Chapter 12
Numerical and Experimental Study of Parabolic Trough Solar Collector

Belkacem Agagna and Arezki Smaili

Abstract The Solar power plants with parabolic trough collectors (PTCs) represent more than 92% of the solar thermal power plants currently in operation. It is also the most suitable technology for most plants under construction and planned [1]. The present study evaluates the thermal performance of LS-2 parabolic trough collector of Solar Electric Generating System (SEGS VI) plant. A thermal model is developed for the solar receiver based on comprehensive energy analysis. The model has been validated with the experimental data from Sandia National Laboratory. Besides, the predictions from the model were compared with the published modeling studies (both on sun and off sun tests have been considered). In all cases, the results of the developed model show a good agreement with the experimental tests and the prediction results of the other models. Particular interest has been given for the impact of the glass cover diameter on the efficiency of the solar receiver.

Keywords Parabolic trough solar collector · Solar receiver · Heat transfer analysis · Modeling

12.1 Introduction

The solar receiver is the main component in the PTC, it absorbs the incident solar radiation and converts it into useful heat. The improvement of the performance of this component requires an accurate analysis of the thermal losses. This can be done through a comprehensive heat transfer analysis. In this work, we are interested in the solar collector of the power plant (SEGS VI) [2, 3]. The collector used in this plant (LS-2) has been tested in SNL laboratories (Sandia National Laboratories), Albuquerque (New Mexico) site. The results of the experiments were published by

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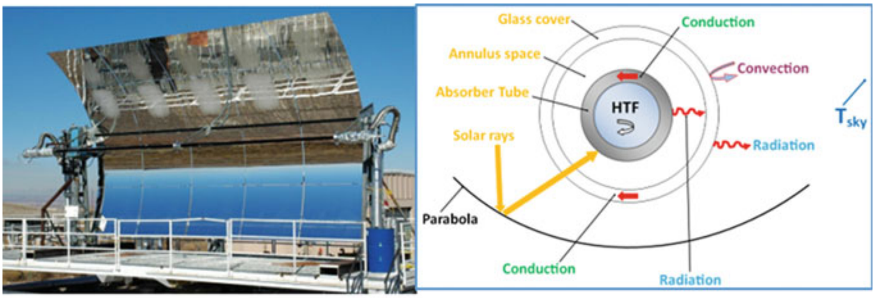
Dudley [2]. Forristall [3] developed a 1D model implemented in EES software (Engineering Equation Solver) and validated it through a detailed comparison with SNL experimental data. García-Valladares et al. [4] have developed a numerical model for a single-pass receiver. They are also proposed a double pass type absorber. Cheng et al. [5] combined the optical analysis with CFD to simulate the complex heat transfer problem of a PTC. Kalogirou [6] has established the energy balance of the receiver and developed a model written in SEA language. The model has been validated on the basis of Dudley's experiments [2].

The present paper presents comprehensive modeling of the solar receiver. It is organized in two parts. The first part present the model and its validation by comparison with experimental data from Dudley [2]. In the second part, a comparative study with previous work is presented, with a focus on the impact of variation in the diameter of the glass cover on system performance.

12.2 Mathematical Modeling

The absorber tube is often coated with a selective layer and surrounded by a transparent glass envelope. It is placed on the focal line of the PTC, as shown in Fig. 12.1a. The incident solar energy absorbed, is not entirely transmitted to the heat transfer fluid because a part is dissipated in the form of heat losses. Figure 12.1b illustrates the heat transfer in LS2 solar receiver. To analysis the heat transfer, we have developed a model, which allows energy analysis of the parabolic trough concentrator and the prediction of the thermal performance.



(a) (b)

Fig. 12.1 LS2 PTC tested at SNL (left). Heat transfer model of the PTC (right)

12.2.1 Hypothesis

When calculating the energy balance for a turbulent regime, the following assumptions have been taken into account: (i) The regime is permanent; (ii) The thicknesses of the glass and the absorber are negligible [2, 4, 6]; (iii) The solar flux at the absorber is uniform; (iv) For a vacuum annulus, convective exchanges are negligible.

12.2.2 Thermal Model

Convection Heat Transfer Between the HTF and the Absorber Tube. The convection heat transfer from the receiver tube to the HTF can be given by:

$$q_{abs} = h_i.\pi D_{r,i}\left(T_r - T_f\right).L \tag{12.1}$$

With:

$$h_i = \frac{Nu.k_f}{D_{r,i}} \tag{12.2}$$

where h_i is the HTF convection heat transfer and is evaluated as a function of the Nusselt number N_u, and k_f represents the thermal conductance of the HTF [6].

Thermal Radiation Heat Transfer in the Annulus Space. The radiation heat transfer between the receiver tube and glass envelope is estimated:

$$q_{loss\,r,an} = h_{r,an}.A_{r,o}.\left(T_r - T_g\right) \tag{12.3}$$

With

$$h_{r,an} = \frac{\sigma\left(T_r^2 + T_g^2\right)\left(T_r + T_g\right)}{\frac{1}{\varepsilon_r} + \left(\frac{A_{r,o}}{A_{gl,i}}\right)\left(\frac{1}{\varepsilon_g} - 1\right)} \tag{12.4}$$

where, Tg is the glass envelope temperature and $h_{r,an}$ is the radiative heat transfer coefficient between the receiver tube and the glass envelope.

With: ε_r, ε_g and σ are the emissivity of the receiver, the emissivity of the glass envelope and the Stéfan-Boltzmann constant, respectively. $A_{gl,i}$, $A_{r,o}$ are the inner surface of the glass cover (m^2), the outer surface of the absorber tube (m^2), respectively.

Convection Heat Transfer from the Glass-Cover to the Ambient. The convection heat transfer from the glass envelope to the ambient is the largest source of heat loss, especially if there is wind. From Newton's law of cooling [6]:

$$q_{loss\,conv,ext} = h_w.A_{gl,o}.\left(T_g - T_a\right) \tag{12.5}$$

With

$$h_w = (Nu.k)/D_{gl,o} \tag{12.6}$$

Nusselt number can be estimated as function of Reynolds number [3, 6].

Radiation Heat Transfer from Glass Cover to the Sky. The net radiation transfer between the glass and the sky can be given by [3, 6]

$$q_{loss\,r,ext} = h_{rca}.A_{gl,o}.\left(T_g - T_a\right) \tag{12.7}$$

With

$$h_{rca} = \sigma\varepsilon_g\left(T_g + T_a\right)\left(T_g^2 + T_a^2\right) \tag{12.8}$$

12.3 Results and Discussion

12.3.1 Estimation of the Thermal Efficiency

Table 12.1 illustrates the operating conditions [2] and the results obtained.

Table 12.1 Operating conditions and obtained results

Case	Gb (W/m^2)	Wind (m/s)	Flow (l/mn)	T_{amb} (°C)	T_{in} (°C) HTF	Delta Air (°C)	η_{Dudley} (%) Exp [2]
1	933.7	2.6	47.7	21.2	102.2	91.9	72.51
2	968.2	3.7	47.8	22.4	151.0	139.8	70.90
3	982.3	2.5	49.1	24.3	197.5	184.3	70.17
4	909.5	3.3	54.7	26.2	250.7	233.9	70.25
5	937.9	1.0	55.5	28.8	297.8	278.6	67.98
6	880.6	2.9	55.6	27.5	299.0	280.7	68.92
7	903.2	4.2	56.3	31.1	355.9	334.1	63.82
8	920.9	2.6	56.8	29.5	379.5	359.4	62.34

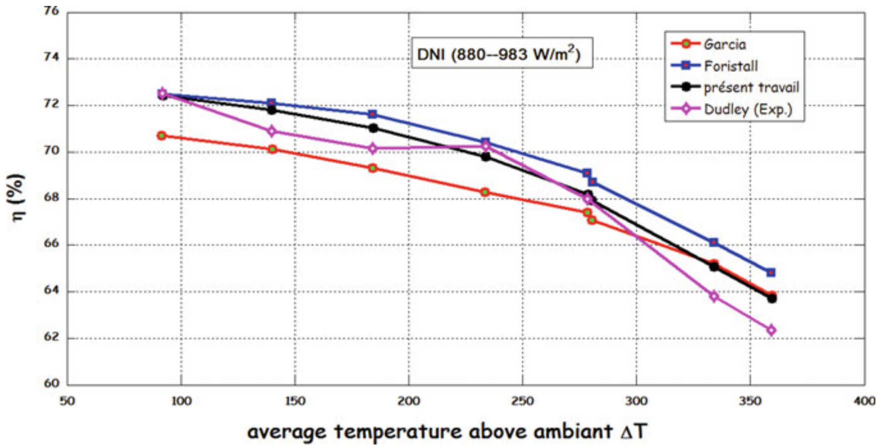


Fig. 12.2 Thermal efficiency predictions of the PTC vs. ΔT

From Fig. 12.2, it is noted that the efficiency of the system is maximum where the temperature difference ΔT is low. Besides, the efficiency of the solar concentrator decreases remarkably with the increase of ΔT. This is because heat losses increase with the increase in the operating temperature.

12.3.2 Estimation of Heat Losses

Table 12.2 shows the operating conditions [2] with solar irradiation of zero W/m^2. The results show an increase in heat losses with the rise of the temperature difference ΔT. High operating temperature increases exponentially the radiation heat losses to the ambient, which explains clearly the decrease of the yield in the preceding cases.

Table 12.2 Operating conditions and obtained results

Case	Wind (m/s)	Flow (l/mn)	T_{amb} (°C)	T_{in} (°C)	T_{out} (°C) Exp	q''_{Dudley} (W/m^2) Exp [2]	$q''_{present\ model}$ (W/m^2)	q''_{Garcia} (W/m^2) [4]
1	3.2	27.4	26.3	99.55	99.54	0.3	3.75	3.81
2	2.9	27.4	25.4	100	99.97	0.85	3.82	3.91
3	1.1	53.6	19.9	153.4	153.3	5.3	8.55	8.68
4	0.1	54.7	22.5	199.4	199.0	14.04	13.71	13.57
5	1.5	55.6	24.2	253.8	229.2	23.4	22.91	23.04
6	2.0	56.0	26.7	299.0	297.9	36.7	32.68	32.6
7	0.6	56.8	27.6	348.3	319.9	55.8	45.57	45.65

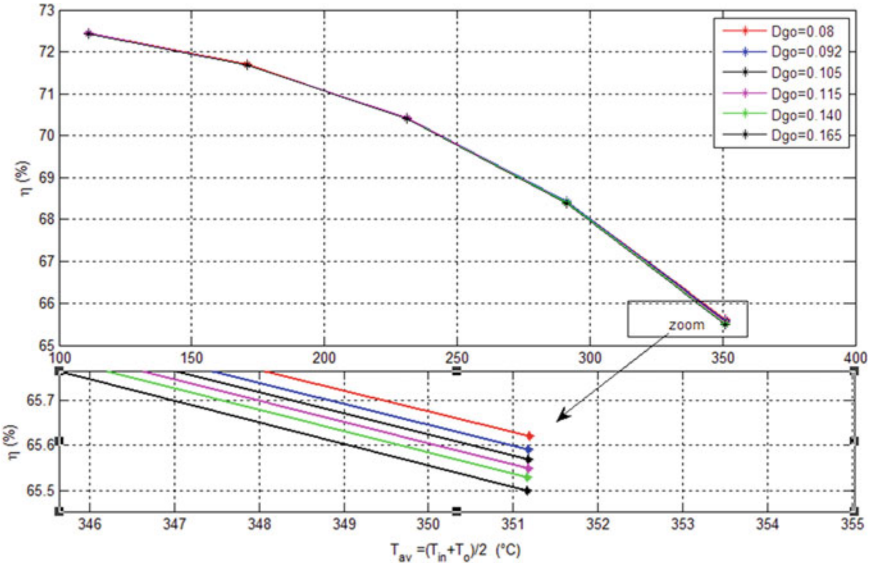


Fig. 12.3 Variation of the efficiency as a function of ΔT for different cover glass diameters

12.3.3 *The Sensitivity of the Performance to the Diameter of the Glass Cover*

The operating conditions including constant solar irradiation of 933.7 W/m^2 [2], are summarized with the following parameters: $V_w = 2.6$ m/s, V = 47.7 l/mn and $T_{air} = 21.2$ °C. For all the considered cases, a thickness of 0.003 m is used. Figure 12.3 illustrates the variation of the efficiency as a function of ΔT. It is clear that the efficiency decreases with the increase of the temperature difference. It has been observed that the efficiency decreases slightly with increasing the outer diameter of the envelope.

Figure 12.4 shows the variation of the thermal losses as a function of ΔT for different cover glass diameters. The increase in the outer diameter induces a slight increase in thermal losses. It can be seen from Figs. 12.3 and 12.4 that the variation of the outer diameters of the glass envelope does not have much effect on the performance of the solar receiver.

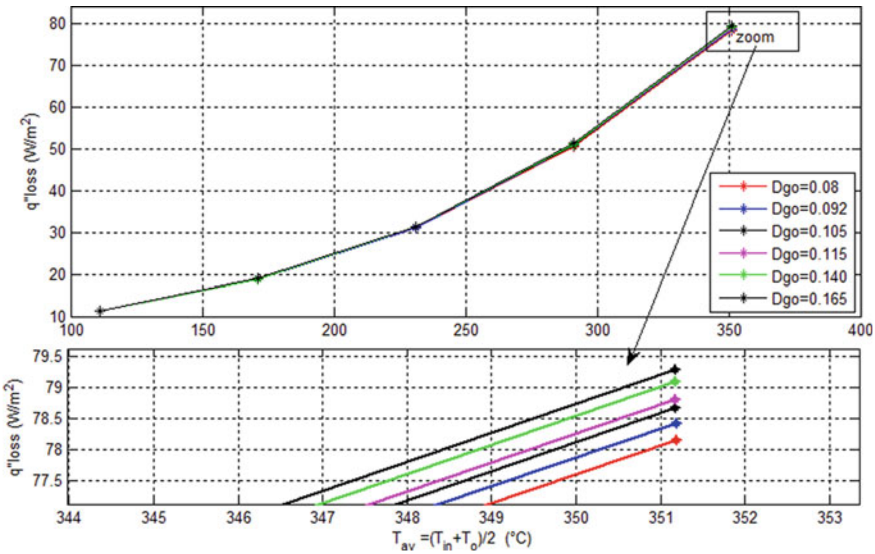


Fig. 12.4 Variation of the thermal losses as a function of ΔT for different cover glass diameters

12.4 Conclusion

The present study focused on the heat transfer analysis of the receiver of PTC. A simplified one-dimensional model has been established and used to analyze the thermal performance. The predictions of the model have been successively compared with the experimental results of Dudley et al. [2]. Besides, good agreement has been observed when comparing the predictions with the previously published studies. The sensitivity analysis indicates that the increase in the outer diameter of the glass cover does not influence the performance of the PTC.

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