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Analytic modeling of parabolic trough solar thermal power plants

Germán A. Salazar [a, b], Naum Fraidenraich [a], Carlos Antonio Alves de Oliveira [c], Olga de Castro Vilela [a], Marcos Hongn [b], Jeffrey M. Gordon [d, *]

[a] Departamento de Energia Nuclear, Grupo de Pesquisas em Fontes Alternativas de Energia, Universidade Federal de Pernambuco, Av. Prof. Luiz Freire, 1000-DCU, CEP 50740-540, Recife, PE, Brazil
[b] Instituto de Investigaciones en Energias No Convencionales, Av. Bolivia 5150, A4408FVY, Salta, Argentina
[c] Instituto Federal de Alagoas, Campus Palmeira dos Índios, Av. Alagoas, Palmeira de Fora, 57608-180, Palmeira dos Índios, AL, Brazil
[d] Department of Solar Energy and Environmental Physics, Blaustein Institutes for Desert Research, Ben-Gurion University of the Negev, Sede Boqer Campus, 84990, Israel

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A B S T R A C T

We derive, evaluate and validate comprehensive analytic modeling of the energy flows in parabolic trough solar thermal power plants. The analytic formulae are straightforward to implement and evaluate, relating to the heat transfer within and from the solar concentrators (including transients, mainly overnight heat losses), and the impact of solar field operation on turbine power and efficiency. Prior numerical simulations used to design solar thermal power systems have either been proprietary or devoid of a fully-reported source code - hence inaccessible or problematic for widespread use. Also, the dependence of these simulations on extensive numerical procedures does not provide a transparent physical picture that grants a clear understanding of how component and system performance vary with the principal operating and input variables - a drawback overcome by the analytic approach presented here. Published experimental measurements of sufficient extent to permit meaningful comparisons between theory and experiment for such solar thermal power plants are exceptionally limited. This narrow data base is used for model validation on both a monthly and an hourly basis. The analytic model is then applied to evaluating a solar power plant now being planned for northeast Brazil, also highlighting the energy-delivery advantages of low-latitude locations.

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1. Introduction

Of the roughly 5 GW of operational commercial solar thermal power plants, about 85% comprise single-axis-tracking parabolic trough concentrators driving steam turbines [1] (Fig. 1). Almost all systems use an oil coolant pumped through (a) evacuated receiver tubes at a variable flow rate that maintains a fixed collector outlet temperature, followed by (b) a heat exchanger the secondary of which feeds the power block. Gas-fired backup operates in parallel with solar so as to ensure the turbine receives a prescribed constant flow rate of fixed-temperature steam, thereby achieving uninterrupted electricity production independent of solar intermittency. The overwhelming majority of these installations do not employ thermal storage [1,2].

Systems installed to date have been designed with large-scale numerical computer simulations that compute all energy flows as a function of the meteorological and operating variables. Most of these simulations are proprietary or lacking a full source code - hence problematic to implement [2]. Other simulations are based on cumbersome encoded and unalterable numerical procedures that preclude a clear physical understanding of how each facet of system performance varies with the input and operational variables [3–8]. Moreover, in addition to their complexity, most available simulations are rather focused on specific systems.

Ongoing research in this area covers the modeling, design and evaluation of system components - as well as overall system performance assessments - performed with the types of complex numerical procedures noted above. Recent papers have addressed pivotal issues that include (1) collector and turbine properties, sizing, and control strategies [9–14], (2) how short-term and long-term solar availability affects system yield [15], (3) the influence of thermal storage [14,15], and even (4) how artificial neural networks can facilitate system design and evaluation [16].

* Corresponding author.
E-mail address: jeff@bgu.ac.il (J.M. Gordon).

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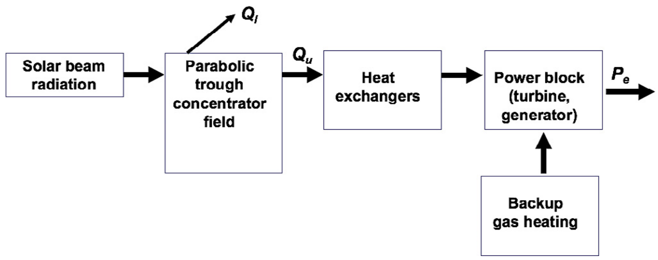

Fig. 1. Schematic energy flow diagram for a solar thermal power plant generating AC electrical power P_e, with parallel gas-fired backup ensuring a specified steam flow rate to the turbine. Q_u is useful thermal power delivery. Q_l is heat loss to ambient. P_e is net electricity generation.

Comparing model predictions against field performance turns out to have considerable limitations. *Published* experimental measurements of both solar beam irradiance and detailed solar thermal power plant performance (including separate monitoring of the collector field and the turbine block) appear to be surprisingly uncommon. To our knowledge, verifiable monthly-average figures for all required measurements are restricted to a single year. Additionally, hour-by-hour data are limited to a single clear day. Both come from one particular large-scale installation in Kramer Junction, CA, US [2,17]. These data provide the basis for our comparisons of theory versus experiment.

As for analytic modeling of system performance, a first step was taken in Ref. [18], but (a) did not relate to transients (most notably collector nighttime cool-down losses), (b) did not account for the sizable gap between instantaneous and long-term performance, and (c) did not accommodate different collector flow strategies.

In overcoming these limitations, the analytic model developed here (1) offers physical transparency for the main processes impacting collector and turbine performance (as opposed to the modeling being tacitly embedded within complicated numerical simulation procedures, *vide infra*), (2) can readily be applied to a variety of solar thermal power systems, and (3) is amenable to evaluation by any user via straightforward calculations.

The rest of the paper is organized as follows. Section 2 presents the general method with full details on collector, system and turbine modeling. This includes modeling the optical and thermal energy flows of the solar collector field, how the state of the steam produced by the solar field impacts turbine performance, and collector nighttime cool-down losses. Section 3 comprises the results and discussion: a case study with comparisons of theory vs. experiment for both solar collector efficiency and overall system electrical efficiency, plus the design of a solar thermal power plant currently being planned. Section 4 summarizes our conclusions, emphasizing the new added value of the analytic modeling for analyzing and designing solar thermal power plants.

2. General method and modeling

2.1. Modeling collector optical gains

The solar power absorbed by the collectors per absorber area, Q_{abs}, is

$$Q_{abs} = I_{bn}\cos(\theta)\,FS\,K(\theta)\,\eta_o\,C_g = I_{coll}\,C_g \tag{1}$$

where I_{bn} is the incident *normal* solar beam irradiance (per aperture area), θ is the solar incidence angle on the aperture [19], *FS* is the unshaded fraction of collector aperture (a function of solar geometry and field layout, for which the closed-form expression is presented in Ref. [20]), and I_{coll} denotes the *collected* solar irradiance per aperture area. In contrast, "collectible" radiation refers just to the product $I_{bn}\cos(\theta)$, which is the solar beam irradiance incident on the aperture. Geometric concentration ratio C_g is typically ~20–25 (defined as the ratio of collector aperture area A_{coll} to absorber area A_{abs}, with A_{abs} relating to absorber tube circumference W times its length L). η_o is the collector optical efficiency at normal incidence, comprising the product of mirror reflectivity, receiver tube transmittance, receiver coating absorptance, and receiver intercept factor [19]. $K(\theta)$ is the incidence angle modifier [19], which measures how optical gains vary with θ relative to their value at normal incidence ($K(0) \equiv 1$). $K(\theta)$ subsumes how the transmission of the glazing, the absorption of the absorber coating, and the width of the sun's image projected onto the receiver tube by the concentrator vary with incidence angle.

2.2. Modeling collector heat transfer

The instantaneous useful thermal power delivery per unit length (in W/m), as a function of position x along the collectors, $Q_u(x)$ ($0 \le x \le L$ from entry to exit) is proportional to the difference between the local absorber temperature $T_{abs}(x)$ and fluid temperature $T_f(x)$:

$$Q_u(x) = Wh\left(T_{abs}(x) - T_f(x)\right). \tag{2a}$$

The convective heat transfer coefficient h between the coolant and the absorber can be obtained from the dimensionless Nusselt number Nu

$$h = \frac{k}{D_i}Nu \tag{2b}$$

where k is the thermal conductivity of the fluid, and D_i is the internal diameter of the absorber tube ($D_i = 0.066$ m for the example considered here, with the tube outer diameter being 0.070 m). For fully-developed turbulent flow (the dimensionless Reynolds number $Re > 10{,}000$) in long tubes (length/diameter > 60), and for the dimensionless Prandtl number Pr in the range $0.7 < Pr < 160$, Nu is given by the Dittus-Boelter correlation [21].

$$Nu = 0.023Pr^{0.4}Re^{0.8}. \tag{2c}$$

$\mathrm{Pr} = \nu/\alpha = \mu/(\rho\,\alpha)$, where ν is the kinematic viscosity of the fluid, μ is the dynamic viscosity, and ρ is the density. α is the thermal diffusivity $\alpha = k/(\rho\,C_p)$ where C_p is the specific heat of the fluid. $\mathrm{Re} = v\,D_i/\nu$, where v is the linear flow velocity. For example, during solar power generation, and for the physical properties of the collector fluid [22], one obtains $\mathrm{Pr} = 5.5$ and $\mathrm{Re} = 4.0\cdot10^5$, yielding $h = 1.8\cdot10^3$ W/(K-m$^2_{\text{inner tube}}$). In all our equations for collector energy balance, h is expressed per unit area of *outer* tube, i.e., modified by the ratio of tube inner-to-outer area, $(0.066/0.070)^2 = 0.889$.

Equation (2) relates to heat transfer *within* the collector. For heat transfer *from* the collector *to* the environment, we base the analysis on experimental measurements from the evacuated selectively-coated receiver tube considered here [23]. The data reported in Ref. [23] also included measurements for non-evacuated, air-filled, and unglazed receiver tubes, as well as for a different selective coating, but the relevant results here are those pertaining to the receiver tubes installed in the solar field for which system performance data were available.

A simulation model for evaluating the heat losses for several configurations of glazed receiver tubes was derived in Ref. [8]. The results were empirically correlated as third-order polynomials in the average temperature of the collector coolant [24], and

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