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Numerical Simulation of Thermal Performances of a solar PTC receiver using TRNSYS software

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Abstract

Solar energy is a key renewable energy source and the most abundant energy source on the globe. Solar energy can be converted into electric energy by using two different processes: by means of photovoltaic (PV) conversion and the thermodynamic cycles. Concentrated solar power (CSP) is viewed as one of the most promising alternatives in the field of solar energy utilization. The objective of this paper is to investigate numerically the thermal performance of a solar Parabolic Trough Collector (PTC). The effect of some design and operation parameters, which included the mass flow rate and the natureof HTFon thermal performance of the solar PTC were analyzed.

Keywords: *Solar PTC receiver;thermal performance; solar power; TRNSYS.*

I. INTRODUCTION

Solar power plants are a relatively new technology with considerable potential for development. They provide an opportunity for sunny countries to produce electricity. Several technologies exist in this field. Fig. 1 shows the particular case of a solar PTC receiver. There is the solar field used to collect the energy to heat, than the heat transfer fluid which then enters the factory block to generate the electrical energy.

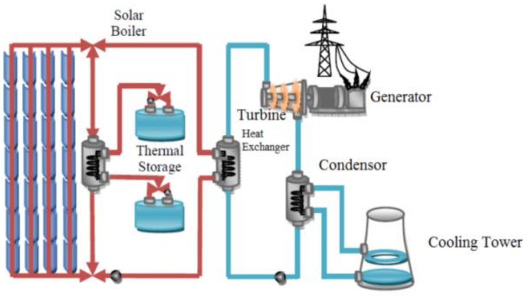

Fig .1: Schematic diagram of parabolic concentrated solar thermal power plant

Morocco, being one of largest energy importer in Middle East and North Africa, is making concerted efforts to reduce its reliance on imported fossil fuels. Renewable energy is an attractive proposition as Morocco has almost complete dependence on imported energy carriers. In 2012, Morocco spent around US$10 billion on all energy imports (crude oil and oil products, coal, natural gas and electricity). Annual electricity consumption in Morocco was 33.5 TWh in 2014, and is steadily increasing at a rate of around 7 percent each year.

Morocco has launched one of the world's largest and most ambitious solar energy plan with investment of USD 9billion. The Moroccan Solar Plan is regarded as a milestone on the country's path towards a secure and sustainable energy supply which is clean, green and affordable. The aim of the plan is to generate 2,000 MW (or 2 GW) of solar power by the year 2020 by building mega-scale solar

January - February 2020
ISSN: 0193 - 4120 Page No. 136 - 142

power projects at five location — Laayoune (Sahara), Boujdour (Western Sahara), Tarfaya (south of Agadir), AinBeniMathar (center) and Ouarzazate — with modern solar thermal, photovoltaic and concentrated solar power mechanisms. Morocco, the only African country to have a power cable link to Europe, is also a key player in Mediterranean Solar Plan and Desertec Industrial Initiative [1]

II. RELATED WORK

In this section, we will discuss some approaches for modeling of a solar station with the parabolic trough solar receiver. Hajjaj.M et al[2] have presented a new mathematical model of solar PTC.Stuetzle T.A et al [3] have established a collector field model as coupled a partial differential equations. B. Zeroual et al [4] have used a synthetic-oil heat transfer fluid (Therminol VP-1) in the collector loops to transfer thermal energy to Rankine cycle via heat exchangers train. Bekkouche M.A et al [5] have presented a study for modelling of the thermal behaviour of some solar devices. Y. Marif et al [6] have developed a computer program based on one dimensional implicit finite difference method with energy balance approach, in order to determine the optical and thermal performance of a solar parabolic trough collector under climate conditions of Algerian sahara. M. Ouagued et al [7] have estimated the potential of direct solar irradiance in Algeria and the performance of solar parabolic trough collector (PTC) under the climate conditions of the countries, in order to determine the most promising solar sites in Algeria. Z.D. Cheng et al [8] have used a Three-dimensional numerical simulation of coupled heat transfer characteristics in the receiver tube, is calculated and analyzed by combining the MCRT (Monte Carlo Ray-Trace Method) and the FLUENT software. It is revealed that the non-uniformity of the solar energy flux distribution is very large.B. Lamrani et al [9] have demonstrated that the HTF nature and the wheatear conditions have a significant influence on the thermal performance of the solar PTC. W. Qiliang et al[10] have optimized solar receiverare numerically applied to asmall thermal- collection field with 72m loop using molten salt as HTF to validate enhanced performance.

III. THEORITICAL STUDY

A. Modeling the absorber tube

The modeling of the absorber tube is based on the description of the various thermal exchanges occurring in this system. Assuming the axial symmetry, the thermal exchanges which occur are shown schematically in Fig. 2:

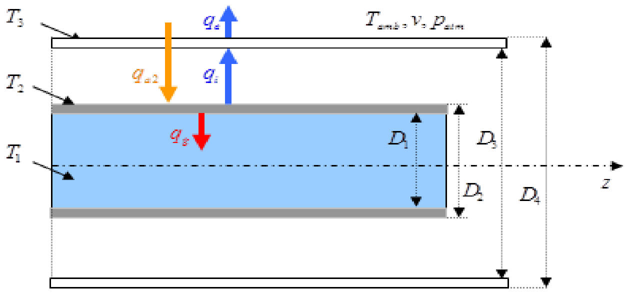

Fig .2: Thermal balance diagram of an element of PTC receiver.

The physical phenomenon is a heat transfer and mass transfer problem whose modeling is made under the following simplifying assumptions.

- The heat transfer fluid is incompressible.
- The shape of the parabola is symmetrical.
- The ambient temperature around the sensor is uniform.
- The glass is considered opaque to infrared radiation.
- The flow of the fluid is one-dimensional.
- Temporal variations in temperature in the thickness of the absorber and in the glass are negligible.
- The absorptivity of the glass is negligible.
- Conduction exchanges in the absorber and the glass are negligible.
- The effect of the shadow of the absorber tube on the mirror is negligible.
- The solar flux at the absorber is uniformly distributed.

TEST Engineering & Management
January - February 2020
ISSN: 0193 - 4120 Page No. 136 - 142

B. Thermal balance equations

1. The thermal balance between the absorber and the heat transfer fluid makes it possible to write [2].

$$\pi\rho_1 C_1 D_1 \frac{\partial T_1}{\partial t}(z,t) = -\rho_1 C_1 \phi \frac{\partial T_1}{\partial z}(z,t) + q_g(z,t)$$
(1)

where T_1 is the temperature within the heat transfer fluid

$\rho_1 = \rho_1(T_1)$: The density of the heat transfer fluid

$C_1 = C_1(T_1)$:the specific heat of the heat transfer fluid

D_1: the internal diameter of the absorber

ϕ: the volumetric flow rate of the heat transfer fluid

$q_g(z,t)$: The amount of energy gained by the heat transfer fluid

z: the coordinate along the length of the tube and the time.

Equation (1) adds the initial conditions and the following limits:

$$T_1(z,0) = T_{amb}(0) \qquad (2)$$

$$T_1(0,t) = T_e(t) \qquad (3)$$

where T_e is the assumed input temperature given and T_{amb} the ambient temperature.

1- *The thermal balance between the absorber and the glass envelope leads to the following equation [2]*

$$\pi\rho_2 C_2 D_2 \frac{\partial T_2}{\partial t}(z,t) = q_{a2}(t) - q_i(z,t) - q_g(z,t) \qquad (4)$$

where:

T_2: is the temperature within the heat transfer fluid,

$\rho_2 = \rho_2(T_2)$: The density of the heat transfer fluid

$C_2 = C_2(T_2)$: The specific heat of the heat transfer fluid

D_2: the internal diameter of the absorber

$q_{a2}(t)$:amount of solar energy absorbed by the tube

$q_i(z,t)$: amount of energy indicating the heat transfer between the absorber tube and the glass envelope.

The amount of solar energy absorbed by the absorber can be written as [1]:

$$q_{a2}(t) = \tau\beta\gamma\alpha_2 L I_b(t) \qquad (5)$$

where:

α_2: The absorptivity of the absorber

γ: The reflectivity of the concentrator reflecting surface

τ : The transmitivity of the glass

β : The optical factor of the sensor

L: The length of opening of the concentrator and

$I_b(t)$: The solar radiation received.

Equation (1.4) is subject to the following initial condition:

$$T_2(z,0) = T_{amb}(0) \qquad (6)$$

2- *The thermal balance between the glass envelope and the environment is written [2]*

$$\pi\rho_3 C_3 D_3 \frac{\partial T_3}{\partial t}(z,t) = q_i(z,t) - q_e(z,t) \qquad (7)$$

where:

T_3: is the temperature within the heat transfer fluid,

$\rho_3 = \rho_3(T_3)$: The density of the heat transfer fluid

$C_3 = C_3(T_3)$: The specific heat of the heat transfer fluid

D_3: The internal diameter of the absorber

$q_{a3}(t)$: The amount of solar energy absorbed by the glass

$q_e(z,t)$: The amount of energy indicating the heat transfer between the glass and the ambient air.

Equation (1.7) is subject to the following initial condition:

$$T_3(z,0) = T_{amb}(0) \qquad (8)$$

C. Calculation of the quantities of energy exchanged

Considering the forced convection of the heat transfer fluid in the absorber tube:

$q_g(z,t)$ is calculated by the following equation which applies to a fully developed flow in a smooth circular tube.

January - February 2020
ISSN: 0193 - 4120 Page No. 136 - 142

$$q_g(z,t)=\pi h_1 D_1\left(T_2(z,t)-T_1(z,t)\right) \quad (9)$$

where h_1 is the coefficient of convective exchange between the heat transfer fluid and the absorber.

The coefficient is given by:

$$h_1=\frac{N_{u1}\lambda_1}{D_1} \quad (10)$$

where: $\lambda_1=\lambda_1(T_1)$ is the thermal conductivity of the heat transfer fluid

The local Nusselt number is given by [2]

$$N_{u1}=0.023R_{e1}^{4/5}P_{r1}^{n} \quad (11)$$

with $n=0.4$ for $T_2 \succ T_1$ and $n=0.3$ for $T_2 \prec T_1$ and where R_{e1} the Reynolds number and P_{r1} Prandtl number.

The Reynolds number is defined by:

$$R_{e1}=\frac{4\rho_1\phi}{\pi\mu_1 D_1} \quad (12)$$

where: $\mu_1=\mu_1(T_1)$ is the dynamic viscosity of the heat transfer fluid. The number of Prandtl is defined by:

$$P_{r1}=\frac{\mu_1 C_1}{\lambda_1} \quad (13)$$

The heat transfer between the absorber and the glass envelope is essentially achieved by radiation between the absorber tube and the glass envelope.

The amount of heat exchanged is [1]

$$q_i(z,t)=\frac{\pi\sigma D_2\left(T_2^4(z,t)-T_3^4(z,t)\right)}{\frac{1}{\varepsilon_2}+\frac{1-\varepsilon_3}{\varepsilon_3}\frac{D_2}{D_3}} \quad (14)$$

where $\varepsilon_2=\varepsilon_2(T_2)$ is the emissivity of the absorber, ε_3 the emissivity of the glass and σ the constant of Stefan Boltzmann.

The heat transfer between the glass envelope and the environment is achieved by convection and radiation. The quantity of heat exchanged is written:

$$q_e(z,t)=\pi h_3 D_4\left(T_3(z,t)-T_{am}(t)\right)+\pi\sigma\varepsilon_3 D_4\left(T_3^4(z,t)-T_{amb}^4(t)\right) \quad (15)$$

where h_3 is the convection transfer coefficient between the glass envelope and the ambient air.

The coefficient of transfer by convection is written:

$$h_3=\frac{N_{u3}\lambda_{air}}{D_4} \quad (16)$$

The number of Nusselt is given by [1]:

$$N_{u3}=0.3+\frac{0.62R_{e3}^{1/2}P_{r3}^{1/3}}{\left(1+\left(\frac{0.4}{P_{r3}}\right)^{2/3}\right)^{1/4}}\left(1+\left(\frac{R_{e3}}{282000}\right)^{5/8}\right)^{4/5} \quad (17)$$

The number of Reynolds is given by:

$$R_{e3}=\frac{\rho_{air}vD_4}{\mu_{air}} \quad (18)$$

where: v is the velocity of the wind, $\rho_{air}=\rho_{air}(T_{moy})$ the density of the air, - $\mu_{air}=\mu_{air}(T_{moy})$ is the kinematic viscosity of the air.

The mean air temperature is estimated using the following formula:

$$T_{moy}=\frac{T_3+T_{amb}}{2} \quad (19)$$

The number of Prandtl is given by:

$$P_{r3}=\frac{\mu_{air}C_{air}}{\lambda_{air}} \quad (20)$$

IV. NUMERICAL METHOD

1. Diagram in TRNSYS software

After its implementation in the TRNSYS16 TESS library, the model (type 536), was integrated into an evaluation project under different climatic zones in Morocco shown in Fig.3.

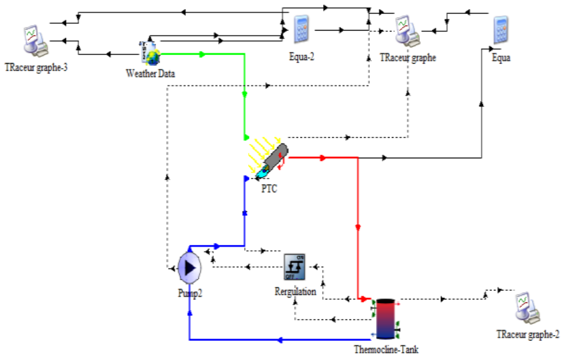

Fig.3: Simulation by TRNSYS of a solar PTC with Thermocline- Tank.

January - February 2020
ISSN: 0193 - 4120 Page No. 136 - 142

We use the TRNSYS16 software to simulate PTC system using solar salt as the HTF in the first case. All parameters of the PTC were introduced in the TRNSYS model shown in Fig. 2. This model consists of a component for the weather data measurement of different climatic zones in Morocco, related to the type (536) for the CSP collector. The concentrator is connected in closed loop with a pump (Flow rate = 275 [kg/h]) and a tank volume = 0.3 [m^3], specific heat of the fluid = 1.495 [kJ/kg K]).Several points in the model are connected to a plotter in order to visualize different parameters, namely the HTF inlet and outlet temperature, the useful power and the direct irradiation in addition to a printer to output the obtained results.

2. *Meteorological data*

Typical Meteorological Year (TMY) for a specific location is employed to generate meteorological data. This type under TRNSYS software is generated from a database in order to present the range of weather for the specific location by giving the average data for a year which are compatible with the long-term averages for the same location.

According to the objective of the present study that investigates the thermal performance of a PTC receiver under the Moroccan meteorological data, the selected location is Errachidia city. The geographic location of Errachidia represents the best climatic conditions such as the abundant sunshine throughout the year, low humidity and precipitation.Fig. 4 shows the variation of the ambient temperature and the wind velocity throughout the year for Errachidia site. The evolution of direct normal irradiance (DNI) is represented in Fig. 5.

According to the objective of the present study that investigates the thermal performance of a solar PTC under the Moroccan meteorological data, the selected location is Errachidia city, which is the second most important insolation region in Morocco.

In the following section, the dynamic results are presented during a representative week (first week of July) and the variation of direct normal irradiation is summarized throughout the year with different HTFs using in the first case solar salt.

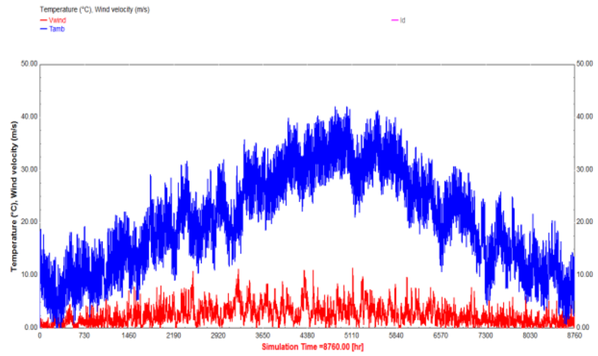

Fig.4: Annual ambient temperature and wind velocity

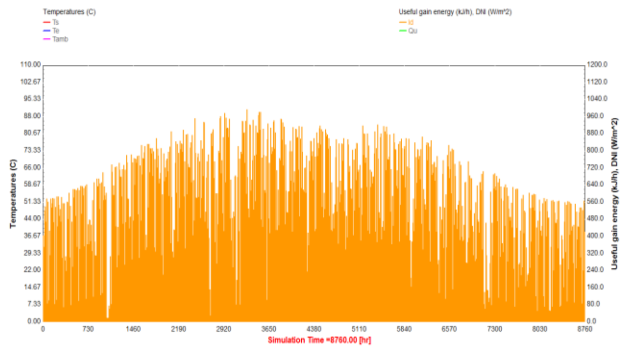

Fig.5. Annual direct normal irradiation

V. RESULT AND DISCUISSION

The hourly variation of the outlet temperature of the PTC collector is represented in Fig.6. It can be observed that HTF outlet temperature at the solar collector varies periodically with time and its minimum value can reach ambient temperature (Tamb) value during de day time.

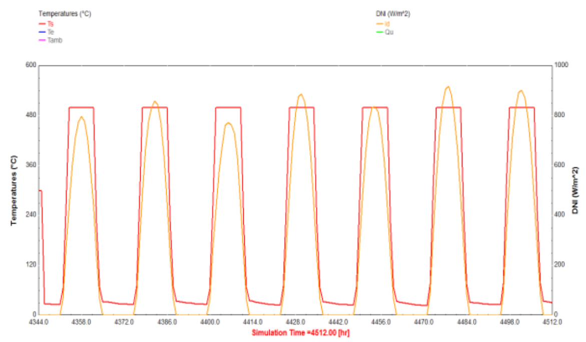

Fig. 6. Variation of Hitec salt outlet temperature

January - February 2020
ISSN: 0193 - 4120 Page No. 136 - 142

according to direct normal irradiation and ambient temperature during the first week of July

It can be observed that HTF outlet temperature at the CSP collector varies periodically with time. In addition required outlet temperature of HTF (Hitec salt) during the operation time (from 6:00 am to 7:00pm) corresponding to 500°C and decreases at the ambient temperature at night are illustrated in Fig.6.

During the operation time of the CSP system, the outlet temperature of solar salt delivered by CSP system increases as the useful heat gained by the solar collector increases. The useful heat gained by the solar collector during the representative week which is ranged between a minimum value of 0.775 kW and maximum value of 0.919 kW. It can be noted also that maximum value of useful heat delivered by the CSP system is obtained at July, 6th.

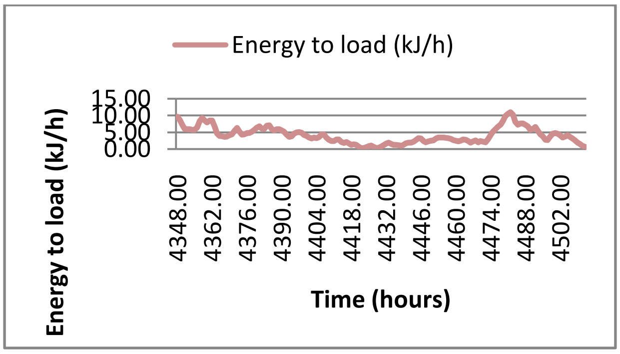

Fig. 7. Energy rate to load during the first week of July

Fig. 7 shows the rate at which energy is removed from the tank to supply the load of the system components. The generation of the energy is greater when the temperature differences are higher. Thus, The rate energy to load gained by the single- tank during the representative week which is ranged between a minimum value of 1.15 kJ/h and maximum value of 11.05 kJ/h.

On the other hand, the effect of the solar radiation intensity on thermal losses is shown in Fig.8 and Fig.9.The difference between provided heat from the sun and the thermal losses at tank level increase with increasing solar radiation intensity during the first week of July.

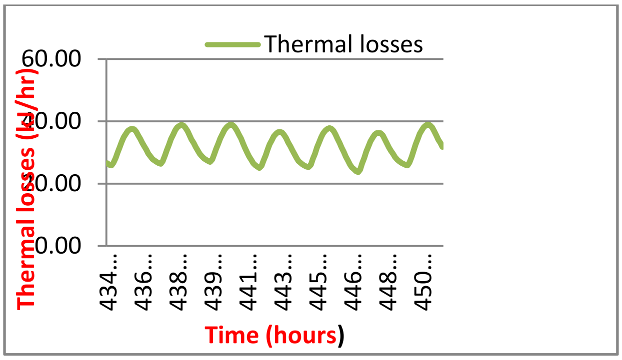

Fig.8. The rate of thermal energy loss to the environment

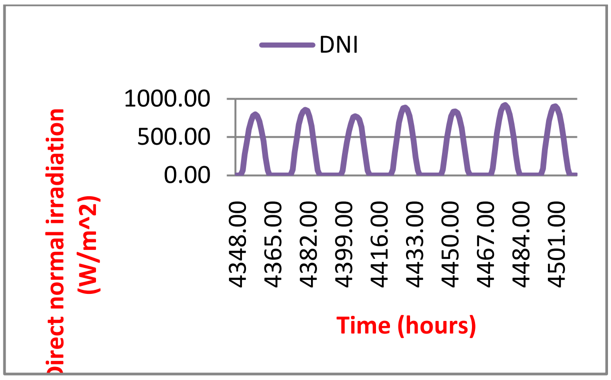

Fig.9. Direct normal irradiation during the first week of July

As noted above, heat loss to the ambient atmosphere has a significant influence on the discharge performance of the TES system. Particularly, when the temperature difference inside and outside is large, the heat loss becomes significant. In order to predict the insulation performance, the heat loss was calculated using a parallel thermal resistance model should be taken into consideration for an even more accurate analysis.

VI. Conclusion

In the present paper, a numerical simulation of a solar Parabolic Trough Collector (PTC) receiver under the new climatic sites of Morocco is presented. A detailed numerical model based on energy balances at the component level of the PTC is developed and simulated by TRNSYS software. We have presented a mathematical model of the PTC receiver capable of predicting the response of the system in terms of the output temperature when considering the following input variables: inlet

January - February 2020
ISSN: 0193 - 4120 Page No. 136 - 142

temperature, heat transfer fluid flow rate, ambient temperature, speed wind and solar irradiation. A non-thermal equilibrium model is used for investigating the effect of different HTFs on the thermal performance of a PTC system using Therminol, Molten salt and HITEC respectively as the HTFs and quartzite rock as the filler material in the tank. As future work, we prospect to study the thermal performance of the system and add the control of TES systems in the tank for concentrated solar power plants. And we prospect to study Thermal characteristics, including temperature profiles and discharge effectiveness of storage tank.

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