Modelling and Exergetic Analysis of a Parabolic Trough Solar Collector

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Abstract. A parabolic trough collector was subject of a thermal study considering heat exchange mechanisms involved in the conversion process of solar energy into heat. The physical model presented was developed with assumptions related to steady state conditions. A code, developed under Matlab, allowed to estimate, by an iterative calculation, all temperatures encountered in the studied system and to calculate, consequently, various efficiencies: optical, energetic and exergetic. After validation of the model, a parametric analysis was carried out in order to consider the influence of the main model parameters on the exergetic behavior of the solar system.

1. Introduction
Due to its maturity and the relative mastery of its implementation, Parabolic Trough Collector is the most successful advanced technology among concentrated solar systems. This technology is used in several projects around the world, both for pure research purposes and for improvement and testing of industrial processes [1]. Its operating principle consists of concentrating sunlight on a solar heat exchanger (generally à vacuum absorber tube), located along the focal line of the collector. A heat transfer fluid (HTF) is heated after its passage through solar heat exchanger.

Fernández-García A., et al [2] provide an excellent review of the PTCs application areas. Generally, there are two main categories, depending on the temperature levels reached. In the low and medium temperatures range (100°C -200°C), the PTC can be employed as a heat source for various practical processes, ranging from domestic applications such as heating and refrigeration [3-4] to industrial processes such as seawater desalination and solar drying of agricultural products [4]. Solar power plants are the other category of PTC’s applications and the most important in terms of projects carried out or in progress [1]. The temperatures reached are between 300 °C and 450°C, therefore they are able to produce high superheated steam for solar thermodynamic electricity generation. The steam is produced directly [5] in a solar field (DSG), or indirectly by transporting the HTF either to a solar steamer [6] or a recovery boiler [7]. For readers interested in PTC’s applications topic, A. Fernandez-Garcia et al [8] and V.K. Debasing et al [9] gave an excellent global and recent review.

Assuming that heat-loss rate is a linear function of the absorber temperature, the problem of the solar radiation conversion into a useful heat, in a linear collector, received a first analytical solution in the Hottel's pioneering work [10]. Making some improvements in heat transfer correlations usually encountered in the study of PTC, Padilla and al [11] introduced a novel one-dimensional heat transfer
model. Seyed Ebrahim Ghasemi et al [12] have developed three-dimensional thermo-fluid steady model of PTC where the governing equations are solved with Finite Volume Method (FVM) and implemented using CFD software (Fluent). On the other hand, many approaches [13,14,15]) were developed to determine the thermal implications induced by the fact that solar flux distribution on the absorber-tube is not uniform.

In this work, the parabolic trough collector is subject to a detailed study considering heat exchange mechanisms involved in the conversion of solar radiation into heat. An analytical model, as well as a novel computation code, developed under Matlab, is proposed. The overall heat loss coefficient derived analytically is modified by a correction factor estimated from experimental work available in literature. A model, which is successfully validated by Sandia National test results in term of HTF outlet temperature and energy efficiency, is used to identify and examine the impact of operational and environmental parameters on exergy collector performances. Finally and as a demonstration case, the present investigation is completed by PTC year-round exergy performance evaluation types working under climatic conditions of four Algerian regions.

2. Mathematical model

We propose to determine the HTF outlet temperature \( T_{fo} \) passing through the PTC’s absorber-tube whose flow mass rate \( m_f \) and inlet temperature \( T_{fi} \) are controlled.

The energy efficiency of the system is given by the following equation:

\[
\eta_g = \frac{Q_u}{I A_o}
\]  
(1)

Where:

- \( Q_u \): Useful thermal
- \( I \): Direct normal irradiance
- \( A_o \): Aperture area

The ratio between the HTF exergy gained and the solar exergy received defines the exergy efficiency. Neglecting the destroyed exergy due to the HTF frictions, the gained exergy expression is given by (equation 2).

The exergy of the solar radiation arriving on the concentrator is quantified according to Petela’s formula [15].

\[
\text{Ex}_{\text{gain, fluid}} = m_f \left[ \int_{T_{fi}}^{T_{fo}} C_p f T \, dT - T_a \int_{T_{fi}}^{T_{fo}} C_p f \frac{T_a}{T} \, dT \right]
\]  
(2)

An analytical model based on the steady state balance equations of the absorber tube was selected in this study.

![Figure 1: Heat exchange at the receiver](image)

2.1. Thermal gain

The useful energy \( Q_u \) transmitted by the tube absorber depends on the optical efficiency of the system and the heat losses towards the environment.

\[
Q_u = F_R A_o \left[ \eta_{opt} - \frac{U}{C} (T_{fi} - T_a) \right]
\]  
(3)
\( F_R \): Heat removal factor function, among others, of the convective exchange coefficient \( h_f \) between the tube absorber and the HTF; \( \eta_{\text{opt}} \): Optical efficiency; \( U \): Overall heat loss coefficient; \( C \): Concentration ratio; \( T_a \): Ambient temperature.

2.2. Energy absorbed

The solar energy absorbed is given by the product of the solar energy available at the aperture area and the systems optical efficiency (equation 4). The latter is determined by the intrinsic characteristics of the concentrator \( \eta_{\text{optmax}} \) weighted by the function \( E(\theta) \) which represents the joint contribution of the geometric, energetic and isometric effects induced by the incidence angle \( \theta \).

\[
Q_{\text{abs}} = \eta_{\text{opt}}I_A
\]  

(4)

2.3. Thermal losses

Taking as a reference the outside absorber tube surface \( A_{ab,e} \) that is at temperature \( T_{ab,e} \), the expression of the heat losses takes the following form:

\[
Q_L = UA_{ab,e}(T_{ab,e} - T_a)
\]  

(5)

The overall heat loss coefficient is deduced analytically from the heat exchange modes taking place within the absorber tube (see Figure 1).

\[
U = \left[ \frac{D_{ab,e}}{h_{c,e} + h_{r,c,e}}D_{co,e} + \frac{1}{2K_{co}} \ln \left( \frac{D_{co,e}}{D_{co,i}} \right) + \frac{1}{h_{r,co} + h_{c,co}} \right]^{-1}
\]  

(6)

Where:

- \( h_{r,ab-co} \): Radiation heat transfer coefficient between absorber tube and glass cover (Stefan-Boltzmann law).
- \( h_{r,co-a} \): Radiation heat transfer coefficient between the glass cover and the ambience (Stefan-Boltzmann law).
- \( h_{c,ab-co} \): Convection heat transfer coefficient from glass cover to the ambient.
- \( h_{c,co-a} \): Convection heat transfer coefficient from the absorber tube to glass cover.
- \( K_{co} \): Thermal conductivity of the glass cover
- \( D_{co,i}, D_{co,o}, D_{co,i} \) and \( D_{co,o} \) are respectively the outside/inside glass cover and absorber tube diameters.

When the pressure in annulus is greater than 0.013 Pa, heat transfer occurs by natural convection mechanism. In this case, the Raithby and Holland’s correlation in enclosure [16] is recommended to estimate \( h_{c,ab-co} \). All physical properties of the gas are evaluated at the average temperature \( (T_{ab,e} + T_{co,i})/2 \). The convection heat loss from glass cover to ambient, in the presence of wind, is estimated using Zhukauska’s correlation [17] for external forced convection flow normal to an isothermal cylindrical surface. Concerning heat transfer coefficient \( h_f \) from the absorber tube to HTF, we used- for mass flow rates inducing a totally developed turbulent flow- the correlation of Gnielinski [18]. All physical properties and dimensionless numbers involved in \( h_f \) are evaluated at the HTF average temperature except for the Prundt number \( Pr_{ab,i} \) which is evaluated at the inside absorber tube surface temperature.

3. Computational code and model validation

Iterative method is used to solve temperature equations system. Heat transfer is estimated by considering a random value of temperature (close to its boundary temperature), which is then determined by an energy balance equation. If the difference between estimated and calculated absolute temperature is greater than a desired accuracy (0.001 in the simulation), the iterative process is repeated. The mathematical model developed in this study involves five unknown interdependent temperatures \( T_{f_o}, T_{ab,o}, T_{ab,i}, T_{co,i} \) and \( T_{co,o} \). Simulation procedure of steady state model developed in this work is implemented in Matlab environment. The code is validated by experimental measurements carried out by Sandia National Laboratory at the AZIRAK test platform.
Simulation results obtained by our code are very close to the experimental ones published by Dudley V et al as it is possible to observe it on the table 1. Note that the relative average error is 3.27\% for heat transfer fluid outlet temperature and 1.58\% for energy efficiency.

Table 1. Comparison of the outlet temperature and overall efficiency between SNL experimental and model simulation results.

| Test Conditions | THF outlet temperature | Energy efficiency |
|-----------------|------------------------|-------------------|
| DNI (W/m²)  | T_a (°C) | m_f (Kg/s) | T_f,i (°C) | Exp (°C) | Model (°C) | Error (%) | Exp (%) | Model (%) | Error (%) |
|----------------|----------------|----------------|-------------|----------|-----------|-----------|---------|-----------|-----------|
| 933.7          | 21.2          | 47.7           | 102.2       | 124.0    | 123.40    | 2.75      | 72.51   | 71.62     | 1.23      |
| 968.2          | 22.4          | 47.8           | 151.0       | 173.3    | 172.85    | 2.01      | 70.90   | 70.95     | 0.07      |
| 982.3          | 24.3          | 49.1           | 197.5       | 219.5    | 219.06    | 2.00      | 70.17   | 70.00     | 0.24      |
| 909.5          | 26.2          | 54.7           | 250.7       | 269.4    | 268.65    | 4.01      | 70.25   | 68.38     | 2.66      |
| 937.9          | 28.8          | 55.5           | 297.8       | 316.9    | 316.15    | 3.92      | 67.98   | 66.56     | 2.09      |
| 880.6          | 27.5          | 55.6           | 299.0       | 317.2    | 316.20    | 5.49      | 68.92   | 66.16     | 4.00      |
| 903.2          | 31.1          | 56.3           | 355.9       | 374.0    | 373.50    | 2.76      | 63.82   | 63.02     | 1.25      |
| 920.9          | 29.5          | 56.8           | 379.5       | 398.0    | 397.40    | 3.24      | 62.34   | 61.56     | 1.25      |

Average 3.27 1.58

4. Results and discussion
The results presented in this section relate to the PTC LS2 type operating with Syltherm 800 as HTF. This synthetic oil is frequently encountered in installations already operational throughout the world.

4.1. Effect of operating conditions
At figure 2, we observe that exergy efficiency increases as inlet temperature increases and decreases as mass flow increases. When the mass flow rate increases, the entropy and the heat gained increase with, nevertheless, the entropy which increases more considerably. As a result, the exergy gained becomes larger at low flow rates. Thus, for an inlet HTF temperature set at 100 °C, and under the conditions specified in figure 2, the entropy generated, when the flow rate increases from 0.1 kg/s to 1 kg/s, is respectively equal to 53.3 J/K and 63.1 J/K while the gained heat is 23.14 W and 23.97 W for the same flow values. This results in an exgetic efficiency of 23.58\% for a flow rate of 0.1 kg/s and a low efficiency of 16.94\% for a flow rate of 1 kg/s.

Figure 2: Exergy efficiency as a function of mass flow rate and HTF inlet temperature (I = 850W / m², T_a = 25 °C)
It should be noted that the exergy efficiency drops sharply in the range of low mass flow rates but tends towards a quasi-constant value for mass flow rates higher than 1 kg/s (Figure 3). As HTF input temperature increases, the gained heat as well as the generated entropy drop simultaneously but not in the same ratio. Indeed, the generated entropy being less sensitive to HTF inlet temperature than the gained heat, the difference becomes greater as this temperature increases which results in a better exergetic efficiency.

4.2. Style and spacing
As can be seen in figure 4, exergetic efficiency increases as radiation increases and decreases as ambient temperature increases. Fixing ambient temperature at 25°C, the exergetic efficiency is equal to 15.20% for $I=100\, \text{W/m}^2$ and to 21.13% for $I=1000\, \text{W/m}^2$. Although the generated entropy increases when the solar irradiance increases, energy gained increases more so that exergy gain becoming more important which improves significantly the exergetic efficiency.

![Figure 3](image-url) Exergy efficiency as a function of HTF mass flow rate ($I = 850\, \text{W/m}^2$, $T_a = 25\, ^\circ\text{C}$)

![Figure 4](image-url) Exergy efficiency as a function of ambient temperature and direct normal irradiance (m= 0.2Kg/s, $T_h = 100\, ^\circ\text{C}$)
On the other hand, if we set the irradiance $I$ at 850W/m$^2$, the exergy efficiency will be 22.24% for $T_a = 15\, ^\circ C$ and 17.62% for $T_a = 35\, ^\circ C$. We also note (figure 5) that the sensitivity of the exergy efficiency to ambient temperature $T_a$ is almost linear with an almost constant slope (-0.18) whatever a direct normal irradiance level (the three lines are parallel).

The factor $T_a$ appearing in the entropic term of equation 2 causes the fact that the gained exergy decreases with increasing temperature. This relation explains the linear dependence of the exergy efficiency with respect to ambient temperature.

5. Conclusion

The parabolic trough concentrator has been subject to detailed exergy study. The system performance depends on the optical efficiency of the concentrator and the thermal losses at the receiver. These two quantities are determined by system geometry, physical properties of materials, solar irradiance and sun tracking mode. The program can be used to estimate all the temperatures encountered in the study of the parabolic trough collector and consequently to determine the corresponding efficiency.

Effects of model parameters on the exergetic performance of the concentrator were discussed whose mains conclusions are:

- The exergetic efficiency is all the more important as the flow rate is low and the HTF inlet temperature is high;
- The exetical efficiency is all the more important as the DNI is high and the ambient temperature is low.

Finally, the proposed and developed code as part of this work has the advantage of being manageable and easy to implement, which makes it suitable to serve as an engineering simulation tool for sizing and performance estimating studies of PTCs solar fields integrated in power thermal process.

6. Reference

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