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Object-oriented modeling for the transient response simulation of multi-pass shell-and-tube heat exchangers as applied in active indirect thermal energy storage systems for concentrated solar power

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ABSTRACT

This work focuses on the transient numerical modeling of multi-pass shell-and-tube heat exchangers that apply single-phase fluids. A one-dimensional modeling approach is used for the heat exchanger ducts. The governing partial differential equations are solved numerically by applying the finite volume method. In particular, the commonly applied cell-method is used, which is presented in a flexible, intuitive and simulation-platform-independent way. Simulation results are checked for consistency by comparing them to theoretical as well as experimental data available in the literature. Subsequently, the presented modeling approach is used for a specific case study, showing the transient behavior of a typical heat exchanger train configuration currently used at active indirect thermal energy storage systems for CSP (concentrated solar power). Typical process parameters (process gain, dead time and time constant) are given for charging as well as for discharging mode at different heat exchanger loads. Furthermore, transient response simulation results are discussed in detail, providing all used boundary conditions and assumed heat exchanger specifications, thus enabling future model comparison studies. Finally, suitable degrees of discretization are discussed for transient CSP performance simulations on system level, offering a good trade-off between simulation speed and accuracy. Modelica is used as modeling language.

1. Introduction

Solar thermal power, also known as concentrated (or concentrating) solar power (CSP) or STE (solar thermal electricity), is a renewable energy sector with great potential, as it directly harnesses the abundant amount of solar energy incident on planet earth. A rough estimate gives a total of 85 PW of solar power available for terrestrial solar collectors [1]. It has to be emphasized that this is more than 5000 times the current world's power demand of about 15 TW [1]. Furthermore, unlike other renewable energy sectors (like wind or photovoltaic power), solar thermal power plants can provide dispatchable power by means of thermal energy storage and/or hybridization. CSP plants capture the sun's DNI (direct normal irradiation), concentrate it onto a receiving surface and transform the absorbed heat into mechanical work and subsequently electric energy, by using state-of-the-art thermodynamic power cycles.

Today's most mature commercial CSP plants are based on the parabolic trough collector technology. There, the incident solar direct irradiation is focused on receiver tubes that are concentrically placed to the focal lines of parabolic mirrors. A HTF (heat transfer fluid) that is pumped through the receiver tubes collects the thermal energy and delivers it to the steam generator of the plant's power cycle, a conventional subcritical Rankine steam cycle. Today's commercially operated parabolic trough collector plants use thermal oil as HTF. It is a mixture of diphenyl ($C_{12}H_{10}$) and diphenyl oxide ($C_{12}H_{10}O$) and is chemically stable up to about 400 °C [2]. Due to the high costs of this thermal oil, and its high vapor pressure that necessitates the use of pressurized storage vessels [3], an active indirect thermal energy storage system, based on molten salt as storage medium, is the feasible option at commercial parabolic trough collector power plants. The storage medium, the molten salt, is typically a mixture of 60% $NaNO_3$ and 40% KNO_3 (weight percent). This non-eutectic nitrate salt mixture has its solidus temperature at 223 °C and its liquidus temperature at 238 °C [4]. According to a review performed by Bradshaw & Carling [5], the upper design temperature limit is 600 °C, because of the salts' chemical decomposition and the rapidly increasing corrosion rates of piping materials at higher temperatures.

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Nomenclature

A	tube cross sectional area (m^2)
$A_{tube\ inner\ i}$	inner area of heat transfer at each discrete tube segment (m^2)
$A_{tube\ outer\ i}$	outer area of heat transfer at each discrete tube segment (m^2)
AR	amplitude ratio (−) or (K s kg^{-1})
CFD	computational fluid dynamics
CSP	concentrating solar power (or concentrated solar power)
CV	finite control volume
$C_{tube\ i}$	heat capacity of discrete tube section i (J/K)
D_{inner}	tube inner diameter (m)
DAE	differential-algebraic equation
DNI	direct normal irradiation (W/m^2)
f	friction factor (−)
F	heat exchanger rating correction factor (LMTD method) (−)
F_f	friction force acting on the fluid within the control volume i (N)
F_g	gravitational force acting on the fluid within the control volume i (N)
F_p	pressure force acting on the fluid within the control volume i (N)
FEM	finite element method
FVM	finite volume method
$h_{a,i}$	upstream specific enthalpy at the left boundary of control volume i (J/kg)
$h_{b,i}$	upstream specific enthalpy at the right boundary of control volume i (J/kg)
h_i	specific enthalpy of control volume i (J/kg)
h_{fluid}	heat transfer coefficient (W/(m^2 K))
HTF	heat transfer fluid
i	control volume numerator (integer from 1 to n)
IAPWS	International Association for the Properties of Water and Steam
j	conduction model radial section numerator (integer from 1 to 2)
k	thermal conductivity (W/(m K))
k_{tube}	thermal conductivity of the tube material (W/(m K))
K_s	process gain (−) or (K s kg^{-1})
L	tube length (m)
L_i	length of discrete tube segment i (m)
LMTD	logarithmic mean temperature difference (K)
m_i	total fluid mass inside the control volume i (kg)
$\dot{m}_{a,i}$	mass flow at left boundary of control volume i, if entering positive else negative (kg/s)
$\dot{m}_{b,i}$	mass flow at right boundary of control volume i, if leaving positive else negative (kg/s)
MSL	Modelica Standard Library
n	number of finite control volumes, equal to the number of tube segments (integer)
n_t	number of tubes of the bundle (integer)
$n_{t\ lumped}$	number of tubes that are lumped together to one single tube-like object (integer)
Nu_{fluid}	Nusselt number (−)
ODE	ordinary differential equation
p_i	pressure within control volume i (Pa)
Pr	Prandtl number (−)
PDE	partial differential equation
$\dot{Q}_j$	heat flow in cylindrical conduction model radial section j (W)
$\dot{Q}_{net}$	net heat flow over the control volume boundary (W)
r_{center}	cylindrical conduction model center radius (m)
r_{inner}	cylindrical conduction model inner radius (at heat connector a) (m)
r_{outer}	cylindrical conduction model outer radius (at heat connector b) (m)
R_j	thermal resistance of cylindrical conduction model radial section j (K/W)
Re	Reynolds number (−)
RMSE	root-mean-square error (°C)
s	length of discrete flow filament (m); Laplace transform variable
STE	solar thermal electricity
T_i	bulk fluid temperature within control volume i (K)
$T_{tube\ i}$	tube node temperature of discrete segment i (K)
$T_{tube\ inner\ i}$	tube's inner surface temperature of discrete segment i (K)
$T_{tube\ outer\ i}$	tube's outer surface temperature of discrete segment i (K)
t	time (s)
TEMA	Tubular Exchanger Manufacturers Association
u_i	specific internal energy of control volume i (J/kg)
U_i	total internal energy of control volume i (J)
v	flow velocity (m/s)
v_i	flow velocity within control volume i (m/s)
V_i	total volume of the control volume i (m^3)
$\dot{W}_{net}$	net work flow over the control volume boundary (W)
x	coordinate along flow path (m)
z	number of simulated and reference values taken for the RMSE calculation (integer)
Δp	pressure drop due to friction (Pa)
ξ	pressure drop factor (−)
θ	process dead time (s)
ρ_i	density of fluid within control volume i (kg/m^3)
τ	process time constant (s)
ω	excitation frequency (rad/s)

The active indirect two-tank thermal energy storage system (having one hot molten salt tank and one cold molten salt tank) is at the moment the state-of-the-art solution at commercial plants. However, in order to reduce costs a thermocline single-tank approach has been proposed by various authors. In this concept, the hot molten salt tank and the cold molten salt tank is replaced by just one tank containing the hot and the cold salt separated by a thermocline zone, i.e. a temperature gradient zone. A low-cost filler material (packed bed) should prevent convective mixing of the hot and the cold fluid, and furthermore, should provide the bulk of the thermal capacitance of the thermal energy storage [6]. Nevertheless, thermal ratcheting of the storage tank walls remains a significant design concern and further research is required in order to make the thermocline concept applicable at commercial level [7].

In both cases, either the active indirect two-tank or the active indirect single-tank (thermocline) approach, the heat transfer from the thermal oil (the HTF) to the molten salt (the storage medium) and vice versa is accomplished via the use of an oil-to-molten-salt heat exchanger, i.e. as the name already implies, the storage system is charged or discharged indirectly.