat transfer fluid calculated by our code and that obtained by Mokhtar GHODBANE.*

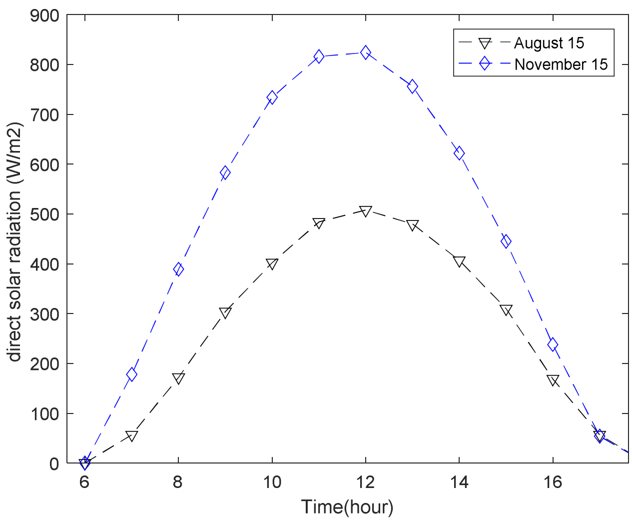

Figure 4. *Evolution of direct solar radiation.*

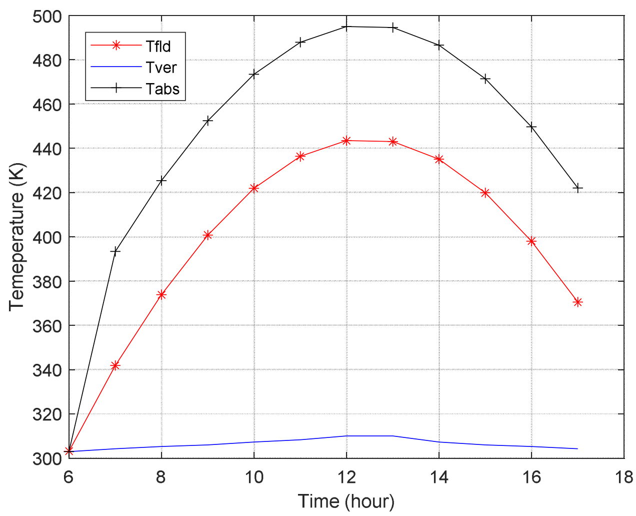

Figure 5. *Evolution of the oulet temperature the therminol Vp-1 (T_{fld}) the absorber (T_{abs}) and the glass (T_{ver}) for the day of August 15.*

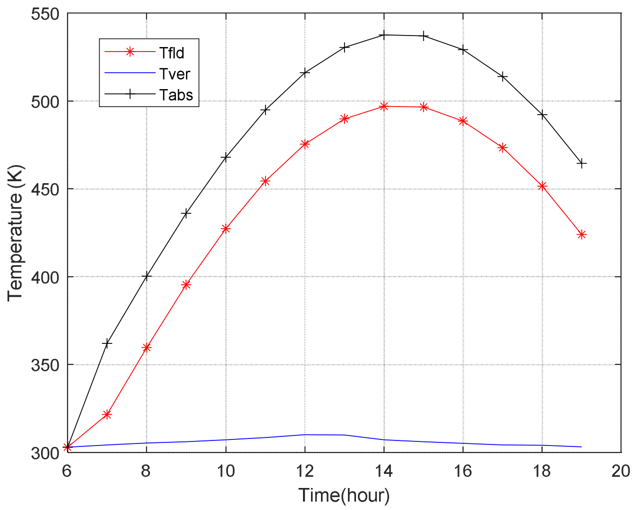

Figure 6. *Evolution of the outlet temperature of the therminol Vp-1 (T_{fld}) the absorber (T_{abs}) and the glass (T_{ver}) for the day of November 15.*

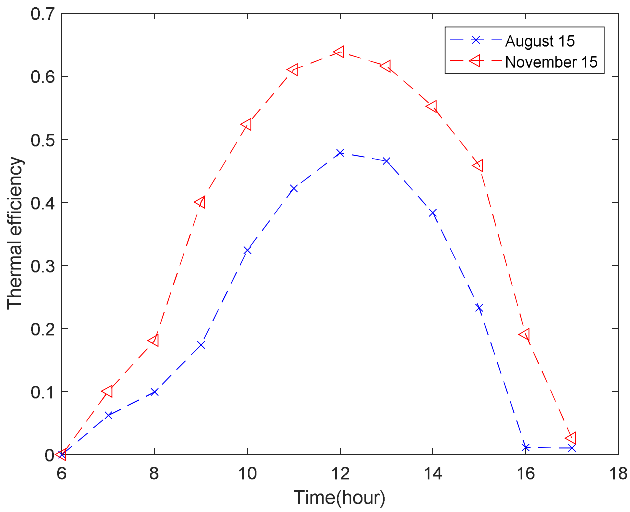

Figure 7. *Temporal evolution of thermal efficiency on two selected days of the year.*

To days the parabolic trough concentrators present a real opportunity for electricity production. That is why this thermodynamic solar power plant is being implemented in the city of Dori in Burkina Faso. The city of Dori was chosen to conduct the study of our concentrator. Dori is located at an altitude of 282 meters, its northern latitude is 4°02′00″, and its longitude is 0° 02′ 00″ East. Therminol Vp-1 [16] was chosen as the heat transfer fluid with a flow rate of 1 kg/s. The absorber is the place of the conversion of concentrated solar radiation into sensible high temperature heat.

Bellos and Tzivanidis investigated combined thermal and optical enhancement techniques based on internal fins and a radiation shield. Using internal fins inside the absorber amplifies turbulence inside the flow by 0.9%, while using a reflective screen increases thermal efficiency by 1. 66%. The final results show that this combination is able to increase the energy efficiency of the collector by up to 2.41% [17].

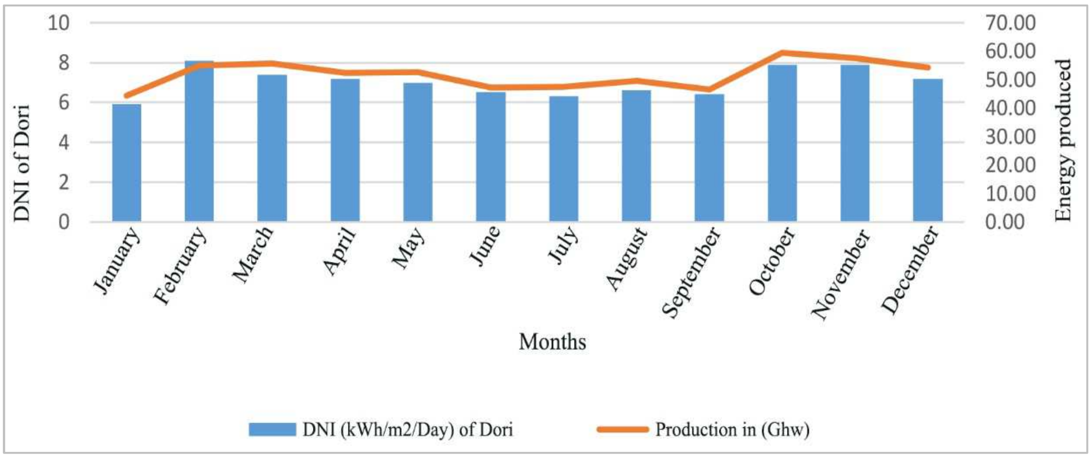

Figure 8. *Electricity production of the power plant during the year.*

For different flow rates from (0.27 to 0.6litre/min) theoretically. The results of Mustafa Al-Dulaimi et al showed that the best optical design is the channel receiver with an optical efficiency of 84% [18].

The figure 4 shows the evolution of the normal direct solar radiation on August 15 and November 15 in Dori. The direct radiation which is 0 W/m^2 at 6 am increases to reach its maximum value of 824 W/m^2 at 12 am for the day of November 15 and 508 W/m^2 for the day of August 15. There is a continuous decrease of solar radiation from 400 W/m^2 to 300W/m^2 from 3 pm onwards for the days of November 15 and August 15 respectively. A drop in irradiation can be observed until 6 pm. The direct radiation is very low on the day of August 15 because of a low lightness index. This is due to a large amount of rainfall during the month of August. On the other hand, the day of November 15 is characterised by a high lightness index.

The figures 5 and 6 show the evolution of the temperature of the heat transfer fluid, the absorber and the glass on the two selected days. It can be seen that the three curves in each figure have the same shape and that the absorber temperature is always higher than the heat transfer fluid temperature owing to its absorption coefficient, which is higher than those of the other components. The temperature of the glass is the lowest, it is always close to the ambient temperature which is 303 K.

The maximum temperature reached during the day of August 15 is 442 K. The drop in direct radiation during this day is felt around 5 pm when the temperature of the fluid drops to 370 K, a difference of 67 K from the beginning of the day. The temperature of the heat transfer fluid, which is initially 303 K at 6am, gradually heats up with the appearance of the first rays of sunlight at 7am and then reaches a temperature of 475 K at 12 am. The maximum temperature reached during the day of November 15 is 498 K at 2 pm. This temperature drops to 425 K at 5 pm and tends towards the ambient temperature when the last direct solar radiation disappears around 18:30. This difference in maximum temperature of the heat transfer fluid is due to the strong variation of direct irradiation between the two selected days of the year which is 316 W/m^2. It should also be noted that August is the least sunny month, when the sky is practically cloud-covered, whereas November is the month when the sky is clear and bright.

In figure 7 the evolution of the thermal efficiency during the two days can be observed. At the beginning of the day, i.e. at 6 o'clock, the thermal performances of the two days are null because of the absence of solar rays. When the solar rays appear, there is a difference between the two performances. The difference becomes visible from 9 am to 4 pm. After 4pm, the thermal performances of these two days become zero because of the absence of solar radiation. The efficiency for these two selected days is respectively 47.83% and 63.83%. The thermal efficiency varies according to the direct irradiation; the higher the irradiation, the higher the thermal efficiency of the collector despite the fact that losses are also high.

The variation of the plant's electricity production over the year is shown in Figure 8. This production is extremely variable during the year. The peak of the production is obtained in October with almost 59.43 GWh. On the other hand, the production is really low in January with 44.38 GWh due to the very important aerosol effects affecting the productivity of the solar thermodynamic plant. Regardless of the sky conditions, the lightness index and the presence of aerosols, the plant has a productivity of 44 GWh every month of the year at the Dori site. Other months such as July, August and September are characterised by a cloudy and degraded sky conducive to thunderstorms and rain, which sometimes prevent the sun from appearing and direct solar radiation from reaching the collectors for energy production, as such a power plant needs direct radiation to heat the heat transfer fluid that flows through the absorber to exchange the heat gained in the steam generator. The annual production of the power plant is estimated to be around 619.99 GWh.

8. Conclusion

At the end of solar power plant thermodynamic modelisation in Dori city, it can stated that our initial objective has been achieved. First, the performance of the parabolic trough concentrator type LS-3 using Therminol vp-1 as heat transfer fluid was determined. This allowed us to simulate the temperature variation of the absorber of the parabolic trough concentrator. Then, the influence of the geographical, physical and geometrical parameters of the concentrator allowed us to determine the respective efficiencies of the concentrator of 63.38% and 47.83%, for the selected days of November 15 and August 15. Maximum temperatures of the heat transfer fluid of 498 K and 442 K were observed for a direct solar radiation of 824 W/m^2 and $508 W/m^2$.

Finally, the thermodynamic solar power plant dimensioning provide 2016 number parabolic trough concentrators spread over an area of 110 hectares. The power plant has an overall efficiency of 22.10% with an annual productivity of 619.99 GWh. Considering the productivity of the plant, it could be a solution to the energy problems of the city of Dori and even of the region.

Nomenclature

ρ_m: Reflection coefficient of the mirror
ρ_v: The glass density
ρ_A: The absorber density
ε_A: The absorber tube emissivity
$D_{A,ext}$: External diameter of the absorber (m)
$D_{A,int}$: Internal diameter of the absorber (m)
A: Absorption coefficient of the absorber
$D_{v,ext}$: External diameter of the glass (m)
$D_{v,int}$: Internal diameter of the glass (m)
Pe: Input pressure (bar)
HPT: High Pressure Turbine
LPT: Low Pressure Turbine
A_{Ab}: absorber surface (m^2)
T_a: Ambient temperature (K)
T_{fld}: Temperature of the heat transfer fluid (K)
T_{abs}: Temperature of absorber (K)
Te: Input temperature
T_{ver}: Temperature of the glass (K)
Cp: Thermal capacity (J/kg k)
C_{ver}: Specific heat of the glass (J/kg.K)
C_{fld}: Heat of the heat transfer fluid (J/kg.K)
q: Amount of heat absorbed by the absorber (w/m)
q_{utile}: Amount of heat exchanged by convection (w/m)
$q_{interne}$: Amount of internal heat (w/m)

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