Available online at www.sciencedirect.com

ScienceDirect

IFAC PapersOnline
CONFERENCE PAPER ARCHIVE

IFAC-PapersOnLine 48-1 (2015) 372–377

Modelling, Simulation and Identification of Heat Loss Mechanisms for Parabolic Trough Receivers Installed in Concentrated Solar Power Plants

Simon Caron*, Marc Röger*

**German Aerospace Center (DLR), Institute for Solar Research, Plataforma Solar de Almería Tabernas 04200, Spain (Tel: +34 950 278 863; e-mail: simon.caron@ dlr.de)*

Abstract: This paper describes a thermodynamic model library developed with the object-oriented language Modelica, which is both implemented for steady-state and transient heat transfer analyses of Parabolic Trough Receivers (PTRs) installed in Concentrated Solar Power (CSP) plants. For the identification of PTR heat loss mechanisms, this heat transfer model is coupled to a derivative-free hybrid optimization routine developed in Matlab, combining a Particle Swarm Optimization (PSO) algorithm with a Nelder-Mead Simplex (NMS) algorithm with search space boundary constraints.

Keywords: Thermodynamic Model, Model Validation, Transient Simulation, Parameter Identification.

1. INTRODUCTION

Parabolic Trough Receivers (PTRs) represent one of the key components in Concentrated Solar Power (CSP) plants, as their thermal performance significantly influences the solar field operating temperature and thus the power plant overall thermal efficiency. The development of an accurate in-situ receiver heat loss measurement method requires a numerical heat transfer model (Price et al., 2006), (Lüpfert et al., 2008) to separate heat loss mechanisms (Röger et al., 2014).

Several receiver heat transfer models have been published in the literature. Two-dimensional heat transfer models based on thermal resistance networks have been implemented with Engineering Equation Solver (EES) and validated under steady-state conditions (Forristall, 2003), (Kalogirou, 2012). More detailed three-dimensional models combining Finite Element Method (FEM), Computational Fluid Dynamics (CFD), and Monte Carlo Ray-Tracing (MCRT) have also been implemented with ANSYS (Wirz et al, 2012) and are more suitable for sun irradiated receivers (Eck et al., 2010).

This paper describes a three-dimensional receiver model library which has been extended from two-dimensional models (Forristall, 2003), (Kalogirou, 2012). This model has been implemented with Modelica, an object-oriented language designed for modelling complex physical systems. Steady-state and transient simulations have been performed for single receivers within Dymola simulation environment.

This receiver model has been validated for steady-state temperature conditions and has been coupled to a derivative-free hybrid optimization routine developed in Matlab in order to identify receiver heat loss mechanisms on the basis of transient measurements. The optimization routine combines a Particle Swarm Optimization (PSO) algorithm and a Nelder-Mead Simplex (NMS) optimization algorithm including search space boundary constraints.

2. THERMODYNAMIC MODEL

2.1 Heat loss balance

A typical PTR is made of two concentrical tubes. The inner stainless steel tube absorbs concentrated solar irradiation and transfers the heat to a fluid. The outer borosilicate glass envelope transmits solar irradiation and protects the absorber tube. A selective coating is typically applied on the outer absorber surface to reduce thermal radiation exchange while absorbing as much solar irradiation as possible. The annulus between both tubes is evacuated to reduce convective thermal losses. Both tubes are sealed with vacuum tight bellows on each ends, which compensate for the thermal expansion of the absorber at high operating temperatures.

A radial cross-section of a non-irradiated PTR is illustrated in Fig.1 (Lei et al., 2013) with radial heat flows.

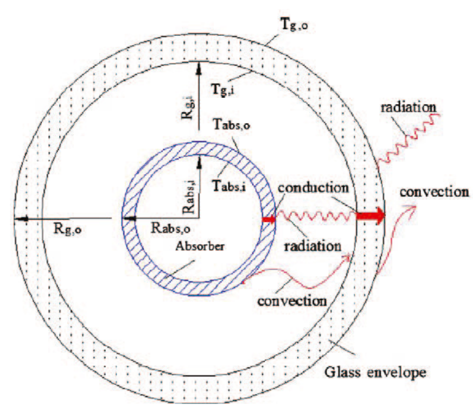

Fig. 1: Receiver cross-section and radial heat flows.

The model includes the radial heat flows illustrated in Fig.1 as well as circumferential and longitudinal conduction terms. The PTR geometry is discretized in cylindrical coordinates. Relevant heat flows are listed in Table 1. Heat flow model assumptions are described in (Röger et al., 2014).

Peer review under responsibility of International Federation of Automatic Control.
10.1016/j.ifacol.2015.05.058

Table 1. List of modeled heat flows

Heat flow	Heat Transfer from … to…		Mechanism
$\dot{q}_{cond,abs}$ (W)	Absorber (inner to outer surface) Temperatures: $T_{abs,i}$ (K); $T_{abs,o}$ (K)		Conduction (3D)
$\dot{q}_{rad,abs\text{-}gl}$ (W)	Absorber (outer surface) $T_{abs,o}$ (K)	Envelope (inner surface) $T_{gl,i}$ (K)	Radiation (1D)
$\dot{q}_{rad,abs\text{-}amb}$ (W)	Absorber (outer surface) $T_{abs,o}$ (K)	Ambient (Sky temp.) T_{sky} (K)	Radiation (1D)
$\dot{q}_{conv,abs\text{-}gl}$ (W)	Absorber (outer surface) $T_{abs,o}$ (K)	Envelope (inner surface) $T_{gl,i}$ (K)	Convection (1D)
$\dot{q}_{cond,gl}$ (W)	Envelope (inner to outer surface) Temperatures: $T_{gl,i}$ (K); $T_{gl,o}$ (K)		Conduction (3D)
$\dot{q}_{rad,gl\text{-}amb}$ (W)	Envelope (outer surface) $T_{gl,o}$ (K)	Ambient (Sky temp.) T_{sky} (K)	Radiation (1D)
$\dot{q}_{conv,gl\text{-}amb}$ (W)	Envelope (outer surface) $T_{gl,o}$ (K)	Ambient (air temp.) T_{air} (K)	Convection (1D)

The heat loss balance is expressed in the equation set (1-4) for a non-irradiated PTR with isolated bellows and a semi-transparent glass envelope.

$$\dot{q}_{cond,abs} = \dot{q}_{rad,abs\text{-}gl} + \dot{q}_{conv,abs\text{-}gl} \quad (1)$$

$$\dot{q}_{rad,abs\text{-}gl} + \dot{q}_{conv,abs\text{-}gl} = \dot{q}_{cond,gl} \quad (2)$$

$$\dot{q}_{cond,gl} = \dot{q}_{rad,gl\text{-}amb} + \dot{q}_{conv,gl\text{-}amb} \quad (3)$$

$$\dot{q}_{rad,abs\text{-}amb} + \dot{q}_{cond,gl} = \dot{q}_{loss} \quad (4)$$

where $\dot{q}_{loss}$ (W) corresponds to the PTR overall heat loss. This heat loss can be normalized by the PTR nominal length L (m) and is identified as the specific PTR heat loss $\dot{q}'_{loss}$ (W/m).

The transient heat loss balance for the glass envelope is expressed in Equation (5-7):

$$\rho_g\, c_{p,g}\, V_g \frac{dT_{gl}}{dt} = \dot{Q}_i + \dot{Q}_o \quad (5)$$

$$\dot{Q}_i = \dot{q}_{rad,abs\text{-}gl} + \dot{q}_{conv,abs\text{-}gl} = \dot{q}_{cond,gl} \quad (6)$$

$$\dot{Q}_o = \dot{q}_{rad,gl\text{-}amb} + \dot{q}_{conv,gl\text{-}amb} = \dot{q}_{cond,gl} \quad (7)$$

where ρ_g, $c_{p,g}$ and V_g respectively correspond to the glass density (kg/m^3), the glass specific heat capacity (J/kg.K) and the glass volume (m^3). Equations (6-8) are implemented in Modelica to derive the glass envelope temperature.

2.2 Internal heat loss mechanisms

Combining equations (1-5), the PTR overall heat loss can be expressed as the sum of three internal heat loss mechanisms: (i) thermal radiation exchange between the absorber and the envelope $\dot{q}_{rad,abs\text{-}gl}$ (W), (ii) annulus convection $\dot{q}_{conv,abs\text{-}gl}$ (W) and (iii) thermal radiation exchange between the absorber and the ambient $\dot{q}_{rad,abs\text{-}amb}$ (W).

The thermal radiation exchange $\dot{q}_{rad,abs\text{-}gl}$ is expressed for diffuse radiating surfaces (Siegel et al., 1981) in Eq. (8):

$$\dot{q}_{rad,abs\text{-}gl} = \frac{2\pi\, R_{abs,o}\, L\, \varepsilon_{abs}\, \varepsilon_{gl}\, \sigma}{\varepsilon_{gl} + \varepsilon_{abs}(1-\varepsilon_{gl}).\frac{R_{abs,o}}{R_{gl,i}}}\left(T_{abs,o}^4 - T_{gl,i}^4\right) \quad (8)$$

where σ is Stefan-Boltzmann constant, ε denotes emittance values and R denotes geometrical radii for the corresponding surface. The absorber thermal emittance ε_{abs} (%) is a key PTR thermal property. This property is a function of the absorber temperature T_{abs} (K).

The annulus convection $\dot{q}_{conv,abs\text{-}gl}$ is expressed in Eq. (9):

$$\dot{q}_{conv,abs\text{-}gl} = 2\pi\, R_{abs,o}\, L\, h_{ann}.\left(T_{abs,o} - T_{gl,i}\right) \quad (9)$$

where h_{ann} (W/m^2.K) is a key PTR thermal property corresponding to the annulus heat transfer coefficient. This coefficient depends on the annulus pressure (Ratzel et al., 1979) and also on gas thermophysical properties (Burkholder, 2011). The nonlinear relationship between the annulus heat transfer coefficient and the annulus pressure is illustrated for four different gases in Fig. 2.

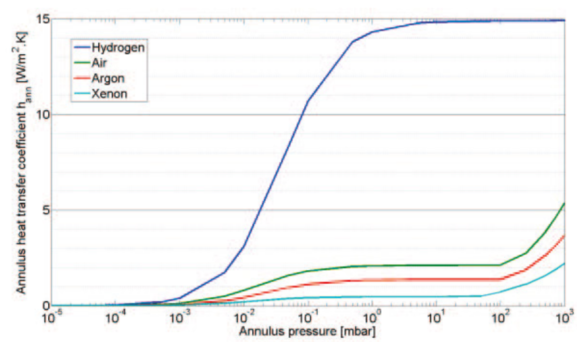

Fig.2: Simulation of annulus convective heat transfer. Fluid thermophysical properties derived from (Kleiber et al., 2010). Boundary conditions: T_{abs}=350°C, ε_{abs} = 10%, T_{amb} = 25°C, v_{wind} = 0 m/s. Receiver cross-sectional geometry: $R_{abs,i}$ = 33 mm; $R_{abs,o}$ = 35 mm; $R_{gl,i}$ = 59.5 mm; $R_{gl,o}$ = 62.5 mm.

3. MODELICA LIBRARY

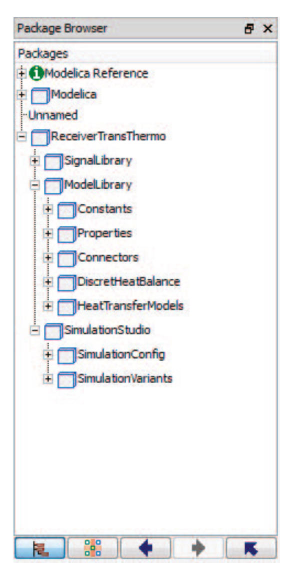

Fig. 3: ReceiverTransThermo Modelica Library.

3.1 Library components

The PTR Modelica library is named *ReceiverTransThermo* and its structure is displayed above in Fig.3. This library includes a first package *SignalLibrary* for transient analyses