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A new approach for the prediction of thermal efficiency in solar receivers

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ABSTRACT

Optimization of solar concentration receiver designs requires of models that characterize thermal balance at receiver wall. This problem depends on external heat transfer coefficients that are a function of the third power of the temperature at the absorber wall. This nonlinearity introduces a difficulty in obtaining analytical solutions for the balance differential equations. So, nowadays, several approximations consider these heat transfer coefficients as a constant or suggest a linear dependence. These hypotheses suppose an important limitation for their application.

This paper describes a new approach that allows the use of an analytical expression obtained from the heat balance differential equation. Two simplifications based on this model can be made in order to obtain other much simpler equations that adequately characterize collector performance for the majority of solar technologies. These new equations allow the explicit calculation of the efficiency as a function of some characteristic parameters of the receiver. This explicit calculation introduces some advantages in the receiver optimization process because iteration processes are avoided during the calculations. Validation of the proposed models was made by the use of the experimental measurements reported by Sandia National Laboratories (SNL) for the trough collector design LS-2.

1. Introduction

Several numerical models have been developed in order to simulate the thermal performance of solar concentrating systems. Some examples are exposed for the main Concentrating Solar Power (CSP) systems.

Parabolic Troughs Collectors (PTC) is the most mature technology of CSP systems; so many thermal models can be found in the technical literature for these collectors. The model developed by Forristall [1] is considered as a reference in the field. It is based in a two dimensional energy balance in the receiver, using state of the art correlations for the heat transfer coefficients. Results of this model were validated satisfactorily with experimental results. Some authors used this model as basis for their own models, as for example Montes et al. [2] who extended its application to Direct Steam Generation (DSG) inside the receiver. Kalogirou [3] followed a similar strategy, performing a detailed thermal model which was solved in Engineering Equation Solver (EES), like the Forristall's one.

Some authors have also enhanced this reference model. Thus Yilmaz and Soylemez [4] focused on radiation losses from the glass envelope and conduction losses trough support brackets, performing a more detailed analysis for these heat transfer modes. Padilla et al. [5] discretized axially the absorber tube and the envelope considering heat conduction in axial direction and estimating the view factors between nodes for radiation heat transfer. Hachicha et al. [6] broadened this discretization to the azimuthal direction in order to consider non-uniformities on heat absorbed at the receiver.

On the other hand, detailed numerical simulations for the thermal and fluid-dynamic behavior, considering Navier Stokes Equations, were carried out both in 1-D [7] and 3-D [8].

Some of the models developed for Linear Fresnel Reflector (LFR) are also based on the Forristall's one for PTC. Thus, Heimsath et al. [9] adapted the energy balance for a single-tube receiver. It is important to note that while in PTC heat transfer coefficients can be characterized by existing correlations, as the involved geometries are widely studied, in LFR systems these geometries are more complex and there are no correlations. At this regard, Dey [10] and Pye [11] used Finite Element Analysis (FEA) to calculate heat conduction in absorber plates and Computational Fluid Dynamics (CFD) to estimate heat transfer by convection inside the cavity of multi-tube receivers. Reynolds [12] also validated qualitatively those CFD calculations with experimental data, finding significant deviations. On the other hand, other studies [13] are based on an

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Nomenclature

A area (m^2)
C concentration factor (–)
c_p specific heat capacity at constant pressure ($J\,kg^{-1}\,K^{-1}$)
D diameter (m)
f_0, f_1, f_2, f_3, f_4 dimensionless factors of the model (–)
F′ dimensionless coefficient of the model, which represents a rate between heat transfer coefficients (–)
F'_v dimensionless coefficient of the model defined by Fraidenraich et al. [19] (–)
g(Z) characteristic function of the model (–)
h convective heat transfer coefficient ($W/m^2\,K^{-1}$)
k conductivity ($W/m\,K^{-1}$)
L receiver length (m)
$\dot{m}$ thermal fluid mass flow ($kg\,s^{-1}$)
n number of collector support brackets (–)
P_b perimeter of the support section (m)
Pr Prandtl number (–)
Q fluid flow
Re Reynolds number (–)
T temperature (K)
$\overline{T^3_{ro,ext}}$ average value of the third order polynomial of receiver outer wall and external temperatures (K^3)
$\dot{q}''$ power per unit area (W/m^2)
$\dot{q}'$ power per unit length of the collector (W/m)
$\dot{q}''_{crit}$ critical heat flux (assuming wall temperature equal to fluid one) corresponding to null efficiency (W/m^2)
U heat transfer coefficient ($W/m^2\,K^{-1}$)
U_{crit} critical internal heat transfer coefficient corresponding to half of maximum efficiency ($W/m^2\,K^{-1}$)
U_{ext} external heat transfer coefficient ($W/m^2\,K^{-1}$)
U_t global heat transfer coefficient (from the fluid to the ambient) considering null radiation focused on the receiver ($W/m^2\,K^{-1}$)
U′ heat loss function coefficient as defined by Fraidenraich et al. [19] ($W/m^2\,K^{-1}$)
W perimeter (m)
x longitudinal coordinate (m)
x* dimensionless longitudinal coordinate (–)
Z dimensionless variable of the model (–)

Greek symbols
α radiation absorptivity
τ radiation transmissivity
ρ radiation reflectivity
γ interception factor
Δ Increment
ε_{ext} outer surface emissivity (–)
η cumulative thermal efficiency (–)
η′ cumulative thermal efficiency derivative (m^{-1})
η_x, η_{x^*} local thermal efficiency for x dimension or dimensionless length (–)
η_0 local thermal efficiency at tube inlet (–)
η_t cumulative thermal efficiency (–)
σ Boltzmann constant ($W/m^2\,K^{-4}$)

Subscripts
abs absorbed at the surface
ac annular convection
ar annular radiation
b support bracket
base connection region of the support brackets with the absorber
crit critical running conditions
cs, b cross section of the support brackets connection arms
ec external convection
er external radiation
ext external conditions
fe inlet flow conditions
g glass
gi glass inner surface
go glass outer surface
int internal
loss loss
opt optical
rec receiver
ri receiver inner surface
ro receiver outer surface
s stagnation conditions (null efficiency at high fluid temperatures)
u useful
Tot total

Acronyms
AZTRACK AZimutal TRACKing. Experimental facilities at SANDIA Laboratories
CFD Computational Fluid Dynamics
CRS Central Receiver Systems
CSP Concentrating Solar Power
DSG Direct Steam Generation
DNI Direct Normal Irradiation (W/m^2)
EES Engineering Equation Solver
FEA Finite Element Analysis
HCE Heat Collection Element
HTF Heat Transfer Fluid
LFR Linear Fresnel Reflectors
NTU Number of Transfer Units (–)
PTC Parabolic Trough Collector
SNL Sandia National Laboratories

experimental sensitivity analysis to find estimations for the overall heat loss coefficient.

Finally, numerical models for Central Receiver Systems (CRS) involve the characterization of heat transfer coefficients in complex situations, due to non-elementary geometries and boundary conditions. Montes [14] used a combination of analytical correlations based on experimental data and Li et al. [15] simplified the problem using correlations derived for other technologies and/or applications.

Almost all the mentioned works are based in numerical approaches, but some other authors develop analytical studies. Analytical thermal models are based on the energy balance equation. In this case, heat radiation introduces a non-linear behavior which makes it difficult to obtain an expression for the solution. The most important approximations found in the bibliography are summarized below.

Hottel and Whillier [16] and Bliss [17] obtained a model assuming that the heat loss coefficient does not depend on temperature and, therefore, it is considered as a constant along the tube, yielding to the following expression for the efficiency:

$$\eta(x) = \left[1 - \frac{U_{ext}(T_{fe} - T_{ext})}{\dot{q}''_{abs}}\right]\dot{m}C_p/WxU_{ext}(1 - e^{-F'WxU_{ext}/\dot{m}C_p}) \quad (1)$$