Applied Energy 106 (2013) 287–300

Contents lists available at SciVerse ScienceDirect

Applied Energy

journal homepage: www.elsevier.com/locate/apenergy

Modeling and co-simulation of a parabolic trough solar plant for industrial process heat

R. Silva [a,*], M. Pérez [a], A. Fernández-Garcia [b]

[a] CIESOL Centro Mixto Universidad de Almería – CIEMAT, Almería 04120, Spain
[b] CIEMAT – Plataforma Solar de Almería, Ctra. Senés, km 4, E04200 Tabernas, Almería, Spain

HIGHLIGHTS

- A tri-dimensional dynamic complex model of a parabolic-trough collector is proposed.
- The collector model was validated with experimental data and agrees very well.
- An innovative co-simulation integration environment for PTC plants is developed.
- Co-simulations with complex dynamic and simplified stationary models were compared.
- Co-simulations for a reference solar industrial process heat scenario are presented.

ARTICLE INFO

Article history:
Received 24 June 2012
Received in revised form 14 January 2013
Accepted 16 January 2013

Keywords:
Parabolic-trough collectors
Process heat
Co-simulation
TISC
Modelica
TRNSYS

ABSTRACT

In the present paper a tri-dimensional non-linear dynamic thermohydraulic model of a parabolic trough collector was developed in the high-level acausal object-oriented language Modelica and coupled to a solar industrial process heat plant modeled in TRNSYS. The integration is performed in an innovative co-simulation environment based on the TLK interconnect software connector middleware. A discrete Monte Carlo ray-tracing model was developed in SolTrace to compute the solar radiation heterogeneous local concentration ratio in the parabolic trough collector absorber outer surface. The obtained results show that the efficiency predicted by the model agrees well with experimental data with a root mean square error of 1.2%. The dynamic performance was validated with experimental data from the Acurex solar field, located at the Plataforma Solar de Almeria, South-East Spain, and presents a good agreement. An optimization of the IST collector mass flow rate was performed based on the minimization of an energy loss cost function showing an optimal mass flow rate of 0.22 kg/s m^2. A parametric analysis showed the influence on collector efficiency of several design properties, such as the absorber emittance and absorptance. Different parabolic trough solar field model structures were compared showing that, from a thermal point of view, the one-dimensional model performs close to the bi-dimensional. Co-simulations conducted on a reference industrial process heat scenario on a South European climate show an annual solar fraction of 67% for a solar plant consisting on a solar field of 1000 m^2, with thermal energy storage, coupled to a continuous industrial thermal demand of 100 kW.

1. Introduction

Co-simulation is an innovative simulation concept that consists on coupling distributed parallel simulation tools in an integrated environment that manages the data flow and synchronization between them [1–3]. This modeling and simulation philosophy explores the synergies of combining different tools in a cooperative way, hence allowing the development of more complex overall models in a shorter time. In spite of these advantages there are still at present no known studies of co-simulation applied to parabolic trough solar plants.

The classical parabolic trough solar plant annual simulation approach typically relies on simplified stationary collector models that are built from empiric efficiency data. This type of approach, however, has a major drawback in the fact that it does not model many important physical phenomena that occur in the collector, such as the dynamic effects, e.g. time constant and transport delay, fluid velocity and wind speed influence on efficiency. Furthermore, it does not contemplate specific solar field geometry in detail, such as the row and series bi-dimensional distribution, typically considering the entire solar

---

* Corresponding author. Tel.: +351 966341445.
E-mail address: ricardosilva@ual.es (R. Silva).

0306-2619/$ - see front matter © 2013 Elsevier Ltd. All rights reserved.
http://dx.doi.org/10.1016/j.apenergy.2013.01.069

Nomenclature

A_a	absorber cross-sectional area, m^2
A_c	collector aperture area, m^2
A_f	fluid cross-sectional area, m^2
A_g	glass envelope cross-sectional area, m^2
A_l	fluid lateral area, m^2
C_f	thermo-hydraulic cost function
C_p	fluid specific heat capacity at constant pressure, $J\ kg^{-1}$ - K^{-1}
D	absorber tube internal diameter, m
D_e	absorber tube external diameter, m
D_g	glass tube internal diameter, m
D_{ge}	glass tube external diameter, m
dq	infinitesimal heat transfer, W
dx	differential of the x-coordinate, m
$d\theta$	differential of the θ-coordinate, rad
E_i	absolute error of collector efficiency for an individual test point, %
E_{max}	maximum absolute error of collector efficiency for an individual test point, %
E_t	transient energy accumulation rate on infinitesimal fluid element, W
f	friction factor
f_{bs}	fraction of collector aperture area shaded
G_b	normal direct solar irradiance, $W\ m^{-2}$
h_{af}	heat convection coefficient between fluid and absorber, $W\ m^{-2}\ K^{-1}$
h_{gc}	heat convection coefficient between glass envelope and ambient, $W\ m^{-2}\ K^{-1}$
k_a	absorber thermal conductivity, $W\ m^{-1}\ K^{-1}$
k_f	fluid thermal conductivity, $W\ m^{-1}\ K^{-1}$
K_o	dust, misalignment, and imperfections optical coefficient
K_θ	incidence angle modifier coefficient
L	tube length, m
L_p	spacing between parallel rows, m
$\dot{m}$	mass flow rate, $kg\ s^{-1}$
N	total number of test points
N_p	number of collector in parallel
N_s	number of collector in series
Nu_D	Nusselt number of the interior fluid
Nu_{DGC}	Nusselt number of the exterior wind
p_i	inlet pressure, Pa
P_i	internal tube perimeter, m
p_o	outlet pressure, Pa
Pr	Prandtl number
$\dot{Q}_L$	collector total thermal losses to environment, W
$\dot{Q}_H$	collector hydraulic power consumed, W
$q_{12,conv}$	heat flow rate exchanged by convection between fluid and absorber, W
$q_{2,cond}$	heat transfer by conduction in the absorber, W
$q_{2,irr}$	irradiance absorbed in the absorber, W
$q_{21,conv}$	heat flow rate exchanged by convection between absorber and fluid, W
$q_{23,rad}$	heat flow rate by radiation between absorber and glass envelope, W
$q_{3,irr}$	irradiance absorbed in the glass envelope, W
$q_{3a,conv}$	heat flow rate by convection from glass envelope to ambient, W
$q_{3s,rad}$	heat flow rate by radiation from glass envelope to sky, W
$q_{e,in}$	fluid element inlet enthalpy flow rate, W
$q_{e,out}$	fluid element outlet enthalpy flow rate, W
Ra_D	Rayleigh number
Re_D	Reynolds number
T_1	fluid average temperature, K
T_2	absorber average temperature, K
T_3	glass envelope temperature, K
T_a	absorber temperature, K
T_{amb}	ambient temperature, K
T_f	fluid temperature, K
T_m	fluid average temperature, K
T_s	sky temperature, K
V	fluid infinitesimal element volume, m^3
V_f	fluid velocity, $m\ s^{-1}$
W	aperture width, m
X_i	vector of test conditions for point i
α	thermal diffusivity of annular air, $m^2\ s^{-1}$
α_g	glass tube absorptance
α_o	absorber absorptance
α_o	absorber absorptance
β	collector slope, rad
ΔT	average fluid temperature above ambient temperature, K
ε	absolute roughness, m
ε_a	absorber emittance
ε_{Gt}	glass tube emittance
η_{exp}	measured efficiency at test point, %
η_{model}	model predicted efficiency at test point, %
θ	incidence angle, rad
ν	kinematic viscosity, $m^2\ s^{-1}$
ρ	fluid density, $kg\ m^{-3}$
ρ_o	mirror reflectance
τ_o	glass envelope transmittance

Abbreviations

DAE	differential algebraic equations
FDM	finite difference method
FVM	finite volume method
IST	industrial solar technology
LCR	local concentration ratio
MCRT	Monte Carlo ray tracing
NEP	new energy partners
ODE	ordinary differential equation
PDE	partial differential equation
PSA	*Plataforma Solar de Almeria*
PTC	parabolic-trough collector
RMSE	root mean square error
SHC	solar heating and cooling
SIPH	solar industrial process heat
TISC	TLK inter software connector

field area to be concentrated in a unique algebraic equation. On the other hand, the studies that address parabolic trough collector modeling in detail are mostly limited to short-term simulation, not having the necessary integration level to perform annual plant simulations. Forristal et al. [4] developed a steady-state parabolic trough collector (PTC) model based on energy balances and constitutive relationships. He et al. [5] developed a finite volume method PTC model coupled with a Monte Carlo ray-tracing method. Padilla et al. [6] developed a one dimensional numerical model of a PTC. Yebra et al. [7] developed a one dimensional finite volume method PTC model with direct-steam generation using an experimentally identified heat loss curve. Camacho et al. [8] developed a one dimensional dynamic model coupled with an experimentally identified heat loss curve.