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Dynamic performance and stress analysis of the steam generator of parabolic trough solar power plants

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Abstract

The thermal stress on thick-walled components, such as tubesheets and steam drums, limits both the temperature ramp-up rates and the temperature differences between outer and inner walls. The cyclic operation of concentrating solar power plants may lead to fatigue damage. For these reasons, a stress analysis of the steam generator is required to assure its lifetime.

A methodology is presented for the thermo-mechanical analysis of the steam generator for a parabolic trough power plant. This methodology consists of coupling transient thermodynamic and stress models of the heat exchangers in order to calculate the stress. Besides the heat exchanger models, a transient model for a TEMA H heat exchanger is proposed. Finite element simulations are carried out to calculate the deviations of the simplified analytical models. In this way, a powerful tool that allows the analysis and optimization of the steam generator operation is proposed.

The results suggest that U-tube tubesheets are exposed to high thermal stresses on the no-tube-lane zone, especially in the reheater. The steam generator start-up can be accomplished in around 45 minutes using 36.4 MWh_{th}. Furthermore, the TEMA X evaporator presents a thermal stress reduction of 35% compared to the kettle evaporator.

Key words: *Steam generator; Start-up; Thermal Stress; Parabolic trough power plant; Solar thermal power.*

Nomenclature

Abbreviations

CCPs : combined-cycle plants

CSP : concentrating solar plants.

CT : cold tank.

DDTJ : steam drum-downcomer T-junction.

EV : evaporator.

36 FW : feed-water.
37 $HNTJ$: head-nozzle T-junction.
38 HP : high pressure.
39 HPT : high pressure turbine.
40 $HRSG$: heat recovery steam generator.
41 HTF : heat transfer fluid.
42 HT : hot tank.
43 H_x : heat exchanger.
44 LP : low pressure.
45 LPT : low pressure turbine.
46 NTL : no-tube-lane.
47 PH : preheater.
48 $PTPP$: parabolic trough power plant
49 RH : reheater.
50 SG : steam generator.
51 SH : superheater.
52 TSJ : tubesheet junction.
53 $TSJ - HS$: tubesheet junction at the head-side.
54 $TSJ - SS$: tubesheet junction at the shell-side.
55 UTS : ultimate tensile strength.

56

57 **Symbols**

58 A : heat transfer area (m^2)
59 A_c : cross section area (m^2)
60 a : thermal diffusivity (m/s^2)
61 C_m^* : dimensionless thermal capacitance of metal wall (-)
62 D : diameter (m), dimensionless time (-)
63 E : modulus of elasticity (MPa)
64 E^* : equivalent modulus of elasticity (MPa)
65 F : force (N)
66 h : convective coefficient ($W/m^2\ ^\circ C$)
67 I : moment of inertia (m^4)
68 i : specific enthalpy (J/kg)
69 K : stiffness (N/m)
70 K' : inverse of stiffness (m/N)
71 L : length (m), dimensionless length (-), level (m)

72	L_p : tube pitch (mm)
73	L_s : tubesheet thickness (m)
74	M : mass (kg), moment (N m)
75	$\dot{m}$: mass flow rate (kg/s)
76	P : pressure (Pa)
77	R : radius (m)
78	R_{ol} : radius of a circle circumscribed to outermost tube of a bundle (m)
79	r : radial coordinate (-)
80	T : temperature (°C)
81	T^* : dimensionless temperature (-)
82	t : time(s)
83	t_j : thickness of component j (m)
84	t^* : dimensionless time(-)
85	u : specific internal energy (kJ/kg)
86	V : volume (m ³), vertical force (N)
87	W_t : turbine power (MWe)
88	y : vertical displacement (m)
89	z : axial coordinate (-)

90

Greek Symbols

92	α_t : thermal stress concentration factor (-)
93	α_m :pressure stress concentration factor (-)
94	β_t :linear thermal expansion coefficient (mm/mm/K)
95	θ : rotation angle (rad), circumferential coordinate (-)
96	ν : Poisson's ratio (-)
97	ρ : density (kg/m ³)
98	σ : stress (MPa)
99	τ : shear stress (MPa)
100	ψ : U-tube stress concentration factor (-)

101

Subscripts

103	0 : nominal conditions
104	a : allowable
105	ave : average

106 *atemp* : attemperator
 107 *b* : bending
 108 *cyl* : cylinder
 109 *d* : drum
 110 *dc* : downcomer
 111 *h* : head
 112 *i* : inlet, inner
 113 *ini* : initial
 114 *m* : metal
 115 *o* : outlet, outer
 116 *r* : riser
 117 *s* : shell
 118 *sat* : saturated
 119 *st* : steam
 120 *su* : surface
 121 *t* : tube
 122 *ts* : tubesheet
 123 *w* : water
 124

125 **1. Introduction**

126 Concentrating solar power (CSP) plants with thermal energy storage have become promising
 127 alternatives to traditional power plants to generate electricity. In order to further enhance the use
 128 of CSP plants in grid regulation services and increase the annual electricity production, it is
 129 necessary to increase their flexibility by means of fast start-ups or load changes [1,2].
 130 Nevertheless, fast transient operations induce high thermal stresses on thick-walled components.
 131 And combining them with the typical operation of CSP plants, with daily start-ups, shut-downs
 132 and load changes, may compromise the structural integrity of critical components like the steam
 133 generator (SG) [3]. Under these operational conditions the use of on-line monitoring control is
 134 required [4]. Sometimes the thermal stress cannot be measured directly and then it has to be
 135 calculated by means of analytical models. In this way, the stress estimation becomes the first
 136 step to control the lifetime reduction of thermal units like SG.

137 Several studies can be found about the stress analysis of the start-up of conventional power
 138 plants [5,6]. In these plants the steam drums and superheater headers are normally considered
 139 the most critical parts. The stresses of these components are typically estimated using EN
 140 12952-3 [7] by means of simplified analytical models due to the relative low complexity of their
 141 geometry. For example, Taler et al. [8] developed an innovative method to optimize the start-up
 142 of fired-boilers focused on the steam drum stresses. This method consists of the construction of
 143 an analytic stress function of the steam drum by means of finite element analysis results. Then,
 144 the negative thermal stress, which compensates for the positive pressure stresses, can be
 145 calculated by using an analytic function. As a result, a meaningful reduction of the start-up time
 146 was obtained. Sun et al. [9] performed a stress analysis of the U-tube steam generator of a

nuclear power plant. Simulations by finite element method were carried out using ANSYS software, coupling the mechanical and thermal loads. The results indicated that the highest stresses are obtained at the contact points of the tube support plates, which are highly susceptible of stress corrosion and fatigue damage.

Similar studies about the dynamic behavior of heat recovery steam generators (HRSGs) of combined cycled plants (CCPs) are also available. For example, Kim et al. [10] analyzed the dynamic behavior of a HRSG during a cold start-up. Only the steam drum was considered as the critical stress point. Different strategies were developed by means of exhaust gas turbine bypass to reduce the start-up time and maintaining the steam drum stress in allowable ranges. Sindareh-Esfahani et al. [11] studied the cold startup of an HRSG and optimized the start-up time based on a model predictive control approach. In order to assure the lifetime of the HRSG components, the model considered the thermo-mechanical constraints of the steam drum, superheater and economizer.

Normally the SG of CSP plants consists of shell-and-tube heat exchangers as was shown in Refs. [12,13]. Extensively used methods for the heat exchanger design are TEMA standards [14] and ASME Section VIII Div 1 [15], which consider continuous operation and do not consider fatigue due to cyclic operation [16]. In addition, these standards do not address specific guidelines for the stress estimation of typical loads of a SG, e.g. thermal loads on U-tube tubesheets [13] and temperature gradients on the tubesheet junctions [17]. In such cases it is necessary to carry out a finite element analysis in order to assure an appropriate SG design.

Several studies can be found about the dynamic performance of CSP plants. For instance, Al-Maliki et al. [18] investigated the dynamic performance of a parabolic trough power plant (PTPP) under extremely cloudy days. The model was developed in APROS (Advance Process Simulation Software) and validated with operational data from Andasol II. Several details of the control system for the solar field and power block were considered. However, thermo-mechanical constraints of the steam generator or steam turbine were not shown. Ferruzza et al. [19] studied the impact on the yearly energy production of a PTPP for various heating rates of the evaporator, superheater and turbine. The results indicated that the optimal evaporator heating rate should be in the range from 7 °C/min to 10 °C/min. However, details of the SG design, control or dynamic modeling are not available.

In this work, a novel methodology for the thermo-mechanical analysis of the SG of a PTPP is proposed. For the first time, a dynamic model of a TEMA H heat exchanger is presented. Dynamic models for heat exchangers TEMA F, and TEMA X recirculation evaporator with steam drum are also presented. Several analytical models were developed to estimate the stresses on the critical parts of the SG. The methodology is based on models with short computational time, thereby providing a powerful tool for the evaluation of technical constraints of the SG design and optimization of the SG operation. The accuracy of the analytic stress models were evaluated by comparing their results with those of a finite element analysis software (ABAQUS/CAE software [20]). A PID control system was developed to guide the SG start-up. A stress analysis was performed for the start-up of the SG according EN 12952-3[7] and ASME VIII Div 2 [15]. Finally, a stress analysis was made to compare the TEMA X and kettle evaporators. This methodology has been used as a basis for the fatigue analysis of the SG of a PTPP presented in González-Gómez et al. [21].

The primary novel contributions of the paper are the following: i) a methodology, based on models with short computational time, for the evaluation of stresses in steam generators for parabolic trough power plants experiencing highly fluctuating operation; ii) a complete stress

analysis of the steam generator for parabolic trough power plants identifying the critical points with respect to stresses; iii) a dynamic model of the TEMA H heat exchanger.

2. Plant description

A layout similar to the contemporary European parabolic trough power plants is selected for the study [22]. The main subsystems are the following: solar field, storage system, steam generator and power block. The solar field is composed of parallel loops of parabolic trough collectors, where solar radiation is concentrated to heat-up a thermal oil to 393 °C. The solar field is sized to obtain a solar multiple of 2.

The thermal storage consists of an indirect system with molten salt formed by two tanks (cold and hot). The thermal storage system is sized to 7.5 hours of full load operation of the power block. The thermal energy from the solar field is transferred by the thermal oil to the molten salt using oil-to-salt heat exchangers. As a result of this operation the hot tank is charged with hot salt. The flow direction can be reversed for discharge mode and delivered to the power block in a controlled way.

The power block is based on a regenerative reheat Rankine cycle with a 55 MWe (gross) turbine power. Table 1 shows the main design parameters of the power block and SG. The feed-water preheating system consists of five closed heaters (2 high-pressure, 3 low-pressure) and a deaerator. The low-pressure turbine steam outlet is sent to the condenser where heat is rejected. This configuration attains a power block efficiency of 37.5% [12].

Table 1
Power block and steam generator design parameters.

Parameter	Value
Steam turbine power	55 MWe
Thermal oil inlet temperature	393 °C
Thermal oil mass flow rate	593 kg/s
Thermal oil outlet temperature	293 °C
Main steam temperature	377 °C
Main steam mass flow rate	61.91 kg/s
Main steam pressure	10.6 MPa
Reheat steam temperature	378 °C
Reheat steam mass flow rate	50.19 kg/s
Reheat steam pressure	1.83 MPa
Feed-water temperature	249.2 °C
Condenser pressure	0.0078 MPa

The SG consists of two parallel trains each one formed by the superheater (SH), reheater (RH), and evaporator (EV) and preheater (PH), respectively. The thermal oil flows on the shell-side in SH, RH and PH whereas the high-pressure water/steam flows on the tube-side [12]. On the contrary, in the EV, the low pressure oil is situated on the tube-side and the water/steam flows on the shell-side [12]. The PH and SH consist of TEMA H shell type heat exchangers. The TEMA H design is selected to reduce the power pump consumption. A TEMA F shell RH type is selected to increase the thermal efficiency since the flows are counter-current. Finally, a U-tube/TEMA X shell type recirculation evaporator with steam drum is selected for the EV. The materials of the SG heat exchangers are the following: i) A266, Gr. 2 for the shell, head and tubesheet, ii) A210, Gr. A1 for the tubes, iii) A516, Gr. 70 for the steam drum.

2.1. Steam generator control system

A control system was developed to model the dynamic behavior of the SG during the start-up operation. In Figure 1 is illustrated a simplified scheme of the control system. A PID control architecture was selected to guide the start-up process. It should be taken into account that the control system has a great influence on the dynamic response on the SG [23], therefore the controller parameters (proportional and integral) were calculated to obtain suitable response velocities.

The SH and RH outlet steam temperatures were kept under feasible temperature range by means of the attenuation control loop. This is made to assure the safe operation of the steam turbine. The RH attenuation system is placed at the inlet of the RH to avoid wet steam at the outlet of the low pressure turbine [24]. A SH and RH thermal oil by-pass control loop was included to keep the energy balance at part-load conditions. The excess of thermal oil is sent to the EV. If the turbine synchronization is not completed, the steam drum pressure increases and must be regulated by the steam by-pass line to the deaerator. When the turbine synchronization is completed the power block operates in sliding pressure mode. Lastly, the steam drum level is regulated by means of the feed-water.

Figure 1. Steam generator control schematic.

2.2. Power block model

The power block transient performance was modeled as series of steady-state simulations runs of different loads. The model calculates the pressure, mass flow rate and temperature of the main variable inputs of the SG: the feed water and reheat inlet steam. The model also provides the turbine power output.

3. Dynamic models for heat exchangers

The models used for the dynamic simulation of heat exchangers are described in this section. The dynamic models for TEMA F and TEMA H heat exchangers are presented. Then, the two-phase flow models of the TEMA X evaporator and steam drum are shown.

3.1. TEMA F transient model for the reheater

A TEMA F shell-type heat exchanger was selected for the RH. The TEMA F shell was modeled as a counter-current heat exchanger using the following assumptions:

- The conduction wall resistance is negligible in comparison with the convection resistance.
- The longitudinal conduction of the tube wall is neglected.
- The heat losses to the ambient are negligible.
- There is an insignificant leakage of fluid or heat transfer through or around the horizontal baffle.
- The heat transfer coefficients of both fluids are constant throughout the heat exchanger length.

For a differential length of the heat exchanger, the energy balances in the HTF side, metal tube wall and steam side are expressed according Equations 1, 2 and 3, respectively:

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

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

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

where $L = \frac{C \dot{m}}{h A}$ is the dimensionless length and $D = \frac{\rho C A_c}{h A}$ is the dimensionless time. The validation of the finite difference model was carried out using the analytical model results obtained by Ansari and Mortazavi [25]. The numerical model showed good agreement with a maximum deviation of 7.5% on the outlet temperatures [26].

3.2. TEMA H transient model for the superheater and preheater

A TEMA H shell heat exchanger was used for the SH and PH. A schematic of the TEMA H shell is shown in Figure 2a. The TEMA H shell is divided into 8 sections, where each one can be classified as counter-current or co-current heat exchanger. A transient model was obtained to describe the heat exchanger following the approach shown for the counter-current RH. Compatibility temperature equations were added to joint counter-current and co-current parts of

the TEMA H heat exchanger. A discretization model of the TEMA H shell heat exchanger is depicted in Figure 2b.

(a)

(b)

Figure 2. Schematic of TEMA H heat exchanger: a) details of the heat exchanger geometry; b) discretization model.

A comparison of the transient responses of the outlet temperatures between TEMA F and TEMA H shell for an inlet temperature step ($T^* = \frac{T - T_{ini}}{T_0 - T_{ini}}$) for different wall capacities is shown

in Figure 3. The comparison is made for different heat capacities ($C_m^* = \frac{C_m \rho_m}{C_{hot} \rho_{hot}}$). As can be seen,

the transient response of the cold fluid for TEMA H shell is slightly slower than that of the TEMA F shell, Figure 3a, while the hot fluid shows the fastest response on the TEMA H shell, Figure 3b.

Figure 3. Transient responses of the outlet temperatures of the TEMA F and TEMA H shell of an inlet temperature step for different wall capacities: a) cold fluid; b) hot fluid.

No experimental nor numerical data are available to validate the transient response of the TEMA H shell. For this reason, a steady-state heat balance for the hot and cold fluids is made to validate the model, obtaining a difference less than $3 \cdot 10^{-5} \%$. The properties of both fluids were re-calculated during the simulation. The heat transfer coefficient on the water/steam side was calculated using the Gnielinski correlation [27]. The heat transfer coefficient on the shell-side was corrected assuming a shell mass flow ratio to the power of 0.6 [28]. The heat transfer coefficient for nominal conditions was calculated taking into account two different zones: (i) cross-flow zone, in which it was calculated according to the Colburn correlations [29]; (ii) window zone, in which it was calculated using the correlation proposed by Singh et al. [30].

3.3 TEMA X recirculation evaporator model

The TEMA X recirculation evaporator and the steam drum were modeled according to Refs. [10,31]. The model considers the inertia of the water/steam mixture and the metal mass. Moreover, the model assumes that the metal wall temperature of the evaporator and the steam drum is equal to the saturation temperature of the steam/water mixture. The fluid properties were determined using method presented in Ref. [32]. The global mass balance of the TEMA X evaporator is expressed according to Equation 4. The global energy balance is expressed by Equations 5 and 6, respectively. The latter is included to satisfy the equilibrium between the heat transfer and HTF enthalpy drop.

$$\frac{d(\rho_w V_w)}{dt} + \frac{d(\rho_{st} V_{st})}{dt} = \dot{m}_w - \dot{m}_{st} \quad (4)$$

$$\frac{d(\rho_w V_w u_w)}{dt} + \frac{d(\rho_{st} V_{st} u_{st})}{dt} + M_{ev} C_m \frac{dT_{m,ev}}{dt} = \dot{m}_w i_w - \dot{m}_{st} i_{st} + \dot{m}_{HTF} i_{HTF,i} - \dot{m}_{HTF} i_{HTF,o} \quad (5)$$

$$\dot{m}_{HTF,i}i_{HTF,i} - \dot{m}_{HTF,o}i_{HTF,o} = h_{HTF,ev}A_{ev}(\bar{T}_{HTF,ev} - T_{m,ev}) \quad (6)$$

The model presented above allows the calculation of the transient response of the pressure and the total water volume although, in order to estimate the steam drum water level, two additional equations are needed [31]. The model was validated with the results presented in Ref. [31]. An example of an open loop response of the steam drum model for a step of 10 kg/s on the outlet steam is shown in Figure 4. The maximum deviation for the pressure (Figure 4a), the water volume (Figure 4b), the steam mass fraction (Figure 4c) and the level drum (Figure 4d) are 2 %, 0.4 %, 1.6 % and 1.2 %, respectively.

Figure 4. Response of the system for a steam mass flow rate step of 10 kg/s: a) steam drum pressure; b) total water volume; c) evaporator outlet steam mass fraction; d) steam drum level.

4. Stress models

In this section the analytical stress models used for the critical points of the heat exchangers are described. Furthermore, a finite element analysis is carried for comparison with the analytical models for the different stress points selected.

4.1 Steam drum and superheater nozzle stress calculations

Typically, in conventional boilers the critical stress points are the thick-walled components such as the SH headers and the steam drum [33]. The European standard EN 12952-3 [7] presents a methodology to calculate thermal and mechanical stresses in the aforementioned components. This standard proposes analytical correlations to calculate thermal and mechanical stress concentration factors for a wide range of geometries of T-junctions, which are very useful to avoid finite element analysis. The stresses are calculated as the sum of thermal and pressure stresses according to Equation 7:

$$\sigma = \sigma^p + \sigma^T = \alpha_m \frac{D}{2 t_j} P_i + \alpha_t \frac{\beta_t E}{1 - \nu} (T_{ave} - T_{su}) \quad (7)$$

where α_t is the thermal stress concentration factor and α_m is the mechanical concentration factor, T_{ave} is the metal average temperature, T_{su} is the inner surface temperature. Normally, the stress on a T-junction is calculated at the point that presents the greatest stresses, P_1 . However, during the start-up, the most limiting stress can be obtained at P_2 [33]. For this reason, in this work the stresses on T-junction are calculated at both points.

The temperature field along the radial coordinate is calculated using the thermal model described by Equation 8, which assumes heat conduction only in the radial direction. When the temperature field is known, the average metal temperature, T_{ave} is calculated following Ref. [10]. The model has two boundary conditions. A convective boundary condition is considered at the inner wall (Equation 9). This is especially important for T-junctions whose heat transfer coefficients vary significantly during the start-up, such as in the SH head. An adiabatic boundary condition is imposed at the external wall (Equation 10).

$$\frac{1}{r} \frac{\partial}{\partial r} \left(r \frac{\partial T}{\partial r} \right) = \frac{1}{a} \frac{\partial T}{\partial t} \quad (8)$$

$$-k \frac{\partial T}{\partial r} \Big|_{r=r_i} = h_j (T_j(t) - T(r_i, t)) \quad (9)$$

$$\frac{\partial T}{\partial r} \Big|_{r=r_o} = 0 \quad (10)$$

A finite element analysis (FEA) using the ABAQUS/CAE software [20] was carried out to compare the results obtained using Equation 7. A coupled heat transfer - thermal stress analysis using ABAQUS/Standard was selected. The calculation procedure used by ABAQUS/Standard can be divided in two steps: i) The heat transfer equations are integrated using a backward-difference scheme, and ii) the coupled system is solved using Newton's method [20]. The FEA is built with element type C3D20RT (20-node thermally coupled brick, triquadratic displacement, trilinear temperature and reduced integration).

Figure 5 shows the FEA results obtained in a SH head-nozzle T-junction (HNTJ) for pressure and thermal load under nominal conditions. The boundary conditions used were: i) zy plane symmetry condition (the displacement and rotation are fixed on the zy plane, and the z and y

directions, respectively); ii) xy plane symmetry condition; iii) the displacement on the y direction is fixed on the nozzle top; iv) pressure load on the end cap.

Table 2 compares the results obtained by FEA and the analytical method. The loads applied on the SH-HNTJ are the temperature field and pressure obtained at nominal conditions. As can be seen, the stress results obtained by FEA and Equation 7 at P_1 are relatively close (8.24 % deviation). This is because the normative EN 12952-3 [7] provides accurate correlations for the estimation of the mechanical stress concentration factor at P_1 . However, correlations to calculate the mechanical stress concentration factor at P_2 are not available in the normative EN 12952-3 [7]. For this reason, in order to estimate the mechanical stress concentration factor at P_2 it is assumed that the geometry consists of an infinite plate with a central hole according to Ref. [34]. The large difference between the analytical model and the FEA results for P_2 is due to the different geometries used for the respective models.

Figure 5. FEA results obtained in the superheater head-nozzle T-junction.

Table 2
Comparison between analytical and FEA results obtained in the superheater head-nozzle T-junction.

Load	Point	Analytical, Maximum stress, (MPa)	FEA, Maximum stress, (MPa)	Difference (%)
Pressure and thermal	Point 1, Equation (7)	230.3	212.5	+8.24

4.2 Tubesheet stress calculation

The tubesheet is one of the most challenging components in the design of the heat exchanger. This is due to its three-dimensional structure of holes and junctions with shell and head [35]. The tubesheet, as a part of the heat exchanger, is generally subjected to a combination of thermal and pressure loads that generate a complex stress state. Conventional methods to calculate the stress on a tubesheet, such as the TEMA standards [14] or ASME Section VIII Div 1 [15], do not provide specific guidelines for thermal loadings such as no-tube-lane (NTL) zone thermal stresses. Thus, in order to consider the aforementioned effects, the method proposed by Ref. [36] was used in this work. The tubesheet stress models proposed by Ref. [36] are summarized in Table 3. The points selected for the stress analysis of the tubesheet are shown in Figure 6.

The stress at tubesheet ligaments due to a pressure load was calculated using Equation 11. First, the equivalent elasticity modulus and the equivalent poisson ratio were calculated, which are mainly dependent on the tube pitch layout and ligament efficiency of the tubesheet. Second, the pressure stress was calculated assuming an equivalent solid plate (σ_1). Finally, a stress concentration factor (K) was calculated as a function of the tubesheet geometry.

Table 3

Tubesheet stress models proposed by Ref. [36].

Load	Equation
Pressure (ligaments)	$\sigma_{eff} = K \frac{R}{h} \sigma_1 $ (11)
Thermal (skin effect)	$\sigma_{max} = \frac{E \beta_s (T_{ave} - T_{su})}{1 - \nu}$ (12)
Thermal (ligaments)	$\sigma_{max} = \frac{K_u E^* \beta_s (T_H - T_C)}{2}$ (13)
Thermal (holes adjacent to the NTL zone)	$\sigma_{max} = \frac{K_D E \beta_s (T_H - T_C)}{2(1 - \nu)}$ (14)

The thermal stress due to "skin effect" was calculated using Equation 12. A two dimensional transient thermal model of the tubesheet ligament (Equation 15) was used to calculate the surface wall temperate (T_{su}) and the average wall temperature (T_{ave}). The model considers three convective boundary conditions at the ligament surface with a fluid contact and an additional symmetric boundary condition.

$$\frac{1}{r} \frac{\partial}{\partial r} \left(r \frac{\partial T}{\partial r} \right) + \frac{\partial^2 T}{\partial z^2} = \frac{1}{a} \frac{\partial T}{\partial t} \quad (15)$$

The thermal stress found at ligaments and holes of the NTL zone of the tubesheet (see Figure 6) was calculated using Equations 13 and 14, respectively. In order to calculate the stress on ligaments the equivalent elasticity modulus (E^*)[36] was used. T_H and T_C were calculated as the bulk temperature of the hot and cold fluids, respectively. Finally, the stress concentration factors K_u and K_D were calculated as a function of the tubesheet geometry.

Figure 6. The five points selected for the tubesheet stress analysis.

A FEA was performed for the RH to check the deviation of the analytical models (Figure 7). Due to the symmetry in geometry and loads, only a sector of 180° is modeled including the shell and head junctions. In order to obtain realistic results the perforated holes of the tubesheet were included in the model. Figure 7a shows the temperature field of the FEA considering the thermal and pressure loads at nominal conditions (the shell side temperatures were not considered). The boundary conditions used were: (i) yx plane symmetry; (ii) zx plane symmetry on the end cap of the head. Figure 4b depicts the FEA stress results at the tubesheet ligaments.

The comparison between FEA and analytical stress results is summarized in Table 4. The comparison suggests that the stresses produced at ligaments near the NTL zone are overestimated by the Equation 13 around 22% compared with the results obtained by FEA. The stresses obtained at the holes adjacent to the NTL zone are 18% higher when using Equation 14 than those of the FEA. For the pressure load case, the results suggest that the stresses are between 33.7% higher when using Equation 11. Finally, as a result of the RH FEA it is expected that the stresses on the tubesheet will be calculated on the safe side. It should be noted that the models proposed by Ref. [36] does not take into account complex effects such in-plane shears [37]. Hence, the accuracy of these models is limited, especially at the interfaces of the perforated and solid regions of the tubesheet. However, the combination of a low computational cost and conservative stress estimation make these models a very attractive solution for the dynamic simulations with a large number of heat exchangers.

Figure 7. FEA of the reheater: a) transfer results of the tubesheet
b) stress results of the tubesheet.

Table 4
Comparison between analytical and FEA results obtained for the reheater.

Load	Point	Analytical, Maximum stress, (MPa)	FEA, Maximum stress, (MPa)	Difference (%)
Pressure	Ligaments, Equation (11)	57.1	42.7	+33.7
Thermal	Ligaments, Equation (13)	320.7	263.4	+21.7
Thermal	Holes adjacent to NTL Equation (14)	509.5	432.6	+17.7

4.3 Tubesheet junction stress calculation

The ASME Pressure vessel code [15] considers six possible junction configurations for U-tube tubesheets, including combinations of integral, gasketed and gasketed extended as a flange. An integral junction was assumed in this work. A simplified analytical model is proposed to estimate the stress on the tubesheet junction (TSJ). Considering elastic conditions, the displacements (x_j) and forces (F_j) are related by the stiffnesses (K_j). The displacements due to thermal loads were neglected. Then, the resulting forces and moments were calculated using Equations 16 and 17.

$$\begin{Bmatrix} y_h \\ \theta_h \\ y_s \\ \theta_s \end{Bmatrix} = \begin{bmatrix} K'_{cyl,My} & -K'_{cyl,Vy} & 0 & 0 \\ K'_{cyl,M\theta} & -K'_{cyl,V\theta} & 0 & 0 \\ 0 & 0 & -K'_{cyl,My} & K'_{cyl,Vy} \\ 0 & 0 & K'_{cyl,M\theta} & -K'_{cyl,V\theta} \end{bmatrix} \begin{Bmatrix} M_h \\ V_h \\ M_s \\ V_s \end{Bmatrix} + \begin{bmatrix} K'_{cyl,p} & 0 \\ 0 & 0 \\ 0 & K'_{cyl,p} \\ 0 & 0 \end{bmatrix} \begin{Bmatrix} P_h \\ P_s \end{Bmatrix} \quad (16)$$

$$\begin{Bmatrix} y_{ts,h} \\ \theta_{ts,h} \\ y_{ts,s} \\ \theta_{ts,s} \end{Bmatrix} = \begin{bmatrix} \frac{L_{ts}}{2} K'_{ts,My} & \frac{L_{ts}}{2} K'_{ts,Vy} & \frac{L_{ts}}{2} K'_{ts,My} & \frac{L_{ts}}{2} K'_{ts,Vy} \\ -K'_{ts,M\theta} & -K'_{ts,V\theta} & -K'_{ts,M\theta} & -K'_{ts,V\theta} \\ -\frac{L_{ts}}{2} K'_{ts,My} & -\frac{L_{ts}}{2} K'_{ts,Vy} & -\frac{L_{ts}}{2} K'_{ts,My} & -\frac{L_{ts}}{2} K'_{ts,Vy} \\ -K'_{ts,M\theta} & -K'_{ts,V\theta} & -K'_{ts,M\theta} & -K'_{ts,V\theta} \end{bmatrix} \begin{Bmatrix} M_h \\ V_h \\ M_s \\ V_s \end{Bmatrix} + \begin{bmatrix} -\frac{L_{ts}}{2} K'_{p,p\theta} \\ K'_{p,p\theta} \\ \frac{L_{ts}}{2} K'_{p,p\theta} \\ K'_{p,p\theta} \end{bmatrix} \{P_h - P_s\} \quad (17)$$

The stiffnesses for pressure loads of the shell and head were calculated as cylinders according to Ref. [34]. The stiffness of the tubesheet was calculated according to ASME VIII Div 2 [15] considering the equivalent solid plate elastic properties modifications (modulus of elasticity and Poisson's ratio). The stress calculation was carried out by estimating the following: (i) the shear forces and moments using the compatibility of the displacements; and (ii) the stress in cylindrical coordinates according the Equation 18 [34]. The von Mises equivalent stress equation [15] was used to consider the stresses in the different directions.

$$\begin{Bmatrix} \sigma_z \\ \sigma_\theta \\ \tau_{rz} \end{Bmatrix} = \begin{Bmatrix} \frac{D^2 P}{4t(D+t_j)} \pm \frac{6M}{t_j^2} \\ \frac{2yE}{D} \pm \frac{6M\nu}{t_j^2} \\ \frac{V}{t_j} \end{Bmatrix} \quad (18)$$

Figure 8. FEA stress results of the superheater tubesheet junction.

Figure 8 shows the FEA results obtained for SH TSJ model considering only pressure loads at nominal conditions. The comparison between the FEA and the analytical stress results of the SH TSJ is summarized in Table 5. The comparison suggests that there is a good agreement between the analytical model (Equation 18) and FEA. In the TSJ at the head-side (TSJ-HS), the stress predicted by the analytical model is only 4 % lower than that predicted by the FEA. However, the stresses obtained in the TSJ at the shell-side (TSJ-SS) using the analytical model are higher than those of the FEA.

In the case that the thermal loads are considered, an important shear stress point is obtained in the TSJ at the NTL zone (TSJ-NTL) (see Figure 6). The stresses at this point can be calculated using Equation 19. This equation is based on the theoretical solution presented by Slot [38] for a solid plate with a temperature difference between upper and lower half parts. The shear stress constant is equal to $K_r = 0.51$ at the junction with head and/or shell zone. This effect is also checked with FEA based on the SH model considering thermal load at nominal conditions. The

results suggest that shear stress is about 14 % lower when using the analytical model than using the FEA.

$$\tau_{r\theta} = K_{\tau} \frac{(T_H - T_C) \beta_t E}{2} \quad (19)$$

Table 5
Comparison between analytical and FEA results obtained for the superheater tubesheet junction.

Load	Point	Analytical model, Equivalent stress, (MPa)	FEA, Equivalent stress, (MPa)	Difference (%)
Pressure	Tubesheet junction head-side Equation (18)	82.9	86.9	-4.6
Pressure	Tubesheet junction shell-side Equation (18)	29.8	25.6	+16.4
Thermal	Tubesheet junction at NTL zone Equation (19)	113.1	132.6	-14.7

The TSJ is formed by the joint of thick metal walls, especially the head and tubesheet. Therefore, it is expected that thermal stresses due to transient operation play a significant role for the TSJ damage. For this reason, a two dimensional transient thermal model based on Equation 15 was performed to determine the temperature field in the TSJ [26]. The model was built assuming axis-symmetric conditions. Convective boundary conditions were used for the surface with fluid contact: head, tube and shell. Insulation conditions were assumed for the external walls. Finally, uni-dimensional heat transfer conditions were assumed for the zones of head and shell of the tubesheet.

Once the temperature field at the TSJ is known, the thermal stress was calculated for TSJ-HS following Ref. [10], considering only the thermal gradient in the radial direction. The thermal stress of the TSJ-SS was calculated according to Equation 12. Once the thermal stresses were calculated, they were combined with the mechanical stresses (Equation 18) taking into account their corresponding sign. Then, the equivalent stresses were calculated using the von Mises equation [15].

4.4 U-tube stress calculation

The thermal stresses on the U-tube due to the differential thermal expansion between cold and hot legs were calculated according to Singh et al. [39]. The calculation procedure is described in the following steps. First, the tube was allowed to expand freely, where the displacement was calculated using Equation 20 and considering a zero axial force. Second, the moments and forces were calculated using an iterative process to accomplish the compatibility of displacements. When the moments and forces are known, the moment law function, $M(\theta)$, can be calculated and then the stresses on the U-bend were computed using Equation 21.

$$\delta = (\overline{\beta_{hot} \Delta T_{hot}} - \overline{\beta_{cold} \Delta T_{cold}}) L - \frac{2F}{AE} L \quad (20)$$

$$\sigma(\theta) = \frac{\psi M(\theta) r_o}{I} \quad (21)$$

5. Application to the steam generator of a 55 MWe parabolic trough power plant

The results of dynamic analysis of the SG include the following:

- An analysis of the SG daily start-up.
- A stress analysis of the heat exchangers, including tubesheets (Figure 6) and T-junctions (Figure 5).
- A comparison of the dynamic response during start-up of two designs of the evaporator of the SG: TEMA X and kettle.

5.1. Steam generator daily start-up simulation

The dynamic analysis of the SG assumes that the storage system and/or solar field can provide the thermal oil inlet conditions to carry out the start-up. Isothermal initial conditions are considered for the SG heat exchangers, matching with the steam drum saturation temperature (at around 240 °C). The start-up procedure was developed based on Refs. [40,41]. The results of the start-up simulation of the SG are shown in Figure 9.

The thermal oil enters to the SG with a mass flow rate around 4% of its nominal value. The thermal oil is sent to the EV at 260 °C (Figure 9a), which is 20 °C above to the saturation temperature of the water in the drum to avoid thermal shocks in thermal oil pipelines. The thermal oil also circulates through the PH. At the same time, the drum water ($T_d \approx 240$ °C, Figure 9c) circulates to heat-up the feed-water from the feed-water-heaters, which is assumed to be equal to 100 °C in the deaerator at the beginning of the start-up. Then, the feed-water enters the PH at a minimum temperature of 180 °C to avoid potential thermal stresses in the tubesheet. Moreover, the steam generated is sent from the steam drum to the deaerator to heat-up the SG feed-water.

When around 4% of the steam generation is produced, the thermal oil mass flow is circulated to SH and RH. Now, the thermal oil mass flow is set to 5% of its nominal value with a temperature ramp-up rate of 8 °C/min. The steam from the steam drum is circulated to the SH and the outlet steam is sent to the RH (Figure 1). Then the RH steam inlet is set to 250 °C to reduce the thermal shock.

After around 20 min the main and reheat steams achieve 320 °C, which is the set point temperature at the inlet to start the turbine synchronization (see Figure 9b/d). The turbine inlet temperature is kept constant by using the attemperation system (Figure 1). The turbine synchronization takes 8 min for a daily start-up [40].

After 28 min, the synchronization of the steam turbine is finished and the HP and LP steam inlet temperatures are increased to its nominal value (see Figure 9b/d). At the same time, the thermal oil mass flow ramp-up rate is set to obtain a ramp-up of 7 °C/min in the evaporator. Then, the steam drum pressure starts to rise and the steam mass flow rate increases as well.

After 45 min, full load is reached and the SG start-up is completed. The energy necessary to carry out the SG start-up is 36.4 MWh_{th}. This value is in the range of the data reported in the literature. For example, according to Schenk et al. [40], the required energy for the start-up of

the SG of a 50 MWe parabolic trough plant is in the range 20.7 MWh_{th} to 44.9 MWh_{th}, depending of the initial conditions.

Figure 9. SG start-up simulation: a) thermal oil temperature and mass flow rate, turbine power output; b) superheater performance; c) evaporator and steam drum performance; d) reheater performance.

5.2. Stress analysis for the start-up procedure

According to ASME VIII Div2 [15] the maximum stress should be limited to prevent ratcheting failure. In this way, the protection against ratcheting considering an elastic stress analysis should be accomplished by using this limitation: the alternating stress amplitude should be lower than the ratcheting stress limit, $\Delta S \leq S_{PS}$. If this limitation is not satisfied, a complex elastic-plastic analysis should be carried out [37]. The alternating stress ΔS is calculated considering a primary membrane, primary bending and secondary stresses ($P_L + P_b + Q$). This means that the stresses due to local structural discontinuities, i.e. the peak stresses (F), must be excluded of this analysis.

The maximum allowable equivalent stress S_{PS} was calculated as the highest value of: (i) three times the ASME VIII Div 1 allowable stress limit S ; (ii) two times the yield stress of the material S_y . This limitation is imposed for the tubesheet and U-tube stress calculations.

In the case of EN 12952-3 [7] the maximum stress is limited to avoid damage of the protective magnetite layer on the metal wall of the component. The maximum allowable stress is calculated as the tangential pressure stress at nominal operating conditions plus 200 MPa: $\sigma_{a,\max} = \sigma_{\theta,p_o} + 200$. The minimum allowable stress is 600 MPa below to the tangential pressure stress at nominal operating conditions: $\sigma_{a,\min} = \sigma_{\theta,p_o} - 600$. As already mentioned, the stress calculations for T-junctions were conducted using EN 12952-3 [7].

Figure 10 shows the results of the stress analysis of the SG during the start-up. The vertical line at $t = 45$ min points out the end of the start-up process while the horizontal line identifies the maximum allowable stress. As it can be seen, the stress evolution in all components is within suitable stress ranges according to the aforementioned limits.

According to the stress nature, two stress groups can be identified. The first group considers the stress points due to the thermal inertia. This group is formed by the following points: the TSJ-HS, the TSJ-SS, the DDTJ and the SH-HNTJ (Figure 10b/c/g/h). This group is more susceptible to the thermal stresses due to temperature ramp-up rates, and therefore establishes the minimum start-up time.

The second group considers the stress points where the thermal inertia can be neglected and then the stress can be estimated using the temperature and pressure states. This group is formed by the tubesheet ligaments, the TSJ-NTL zone and the U-bend region (Figure 10a/d/e/f).

During the daily start-up, the highest stress value is obtained in the U-bend of the SH (Figure 10f). Nevertheless, this value does not represent a constrictive limitation since this stress can be reduced increasing the U-bend radius and/or the clearance in the last baffle. It is important to take into account that the U-bend radius design was accomplished considering nominal operation conditions and for this reason the resulting stresses on the SH U-bend reach the safety value of 0.3 UTS.

The most critical stress points are obtained at the tubesheet ligaments and the tubesheet holes adjacent NTL zone (Figure 10a/d). It is worth to mention that the ligaments that present the highest stress are those located near the NTL (Figure 6). In contrast, the most stressed point at the tubesheet holes at the NTL zone is obtained for the RH in nominal conditions, when a high difference in temperature between the tube inlet and outlet is reached (Figure 10d).

The most stressed points belong to the second group, in which the thermal inertia can be neglected, suggesting that higher temperature ramp-up rates could be used in the SG start-up. However, the temperature ramp-up rate selected for the thermal oil inlet (8°C/min) is already high compared with the literature [40,42]. Therefore, an increment of the temperature ramp-up rate is considered as an unrealistic scenario.

A ramp-up rate of 7 °C/min was selected for the evaporator. This value is within the optimal range of the ramp-up rates suggested by Ferruzza et al. [3], suggesting that higher evaporator ramp-up rates will not lead to a significant increment of the annual energy production.

The high thermal stresses obtained in the RH even under design conditions (Figure 10d) suggest that TEMA F design is sensitive to fatigue cycle failure. Therefore, another possible design option can be the hairpin type heat exchanger. Since this design has two tubesheets, the stresses at the NTL are eliminated. Another possible option consists of splitting the RH in two heat exchangers in series, reducing the temperature difference in the tubesheet. This design option is proposed by Aalborg CSP [43].

Figure 10. Stress analysis of the steam generator start-up: a) tubesheet ligament; b) tubesheet junction at head side; c) tubesheet junction at shell side d) tubesheet holes adjacent of the no-tube-lane zone; e) tubesheet junction at the no-tube-lane zone; f) U-tube; g) drum-downcomer T-junction; h) superheater head-nozzle T-junction.

5.3. Comparison between kettle and TEMA X recirculation evaporator

Kettle evaporators have been used in pioneering PTPPs due to their relatively low cost and successful operation [44]. For these reasons, this design has been installed in recent PTPPs [45,46]. However, other references suggest that kettle evaporators are susceptible of high stresses due to the high wall thicknesses (shell and tubesheet), and thus, they have high risk of premature fatigue failure [43].

A new simulation was carried out to compare the dynamic behavior of TEMA X and kettle evaporators. Figure 11 shows the critical points selected for the stress analysis. The kettle evaporator design was obtained for Ref. [12]. The dynamic model for kettle was developed using a simplified second-order model based on Equations 4, 5 and 6. This model captures very well the dynamic response of the pressure for input changes of load, steam flow or feed-water flow [31]. Following the previous simulation, the ramp-up rates selected are 8 °C/min for thermal oil inlet and 7 °C/min for evaporator.

Figure 11. Critical points selected for the stress analysis of the TEMA X recirculation and kettle evaporators.

Figure 12 illustrates the results of the stress analysis of the TEMA X and kettle evaporators. The results indicate that the stresses obtained are placed within allowable ranges for both designs. The TEMA X evaporator obtains a stress reduction of 35% compared to the kettle evaporator, see Figure 12a at $t = 40$ minutes. The maximum stress on the tubesheet is in the same order of magnitude for both designs, see Figure 12b. However, the higher thermal inertia of the ligaments of the kettle tubesheet leads to higher transient thermal stresses as it can be seen in Figure 20-b at $t = 35$ minutes.

Figure 12. TEMA X and kettle evaporators stress analysis: a) drum-downcomer T-junction (shell for kettle); b) tubesheet ligament.

6. Conclusions

In this paper a dynamic analysis of the steam generator of a parabolic trough power plant was carried out. Different transient models were proposed for the single phase heat exchangers TEMA F and TEMA H, and the TEMA X recirculation evaporator with steam drum. The thermal stresses were calculated at the critical parts of steam generator, including tubesheets, steam drum and U-bend regions. Furthermore, finite element simulations of different parts of the steam generator were developed to check the deviation of the simplified analytical stress models.

The main conclusions of this work can be summarized as follows:

- The proposed steam generator design fulfils the ratcheting stress limits according to ASME Section VIII Div 2 [15] and the allowable stress limits of EN 12952-3 [7]. This means that the proposed design of the steam generator is valid to carry out a fatigue analysis to estimate its lifetime.
- The results suggest that the start-up of the steam generator takes around 45 minutes and the energy needed for the start-up is around 36.4 MWh_{th}.
- The highest thermal stresses are obtained at the no-tube-lane of the U-tube tubesheets, especially in the reheater.
- The TEMA X evaporator obtains a 35% lower stress compared to the kettle evaporator for the same temperature ramp rates.

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