SolarPACES 2013

Transient behavior of an active indirect two-tank thermal energy storage system during changes in operating mode – An application of an experimentally validated numerical model

F. Zaversky\textsuperscript{a,*}, M.M. Rodríguez-García\textsuperscript{b}, J. García-Barberena\textsuperscript{a}, M. Sánchez\textsuperscript{a}, D. Astrain\textsuperscript{c}

\textsuperscript{a}National Renewable Energy Center (CENER), Solar Thermal Energy Department, c/ Ciudad de la Innovación 7, Sarriguren (Navarre), Spain
\textsuperscript{b}CIEMAT-Plataforma Solar de Almería, E-04200 Tabernas, Almería, Spain
\textsuperscript{c}Public University of Navarre (UPNA), Department of Mechanical, Thermal and Materials Engineering, Campus Arrosadia s/n CP 31006, Pamplona, Spain

Abstract

Solar thermal power plants are a promising way of providing renewable electricity, particularly, due to their ability to store thermal energy. At today’s parabolic trough collector plants, the only commercially applied thermal energy storage system is an active indirect two-tank thermal energy storage, based on molten salts. This work focuses on the transient response simulation of such a thermal energy storage system during changes in operation. Modelica is used as modeling language. Simulation results show that there are certain limitations regarding instantaneous thermal power demand or supply, which still requires heat transfer fluid buffer storage in order to provide the thermodynamic power cycle of the plant with a continuous thermal power input.

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Selection and peer review by the scientific conference committee of SolarPACES 2013 under responsibility of PSE AG.
Final manuscript published as received without editorial corrections.
Keywords: CSP; Thermal energy storage; Molten salt; Transient response; Modelica

1. Introduction

Solar thermal power, also known as concentrating (also: concentrated) solar power (CSP) or solar thermal electricity (STE), 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 direct normal irradiation (DNI), 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.

<table>
<thead>
<tr>
<th>Nomenclature</th>
<th>Description</th>
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<tr>
<td>CSP</td>
<td>concentrated (or concentrating) solar power</td>
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<tr>
<td>DAE</td>
<td>differential-algebraic equation</td>
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<tr>
<td>DNI</td>
<td>direct normal irradiation</td>
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<tr>
<td>FVM</td>
<td>finite volume method</td>
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<tr>
<td>HTF</td>
<td>heat transfer fluid</td>
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<td>MSL</td>
<td>Modelica standard library</td>
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<td>STE</td>
<td>solar thermal electricity</td>
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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 heat transfer fluid (HTF) 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. Already commercially operated parabolic trough collector plants use thermal oil as HTF. It is a mixture of diphenyl (C₁₂H₁₀) and diphenyl oxide (C₁₂H₁₀O) 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 two-tank thermal energy storage system, based on molten salt as storage medium, is applied at commercial parabolic trough collector power plants. There, the molten salt, a mixture of 60% NaNO₃ and 40% KNO₃ (weight percent), is stored in two large tanks, at two temperature levels and at ambient pressure, providing a certain temperature difference in order to charge or discharge sensible heat. 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. Due to the needed temperature differences for the heat transfer in the heat exchanger, the design temperature levels of the hot and the cold tank are typically 386 °C and 292 °C, respectively.

The typical heat exchanger setup used at commercial parabolic trough collector plants is a counter flow shell-and-tube heat exchanger design having the thermal oil on the tube-side and the molten salt on the shell-side [3]. This fluid assignment is primarily due to the relatively high system pressure of the thermal oil (HTF) circuit. The vapor pressure of the thermal oil is approximately 10 bar at the nominal solar field outlet temperature of 391 °C. Thus, taking the piping and solar field pressure drop into account, the maximum system pressure of the thermal oil circuit is usually around 25 to 30 bar. On the other hand, the molten salt features a very low vapor pressure. It can be stored in the tanks at ambient air pressure, and is thus placed on the heat exchanger’s shell-side.

This work focuses on the transient simulation of a typical active indirect two-tank thermal energy storage system (used at 50 MWe parabolic trough collector power plants) during changes in operating mode and load. In particular, this work describes the transient responses of the oil-to-molten-salt heat exchanger train to abrupt changes in load and to transitions between charging and discharging mode. The molten salt storage tank model is based on a recently published work [4] of the authors. The shell-and-tube heat exchanger model is based on a one-dimensional finite volume approach that builds upon standard Modelica modeling approaches as proposed by the Modelica standard library [5] (MSL). It has been experimentally validated beforehand against measured data obtained at CIEMAT-Plataforma Solar de Almeria. A description of the validation facility can be found in [6]. In summary, it can be said that the model agrees well with the performed experiments. However, a thorough presentation of the performed validation and a detailed model description do lie beyond the scope of the present work, and thus, will be subject of a separate paper. This work gives an example of the model’s application.
2. The methodology and the overview of the modeling approach

This work presents the application of a Modelica-based model of an active indirect two-tank thermal energy storage system. Modelica is a multi-purpose physical system modeling language and has been developed in an international effort in order to unify already existing similar modeling approaches and to enable developed models and model libraries to be easily exchanged. The concept is based on non-causal models featuring true ordinary differential and algebraic equations, i.e. differential-algebraic equation (DAE) systems [7].

2.1. The storage system layout and dimensions

Fig. 1 shows the general setup of an active indirect two-tank thermal energy storage system for CSP. The basic components are the molten salt storage tanks (the hot tank and the cold tank) and the oil-to-molten-salt heat exchanger train, which typically consists of 3 shell-and-tube subunits connected in series [8]. Each shell-and-tube subunit is considered to have two shell passes with a longitudinal baffle and two tube passes in U-tube design, which corresponds to a TEMA-F [9] type design. Note, that the molten salt suction pumps as well as the distribution headers are not displayed in the storage tank schemes.

![Diagram of active indirect two-tank thermal energy storage system](attachment:diagram.png)

Fig. 1. Active indirect two-tank thermal energy storage system scheme (charging mode)

The dimensions of the system are set to values currently applied at 50 MWe parabolic trough collector plants operated in Spain. The diameter and height of the storage tanks’ steel container is set to 38.5 m and 14 m, respectively. The maximum absolute molten salt level height is set to 13 m. The height of the suction pump inlet is set to 0.7 m. Thus, an empty cold or hot tank has still an absolute molten salt level height of 0.7 m, which defines the remaining molten salt mass during cool-down.

The dimensions of each of the three shell-and-tube subunits are assumed as follows: The inner shell diameter is set to 2 m, the shell length is set to 10 m, the tubes’ outer and inner diameter is set to 19.05 mm and 15 mm, respectively. Furthermore, a staggered 45° square tube bundle arrangement is assumed and the tube pitch is set to 28.6 mm. This yields according to Shah and Sekulic [10] an estimated total number of tubes of 3794 per subunit. Finally, a vertical baffle cut of 40 cm is assumed, the baffle center spacing is set to 0.5 m, the baffle inlet and outlet spacing is set to 0.8 m.
2.2. The molten salt storage tank sub-model

The storage tank model is based on the assumption of having one representative molten salt temperature. Hence, the molten salt inventory within the tank is modeled via a single lumped control volume, defining an ideally mixed energy balance. The heat loss to the ambient is modeled in transient mode, by taking the thermal inertia of the tank’s steel container as well as that of the insulation material into account. Convective heat losses via the tank’s gas atmosphere (nitrogen at ambient pressure) above the molten salt surface are neglected. The important radiative heat transfer between the surface of the molten salt and the non-wetted parts of the tank’s steel jacket is considered assuming an ideal cylindrical geometry. The convective heat transfer coefficients at the wetted inner surfaces of the tank’s steel jacket are assumed to be constant. The heat transfer at the tank’s outer surfaces is split into the convective and the radiative part, also considering the solar absorption, which allows for a reasonable approximation of the influence of altering environmental boundary conditions. It has to be noted that the typical area of application of this storage tank model are CSP performance simulations, rather than short transient response simulations for control design purpose, where the molten salt temperature within the tank may be assumed to be constant. For a detailed description and derivation of the model, the interested reader is referred to a previously published work [4] of the authors. Fig. 2 displays the corresponding molten salt storage tank model scheme, also featuring the gas atmosphere control volume, which is, however, not considered in this work.

Fig. 2. Molten salt storage tank model scheme (also considering the convective heat loss via the gas atmosphere)

2.3. The shell-and-tube heat exchanger sub-model

Given the importance of shell-and-tube heat exchangers in process engineering, their performance modeling has already been subject of numerous publications. In summary, already proposed heat exchanger models can be subdivided into two groups, namely (i) three-dimensional numerical models with high spatial resolution, also referred to as computational fluid dynamics (CFD), and (ii) numerical as well as analytical models of much lower
spatial resolution, e.g. approaches using one-dimensional fluid flow models for each of the heat exchanger ducts. Whereas detailed 3-D numerical models [11-14] (category i) are important for improving and optimizing heat exchanger design, models of much lower spatial resolution [15-19] (category ii), and thus, much less computational effort, are applied for transient response simulations that are required for the evaluation of heat exchanger operation and control strategies, i.e. the subject-matter of this study.

This work presents the application of an object-oriented Modelica-based one-dimensional shell-and-tube heat exchanger model. The general approach is similar to that already proposed by Botsch et al. [16]. However, the model, applied in this work, is built upon standard modeling approaches as proposed by the Modelica standard library [5]. Thus, the MSL’s rigorous implementation of a one-dimensional fluid flow, according to the finite volume method (FVM) [20], forms the core feature of the here applied model. In summary, the total flow volume on the shell side and on the tube side is discretized in a certain number of finite control volumes along the flow direction, obtaining a one-dimensional model of the flow on both heat exchanger sides. Furthermore, a certain number of tubes of the U-tube bundle that have similar thermal boundary conditions are lumped together, obtaining a relatively simple transient model of the tube-side flow and the tube wall (lumped capacity 1-D radial heat conduction equations), without modeling each tube of the bundle individually. The shell-side flow may be described by a row of discrete zones, determined by the installed vertical segmental baffles (see Fig. 3), which can be written as follows: inlet zone – window zone – cross-flow zone – window zone – cross-flow zone – window zone – etc. – outlet zone. Hence, the shell-side flow is modeled by assigning each zone one finite control volume. The tube-side and the shell-side control volumes are correspondingly linked in order to represent the real flow setup (two passes on the shell side and two passes on the tube side), which is also referred to as “cell-method” [21].

The forced convective heat transfer on the tube-side is modeled using the widely accepted Nusselt number correlation proposed by Gnielinski [22], valid for transitional and turbulent flow in smooth tubes. The interested reader is referred to Eq. number (14) and (15) of his original work. The forced convective heat transfer on the shell-side is considered according to the Bell-Delaware method [10] that estimates the shell-side heat transfer coefficient by multiplying the ideal value for cross-flow over a tube bundle by correction factors that take non-ideal flow conditions into account. The ideal shell-side heat transfer coefficient for cross-flow over a tube bundle is evaluated according to Gnielinski [23], who provided Nusselt number correlations for single tube rows and tube bundles under cross-flow, based on numerous measured values.

The momentum balance is reduced to its steady state formulation, considering only the pressure drop due to friction, which is calculated according to Moody [24] and Idelchik [25] on the tube-side, and according to Gaddis and Gnielinski [26] on the shell-side.

A detailed description of the model and its experimental validation against measurement data obtained at CIEMAT- Plataforma Solar de Almeria will be subject of a future publication of the authors.

Fig. 3. Shell-and-tube heat exchanger scheme (TEMA-F type design)
2.4. The fluid properties

Within the MSL, all specific media property functions are decoupled from the library components by defining a replaceable “medium package” in each of them. Basically, all fluid property function names and interfaces are defined within the base class “partial medium”. In order to allow for a full replaceability, each specific medium model extends from this base class the “partial medium” and defines the specific medium related relationships by re-declaring each necessary medium property function. Thus, every single component of the library that covers the modeling of fluid is not limited to a single specific medium. In fact, it can easily be adapted for the use of different media, by simply replacing the default medium package when instantiating the final model.

Since the media properties for solar salt as well as that for thermal oil are not yet available in open Modelica libraries, these features have been newly implemented. Both fluids are modeled as incompressible at liquid state according to [27] and [2], respectively. In order to allow for efficient simulation code generation, inverse and derivative functions are implemented as well.

2.5. The final storage system model and its control

The final storage system model consists of 2 instances of the molten salt storage tank model class (i.e. one for the hot tank and one for the cold tank), and 3 instances of a shell-and-tube heat exchanger model class that are connected in series (see Fig. 1). All connecting pipes are assumed to be ideally insulated.

The HTF mass flow rate is a given boundary condition of the model. The same holds for the HTF inlet temperatures. They are set to the constant values of 391 °C, for charging (solar field outlet temperature), and 285 °C, for discharging (power block outlet temperature). The molten salt mass flow rate is controlled in such a way that either the molten salt outlet temperature (charging), or the HTF outlet temperature (discharging) keeps to the assigned set-point. In particular, during storage system charging mode, the molten salt outlet temperature is kept to the hot tank’s design temperature. On the other hand, during storage system discharging mode, the HTF outlet temperature is kept to the desired feed temperature of the CSP plant’s steam generator (in this work 376 °C). In this way, the feed temperature of the cold molten salt tank varies according to the thermal characteristic of the heat exchanger train, i.e. it is not controlled to keep to the cold tank’s design temperature of 292 °C. Continuous PI controllers are applied for both the charging, and the discharging mass flow control. The inertia of the control system’s actuator circuit (including the molten salt pumps) is modeled via an instance of a first-order block having a gain of 1 and a time constant of 30 s, i.e. 63.2% of the final molten salt mass flow rate value is reached after 30 s assuming a step-change in the control signal.

Since the HTF inlet temperatures are assumed to be constant while charging and discharging, the HTF mass flow rate variation is the only major disturbance caused. This also includes changes in flow direction when switching from charging to discharging mode, or vice versa. The controller settings have been obtained via a so-called process reaction curve method, which can be applied to self-regulating processes that feature first-order-plus-time-delay behavior, i.e. that can be described by the three parameters as process gain $K_p$, dead time $\theta$ and time constant $\tau$, as it is the case for the here considered heat exchanger in the range close to a certain operating point. In particular, the controller settings have been obtained applying the Chien-Hrones-Reswick method for the aperiodic case and set point step [28]. Evaluating open loop step response simulations and applying the tangent method at the point of inflection [29], yields approximately 0.031 K s kg$^{-1}$ for the process gain $K_p$, 12.3 s for the dead time $\theta$ and 345.6 s for the time constant $\tau$ for charging mode, and 0.056 K s kg$^{-1}$ for the process gain $K_p$, 12.6 s for the dead time $\theta$ and 251.6 s for the time constant $\tau$ for discharging mode. This gives a controller gain $K_p$ of 317.1 kg s$^{-1}$ K$^{-1}$ for the charge controller, and a controller gain $K_p$ of 124 kg s$^{-1}$ K$^{-1}$ for the discharge controller. The integral time $\tau_i$ results in 414.7 s for the charge controller, and in 302 s for the discharge controller. It has to be noted that the controller output in the model is a direct mass flow signal and not a valve position or molten salt pump speed as it is the case in reality. Furthermore, the controller settings, stated above, have been obtained for flow conditions that are close to the nominal ones. However, due to non-linearities of the behavior of the heat exchanger train, its process parameters, on which the controller settings are based, typically vary depending on the actual load (HTF mass flow rate) and may not be ideal for the operation at partial loads.
2.6. The model’s translation and its numerical integration, i.e. simulation

The developed Modelica code has been translated into numerical simulation code using a state-of-the-art commercial Modelica tool, applying its differential-algebraic system solver DASSL [30]. This algorithm applies an implicit method for the numerical integration of the governing ordinary differential equations. In particular, it approximates the derivatives using a \( k \)th order backward differentiation formula, where \( k \) ranges from 1 to 5. At every step it chooses the order and the step size based on the behavior of the solution. Newton’s method is used to solve the resulting equations for the solutions at each discrete point in time [30].

It should be noted that the final model class of this active indirect two-tank thermal energy storage system features 22291 equations and a simulation of a 5 h experiment typically takes around 10 min on a standard desktop computer (3.10 GHz).

3. Discussion of simulation results

In order to show the transient behavior of the thermal energy storage system, the numerical model has been subjected to step changes in HTF mass flow rate. In particular, 4 simulations treat step changes in HTF mass flow rate during charging and discharging mode, respectively (step changes between nominal load and 50% load, and step changes between 50% load and 25% load). In addition to that, 4 simulations treat the complete change in operating mode, including the flow reversal within the heat exchanger ducts, i.e. the switching between storage system charging and discharging mode. These simulations have been performed for 4 load cases (100%, 75%, 50% and 25% load).

Fig. 4 displays the results of the 4 simulations during that no operating mode switching occurs, i.e. there occurs no flow reversal within the heat exchanger ducts, only the mass flow rates change between nominal load (100% load) and 50% load (solid lines), and between 50% load and 25% load (dashed lines). More precisely, the HTF mass flow rate (red) is subjected to ideal step changes between these loads, after 500 s and after 8000 s. Note, that the mass flows are defined positive for charging mode and negative for discharging mode. As can be well observed, the controllers adjust the molten salt mass flow rates (blue) to the new steady state values in order to reach the desired set points. In charging mode, the outlet temperature of the molten salt has to follow its set point, i.e. the hot tank’s design temperature. On the other hand, in discharging mode, the outlet temperature of the HTF has to follow its set point, i.e. the steam generator’s inlet temperature. Thus, while discharging, the feed temperature of the cold molten salt tank is not controlled and varies according to the heat exchanger’s thermal characteristic.

Fig. 5 shows the corresponding heat exchanger duct outlet temperatures in charging mode (molten salt: blue, HTF: red), and the corresponding ideally mixed inventory temperature of the hot molten salt tank (gray). As can be well observed, with the aforementioned controller settings, it takes approximately 25 min to control the system to the steady state conditions of the new operating point. Note, that the difference in temperature response (depending on the load) also slightly influences the molten salt inventory temperature trend. The offset between the solid and the
dashed line (gray) is due to different filling levels (Fig. 5 (a)), since the initial value of the storage system’s state of charge is the same for each simulation run. Lower molten salt filling levels lead to higher temperature drops, due to higher specific heat losses [4].

Fig. 5. (a) hot tank feed and inventory temperatures (charging mode); (b) HTF outlet temperature (charging mode); 100% - 50% load steps: solid, 50% - 25% load steps: dashed

Fig. 6. (a) HTF outlet temperature (discharging mode); (b) cold tank feed and inventory temperatures (discharging mode); 100% - 50% load steps: solid, 50% - 25% load steps: dashed

Fig. 6 shows the heat exchanger duct outlet temperatures in discharging mode (molten salt: blue, HTF: red), and the ideally mixed inventory temperature of the cold molten salt tank (gray). As already mentioned above, in this operating mode, the aim of the controller is to keep the HTF outlet temperature (steam generator feed temperature) at the desired set point. As can be seen in Fig. 6 (a), it takes about 17 min until the HTF outlet temperatures settle at the set point, reaching the new steady state conditions after the step change in HTF mass flow rate (see Fig. 4). Fig. 6 (b) displays the resulting feed temperatures of the cold molten salt tank during discharging mode. The cold tank’s feed temperature is not controlled and depends thus on the operating point. During nominal discharge conditions, the feed temperature of the cold molten salt tank is about 1.5 °C higher than its design temperature of 292 °C, which leads to a slight increase in molten salt inventory temperature (gray solid line). On the other hand, at 25% load, the resulting feed temperature of the cold tank is about 2 °C below its design value, which shows, that the actual temperature of the cold tank’s molten salt inventory varies during operation within a certain range around its design value.

Fig. 7 displays the simulation results of changes between charging and discharging mode. In particular, starting from steady state charging conditions (positive mass flows), the HTF mass flow is inverted at simulation time t = 500 s, which initiates the discharging mode. Then the discharge controller acts in such a way, that the molten salt mass flow also changes its sign in order to settle the HTF outlet temperature at the desired set point, the steam generator’s feed temperature. About 25 min after the flow reversal, the HTF outlet temperature reaches its set point. At t = 8000 s, the HTF mass flow is inverted again, initiating the charging process. With the given controller settings
(see Section 2.5), the molten salt mass flow shows damped oscillations at low partial loads (dotted blue line) when switching from discharging to charging mode, which disappear at higher loads. However, in this case (Fig. 7 (a)) about 30% of overshoot remain, also at nominal loads, which causes the molten salt mass flow rate to be limited to the assumed maximum value of 950 kg/s.

Fig. 7. (a) Mass flows during changes in operating mode; (b) The corresponding temperatures at the hot end of the heat exchanger train; 100% load: solid lines, 75% load: dot-dashed lines, 50% load: dashed lines, 25% load: dotted lines

Summarizing the above presented simulations, it can be said that the thermal inertia of the storage system’s fluid circuit limits the performance during abrupt changes in operating mode, of course, depending on the applied control methodology. Thus, in order to provide the power block with a constant thermal power supply independent of the current solar irradiation, and, to store as much thermal energy as possible when solar irradiation is available in excess, the optimization of control methods and additional HTF buffer storage plays an important role.

4. Conclusions and outlook

This paper presents the application of a detailed one-dimensional numerical model of an active indirect two-tank thermal energy storage system for solar thermal power generation. As presented in this work, the applied model is typically suitable for the evaluation of transient responses caused by the storage system’s disturbance variables as HTF mass flow rate, HTF inlet temperature or molten salt temperatures within the tanks. It thus forms a valuable basis for improving and optimizing the storage system’s operation and control strategy, important tasks related to cost-competitiveness of CSP.

It has been shown that there are certain limitations regarding the storage system’s charging and discharging performance during abrupt changes in operating mode, due to the thermal inertia of the storage system’s fluid circuit. The active indirect two-tank thermal energy storage concept has certain limitations regarding instantaneous thermal power demand or supply. Hence, additional small buffer storage in terms of excess HTF piping volume or small vessels is still important in order to provide the thermodynamic power cycle of the plant with a continuous thermal power input, regardless of the current solar irradiation. Furthermore, it has to be emphasized, that the storage system’s performance strongly depends on the oil-to-molten-salt heat exchanger’s control methodology. Since this study only considers standard PI feedback control loops with constant tuning parameters, the potential of advanced control methods should be treated in future works.

References


