A TRNSYS dynamic simulation model for a parabolic trough solar thermal power plant

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Abstract – This paper presents a validated TRNSYS model for a thermodynamic plant with parabolic trough solar thermal power (PT). The system consist of trough solar collector (PTC) as well as auxiliary components. The simulation of the system has been done during the day (01/01) under the meteorological conditions of Ain Témouchent city (Algeria). The model compared the energy performance of the systems: case (1) - Rankine cycle facility with solar field and case (2)- Rankine cycle facility without solar field. The results showed that the present model has a good agreement with the experimental data of the literature. In case (1), PTC fluid outlet temperature reach the maximum value 330 °C, Work of the steam turbine increase from the 9hr to reach its maximum value 856 KJ/Kg at 13 hr. In case (2), the maximum value of the power remains constant from the beginning of the simulation to 1hr00. Since the flow of fuel (gas natural) consumed does not change throughout the operating period.

Keywords: TRNSYS, Solar thermal power plant, Rankine cycle, parabolic trough power.

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I. Introduction

Electricity generation using a hybrid system combining several renewable energy sources is of great interest to developing countries, such as the Maghreb countries. It is well known that generate energy from a solar thermal system with the following technologies are currently used: 1. Parabolic trough (PTC) collector; 2. Fresnel linear reflector (LFR); 3. Sterling dish and Solar energy tower [1]. Among these technologies, the parabolic collector is more popular and is used in many places around the world. As a result, large concentrated solar power (CSP) facilities are PTC technology. Several works was realized on TRNSYS (Transient Systems Simulation) of solar thermal power station. Jones et al. [2] created a 30 MWe SEGS VI parabolic collector base plant model in TRNSYS to evaluate solar field and power cycle behavior. Kolb [3] evaluates the performance of the 1 MWe Sugurao solar power plants with and without thermal storage using the TRNSYS software. The objective of this work is to present the simulation of the hybrid solar field and power cycle model. The STEC library of the TRNSYS software is used for an Algerian site (Ain Témouchent) and meteorological data is provided by the Meteornorm software [4]. Our goal in this study is the design of a Rankine cycle facility with and without solar field by TRNSYS software.

II. Presentation of the parabolic trough solar thermal power plant used

In the first part, we used the Rankine cycle with the solar field (Figure 1). The components of the steam cycle are steam generators (economizer, evaporator, and overheater), three stage steam turbine, condenser, pump, degasser, subcooler, preheater and solar field (parabolic trough). The objective of the collector field is to produce electricity from the thermal installation. The system consists of a parabolic trough collector type field connected to the Rankine cycle through a series of heat exchangers. The heat transfer fluid (THERMINOL VP-1) is heated by circulating in the absorbers of the parabolic concentrators (solar field) and returns to the Rankine type power cycle, the role of the coolant is to transport the heat from the hot source ( solar rays concentrate in the parabolic trough). The average temperature of the heat transfer fluid changes according to the weather variations from one hour to another during one year (8760 hours). This fluid is pumped with a constant flow to the steam generator, which consists of an economizer, evaporator and superheater.

The heat exchangers are of the counter flow type. Heat transfer fluid (HTF) with Rankine cycle water flow is in opposite directions. The heat transfer fluid enters the superheater at high temperature before passing to the steam generator where the water of the power cycle undergoes a phase change from the liquid state to the vapor state. Then, the heat transfer fluid passes through
the economizer where yields to the water which is in the liquid state (called feeding) its energy. Finally, the fluid is recirculated through the solar field. Water before returning to the economizer to complete the cycle the supply water in the liquid state exiting the condenser passes through degasser and then has preheater.

This plant also contains a 3-stage turbine namely high pressure, medium pressure and low pressure and two steam withdrawals that communicate with the preheaters. This withdrawn steam is used to heat the feed water before it enters the economizer to increase cycle efficiency. The outgoing steam at the low-pressure turbine is condensed in a condenser.

Figure 1. Schematic diagram of the Rankine cycle with solar field

III. System simulation using TRNSYS

TRNSYS (Transient Systems Simulation) is a complete and extensible simulation environment for the transient simulation of systems, including multi-zone buildings. It is used by engineers and researchers around the world to validate new energy concepts, from simple domestic hot water systems to the design and simulation of buildings and their equipment, including control strategies, occupant behavior, alternative energy systems (wind, solar, photovoltaic, hydrogen systems) [5].

The Solar Thermal Electric Component (STEC) libraries were developed by Peter Schwarzbozl (DLR, Germany) and Scott Jones (SNL, New Mexico) in 2002. They are used to simulate thermal systems (solar and conventional) in order to produce electricity. These libraries are not issued as standard with the TRNSYS software [6].

Figure 2 shows the model under Rankine Cycle TRNSYS with solar field and Figure 3 shows the power cycle installation without solar field. Once all the components of the system have been identified, the main components of this model are described and shown in Table 1.

Figure 2. Presentation of the model under TRNSYS of Rankine Cycle with solar field
Figure 3. Presentation of the model under TRNSYS of Rankine Cycle without solar field

Table 1. The components and input design parameters in TRNSYS simulation programs

| Component                          | Type          | Input parameter                                      | Value                      |
|-----------------------------------|---------------|------------------------------------------------------|----------------------------|
| Parabolic trough collector (PTC)  | TYPE396       | Cw - Loss coef.                                      | 0                          |
|                                   |               | D - Loss coef.                                       | -0.096                     |
|                                   |               | Clean Reflectivity                                   | 0.94                       |
|                                   |               | Broken Mirror Fraction                               | 0.0                        |
|                                   |               | Length of SCA (m)                                    | 14.37                      |
|                                   |               | Aperature Width of SCA (m)                           | 1.52                       |
|                                   |               | Focal Length of SCA (m)                              | 0.45                       |
|                                   |               | Rowspsacing(m)                                        | 4.54                       |
|                                   |               | Total Field Area (m²)                                | 200000                     |
| Superheater and economizer        | TYPE 315      | Counter flow mode                                    | 2                          |
|                                   |               | Overall heat transfer coefficient of exchanger (KJ/hr K) | 1037068.37(SUP)             |
|                                   |               | Reference press loss cold side (bar)                 | 3055710.24(ECO)            |
|                                   |               | Reference cold side flow (Kg/s)                      | 1                          |
|                                   |               | Reference cold side flow (Kg/s)                      | 12.14                      |
| Evaporator                        | TYPE 316      | overall heat transfer factor                          | 8164463.48                 |
|                                   |               | reference pressure loss (bar)                         | 1                          |
|                                   |               | reference flow rate (Kg/s)                           | 12.14                      |
| Weather data reading and processing | TYPE109-TMY2 | Ain Temouchent                                       |                            |
| Online plotter                    | TYPE 391      | converting water vapor temperatures to enthalpies     |                            |
| S-split                           | TYPE 389      | Graphical plotter in line with the output file.      |                            |
| Forcing Function Deaerator        | TYPE 14h      | Simulate a vapor separator to allow extraction.      |                            |
|                                   | TYPE 384      | Constant gas flow in time runs.                      |                            |
|                                   |               | Preheater mixer for feed water or steam is mixed with |                            |
|                                   |               | condensate undercooled to produce saturated water at |                            |
|                                   |               | the outlet                                            |                            |
| Turbine                           | TYPE 318      | design inlet pressure (bar)                          | 1                          |
|                                   |               | design outlet pressure (bar)                         | 100-23.89 -2.875           |
|                                   |               | design flow rate(Kg/s)                               | 23.89-2.875-0.08           |
|                                   |               | design inner efficiency                              | 12.14-9.919-8.82           |
|                                   |               |                                                      | 0.84- 0.85- 0.86          |
| Condenser                         | TYPE 383      | dT Cool water out+condensing temp (delta C)          | 4                          |
| Condenser pump and feed pump      | TYPE 300      | Type                                                  | CP                        |
|                                   |               | Maximum flow rate                                     | 8.525                     |
|                                   |               | Fluid specific heat                                   | 4.17814                   |
|                                   |               | Maximum power                                         | 1000                      |
|                                   |               |                                                      | 1000                      |
| Preheater                         | TYPE 317      | cold fluid spec. heat capacity (KJ/Kg K)             | 4.24                      |
|                                   |               | overall heat transfer factor (KJ/hr K)               | 186840                    |
|                                   |               | cold sid ref flow rate (Kg/s)                        | 12.14                     |
| Subcooler                         | TYPE 320      | Specific heat of hot side fluid (KJ/Kg K)            | 3.355                     |
|                                   |               | Specific heat of cold side fluid(KJ/Kg K)            | 4.240                     |
IV. Results and discussion

IV.1. Meteorological data

In order to run a simulation for one year and to predict the performance of the SSH system, a typical Meteorological Year of Ain Témouchent (TMY) is used at GMT1 time zone, latitude 35.306°N and longitude -1.147°E were used. Figure 4a shows the solar radiation, (b) wind speed and dry bulb temperature. The peak Total horizontal radiation is approximately 360 W.m\(^2\) and occurs in the period from June to Jul. This period is characterized by high ambient temperature with the peak 37 °C and maximum wind speed is 15 m/s.

Figure 5 shows the variation of the ambient temperature and annual average humidity. For the variation of the average annual moisture content, we observe that it is less important during the summer season it is balance between 0.48 and 0.63, so is important during the winter season it is balance between 0.63 and 0.82.

IV.2. Thermal performance evaluation with solar field

IV.2.1. PTC fluid inlet and outlet temperature

The curve illustrated in Figure 6 shows that at the beginning of the heating, at \( t = 0 \) the outlet temperature of the coolant is equal to the temperature of the injected fluid, after the sun rises this temperature increases with solar radiation concentrated to the concentrator opening, it reaches 200 °C between 09hr and 10hr while the inlet temperature reaches 177 °C. It continues its increase to reach the maximum value 330 °C to 13hr. During the afternoon, it will decrease until it reaches zero.
IV.2.2. Steam generator temperatures of the fluid on the hot and cold sides

The curve shown in Figure 7 shows that steam generator fluid temperatures on the hot and cold sides increase to a maximum around 13hr at the steam generator inlet. This period corresponds to maximum sunlight. With regard to the (cold side), it is noted that the outlet temperature (cold side) in each exchanger will increase and reach the maximum output value 327 ° C to 13:00h. After 13:00h, the exit temperatures on each side and each exchanger will go down when the irradiation of the sun decreases until 18:00h or it will be zero.

IV.2.3. Transferred power

The curve illustrated in Figure 8 shows the evolution of the transferred power of each exchanger: economizer, evaporator, super heater and condenser. Both types of transferred power ($Q_{in}$ and $Q_{out}$) go into an increase from 8am in the morning to reach a maximum value at 13:00h such that $Q_{in,\max} = 3.9 \times 10^7$ KJ/hr and $Q_{out,\max} = 3.4 \times 10^7$ KJ/hr, then decrease in the next period to reach the zero value at 18:00h.

It is observed that the transferred power used by the system ($Q_{in}$) is greater than the heat lost ($Q_{out}$), this difference is the main advantage for the increase of the efficiency of the installation.

IV.2.4. Work of the steam turbine

Figure 9 shows the evolution of the three stages of the steam turbine and the total work: It is observed that the work begins to increase from the 9:00h to reach its maximum value 856 kJ / kg at 13:00h. Then it starts to decrease after this maximum value so that it is zero at 18:00h. Note also that the intermediate floor work is larger than the floors of the ends.

IV.3. Thermal performance evaluation without solar field

IV.3.1. Power of the steam turbine

In this cycle, the thermal performance is almost constant throughout the week, due to the stability of the turbine power used, if for this reason we will take just one day in this cycle to evaluate the thermal behavior of the installation as we have a week of evaluation. In the second part, we will study a plant of the power cycle by the use of natural gas, and the results of this simulation will be presented by the figures below. The curve shown in Figure 10 shows the evolution of the power of each stage obtained by the steam turbine; we note that the maximum value of the power remains constant from the beginning of the simulation to 01:00h. Since the flow of fuel (gas natural) consumed does not change throughout the operating period.
V. Conclusion

This article aims at the study of an installation of the Rankine cycle powered energetically by the solar field; through a modeling, which was carried out by TRNSYS software, knowing that the weather conditions of Ain Témouchent was used for the purpose of, evaluate the thermal behavior of the installation during one week.

Two cases were studied: case (1) - Rankine cycle facility with solar field and case (2) - Rankine cycle facility without solar field. Results obtained showed:

In case (1), PTC fluid outlet temperature reach the maximum value 330 ° C, Work of the steam turbine increase from the 9hr to reach its maximum value 856 KJ/Kg at 13:00h.

In case (2), the maximum value of the power remains constant from the beginning of the simulation to 1hr00. Since the flow of fuel (gas natural) consumed does not change throughout the operating period. The results showed that the minimum efficiency of the facility is 13% in the winter period. For a power equal to 4MW, the natural gas consumption can be gained of 312.4 kg/hr in the sunny period.

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