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Transient thermo-mechanical analysis of steam generators for solar tower plants

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Abstract

In solar power tower plants, fast start-ups and/or load changes are mandatory to increase the power plant dispatchability. The high temperatures of the working fluids and the partial-load operation will reduce the lifetime of the thick-walled components at the steam generator. Therefore, a proper heat exchanger design should consider the stress evolution during the transient operation of the plant.

This work addresses, for the first time, a methodology to determine the dynamic behavior of all heat exchangers of a steam generator train. The methodology proposed here is a powerful tool for the design of solar power plants. The stress analysis model identifies the most important components of the steam generation train during transient operation. The methodology consists of the combination of analytical models to obtain the coupled response of the steam generation train from the following dynamic variables: temperature, pressure and stress.

An example of this methodology is presented for two start-up initial conditions: the assumption of non-isothermal and isothermal temperature profiles of the heat exchangers. A steam generator train based on conventional shell and tube heat exchangers is analyzed. The analysis shows that the non-isothermal condition takes approximately 50 min to reach nominal conditions, whereas the isothermal condition takes approximately 110 min, requiring 600 tons and 716 tons of hot salt to perform the start-up procedure, respectively.

Key words: *Solar power tower plant; Steam generator; Start-up procedures; Thermal stress analysis.*

Nomenclature

Abbreviations

CCPs: combined-cycle plants

CSP : concentrating solar plants.

EV : evaporator.

FEA : finite element analysis.

FW : feed water.

HP : high pressure.

HPT : high-pressure turbine.

HRSG : heat recovery steam generator.

HTF : heat transfer fluid.

HT : hot tank.

Hx : heat exchanger.

LP : low pressure.

LPT : low-pressure turbine.

PH : preheater.

R : receiver.

RH : reheater.

SG : steam generator.

SH : superheater.

SPTP : solar power tower plant.

TES : thermal energy storage.

UTS : ultimate tensile strength.

Symbols

A : heat transfer area (m²).

A_c : cross-section area (m^2).

C : specific heat capacity (kJ/kg K).

C_m^* : dimensionless thermal capacitance of a metal wall (-).

D : diameter (m), characteristic time constant (-).

E : modulus of elasticity (MPa).

F : force (N).

I : moment of inertia (m^4).

K : stiffness (N/m).

K' : inverse of stiffness (m/N).

L : length (m), characteristic length constant (-), level (m).

L_{tp} : tube pitch (mm).

L_{ts} : tubesheet thickness (m).

M : mass (kg), moment (N m).

P : pressure (Pa).

R : radius (m).

R_{otl} : radius of a circle circumscribed to the outermost tube of a bundle (m).

T : temperature ($^{\circ}\text{C}$).

T^* : dimensionless temperature (-).

V : volume (m^3), vertical force (N).

W_t : turbine power (MWe).

a : thermal diffusivity (m^2/s).

h : convective coefficient ($\text{W/m}^2\text{ }^{\circ}\text{C}$)

i : specific enthalpy (J/kg).

l_{ts} : tubesheet thickness (mm).

$\dot{m}$: mass flow rate (kg/s).

t : time (s),

t_j : thickness of component j (m).

t^* : dimensionless time (-).

r : radial coordinate (-).

u : specific internal energy (kJ/kg).

y : vertical displacement (m).

z : axial coordinate (-).

Greek Symbols

α_t : thermal stress concentration factor (-).

α_m : pressure stress concentration factor (-).

β_t : linear thermal expansion coefficient (1/K).

θ : rotation angle (°).

ν : Poisson's ratio (-).

ρ : density (kg/m³).

σ : stress (MPa).

ψ : U-tube stress concentration factor (-).

Subscripts

0 : nominal conditions.

ave : average.

atemp : attemperator.

b : bending.

cyl : cylinder.

d : drum.

dc : downcomer.

h : head.

i : inlet, inner

m : metal.

i : inlet.

O : outlet, outer

r : riser.

rec : recirculated.

S : shell.

SA : salt.

sat : saturated.

st : steam.

SU : surface

t : tube.

tp : tube pitch.

ts : tubesheet.

W : water.

1. Introduction

In recent years, significant effort has been made to reduce the operation and investment costs of concentrating solar power (CSP) plants. One of the main differential features with respect to other renewable technologies, such as photovoltaics or wind, is the flexibility provided by CSP when integrated with a thermal energy storage (TES) system. This increases the economic competitiveness of these power plants. On the one hand, the increase of the capacity factor diminishes the cost per unit of energy produced. On the other hand, the instantaneous solar source is decoupled from the electricity generation and therefore can be delivered on demand. Then, CSP with a TES system enhances the possibility of participating in grid balancing services [1]. In this manner, these plants may be considered as dispatchable, and therefore, improvements in the flexibility by means of fast start-ups and/or load changes may lead to additional revenue [2]. Moreover, fast start-ups are especially interesting for CSP plants due to the increase of the annual electricity production [3, 4]. However, the start-up and/or load change ramps are limited by the thermal stresses in thick-walled parts of the steam generator (SG)

and/or steam turbine. These thermal changes may produce fatigue and/or creep damage in the SG [5]. For this reason, considerable effort should be made to develop dynamic models that consider the lifetime of the heat exchangers. These models allow the study of different strategies for operating the plant on the safety-side while saving energy during the start-up and shutdown processes.

One of the main features of solar power tower plants (SPTPs) is their higher operating temperatures compared to parabolic trough and linear Fresnel plants [2]. This leads to an increase in the thermal efficiency of the power block, which also reduces the specific costs. Examples of the feasibility of CSP technology are Gemasolar [6], Crescent Dunes [6] and Suncan Dunhuang [6], the three large-scale SPTPs currently in operation. However, other technologies, such as fire-boilers or combined-cycle plants (CCPs), work at similar temperatures to CSP plants. One of the main differences between conventional and CSP plants appears in the SG. The heat exchangers of the SG are based on conventional shell and tube designs. These heat exchangers are sized by means of analytical methods proposed by TEMA standards [7] and/or ASME code Section VIII-Div1 [8] considering the operation at nominal conditions [9]. Then, because the SG will be operated with daily start-ups, load changes and shutdowns, a transient stress analysis is required. Moreover, no specific guidelines are proposed in the TEMA standards and/or ASME code for complex zones, such as the tubesheet, when: transients thermal loadings occur [10], temperature gradients on the junction appear [11] or the tubesheet has a non-standard design [12]. Therefore, these issues must be accomplished by finite element analysis or by complex analytical methods to obtain accurate results.

Start-up and shutdown operations become critical to the creation of operational flexibility. Such flexibility is critical for the necessary adjustments of the plant power output to the electricity demand. In this line, CCPs have become an invaluable technology in the conventional power generation mix, providing both power generation flexibility and energy security to grid balance services. The reliability of CCPs lies in the corresponding dynamical response of the turbines and the heat recovery steam generator (HRSG) system, which exhibits remarkable operational

similarities to the SGs found in SPTPs. In recent years, many efforts have been made to develop simulation models focused on the dynamic behavior of the HRSG, which can be useful for the development of SGs of SPTPs. This interest is mainly motivated by the liberalization of the electricity market and the penetration of renewable technologies, which have forced CCPs to improve their operational flexibility. Several works can be found regarding dynamic simulation of CCPs. For example, Kim et al. [13] developed an HRSG model based on the lumped capacitance method. A cold start-up procedure is studied to keep the thermal stress of the steam drum within the safety limits by means of the exhaust gas turbine bypass. Mertens et al. [14] showed that the transient response of a once-trough-type HRSG is faster than a natural circulation drum-type using an HRSG software tool [15]. Rossi et al. [16] modeled the start-up of a CCP showing that the intermediate pressure rotor stress is slightly higher than the high-pressure rotor stress during a hot start-up. Benato et al. [5] analyzed the lifetime consumption of an HRSG, considering as critical parts the superheater header, the reheater collectors and the steam drum. The lifetime calculation was performed using the European Standard EN 13445 [17]. Their results showed that the highest lifetime consumption is obtained in the superheater header for a 50% load change operation.

Focusing on the CSP plant simulation, the dynamic behavior of the SG has been addressed in a few studies. For instance, Heiko et al. [18] presented a parabolic trough CSP plant model using the Dymola software tool [19]. They studied hot and warm approaches for the power-block start-up. Although the main thermodynamic variables are shown, the technical constraints used for SG, such as stresses or temperatures differences, are missing. Foster Wheeler [20] presented a detailed study regarding the design and performance of molten salt SG for SPTP. Start-up and load-change procedures are described considering several initial conditions. However, in [20], a fixed pressure turbine operation is assumed for all SG loads, while in recent years, sliding pressure turbine operation has been established due to different advantages in control and efficiency under partial-load conditions [21]. Pacheco et al. [22] described different procedures for daily start-up and shutdown operations on the SG based on the experience of the SPTP

experimental facility "Solar Two." Problems related to salt freeze in the evaporator due to the low inlet temperature in the start-up leads to add an additional start-up feedwater to ensure safe operation of the SG. Wagner et al. [23] optimized the operation of an SPTP considering different price market scenarios, showing that the profitability of these plants can be increased by 5-20%. Flueckiger et al. [24] carried out a simulation of an SPTP with a thermocline thermal energy storage system for a molten salt steam generator working under partial-load conditions. Luu et al. [25] presented dynamic modeling of an SPTP with a supercritical CO₂ Brayton power cycle. They developed a start-up scheme to bring the cycle to full-load conditions effectively.

In this work, the thermo-mechanical transient dynamic response of the steam generator of a solar power tower plant is modeled to analyze its start-up process. For the first time, the coupled dynamic behavior of all heat exchangers that comprise the steam generator train of solar power tower plants is analyzed. This analysis is mandatory to obtain the thermo-mechanical variables needed to design the steam generator against fatigue depending on the power plant operation. The proposed methodology computes the stress values on the critical components of the heat exchangers, such as the tubesheet and tubesheet junction, to work below the maximum stress limit. European standard EN 12952-3 [26] is used to estimate thermal and pressure stresses on the superheater head and the steam drum. A control loop has been designed to adjust the dynamic response to the start-up requirements. To illustrate the use of the methodology, two different initial conditions for a start-up are analyzed.

2. System description

A 110-MWe power block subcritical Rankine cycle with regeneration is selected to model the SG system of the SPTP. The plant is composed by a field of heliostats, which reflect the Sun on a solar tower receiver, a steam generator and a power block. The heat transfer fluid (HTF) is molten salt, which is heated up from 290°C to 565°C in the solar tower receiver. A thermal energy storage system (TES) formed by two tanks (one hot and one cold) stores and delivers the energy captured by the solar field in a controlled manner. The steam generator system includes four shell and tube heat exchangers: a superheater (SH), reheater (RH), evaporator (EV) and

preheater (PH). The hot salt is sent to the SG where the thermal energy is transferred to produce main and reheated steam. Additionally, the SG includes, among others, feedwater pumps, hot salt pumps, the steam drum, supporting systems, etc. The SG train designed by González-Gómez et al. [27] for nominal conditions is chosen for the dynamic analysis. The subsystems of the SPTP are represented in Figure 1 with the main design parameters used in the start-up procedure.

Figure 1. Schematic of SPTP.

In the SG train, the molten salt is placed on the shell side, and the steam/water flow is placed on the tube side because of its high pressure. First, the PH, which consists of a U-tube/TEMA F shell heat exchanger type, heats the feedwater up to saturation conditions. The EV consists of a U-tube/TEMA E shell type where the inlet salt nozzle is put far from the tubesheet to minimize its thermal stress. A forced circulation evaporator system is chosen to increase the operability in the start-up process [28]. The steam drum is arranged horizontally to reduce the steam velocity and improve the water droplet separation through the action of chevrons. Finally, the SH and RH are a counter-current U-shell/U-tube heat exchanger type.

The intermittent operation of the SG of SPTPs can produce fatigue damage due to the cyclic stress. Furthermore, the high temperatures and potential impurities of the molten salt in an industrial-grade process may lead to stress corrosion cracking damage. Therefore, material selection is one of the most important points to ensure the reliability of the heat exchangers. Kelly [29] recommends stainless steel Gr347 or 321 for the hot salt piping according to the Solar Two operating experience. Therefore, the materials selected for the SG are shown in Table 1.

Table 1
Selected Materials for the main SG components.

Component	Shell, channel and tubesheet	Tube
Preheater	A516, Gr. 70	A210, Gr. A1
Evaporator	A387, Gr. 91	A213, Gr. 347
Superheater	A240, Gr. 347	A213, Gr. 347
Reheater	A240, Gr. 347	A213, Gr. 347
Steam Drum	A516, Gr. 70	-

2.1. Turbine operation modes

The efficiency of thermal power plants can be increased by changing from the fixed to the sliding pressure operation mode [21,28]. The main advantages of the sliding operation mode are the reduction of the power pump consumption and the efficiency increase of the SG because of the increment of the mean temperature difference in the heat exchangers. An additional benefit of the sliding operation mode is the reduction of throttling losses in the turbine, which may lead to a higher turbine power output under partial-load operation. Alternately, the fixed pressure mode has the advantage of preventing the potential risk of salt freeze in the evaporator. The minimum steam pressure in the fixed pressure mode is 7.4 MPa, avoiding a decrease of the molten salt temperature below its melting safety limit of 290°C. In this work, both fixed and sliding operation models are used for the start-up conditions analyzed. When the steam turbine is operated between 0% and 50% load, a fixed pressure mode is used. From 50% to 100% load, a sliding pressure mode is selected.

The power block transient operation is modeled for partial-load conditions. The model provides the temperature, mass flow rate and pressure of the main inputs of the SG (the feed water and reheat inlet steam) and the turbine power output for design and partial-load conditions.

2.2. Control system

The control system is aimed to guide a stable simulation of the proposed transient models of the SG. Otherwise, it would not be possible to capture the dynamic behavior of the SG system [31]. A PID control strategy has been chosen to guide the transient process for the two example start-ups analyzed. As shown in Figure 2, most of the manipulated variables are operated with proportional and integral actions (PI) in accordance with the proposed start-up scenarios. Only the drum level is driven by a proportional action (P). The controlled variables of the SG are the salt and steam temperatures, the steam flow rate, the steam pressure and the steam drum water level. The manipulated variables are the mass flow rates of the salt, steam flow, SH and RH attemperations systems and variable-speed molten salt pumps.

Figure 2. Schematic of the SG and control system.

The SH and RH attemperation system keeps constant the inlet temperatures of the high-pressure (HP) and low-pressure (LP) turbines for any transient disturbances. This procedure reduces the fatigue damage of the HP/LP turbines. As seen in Figure 2, the SH attemperator is placed at the inlet and outlet, whereas the RH attemperator is only placed at the inlet [20]. The attemperator placed at the inlet of the heat exchangers ensures that the steam is free of water droplets and provides better control of the thermal shock of the pipe walls [32]. In contrast, the attemperator placed at the outlet of the heat exchangers provides direct control with no time lag [33]. Furthermore, the attemperation steam flow is limited to 8% of the total flow [32].

In addition to the attemperation system, the SH and RH outlet temperature are also regulated by salt mass flow valves. The salt flow rate changes under partial-load conditions, making necessary an SH-RH salt by-pass line to send the salt excess flow to the EV. This line increases the evaporator heat duty and the pressure in the steam drum, which may produce an increase in the steam flow rate if the throttle valve position does not change.

The proposed system checks the temperature difference between the salt outlet and the steam inlet in the SH and RH, controlling the salt valves. In this manner, the lower tubesheet of the SH, which is the critical part, works below its safety stress limit.

The steam drum pressure control depends on the turbine operation mode. For fixed pressure mode, an extra flow of salt is sent to the evaporator using the by-pass salt valve to ensure the minimal steam drum pressure. When the turbine is operating in sliding pressure mode, the steam drum pressure is automatically regulated by the saturated steam outlet. Furthermore, the steam drum level is kept in a suitable range by means of the feedwater mass flow rate using a proportional control system.

2.3 Start-up initial conditions

To illustrate the thermo-mechanical analysis of the SG train, two initial conditions are considered at the beginning of the start-up process: daily non-isothermal initial conditions [20] and daily isothermal initial conditions [22]. The salt temperature profiles of the SG are shown in Figure 3 for both starting procedures and for full-load conditions of the plant.

Figure 3. Salt temperature profiles in SG.

- Daily start-up with non-isothermal conditions

The heat exchangers of the SG are closed after the night shutdown and thermally isolated to reduce the heat losses [20]. As they are filled with hot salt, each heat exchanger has different temperatures. This procedure was proposed by Foster Wheeler assuming non-isothermal initial conditions on a daily start-up [20]. The initial temperature profiles of the heat exchangers at the beginning of the daily start-up are calculated from the night shutdown condition, which corresponds to fixed pressure operation mode at 25% partial load with a steam drum pressure of 7.4 MPa and saturation temperature of 290°C. These conditions are set as a freeze salt protection point and correspond to the fixed turbine operation mode. A temperature difference is subtracted for each heat exchanger following the cooling rates published in [20] due to the night cooldown period. It is assumed that there are 8 hours of cooldown time and that the longitudinal heat conduction of the salt can be neglected. The feasibility of this approach is ensured by a salt freeze protection system that consists of an electric heat tracing system in the SG.

- Daily start-up with isothermal conditions

A different strategy is followed by Solar Two SPTP for a daily start-up in which the cold salt circulates during the night through the SG [22]. The SG temperature is kept constant at 290°C at the beginning of the daily start-up. This procedure compensates the ambient energy losses from the heat exchangers and the energy required to heat up the feedwater. However, more energy is needed to heat up the feedwater than to compensate the energy losses. On the one hand, the energy losses during the night are small due to the good insulation systems of the heat exchangers. On the other hand, high energy is needed to maintain the water level in the evaporator due to the steam leakage through the pressure relief valves and the water blowdown.

In contrast, considering the Foster Wheeler [20] approach, the SG heat exchangers filled with salt achieve the isothermal condition as a result of a few days without SG operation. The salt freeze protection system is provided by electric heat tracing to keep the temperature above 290°C in the SG train. It should be noted that the Foster Wheeler approach is considered for a 100-MWe plant, whereas the Solar Two approach is considered for a 12.5-MWe plant.

It is worth mentioning that other heat-up systems were considered in Solar Two for heat exchangers and salt pipelines: electrical heat tracing or intermittent operation of the salt circulating pumps [34]. The first approach results in higher energy consumption than the second approach. However, intermittent operation requires many measurements in the flow circuit, and despite that, it does not avoid the potential for salt freeze due to measurement failure. Moreover, intermittent operation of the salt circulating pumps may lead to fatigue failure. Finally, the overnight temperature control of the SG (heat exchangers and pipelines) is achieved by continuous operation of both electric heat tracing and the cold salt circulating pump.

3. Modeling and validation

This section describes the models used to obtain the transient and dynamic response of the SG for the start-up initial conditions described before. The models either use the theoretical approaches proposed in the literature or are here developed to reflect the transient response of

the SG. The main goal of the models is to find the highest stress values on the SG, identifying the critical part of the heat exchangers. Because the stress and temperature calculations are decoupled, the first step consists of calculating the temperature profiles of the heat exchangers and then obtaining the stress values.

3.1. Single-phase flow heat exchangers

The SH, RH and PH are modeled as counter-current heat exchangers [35]. Assuming negligible wall resistance and heat transfer to the outer wall, the heat balance can be expressed as follows:

$$D_{sa} \frac{\partial T_{sa}}{\partial t} + L_{sa} \frac{\partial T_{sa}}{\partial x} + T_{sa} = T_m \quad (1)$$

$$D_m \frac{\partial T_m}{\partial t} + T_m = \frac{h_{sa} A_{sa} T_{sa} + h_{st} A_{st} T_{st}}{h_{sa} A_{sa} + h_{st} A_{st}} \quad (2)$$

$$D_{st} \frac{\partial T_{st}}{\partial t} - L_{st} \frac{\partial T_{st}}{\partial x} + T_{st} = T_m \quad (3)$$

where $L = \frac{C \dot{m}}{h A}$ is the characteristic length and $D = \frac{\rho C A_c}{h A}$ is the characteristic time. The above

equation system can be solved numerically using an implicit finite difference scheme. Another means by which to solve the equation system above is the method proposed by Ansari and

Mortazavi [36]. This method is based on the analytical solution of Equations (1) and (3)

assuming that wall temperature variation is a first-order system for one cell. Then, the analytic solutions are used to obtain the outlet temperatures on each cell. Figure 4 shows a comparison

of the solutions for different heat capacities of the metal wall ($C_m^* = \frac{C_m \rho_m}{C_{hot} \rho_{hot}}$) by the analytical

and numerical approaches. The temperatures are expressed in a dimensionless form (

$T^* = \frac{T - T_i}{T_o - T_i}$). The results show that the numerical solution presents similar values to the

analytical model, and thus the numerical model is employed in this work. The heat exchangers are discretized along the tube length to the baffle spacing. The time step is calculated to fulfill the Courant condition.

Figure 4. Response comparison between numeric and analytical methods for a temperature step change in the hot fluid: a) cold fluid outlet and b) hot fluid outlet.

Additionally, the fluid properties are calculated in each time step, and the heat transfer coefficient on the water/steam side in PH, SH and RH is calculated according to Gnielinski correlation [37]. The heat transfer coefficient on the shell side is calculated using the following relation:

$$h_s = h_{s,0} \left(\frac{\dot{m}_{sa}}{\dot{m}_{sa,0}} \right)^{0.6} \quad (4)$$

where $h_{s,0}$ is the heat transfer coefficient of the shell at nominal conditions, obtained as a function of the heat transfer coefficient in the cross-flow zone using the Colburn correlation [38]

and heat transfer coefficient in the window zone using the Singh et al. [39] correlation. Both terms are weighted with respect to their respective heat transfer areas.

3.2. Two-phase flow model for the evaporator and the steam drum

A mathematical model is proposed for the evaporator and the steam drum [13,40]. The thermal inertia of the water/steam mixture and the metal mass are considered. The metal wall temperature of the evaporator and steam drum is assumed to be equal to the steam/water mixture, i.e., the saturation temperature. Similarly, the steam outlet enthalpy is assumed to be equal to the vapor enthalpy. The mass and energy balances in the evaporator-drum control volume lead to the following Equations (5), (6) and (7).

$$\frac{d(\rho_w V_w)}{dt} + \frac{d(\rho_{st} V_{st})}{dt} = \dot{m}_w - \dot{m}_{st} \quad (5)$$

$$\frac{d(\rho_w V_w u_w)}{dt} + \frac{d(\rho_{st} V_{st} u_{st})}{dt} + M_{ev} C_m \frac{dT_{m,ev}}{dt} = \dot{m}_w i_w - \dot{m}_{st} i_{st} + \dot{m}_{sa} i_{sa,i} - \dot{m}_{sa} i_{sa,o} \quad (6)$$

$$\dot{m}_{sa} i_{sa,i} - \dot{m}_{sa} i_{sa,o} = h_{sa,ev} A_{ev} (\bar{T}_{sa,ev} - T_{m,ev}) \quad (7)$$

The saturated water and steam properties are calculated using the XSteam tool [41], which is based on the industrial formulation of the IAPWS IF-97 [42]. The equation system presented above is valid to calculate the dynamic behavior of the steam drum pressure. However, to calculate the dynamic behavior of the steam drum water level, two additional equations are needed [40]. The first equation is obtained from an energy balance in the evaporator. A lumped parameter model is used to describe the dynamic behavior of the tube bundle of the evaporator.

The second equation is obtained by a mass balance on the steam under the liquid level of the steam drum. Finally, a four-equation non-linear system is obtained, which is solved by an explicit scheme using the Runge-Kutta method. A time step of 1 second is used to ensure the numerical stability. Figure 5 shows the model response for different two-step disturbances [40]. Figure 5 shows the model response for a heat step of 10 MWth. The heat step results in an increase of the steam mass fraction (Figure 5-c), which leads to a pressure increase (Figure 5-a). The increase of the pressure (Figure 5-a) produces high steam condensation, as shown in Figure 5-b when the water volume increases. From the drum-level control point of view, the dynamic response of the two-phase problem is illustrated in Figure 5-d. Thermodynamically, the drum-level response is determined by two opposite effects: on the one hand, the reduction of the steam volume during the pressure increase, and on the other hand, the steam condensation increase.

Figure 5. Response of the system for a heat step of 10 MWth: a) steam drum pressure; b) total water volume; c) evaporator outlet steam mass fraction; d) steam drum level.

3.3. Stresses on the steam drum and SH header

A widely used methodology to estimate the maximum heating rates of thick-walled components is proposed by European standard EN 12952-3 [26], in which several concentration factors for different geometries, connections shapes, and loadings can be found. Usually, in conventional boilers, the limiting components are the heads of SH and RH, the steam drum, and T- and Y-shaped junctions in steam pipelines [9]. The total tangential stress is calculated by combining the pressure and thermal stresses using their respective concentration factors:

$$\sigma = \sigma^p + \sigma^T = \alpha_m \frac{D}{2t_j} P_j + \alpha_t \frac{\beta_t E}{1-\nu} (T_{ave} - T_{su}) \quad (8)$$

where T_{ave} is the metal average temperature and T_{su} is the inner surface temperature. The thermal concentration factor (α_t) is a function of the working fluid (steam or water) and the diameter ratio between the main component and the outlet/inlet pipeline. The pressure concentration factor (α_m) mainly depends on the ratio between the diameter and thickness of the main component and the outlet/inlet pipeline. According to Taler et al. [43], two points should be considered in a T-junction connection, as shown in Figure 6. This approach is used to estimate the stress on the T-junction formed by the steam drum-downcomer junction and SH head-nozzle junction.

Figure 6. T-junction points selected.

A transient temperature model is created to estimate the average and inner metal wall temperatures for the steam drum and the SH head (Equation (9)). The model considers that the temperature varies only in the radial direction. A heat transfer boundary condition is used on the inner surface, whereas an insulated boundary condition is considered on the outer surface.

$$\begin{aligned}
\frac{1}{r} \frac{\partial}{\partial r} \left(r \frac{\partial T}{\partial r} \right) &= \frac{1}{a} \frac{\partial T}{\partial t} \\
T(r, 0) &= T_0 \\
-k \frac{\partial T}{\partial r} \Big|_{r=r_i} &= h_j (T_j(t) - T(r_i, t)) \\
\frac{\partial T}{\partial r} \Big|_{r=r_o} &= 0
\end{aligned} \tag{9}$$

Equation (9) is solved numerically using the Crank-Nicolson method. The wall thickness is discretized along the radial coordinate to obtain increments smaller than $5 \cdot 10^{-3}$ m. The time step is calculated according to the von Neumann stability criterion. Once the radial temperature distribution of the metal wall is known, the average wall temperature is calculated according to Kim et al. [13].

3.4. Stresses on the tubesheet

One of the most important parts of the shell and tube heat exchanger is the tubesheet due to its complex geometry [44]. Tubesheet calculations for heat exchangers for standard industrial applications are typically performed according to ASME Section VIII-Division 1 or/and TEMA standards [7]. However, no specific guidelines are shown in TEMA standards or/and ASME code for complex loadings, such as transients thermal loadings [10] or thermal gradient on the junction behavior [11]. For this reason, the method proposed by O'Donnell et al. [45] is used for tubesheet stress calculation. A schematic of the main tubesheet zones is illustrated in [27].

The main equations for the stress calculations proposed by O'Donnell et al. [45] are shown in Table 2. In the first step, the effective elastic constants, which are functions of the ligament efficiency and tube pitch layout, are calculated. In the second step, the pressure stresses are calculated considering an equivalent solid plate [46]. The ligament stress is calculated by combining the pressure stress using Equation (10) and the thermal stress using Equation (14),

considering their stress sign. A considerable temperature difference in the no-tube-lane zone is found in PH, and therefore the stress must be calculated using Equations (15) and (16).

Table 2

Main equations for stress calculations proposed by O'Donnell et al. [45].

Load	Stress intensity	Equation
Pressure and thermal	Average across the ligament at either surface of the plate	$\sigma_{eff} = K \frac{R}{h} \sigma_1 $ (10)
Pressure	Average across the ligament and through thickness	$S_{eff} = \frac{R}{h} \left[\left(\frac{\Delta P r}{H} \right)^2 + (\sigma_r)^2 \right]$ (11)
Pressure and thermal	Peak in ligaments	$\sigma_{max} = Y \sigma_1 + P$ (12)
Pressure and thermal	Peak at perforations adjacent to the rim	$\sigma_{max} = K_r \sigma_{rim} + P$ (13)
Thermal (skin effect)	Peak at the surface	$\sigma_{max} = \frac{E \beta_t (T_{ave} - T_{su})}{1 - \nu}$ (14)
Thermal (temperature difference across the no-tube-lane zone)	Peak in ligaments	$\sigma_{max} = \frac{K_u E^* \beta_t (T_H - T_C)}{2}$ (15)
Thermal (temperature difference across the no-tube-lane zone)	Peak at holes adjacent to the no-tube-lane zone	$\sigma_{max} = \frac{K_D E \beta_t (T_H - T_C)}{2(1 - \nu)}$ (16)

3.4.1. Thermal tubesheet ligament analysis

Under start-up conditions, the Biot number varies from 0.03 to 0.25 in the surface of the tubesheet ligament. When the Biot number is higher than 0.05 in the cylinders, the transient conduction analysis would be convenient [47]. Assuming an equilateral triangle pitch layout, an isolated cylinder is analyzed as shown in Figure 7.

The transient temperature field of the simplified tubesheet ligament model is calculated using Equation (17). Symmetric boundary conditions are considered for the outer wall of the simplified ligament model. The boundary conditions used on the tube, head and shell side are described in Figure 7.

Figure 7. Tubesheet ligament boundary conditions.

$$\frac{1}{r} \frac{\partial}{\partial r} \left(r \frac{\partial T}{\partial r} \right) + \frac{\partial^2 T}{\partial z^2} = \frac{1}{a} \frac{\partial T}{\partial t} \quad (17)$$

Equation (17) is discretized in a grid of nodes where the radial and axial increments are smaller than $5 \cdot 10^{-3}$ m. Equation (17) is solved numerically by a time implicit finite difference scheme. An example of the transient temperature field evolution on the tubesheet ligament is shown in Figure 8. At this point, the surface temperature (T_{su}) and average wall temperature (T_{ave}) are

known, and then the thermal stresses on the tubesheet ligament can be calculated using Equation (14). This approach guarantees a conservative calculation because it is assumed that the tubesheet has perfectly clamped edges.

Figure 8. Transient temperature field in the tubesheet ligament zone.

3.4.2. Tubesheet junction

According to Soler et al. [48], three tubesheet junction types are possible: two-side gasketed, two-side integral and one-side integral with the other side gasketed. In this work, an integral junction type is considered for both the head and shell sides. The stresses on the junctions of the tubesheet are normally estimated using Finite Element Analysis (FEA) because of the complex geometry of the head [44,49]. FEA is outside the scope of this work since the goal of this analysis is to determine the startup and load change times through low-time-consumption simplified models. Therefore, an analytical model is proposed to estimate the stresses of the tubesheet junction. A schematic of the head-tubesheet-shell junction forces and moments is shown in Figure 9. The displacements and forces can be related as a function of their respective

stiffness values as: $F_j = K_j x_j$ or $x_j = K_j^{-1} F_j = K'_j F_j$. Then, the problem can be written in matrix form, and the displacements of the shell and the head can be calculated using Equation (18). The displacements of the tubesheet are calculated using Equation (19).

Figure 9. Schematic of the head-tubesheet-shell junction forces.

$$\begin{Bmatrix} y_h \\ \theta_h \\ y_s \\ \theta_s \end{Bmatrix} = \begin{bmatrix} K'_{cyl,My} & K'_{cyl,Vy} & 0 & 0 \\ K'_{cyl,M\theta} & K'_{cyl,V\theta} & 0 & 0 \\ 0 & 0 & K'_{cyl,My} & K'_{cyl,Vy} \\ 0 & 0 & K'_{cyl,M\theta} & K'_{cyl,V\theta} \end{bmatrix} \begin{Bmatrix} M_h \\ V_h \\ M_s \\ V_s \end{Bmatrix} + \begin{bmatrix} K'_{cyl,t} & 0 \\ 0 & 0 \\ 0 & K'_{cyl,t} \\ 0 & 0 \end{bmatrix} \begin{Bmatrix} T_{su,h} \\ T_{su,s} \end{Bmatrix} + \begin{bmatrix} K'_{cyl,p} & 0 \\ 0 & 0 \\ 0 & K'_{cyl,p} \\ 0 & 0 \end{bmatrix} \begin{Bmatrix} P_h \\ P_s \end{Bmatrix} \quad (18)$$

$$\begin{Bmatrix} y_{ts,h} \\ \theta_{ts,h} \\ y_{ts,s} \\ \theta_{ts,s} \end{Bmatrix} = \begin{bmatrix} 0 & 0 & 0 & 0 \\ -K'_{ts,M} & -\frac{L_{ts}}{2} K'_{ts,M} & -K'_{ts,M} & -\frac{L_{ts}}{2} K'_{ts,M} \\ 0 & 0 & 0 & 0 \\ -K'_{ts,M} & -\frac{L_{ts}}{2} K'_{ts,M} & -K'_{ts,M} & -\frac{L_{ts}}{2} K'_{ts,M} \end{bmatrix} \begin{Bmatrix} M_h \\ V_h \\ M_s \\ V_s \end{Bmatrix} + \begin{bmatrix} K'_{ts,t,y} & 0 \\ K'_{ts,t,\theta} & -K'_{ts,t,\theta} \\ 0 & K'_{ts,t,y} \\ K'_{ts,t,\theta} & -K'_{ts,t,\theta} \end{bmatrix} \begin{Bmatrix} T_{su,h} \\ T_{su,s} \end{Bmatrix} + \begin{bmatrix} \frac{L_{ts}}{2} K'_{ts,p} \\ K'_{ts,p} \\ -\frac{L_{ts}}{2} K'_{ts,p} \\ K'_{ts,p} \end{bmatrix} (P_h - P_s) \quad (19)$$

The stiffnesses, K_i , for a cylinder and solid disk for pressure and thermal loads are calculated according to [50, 51]. The stiffness of the tubesheet is modified as an equivalent solid plate using the equivalent elastic modulus and Poisson's ratio as a function of the ligament efficiency and pitch layout according to ASME code [8]. In the first step, the free displacements (y_j, θ_j) on the tubesheet, head and shell are calculated with consideration of only the pressure and

thermal loads, i.e., without the interaction of the moments and forces. In the second step, the moments and forces are calculated to achieve the compatibility of the displacements.

3.4.3. Thermal tubesheet junction analysis

A numerical model based on Equation (17) is also carried out to calculate the transient temperature field on the tubesheet junction. A schematic of the boundary conditions is shown in Figure 10, whereas an example of the transient temperature field evolution on the tubesheet junction is shown in Figure 11. As seen, the transient effect or characteristic time is longer for the tubesheet junction than the tubesheet ligament. This leads to higher variations between the peak and valley of thermal stresses during the start-up or load change processes.

Figure 10. Schematic of the tubesheet junction boundary conditions.

Figure 11. Transient temperature field in the tubesheet junction.

The surface wall temperatures are dependent on the heat transfer coefficient due to the heat transfer boundary conditions used. The complex calculation methods needed to estimate the heat transfer coefficients on the head and shell side make it necessary to simplify its estimation. Alternately, the heat transfer coefficient on the tubesheet of the shell sidewall is calculated as

one order of magnitude lower than the cross flow on the shell side. However, the heat transfer coefficient on the tubesheet of the head sidewall is considered to be one order of magnitude lower than the heat transfer coefficient of the tube side. These assumptions are in accordance with the tubesheet simulation performed by Shen et al. [49].

The membrane stresses of the head and shell at the junction are calculated using the equations presented in [51]. Once the moments are known, the bending stresses of the head and shell can be estimated using Equation (20). The stresses at the tubesheet-head junction are calculated using a conservative approach combining the tangential head transient thermal stress with the axial head bending stresses using the Von Misses equation [8]. The stresses suffered on the tubesheet-shell junction are calculated considering only the peak thermal stress using Equation (14).

$$\sigma_b = \pm \frac{6M}{t^2} \quad (20)$$

3.5. Stresses on U-tubes

Although U-tube design is used to avoid the differential thermal expansion between the shell and tubes, thermal stresses in the U-bend appear since the tubes are supported by segmental baffles. Because the tubes are not free to expand, the tubes lose their straight shape, producing contacts with the segmental baffle supports, as shown in Figure 12. The minimal U-bend radius can be calculated such that it does not overpass the maximum stress limit working at design conditions. However, this stress level is strongly dependent on the temperature profiles of the hot and cold fluids along the heat exchanger, which change during transient operation. For this reason, transient analysis is required for the U-tubes.

Figure 12. U-tube deformation by thermal expansion.

An analytical model proposed by Singh et al. [52] is used to calculate the thermal stress on the U-tubes. The model can be written in matrix form based on the stiffness values of the U-tubes, which are mainly functions of the displacements, baffle spacing, tube length, modulus of elasticity and moment of inertia. The displacement of the hot leg over the cold leg can be calculated using Equation (21), assuming the axial force $F = 0$. The problem can be solved via an iterative process until convergences of the displacements, forces and moments are reached.

$$\delta = (\overline{\beta_{hot} \Delta T_{hot}} - \overline{\beta_{cold} \Delta T_{cold}}) L - \frac{2F}{AE} L \quad (21)$$

Once the forces and moments are known, the moment function ($M(\theta)$) in the U-bend can be calculated. Then, the stress is calculated as follows:

$$\sigma(\theta) = \frac{\psi M(\theta) r_o}{I} \quad (22)$$

where ψ is the stress intensification factor for the U-bend and r_o is the outer-tube radius. An example of the SH U-bend thermal stress along the angle θ for different temperature differences between the hot leg and cold leg is shown in Figure 13. As seen, the maximum stress on the SH U-bend is obtained for $\theta = 0^\circ$.

Figure 13. SH U-bend stress for different temperature differences between the hot leg and cold leg.

4. Results

The results of applying the proposed methodology to an SG for two start-up conditions are presented in this section. First, an analysis of the SG performance for two different feedwater heat-up systems is carried out: i) with water recirculation from a steam drum; and ii) with steam recirculation from a steam drum using a start-up feedwater heater. Then, the results of the daily start-up with non-isothermal conditions and isothermal conditions are presented. For both start-up procedures, an analysis of the stresses of the main components is also presented.

4.1. Feedwater temperature

Under start-up conditions, the feedwater heater system is not able to provide the feedwater at its full-load temperature set point. To solve this issue, a back-up system, such as auxiliary heaters, may be installed [53]. In the first step, the feedwater heating system described in [20] is adopted. This procedure basically consists of heating the feedwater at 245°C before the preheater inlet by mixing hot water from the drum (290°C). As seen in Figure 14-a/b, high water

mass flow rate ($\dot{m}_{w,rec}$) must be circulated from the drum to maintain the preheater inlet-temperature set-point. This high mass flow rate excessively reduces the temperature of the preheater salt outlet up to ($T_{PH,sa,o} \approx 250^{\circ}\text{C}$), increasing the potential risk of salt freezing. For this reason, a different feedwater heating system is needed.

Based on the Solar Two operation experience, Pacheco et al. [22] proposed the use of a start-up feedwater heater (Figure 2), in which auxiliary steam is extracted from a steam drum to heat the feedwater. Furthermore, the preheater inlet temperature set point is increased from 245°C to 260°C . This alternative mode is implemented, and the results are shown in Figure 14-c/d. As seen in Figure 14-c/d, the circulated steam mass flow from the drum ($\dot{m}_{st,rec}$) is considerably lower than the water circulation system ($\dot{m}_{FW}$), although the preheater inlet temperature set point has been increased. Now, the preheater salt outlet temperature is approximately $T_{PH,sa,o} \approx 270^{\circ}\text{C}$, which can be considered safe operation.

Figure 14. Feedwater system performance: a, b) without a start-up feedwater heater; c, d) with a start-up feedwater heater.

4.2 Daily start-up with non-isothermal initial conditions

For daily non-isothermal initial conditions, the heat exchangers of the SG are closed after the nightly shutdown and thermally isolated to reduce the heat losses [20]. Because they are filled with hot salt, each heat exchanger has different temperatures, as was described in Figure 3.

Figure 16 shows the time evolution of the main thermodynamic variables during start-up and steady-state of the SG. In addition, a schematic of the startup procedure, where the main steps are described, is shown in Figure 15. Both Figures 16 and 17 differentiate between the start-up and nominal conditions by a vertical line. It is assumed that the TES system has enough energy to initialize the start-up of the SG.

In the first step, the hot and cold salt pumps provide a mass flow rate of 4% of the nominal flow to the SG [20]. The salt is sent to EV at 448°C by the bypass line, which is the design salt inlet temperature. The salt also circulates to the PH. The steam generated is sent to the start-up

feedwater heater, in which the feedwater is heated to 260°C when it enters the PH. It is assumed that the feedwater available at the beginning of the start-up is at 100°C from the deaerator. The steam drum pressure is kept constant at 7.4 MPa until the turbine achieves the sliding pressure conditions (Figure 16-c).

After $t = 4$ min, the salt is sent to SH and RH at 565 °C (Figure 16-a), when the steam generation is established at 4%. At the same time, the saturated steam from the steam drum is sent to the SH, and outlet steam is sent to the RH by the SH-RH by-pass line, where the attenuation system keeps the RH steam inlet temperature at 365°C. Moreover, an additional valve is used in the SH-RH by-pass line to reduce the RH inlet pressure to an allowable range. The RH outlet steam is sent to the deaerator. During this process, the salt flow rate is increased to 5%.

When the steam generation is set to 5% and the SH and RH outlet steam temperature is maintained at 496°C (Figure 16-b/d), the steam is sent to the HP and LP turbines, respectively [20]. Then, the SH to RH steam by-pass line is closed. This process is continued until the turbine synchronization is completed, which takes approximately 15 minutes [20].

At $t = 24$ min, the turbine synchronization is completed, and the SH and RH steam outlet temperature is increased from 496°C to 550°C (Figure 16-b/d). The salt mass flow rate is increased by a ramp of 5%/min until it reaches 50% of its nominal value (Figure 16-a). At this load point, the turbine is switched from fixed pressure mode to sliding pressure mode. Now, the salt flow rate ramp is kept to 5%/min until the salt flow rate reaches its nominal value. At the same time, the pressure on the steam drum starts to increase, and as a consequence of the sliding pressure mode, the steam mass flow rate increases too (Figure 16-c).

Full-load conditions are achieved after 50 min. The energy delivered from the TES system and the required mass of hot salt to start up with non-isothermal initial conditions is approximately 70 MWth and 600 tons, respectively. The energy consumption of the salt pumps is approximately 45 kWhe.

Figure 15. Schematic of the procedure of the start-up with non-isothermal initial conditions.

Figure 16. Non-isothermal start-up simulation: a) salt temperature and mass flow rate, turbine power output; b) SH performance; c) steam drum performance; d) RH performance.

Typically, the stress is limited to avoid material failure over its operation lifetime. Following ASME code [8], two different mechanisms of damage are calculated: creep and fatigue. The fatigue damage is related to the magnitude of the stress cyclic variations, and the number of cycles to failure is computed as a function of the equivalent strain range [8]. The creep damage is a function of the stress level and the temperature. However, detailed calculation of the material life requires the operating history data and this is outside the scope of this work. Therefore, a conservative stress limit equal to 40% of the Ultimate Tensile Strength (UTS) is imposed. This stress limit value considers both fatigue and creep damages.

The stress evolution of the SG components is shown in Figure 17. The stress evolution can be divided into two groups. Group 1 (Figure 17-a/b/c/e/f/g) refers to the parts where the thermal inertia is significant, and therefore a delay is found between the wall inner surface temperature and the average wall temperature. These stresses are produced in the tubesheet ligament, tubesheet-head junction, tubesheet-shell junction, steam drum and inlet and outlet SH heads. Group 2 (Figure 17-d/h) refers to the parts where the thermal inertia can be neglected. These stresses are produced in the no-tube-lane tubesheet zone and U-tubes.

For the non-isothermal start-up, the most limiting stresses are marked with a red circle in Figure 17. These stresses are obtained in the SH lower tubesheet-shell junction (Figure 17-c) when the pressure increases on the steam drum (Figure 16-c). Although the highest stresses are obtained in the SH and RH U-tubes (Figure 17-h), these values do not limit the start-up ramp. First, it should be noted that the highest U-tube stresses correspond to U-tubes with the lowest radius. Second, this problem is independent of the temperature change rate since it is only a function of the temperature profile state along the heat exchanger. Then, this issue can be solved by increasing the minimal U-bend radius. However, the most interesting result is that the SH and RH U-tubes are more stressed during partial-load operation than during full-load operation. This is caused by the higher temperature differences between the hot leg and cold leg under partial-load conditions.

Figure 17. Stress evolution of the main components during non-isothermal start-up: a) tubesheet ligament stresses; b) tubesheet head junction stresses; c) tubesheet shell junction stresses; d) tubesheet no-tube-lane stresses; e) SH inlet head nozzle junction stresses; f) SH outlet head nozzle junction stresses; g) steam drum downcomer junction stresses; h) U-tube stresses.

4.3 Daily start-up with isothermal initial conditions

For a daily start-up with isothermal initial conditions, it is considered that, after the plant shutdown, cold salt is pumped through the heat exchangers during the night. In this manner, the temperature is kept constant at 290°C in the entire SG at the beginning of the start-up process. The steam drum pressure is kept constant at 7.4 MPa. During this period, the potential condensation of the steam SH and RH should be taken to the deaerator. A schematic of the start-up procedure and the time evolution of the main parameters are shown in Figures 18 and 19, respectively. The isothermal and non-isothermal SG conditions have practically the same start-up procedure since both are considered as a daily start-up, and thus the turbine inlet conditions are the same.

In the first step, salt is sent to the EV through the by-pass line at 350 °C to start the heat-up process of the EV and PH. When the steam production is established at approximately 4%, the salt and steam valves are opened in the RH and SH. The salt temperature change rate is set to 150°C/hr (Figure 19-a).

At $t = 67$ min, the SH and RH steam temperatures reach 496 °C, and these steam flows are sent to the HP and LP turbine stages to start the synchronization. At the end of the turbine synchronization, the steam outlet temperatures have increased to 550°C (Figure 19-b/d). At this point, the salt flow rate is increased by a ramp of 5%/min until the sliding pressure turbine operation condition is achieved. Then, the salt flow rate ramp is increased by 5%/min until full-load conditions are achieved.

The start-up process finishes after 110 min when full-load conditions are obtained. The energy delivered from the TES system to accomplish the start-up with isothermal initial conditions is

approximately 85 MWth, whereas the hot-salt mass required is 716 tons. The energy consumption of the salt pumps is approximately 53 kWhe.

Figure 18. Schematic of the start-up procedure with non-isothermal initial conditions.

Figure 19. Isothermal start-up simulation: a) salt temperature and mass flow rate, turbine power output; b) SH performance; c) steam drum performance; d) RH performance.

The stress evolutions of the main components of the SG are depicted in Figure 20. As seen in Figure 20, the highest stress values are obtained in the SH upper tubesheet-head junction (Figure 20-b), the SH lower tubesheet-shell junction (Figure 20-c) and the U-tubes in the SH and the RH (Figure 20-h).

At the beginning of the start-up, the limiting stress is found in the SH upper tubesheet-head junction (Figure 20-b). Although the salt enters the RH at a high temperature, the greater thickness of the SH leads to higher thermal stresses in the SH than in the RH. It should be noted that the stresses on the SH tubesheet-head junction are higher than in the ligament. This is due

to the higher thermal inertia values of the SH tubesheet-head junction, which leads to higher delay between the surface temperature and the average wall temperature.

When the sliding pressure condition is achieved at $t = 93$ min, the pressure in the steam drum begins to increase, and as a result, the stresses on the steam drum and SH lower tubesheet grow (Figure 20-c). At this point, the SH lower tubesheet-shell junction is the most limiting part. Moreover, the transient thermal stresses are higher in the tubesheet junction (Figure 20-c) than in the head at the nozzle point (Figure 20-e). This is caused by the greater thickness on the head at the tubesheet junction than in the head at the nozzle point, which is made to reduce the bending stress of the head at the tubesheet junction point. The highest U-tube stress values in the SH and the RH (Figure 20-h) are obtained when the highest temperature difference between the salt inlet and the steam inlet appears.

As exhibited by the non-isothermal start-up, the maximum allowable stress limit of 40% of UTS is overpassed. However, as mentioned before, the stresses of the U-tube do not limit the start-up ramp since this problem can be easily solved by increasing the U-bend radius. Another possible solution would be an increase of the outside tube diameter, followed by the tube thickness and their stiffness values.

Figure 20. Stress evolution of the main components during isothermal start-up: a) tubesheet ligament stresses; b) tubesheet head junction stresses; c) tubesheet shell junction stresses; d) tubesheet no-tube-lane stresses; e) SH inlet head nozzle junction stresses; f) SH outlet head nozzle junction stresses; g) steam drum-downcomer junction stresses; h) U-tube stresses.

Suitable design of the steam generator is highly conditioned by the plant operation. The example SG considered in this work employs similar operating conditions to Crescent Dunes [6] in terms of turbine power and operation with daily start-ups. The start-up procedure proposed, with conservative ramp rates, can be considered suitable for a baseload SPTP that operates with a fixed daytime tariff.

In the opposite situation are the plants with large thermal storage tanks that can work at full load 24 h per day, such as Gemasolar [6]. This operational condition significantly reduces the number of start-ups and thus increases the lifetime of the steam generator components.

Solar power tower plants that operate to meet their electricity production to peak demands increase their dispatchability by employing fast ramps in the SG train. To obtain realistic results, the reduction of the lifetime due to fatigue failure should be considered together with the increment of the profits obtained by the higher dispatchability. The proposed methodology can be used as a powerful tool to either design or optimize the plant operation from the thermo-mechanical response of the SG. This allows the study of different strategies that can reduce the stress levels in the SG for the plant requirements. In addition, the low computational costs of the models enhance their use in real-time monitoring to help in the development of feedback control architectures.

5. Conclusions

A thermos-mechanical methodology for the analysis of steam generators for solar power tower plants is proposed. The steam generator design consists of conventional shell and tube heat exchangers based on TEMA and ASME standards. For the first time, a complete methodology is proposed to solve the coupled dynamic behavior of the steam generator train. The stresses can be calculated in real-time since they are based on low-time-consumption analytical models. This can be used as a powerful tool to calculate the lifetime of the steam generator components and then optimize their design and operation.

Different dynamic models of the main components of the steam generator are developed to estimate the temperature field and the stresses of the most important parts of the shell and tube

heat exchangers, such as the tubesheet and the tubesheet junction. The European standard EN 12952-3 [26], which is normally used for conventional boilers, is used to calculate the stress values of the steam drum and superheater heads. A simple simulation control scheme is applied to obtain the transient response of the system.

Two start-up strategies are studied as examples of the proposed methodology. The first one is based on the Foster Wheeler manufacturer approach [20], in which the heat exchanger valves are closed after the plant shutdown, and thus the initial temperature profile is non-isothermal. The results show that the steam generator start-up takes approximately 50 minutes to achieve full-load conditions. A mass of hot salt of 600 tons (70 MWth) and a consumption of 45 kWhe in the molten salt pumps are needed for this start-up.

The second strategy studied is based on the operating experience of the solar power tower plant "Solar Two" [22]. Here, the steam-generator temperature profile is isothermal, 290°C at the beginning of the start-up. In this case, cold salt is circulated through the steam generator during the night to reduce the risk of salt freeze. The results show that the steam generator needs approximately 110 minutes to achieve full-load conditions. A mass of hot salt of 716 tons (80 MWth) and a parasitic consumption of 53 kWhe of the molten salt pumps are needed.

The main input variables, such as the mass flow and the temperature change rates, are calculated for both strategies to avoid the overpassing of the stress limits on the proposed steam generator. In both start-up strategies, it is found that the most constrained part of the steam generator is the SH tubesheet junction.

Furthermore, two start-up feedwater heating strategies are compared. The first is based on water drum recirculation, and the second is based on steam recirculation. Although the second requires an additional start-up feedwater heater, the reduction of the potential salt freeze risk due to the higher outlet preheater salt temperature may compensate the initial investment. Moreover, high mass flow rates of water are required in the drum circulation systems to achieve the SG feedwater inlet temperature, and this may lead to higher operational pump costs.

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