Simulation of different modes of heat transfer on a parabolic trough solar collector

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Abstract- *The development of solar concentrator technology has just reached a very significant level. Using reflectors to concentrate the sun's rays on the absorber dramatically reduces the size of the absorber, reducing heat loss and increasing its efficiency at high temperatures. Another advantage of this system is that the reflectors are significantly less expensive, per unit area, than the flat collectors. To determine the performances of a cylindrical-parabolic concentrator, mathematical modeling of the heat balance on the absorber, the coolant, and the glass envelope was established using Matlab. The system of equations obtained is solved by the finite difference method. The results for a typical day are the variation in the temperature of the heat transfer fluid, the absorber tube, and the glass envelope. Thus, we examine the effect of the wind speed, flow rate on the temperature distribution of the coolant at the outlet. However, for a mass flow rate of the fluid of 0.1 kg / s, the outlet temperature of the fluid is 85 ° C with a thermal efficiency of 73%. Excluding the energy absorbed by the absorber tube is 75% of the solar intensity received on the reflector.*

Keywords: *Parabolic trough collector, Solar thermal energy, Simulation, Heat transfer, Solar concentrator.*

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I. Introduction

Renewable energies have experienced the first phase of development during the oil shocks of 1973 and 1978. Then a period of decline after the counter-shock of 1986, before regaining a new lease of life in 1998 following the signing of the Kyoto protocol, which predicts, in particular, a 5.2% drop in greenhouse gas emissions from wealthy countries over the period 2002-2012 compared to 1990 [1].

Solar energy can be used to generate power using concentrated solar systems. It is based on spherical, parabolic or Fresnel lens-based mirror concentrators [2], which have the principle of focusing the incident solar radiation in a point or a line. The conversion efficiency is high, and Temperature can easily surpass 500 °C. These technologies offer a natural alternative to the consumption of fossil resources with a low environmental impact and a high potential for cost reduction, as well as the possibility of hybridization of these facilities, by utilizing direct solar radiation, which is considered the primary resource and is very significant on a planetary scale [3]. When it comes to the specific challenges of electricity generation, transmission, and distribution, alternative sources, as opposed to conventional ones, can be a feasible answer. In areas where electrical energy delivery is unreliable, including alternative energy generation sources, such as solar, can be a viable choice [4].

Several studies have been conducted to anticipate, analyse, and estimate the performance of parabolic trough collectors under a variety of weather conditions and configurations. A comprehensive review is presented, that examines various models and simulation approaches of PTCs. This study constitutes a complementary application contribution to parabolic trough concentrators [5]. According to the literature a detailed heat transfer solar receiver model has been performed [6]. The proposed models are based on the detailed analysis of one-dimensional or two-dimensional heat transfer processes assumed by trough collectors. Other researchers [7] performed precise two-dimensional digital heat transfer of PTC using synthetic oil Therminol VP1 as heat transfer fluid (HTF). They came up with two software options based on the properties of the solar PTC they were using. The first was written in Matlab and was used to calculate the annual solar energy collected on the absorber pipe using several tracking modes that were available in the area. Because (EES) automatically detects all unknowns and groups of equations for the most efficient solutions, the second program code is written using a simultaneous equation solving software (EES) to evaluate the performance of the heat collector element (HCE) [8].

In the current approach, a simulation program will be developed under Matlab to deal with the different modes of heat transfer in the concentrator absorber by determining the evolution of the system's temperature. Different parameters are studied to explain their importance in the temperature variation. The results taken are for a sunny day of June 21 in the region of Tlemcen, north-western Algeria, and water as the heat transfer fluid.

II. Material and methods

The absorber was the essential element in the composition of the cylindrical-parabolic concentrator, whose role is to absorb the incident solar radiation for its conversion into heat before transmitted to the heat transfer fluid. We report in Figure1 the diagram of the process explaining the heat flow exchanged between components. However, the heat exchange coefficients are considered as known while considering the following assumptions:

- The transfer by conduction between the absorber and the glass is considered negligible.
- The flow of the incompressible fluid is in a one-dimensional direction.

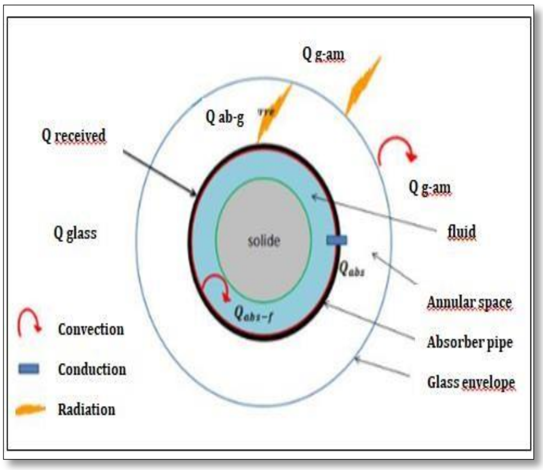

Figure 1. Different heat transfer modes

II.1. Incident radiation

Our study will focus on the equations of direct incident solar irradiance, as it is the resource of concentrating systems [9]. Modeling is done on a horizontal plane using two models. The data considered for the simulation in the Mediterranean countries provide from the site of Tlemcen city having the following coordinates Latitude 34.53 °, Altitude 806m, Longitude 1.33 ° [10]

Direct solar radiation is calculated according to the Capderou model, and the expression gives it: [11]

$$I_d = I_0 . s_0 . sinh_s . \exp(-T_L . m_a . \delta_R) \quad (1)$$

Where ε_0 represents the earth-sun distance correction, it is expressed as follows [12]:

$$s_0 = 1 + 0.034 * \cos\left(\frac{360}{365} * (N-2)\right) \quad (2)$$

With hs is the sun elevation angle, N is the number of days, and I_0 is the constant solar (1367W/m²)

The atmospheric mass m_a , as well as the Rayleigh thickness δR, is given by [11, 12]:

$$m_a = [\sin(h_s) + 9.4 * 10^{-4}(\sin(h_s) + 0.0678)^{-1.253}]^{-1} \quad (3)$$

$$\delta_R^{-1} = 6.6296 + (1.7513 \times m_a) - (0.1202 \times m_a^2) + (0.0065 \times m_a^3) - (0.00013 \times m_a^4) \quad (4)$$

Or the Linke factor $T_L$is given by the expression [13]:

$$T_L = T_0 + T_1 + T_2 \quad (5)$$

Where:

$$T_0 = (2.4 - 0.9 * sin\varphi) + 0.1 * (2 + sin\varphi) - (0.2 * z) - (1.22 + 0.14 * A_{he}) * (1 - sin(h_s)) \quad (6)$$

$$T_1 = 0.89^z \quad (7)$$

$$T_2 = [0.9 + (0.4 * A_{he})] * 0.63^z \quad (8)$$

With z is altitude, φ is latitude and A_{he} winter-summer alternation.

However, the expression of the energy absorbed by the absorber is given by the expression: [14]

$$q_{abs} = A_0 . I_d . \alpha_0 . \rho_0 . \gamma . K \qquad /A_0 = L * \mathrm{W} \tag{9}$$

The transmission-absorption coefficient $\alpha_0$is given according to the following expression [15]:

$$a_0 = \frac{a_{ab} . \tau_g}{1-(1-a_{ab}).(1-\tau_g)} \tag{10}$$

As for the modified angle of incidence is given by [16]:

$$K = 1 - 3.84.10^{-5}.(\theta) - 143.10^{-6}.(\theta)^2 \tag{11}$$

II.2. Heat balance

The incident solar energy absorbed is not entirely transmitted to the heat transfer fluid. Some are dissipated in the form of thermal losses between the absorber and the glass envelope. Figure 2 shows several heat transfer analyses between collector components and between the collector receiver and its surrounding environment.

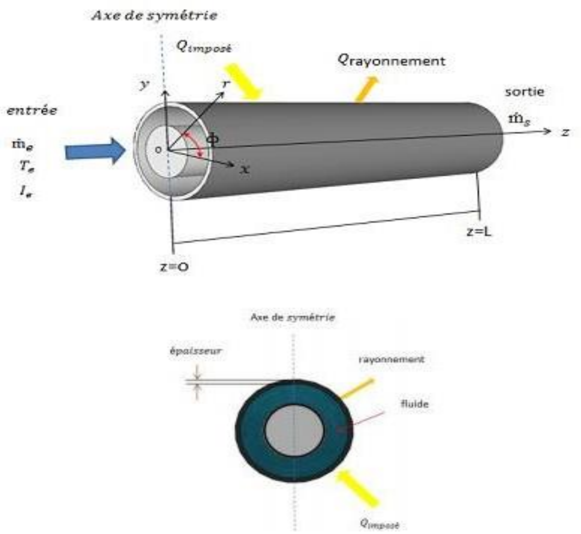

Figure 2. Heat balance of the absorber

- **For the glass envelope:**

$$\mathrm{A_g . \rho_g . Cp_g .} \frac{dT}{dt} = q_{abs.g} + q_{int} - q_{ext} \tag{12}$$

- **For the absorber pipe :**

$$\mathrm{A_{ab} . \rho_{ab} . Cp_{ab} .} \frac{dT}{dt} = q_{abs} - q_{int} - q_u \tag{13}$$

- **For the fluid:**

$$\mathrm{A_f . \rho_f . Cp_f .} \frac{dT}{dt} + \dot{m} .... \frac{dT}{dx} = q_u \tag{14}$$

Whose:

$$q_{abs.g} = A_0 . I_d . \alpha_g . \rho_0 . \gamma . K \tag{15}$$

The boundary condition: $T_{g,0} = T_{am}$; $T_{ab,0} = T_{a,e}$; $T_{f,0} = T_e$ (T_{am} is the ambient temperature).

II.3. The heat transfer coefficients

- *Heat transfer between glass envelope and atmosphere*

Convection and radiation are two transfer techniques for transferring heat from the glass enclosure to the atmosphere; convection requires wind.

The coefficient of radiation is given by [17]:

$$h_{r(ext)} = s_g . \sigma . ((T_{sky} + 273.15)^2 + (T_g + 273.15)^2) * (T_{sky} + T_g + 546.3) \tag{16}$$

With the expression of the sky temperature [18]:

$$T_{sky} = 0.0552 T_{am}^{1.5} \tag{17}$$

Then the coefficient of convection is given by [19]:

$$h_{c(ext)} = \left[0.6 + 3.87 * \left[\frac{Ra_a^{\ 1/6}}{\left(1+\left(\frac{0.559}{Pr_a}\right)^{9/16}\right)^{16/9}} \right] \right]^2 * \frac{a}{D_{ge}} \tag{18}$$

- *Heat transfer between the absorber and the glass envelope*

Convection and radiation are the two ways of heat transmission. The annulus pressure affects the convection mechanism. Temperature variations between the outer absorber surface and the inner glass surface cause radiation.

With the expression of the sky temperature [20]:

$$h_{r(in)} = s_{int} . \sigma . ((T_{ab} + 273.15)^2 + (T_g + 273.15)^2) * (T_{ab} + T_g + 546.3) \tag{19}$$

$$\text{With } s_{int} = \frac{1}{\left(\frac{1}{s_{ab}} + \frac{1-s_g}{s_g}\right) * \left(\frac{D_{abe}}{D_{gi}}\right)} \tag{20}$$

Then the coefficient of convection is given by :

$$h_{c(in)} = \frac{2\ {}_{eff}}{D_{abe} . ln\left(\frac{D_{gi}}{D_{abe}}\right)} \tag{21}$$

With:

$$\lambda_{eff} = 0.386 * \lambda_a * \left(\frac{Pr_a}{Pr_a + 0.61}\right)^{1/4} * (Ra_c)^{1/4} \tag{22}$$

$$Ra_c = \frac{\left(ln\left(\frac{D_{gi}}{D_{abe}}\right)\right)^4}{L_{eff}^3 * \left(D_{abe}^{-3/5} + D_{gi}^{-3/5}\right)^5} * Ra_{eff} \tag{23}$$

$$L_{eff} = \frac{D_{gi} - D_{abe}}{2} \tag{24}$$

$$Ra_{eff} = Gr_a * Pr_a \tag{25}$$

- *Heat transfer between the fluid and the absorber*

The flow type affects convective heat transfer from the inside surface of the absorber pipe to the fluid. We consider the case of turbulent flow [21] and is given by:

$$h_u = \frac{\lambda_f * Nu_f}{D_{abi}} \quad (26)$$

With: $$Nu_f = \frac{\left(\frac{f}{8}\right).(Re_f - 1000).\left(1+\left(\frac{D_{abi}}{L}\right)^{2/3}\right).Pr_f}{1+12.7.\sqrt{\frac{f}{8}}\left(Pr_f^{2/3}-1\right)} . \left(\frac{Pr_f}{Pr_{ab}}\right)^{0.11} \quad (27)$$

Knowing that:

$f = \left(1.84.\log(Re_f) - 1.64\right)^{-2}$ for $D_{abi} < L$ and;

$f = \left(1.8.\log(Re_f) - 1.5\right)^{-2}$ for $D_{abi} > L$,

The physical proprieties of fluid (water) are considering variants.

III. Solution procedure, Results and discussions

Matlab did the programming of these equations. We used the finite difference method to solve these equations. The process consists of giving a value of the variable and recalculating this variable with the equation, and finally, we compare the two deals. Depending on the desired precision, if the difference between the two calculated and proposed values is less than the fixed accuracy, this value is taken. Otherwise, the second value is taken, and the calculations are repeated until the difference between these two values becomes inferior to precision.

The characteristics of the solar PTC are mentioned in Table.1

Table 1. Characteristics of the simulated PTC

Absorber length (L)	**7.8m**
Collector width (W)	5m
Focal length (f)	1.84m
The absorber external diameter (D_{abe})	0.07m
The absorber internal diameter of (D_{abi})	0.066m
The glass external diameter (D_{ge})	0.115m
The glass internal diameter (D_{gi})	0.109m
Thermal conductivity of the absorber (λ_{ab})	54W/mK
Thermal conductivity of the glass (λ_v)	1.2W/mK
Absorption of the absorber (α_{ab})	0.906
Glass transmissivity (τ_g)	0.95
Transmissivity-absorbance factor (α_0)	0.864
Emissivity of glass (ε_g)	0.86
Reflection of reflector (ρ_0)	0.93
Interception factor (γ)	0.92

Following the use of the Capderou model for the calculation of the solar intensity and based on a solar tracking system for our solar system, we notice that the solar intensity at the level of the Tlemcen region and for a day sunny, reaches 900W / m²as shown in Figure 3. Moreover, the solar power absorbed by the absorber tube reached 680W / m², Figure 4, which explains the existence of heat losses between the glass envelope and the absorber tube. These losses cause a decrease in the rate of energy absorbed, which is of the order of 75%.

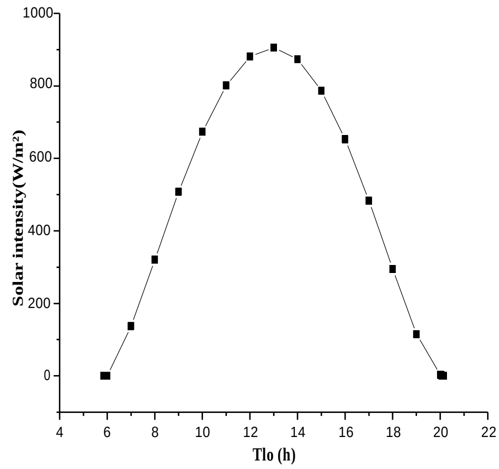

Figure 3. Variation of the solar intensity

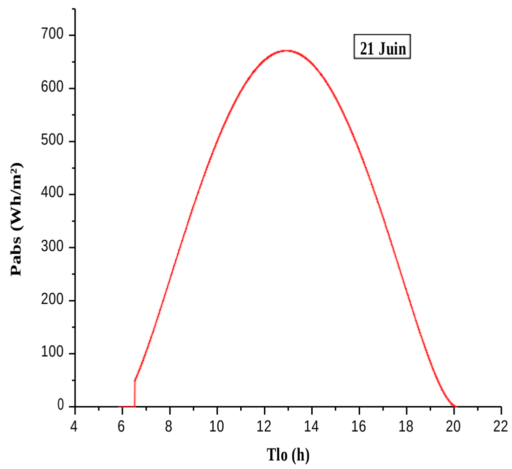

Figure 4. Variation of the absorbed energy

Moreover, Figure 5 shows the variation in the outlet temperatures of the fluid, absorber and glass envelope. It is noted that the fluid outlet temperature is 55 ° C for a mass flow rate of 0.2Kg / s. On the other hand, according to Figure 6, it is deduced that the outlet temperature of the fluid varies as a function of the mass flow rate or the temperature increases by decreasing the latter.

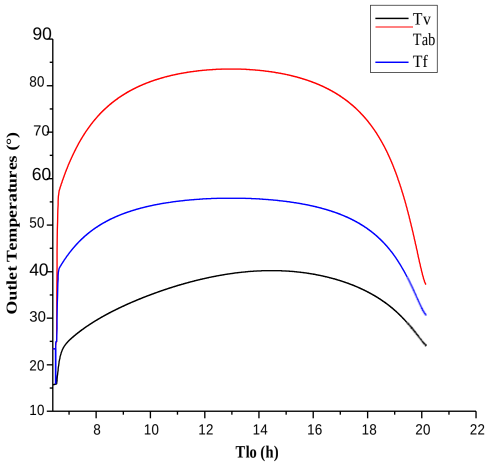

Figure 5. Variation of the outlet Temperature

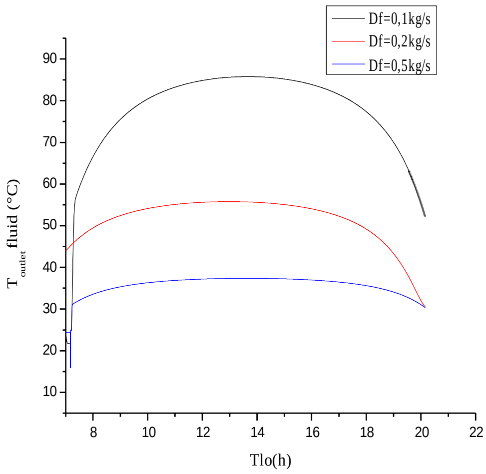

Figure 6. Variation of the outlet Fluid Temperature for different mass flow rate

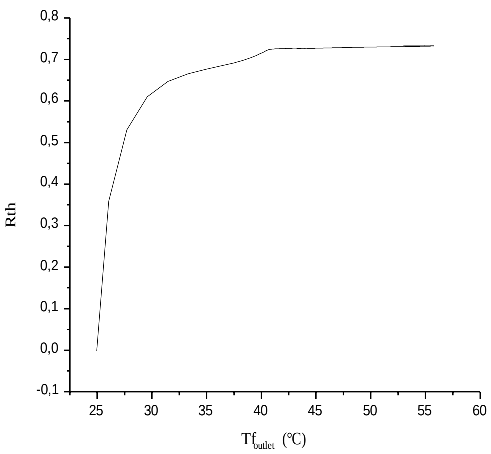

Figure 7. Variation of collector thermal efficiency compared to the outlet temperature of the fluid

In what follows Figure 7, shows the variation of the thermal efficiency as a function of the temperature of the collector which is equal to 73%. This efficiency helps to control the reliability of our system since it is the ratio between the useful flux and the power absorbed through the absorber tube. We note that the thermal efficiency is close to the experimental, whose value is 73.68% in the air in the annular space and validated by SNL. This difference only has to let us think of the existence of thermal losses.

IV. Conclusion

Our study relates to the study of the various existing heat transfers for a PTC sensor. For this, a mathematical simulation in Matlab language is carried out to solve the nonlinear equations. Our results are based on the study of outlet temperatures and influencing parameters.

Among those proposed the mass flow rate of the fluid shows an important parameter introducing into the variation of the outlet temperature or we have seen that the water outlet temperature reaches a value of 85 ° C for a flow rate of 0.1 kg / s. however, the energy absorbed is 75% relative to the solar intensity received on the reflector. Concerning the thermal efficiency which explains the rate of the useful flux linked to the heat transfer fluid, it is 73% for our system under the conditions of the Tlemcen region for a sunny day. In contrast, the wind speed turned out to be negligible.

In addition, the sun tracking system is essential and necessary for the operation of the PTC as it only uses direct solar radiation.

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