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Lifetime analysis of the steam generator of a solar tower plant

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Abstract

Solar tower plants are pushed to operate with fast start-ups and/or load changes to increase their dispatchability and competitiveness. However, this operation mode decreases the lifetime on components like the steam generator due to fatigue damage. Moreover, the high operating temperatures typical of solar tower plants also lead to creep damage, which it is combined with fatigue. Both damage mechanisms may lead to a dramatic reduction of the steam generator lifetime, compromising the power plant economic viability.

In this work the lifetime of the steam generator of a solar tower plant is investigated. For that purpose, an elastic-plastic approach is used to consider the material cyclic hardening. The results show that the steam generator formed by shell-and-tube heat exchangers can operate during 25 years with temperature change rates up to 150°C/h, assuming 300 start-ups per year. The most critical point is the superheater head-nozzle junction for both intermittent and continuous operation regimes. An optimization of the superheater head thickness is performed to maximize its lifetime in both regimes. Finally, it can be concluded that temperature change rates higher than 150°C/h may require a different design of the superheater head and/or a material with better creep-fatigue properties.

Key words: *Steam generator; Stress analysis; Creep-Fatigue analysis; Solar tower plant; Start-up; Concentrating solar power*

Abbreviations

CCP: combined-cycle plants

DDJ: drum-downcomer junction

ECCC: European Creep Collaborative Committee

EV: evaporator

FEA: finite element analysis

HNJ: head-nozzle junction

NTL: no-tube-lane zone

PH: preheater

RH: reheater

SG: steam generator
SH: superheater
STP: solar tower plant
TEMA: Tubular Exchanger Manufacturers Association
TSJ: tubesheet junction
TSL: tubesheet ligaments
UT: U-tube
UTS: ultimate tensile strength

Nomenclature

D_c : damage produced by creep
 D_f : damage produced by fatigue
 D_L : damage limit
 E : modulus of elasticity (GPa)
 E' : dynamic elastic modulus (GPa)
 E_{yf} : modulus of elasticity used to establish the design fatigue curve according to ASME
 E_T : modulus of elasticity calculated at the average temperature of the cycle (GPa)
 k : thermal conductivity (W/mK).
 K' : strain hardening parameter (MPa)
 N : number of cycles
 N_a : number of allowable cycles
 S_{alt} : effective alternative stress (MPa)
 T : temperature (K)
 d_c : damage produced by creep in one year
 d_f : damage produced by fatigue in one year
 h : heat transfer coefficient (W/m² K)
 n' : strain hardening exponent
 t : time (h)
 t_d : hold time (h)
 t_R : time to rupture (h)

Greek symbols

α : thermal expansion coefficient (mm/mm/K)
 ε : strain ($\mu\varepsilon$)
 ε'_f : fatigue ductility ($\mu\varepsilon$)
 ν : Poisson's ratio
 σ : stress (MPa)
 σ'_f : fatigue strength coefficient (MPa)

Subscripts

d : drum
 dc : downcomer
 e : elastic
 h : head
 n : nozzle
 p : plastic
 s : shell
 t : tube
 w : water

1. Introduction

In recent years, European electricity markets have experienced a significant growth of renewable energy sources in order to reduce the CO₂ emissions and fossil fuel dependence. The use of variable renewable energies, such as wind and photovoltaic power systems, has changed the role of conventional thermal power plants as coal power plants and combined-cycles. In this new scenario, variable renewable plants which production is subjected to the weather conditions have preference of dispatch due to environmental policies. In this way, conventional thermal power plants are pushed to act as load-following power plants to balance the variability of aforementioned renewable energy sources. This changes the operation mode of conventional thermal power plants from baseload to load-following with frequent start-ups and shutdowns [1].

Thermal power plants are subjected to lifetime consumption due to accumulation of different damage mechanisms: creep, fatigue, corrosion, erosion, etc. Creep is the main damage mechanism experienced by devices exposed to high temperatures and constant loads. The baseload-power-plants devices are typically designed considering the creep damage as the first lifetime limiting factor. In the case of load-following operation, the devices are subjected to significant stress variations due to startup/shutdowns or load changes, and thus, the fatigue damage becomes important. In the cases where the fatigue and creep damages have a similar order of magnitude, the lifetime consumption of the devices may be dramatically accelerated compromising the power plant economic viability.

Solar tower plants (STPs) may become a renewable solution to play the role of load-following power plants similar to combined-cycle plants (CCPs) [2]. This is possible due to their ability to integrate cost-effective thermal storage systems, which it is a differential feature compared to variable renewable energies like photovoltaic or wind power. Nevertheless, to be considered dispatchable, STPs need to assure enough flexibility to meet the required load change rates [2]. On the one hand, this may lead to a revenue increase, and thus, an increase of its competitiveness. On the other hand, the new role of dispatchable plants with frequent starts-ups/shutdowns and fast load variations may lead to a premature creep-fatigue failure of the high-temperature devices.

Typically, the devices more susceptible to creep-fatigue failure of a thermal power plant are the steam turbine and the steam generator (SG) [3]. In the case of concentrating solar power plants, the SG commonly consists of a train of conventional shell-and-tube heat exchangers, which are typically designed using TEMA (Tubular Exchanger Manufacturers Association) specifications [4]. Such approach is suitable for devices working continuously at nominal conditions or with small temperature variations. Nevertheless, the SG of a STP exchanges a large amount of

thermal energy working under extreme conditions: high temperature (up to 565°C), high pressure (around 12-14 MPa) and cyclic operation with a relatively high temperature change rates (~150 °C/h). In this way, a mandatory step in the design of these plants consists of assuring the lifetime of the SG considering the creep-fatigue damage produced by these extreme working conditions.

Typical processes of thermal power plants like start-ups, shutdowns and load changes involve temperature, pressure and mass flow changes. Then, prior to the thermal stress analysis, it is required to determine the boundary conditions for the heat transfer analysis on a specific part of a device, e. g., the trends of the main thermodynamic variables on the tubesheet of a heat exchanger (mass flow rate, temperature and pressure change rates, etc.). Sometimes the thermal stress analysis is combined with the thermodynamic modeling of the power plant [5–8]. For example, Taler et al. [5] investigated the thermal stresses during the start-up of a fired boiler power plant. They optimized the start-up time considering the steam drum as the most critical stress point. Benato et al. [6] studied the damage experienced in both superheater and reheater headers and steam drum for a load change variation of a combined-cycle power plant. The results showed that the most critical point is the superheater header. Irfan and Chapman [9] studied the thermal stress experienced on W-tube shape heat exchangers for regenerative and recuperative systems. For that purpose, finite element simulations were carried out using ANSYS software [10]. The results indicated that the main contribution to the thermal stresses, in the recuperative systems, was due to the appearance of non-linear thermal gradients in the axial direction. However, regenerative systems showed an almost free stress state due to a comparatively linear temperature gradient in the axial direction. Sun et al. [11] carried out a thorough thermal and structural analysis of the steam generator of a nuclear power plant. Coupled thermal and mechanical stress distributions were evaluated on U-tubes, showing that the critical points from a fatigue point of view are the contact points of the tube support plates. González-Gómez et al. [12] performed a transient thermal and mechanical analysis of the steam generator of a solar tower plant. Simplified stress models were used to study the critical points of the SG during start-up operation. However, in [12] neither the fatigue damage nor the creep damage were considered to determine the critical points. In addition, simplified elastic-stress models were employed to calculate the stress, which do not take into account the complex stress effects in three-dimensional geometries such as tubesheets or head-nozzle junctions. Although, this approach is valid for a first estimation, a finite element analysis is highly recommended to obtain results with higher accuracy to assure the SG lifetime.

The creep-fatigue analysis can be carried out once the thermal stress analysis is performed. In contrast to the thermal stress analysis, few studies are available focusing on the creep-fatigue analysis of the thermal power plant devices. For example, Stoppato et al. [13] carried out a creep-fatigue analysis of the tubes of the superheater of a 320 MW coal thermal power plant. The calculations were made based on ASME Section III Division1 Subsection NH [14] and the British R5 code [15]. In addition to creep-fatigue damage, different mechanisms of damage as: oxidation, corrosion and welding were considered. Finally, two different plant operation strategies were compared by means of an economic analysis. Ando et al. [16] tested the damage caused in the tubesheet of a sodium heat exchanger of a nuclear power plant. They studied different cold and hot transient cases by means of a finite element analysis. The results show that the zone most susceptible to failure is the outermost holes of the tubesheet. The creep-fatigue analysis was performed comparing the elastic and the elastic-plastic approach, showing that the former leads to more conservative results (i. e. the elastic method employs high safety factors that cause higher creep-fatigue damage for the same loading conditions). Keatley et al.

[17] investigated the economic and technical impacts of the cyclic operation of conventional power plants. Different scenarios of variable renewable energy integration were studied. Simplified creep-fatigue models were proposed and combined with a linear damage summation model. Their results showed an increase up to 40% of the operational and maintenance costs of a conventional 400 MWe thermal plant, in the case of high penetration of variable renewable energies. González-Gómez et al. [18,19] analyzed the dynamic performance and the lifetime of the SG of a parabolic trough power plant. The creep damage was neglected due to the low working temperatures of the SG. The most damaged points were found in the superheater head, the steam drum and the reheater tubesheet. However, if it is considered only the damage due to load change operations, the critical point is the steam drum.

Other components susceptible of creep-fatigue failure on thermal power plants are the steam turbines and the solar receivers. Banaszkiewicz [20,21] studied the thermal stress and the lifetime degradation suffered on steam-turbine-rotors. On the one hand, Banaszkiewicz [20] proposed an analytical model based on equivalent Green's function concept to estimate the rotor stresses. The results of the analytical model were compared with operational data showing a relatively good accuracy. On the other hand, Banaszkiewicz [21] analyzed the lifetime of the intermediate-pressure-turbine rotor for hot, warm and cold start-ups. Creep and fatigue damages were taken into account to estimate the number of allowable cycles. The results showed a good agreement between the predicted crack areas and the observed in the turbine inspections. Du et al. [22] analyzed the fatigue fracture of a molten salt receiver using the Crack Tip Opening Displacement method. They developed a three-dimensional tube model for the heat transfer and thermal stress analysis. The results showed that the highest stress is obtained at circumferential direction, and then a potential risk of fatigue fracture is obtained in the axial direction of the tube. Ortega et al. [23] performed a complete structural evaluation of a solar receiver working at supercritical-carbon-dioxide conditions (~ 20 MPa , $\sim 700^\circ\text{C}$). The lifetime of the receiver was also investigated by means of a creep-fatigue analysis following ASME Section III approach [14]. A Larson-Miller model was employed to estimate the time to creep-rupture as a function of the temperature and stress. Their results indicated that the tubes made with Inconel 625 material fulfill the lifetime design condition at around 86% of the nodes.

In this work, the lifetime of a steam generator of a solar tower plant is studied performing a creep-fatigue analysis, which is made following ASME Section III [14]. For this purpose, different elastic-plastic simulations are carried out using ABAQUS/CAE software [24]. The critical stress points of the steam generator obtained on a previous work [12]: the heads and tubesheets of the superheater and reheater, are evaluated following a creep-fatigue analysis. The structural integrity of the steam generator is investigated for two plant operation modes. The former considers intermittent operation with daily start-ups and shut-downs. The second mode considers baseload operation mode with continuous operation. The results show that the most critical creep-fatigue point is the superheater head-nozzle junction for both cases studied. In addition, the lifetime of the superheater head-nozzle junction is optimized for both operation modes.

The main novelties of this work can be summarized as follows: i) this work performs the first study in the literature that analyzes the creep-fatigue damage of the steam generator of a solar tower plant, working with molten salt at high temperatures; ii) a high number of finite element simulations are carried out using an elastic-plastic approach to consider the material cyclic hardening in the: superheater and reheater tubesheets, superheater and reheater head-nozzle junctions and steam drum-downcomer junction.; iii) a simplified methodology is proposed to

optimize the lifetime of the critical point (superheater head-nozzle junction) of the steam generator taking into account both creep and fatigue damages.

2. System description

The SG design belongs to a STP of 110 MWe Rankine cycle. A typical STP layout is illustrated in Figure 1. The SG consists of an indirect steam generator system where the hot fluid is molten salt and the cold fluid is water/steam. The SG is formed by four heat exchangers: superheater (SH), reheater (RH), evaporator (EV) and preheater (PH). The SH heats-up the saturated steam up to the high-pressure turbine inlet conditions whereas the RH heats-up the steam from the high-pressure turbine outlet to the intermediate-pressure turbine inlet conditions. The EV provides the heat necessary to obtain saturated steam from saturated water conditions. Lastly, the PH heats-up the sub-cooled feed water until a point close to saturated water conditions. The SH and RH consist of hairpin type heat exchangers. A forced circulation system with a U-tube/straight shell design is selected for the EV. A U-tube/straight shell heat exchanger design is chosen for the PH. The material selected for the SH, RH and EV is 321 stainless steel (SA-240 Gr. 321 for the shell, head and tubesheet; and SA-213 Gr. 321 for the tubes). The material selected for the PH and steam drum is carbon steel (SA-516 Gr. 70 for the shell, head and tubesheet; and SA-210 Gr. A1 for the tubes). More details of the design procedure of the SG for a STP are available in [25].

Figure 1. Schematic of the typical STP layout.

Figure 2 shows the critical stress points based on the transient thermo-mechanical analysis of the SG start-up [12]. The SG start-up operation involves temperature changes of the working fluids leading to thermal gradients in thick metal parts of heat exchangers. Thermal gradients entail differential thermal expansion leading to daily cyclic thermal-stresses. For SH and RH the critical stress points are the tubesheet junction (TSJ), the tubesheet ligaments (TSL) and the head-nozzle junction (HNJ). Also, the U-Tubes of SH and RH may be considered as potential critical stress points. However, if the part-load performance of the SG is considered, they could be designed in a suitable stress range to prevent a premature creep-fatigue failure. For the EV and PH, the critical stress points are the tubesheet ligament and the no-tube-lane zone (NTL), respectively. Finally, the critical stress point of the steam drum is the drum-downcomer junction (DDJ).

HNJ	Head-nozzle junction	TSJ	Tubesheet junction	NTL	Tubesheet No-tube-lane

TSL	Tubesheet ligament	DDJ	Drum-downcomer junction	UT	U-tubes

Figure 2. Critical stress points of the SG of a 110 MWe STP.

3. Methodology

The lifetime of thermal power plants is reduced due to different mechanisms of damage, such as creep, combination of creep-fatigue, erosion, corrosion, oxidation and others [3]. Typically, the high temperature devices of conventional thermal power plants are designed considering continuous operation at nominal conditions [17]. This means that the temperature and pressure have low variations leading to stress levels that are practically constant. Under these conditions, creep is the main mechanism of damage working at high temperature. In the case of STPs, the typical operation consists of a daily start-up and shut-down [25]. This operation leads to fatigue damage, apart from creep damage, on the devices. Hence, to prevent premature failure, both damage mechanisms should be considered together.

Figure 3 shows the compressive and tensile stress cycles on the SG of a STP. According to the ASME code [14], fatigue damage is calculated considering only the stress/strain ranges whereas creep damage is related with the stress level. During the hold time periods, a stress relaxation may occur. This effect becomes important depending on the working temperature, the hold time and the strain range [26].

Figure 3. Typical compressive and tensile cycles in the loading devices of a STP.

3.1. Cyclic plasticity model

In a STP, the heat exchangers are generally subjected to cyclic thermal and/or mechanical loadings and sometimes the stress level exceeds the yield stress without causing immediate failure. When this occurs, two different cyclic behaviors can be obtained: shakedown or ratcheting. The ratcheting occurs when there are asymmetric stress cycles, i. e., when the mean stress is not zero. This produces progressive plastic strain increments on each cycle in the direction of the mean stress [27]. The shakedown occurs when the plastic strain is stabilized after a few initial cycles (between 5-10) [27].

The plastic behavior of a material subjected to cyclic loading is normally defined by the cyclic stress-strain curve or the hysteresis stress-strain loop. In case of non-cyclic loading, the plastic behavior is modeled by the monotonic stress-strain curve. When the metal is exposed to a cyclic loading, the metal experiences hardening or softening depending on its nature. Typically,

annealed metals experience hardening while cold worked metals show softening [28]. This effect is captured by the cyclic stress-strain curve that connects the stress amplitudes for different strain ranges of the stabilized hysteresis loop. Sometimes the cyclic stress-strain curves are not available, and the monotonic curves are used in place of cyclic stress-strain curves causing different problems. For example, the use of the monotonic curves may underestimate the stress level and then reduce the expected creep damage. Furthermore, the monotonic curves overestimate the plastic strains increasing the fatigue damage. The creep-fatigue analysis can be performed using an elastic approach. However, this approach may lead to a very conservative results due to the high safety factors used [16]. The main advantages of the elastic approach are the easy implementation and low computational cost but at the expense of a lower accuracy.

Figure 4 shows the monotonic curve, the cyclic stress-strain curve and the stabilized hysteresis loop for the 321 stainless steel at 540°C, obtained from the experimental data available in Stoppato et al. [13]. The cyclic stress-strain curve can be expressed by a form of the Ramberg-Osgood relationship:

$$\frac{\Delta \epsilon}{2} = \frac{\Delta \epsilon_e}{2} + \frac{\Delta \epsilon_p}{2} = \frac{\Delta \sigma}{2E} + \left(\frac{\Delta \sigma}{2K'} \right)^{1/n'} \quad (1)$$

where $\Delta \epsilon$ is the total strain range, $\Delta \sigma$ is the stress range, K' is the strain hardening parameter and n' is the strain hardening exponent. K' and n' are calculated by fitting Equation 1 with the experimental data reported in Stoppato et al. [13]. Once the cyclic stress-strain curve is defined, the stabilized hysteresis loop can be obtained using the procedure proposed by Dowling [28]. Table 1 shows the cyclic stress-strain curve data at different temperatures for the SG materials. The carbon steel parameters are obtained from ASME VIII Division 2 [29].

Figure 4. Cyclic, stabilized hysteresis and monotonic stress-strain curves for the 321 stainless steel at 540°C.

Table 1. Cyclic stress-strain curve data.

Material	Temperature (°C)	K' (MPa)	n' (-)
321 stainless steel	540	1928	0.361
	20	2082	0.317
Carbon steel	400	668	0.112
	300	748	0.100
	200	798	0.156

An important step for the elastic-plastic analysis is setting the zone where the plastic strain begins, i. e., the yield stress point. The ASME code [30] express the yield stress of the material for the monotonic stress-strain curve. In the case of cyclic stress-strain curves, it is not clear which is the criterion to define the limit where the plastic zone begins [31]. Here, the yield stress is calculated using a strain offset of $2 \cdot 10^{-5}$ following a conservative approach [32]. The rest of thermal and mechanical properties such as: the Young Modulus (E), Poisson's ratio (ν), thermal expansion (α), thermal conductivity (k) and thermal diffusivity are obtained from ASME code [30]. The properties of SG materials are summarized in Table 2.

Table 2. Properties of SG materials.

Material	Temperature (°C)	Young Modulus (GPa)	Poisson's ratio (-)	Thermal Expansion (mm/mm/ K)	Thermal Conductivity (W/m K)	Thermal diffusivity (m ² /s)
321 Stainless steel	600	151	0.3	21.1e-6	22.6	4.90e-6
	550	156	0.3	20.6e-6	21.9	4.78e-6
	500	160	0.3	20.2e-6	21.2	4.66e-6
	450	165	0.3	19.8e-6	20.5	4.55e-6
	400	169	0.3	19.5e-6	19.7	4.44e-6
Carbon steel	350	179	0.3	15.4e-6	47.0	10.55e-6
	300	185	0.3	14.9e-6	49.2	11.48e-6
	250	189	0.3	14.3e-6	51.4	12.42e-6

3.2. Heat transfer analysis

Before the thermal stress analysis, a heat transfer analysis is mandatory to calculate the temperature field of critical zones of the SG. Figure 5 illustrates the heat transfer boundary conditions used in this work. For tubesheets (Fig. 5-a), head-nozzle junctions (Fig. 5-b) and steam drum-downcomer junctions (Fig. 5-c) convective heat transfer boundary conditions are chosen on the surfaces with working fluid contact. The rest of surfaces are assumed adiabatic. The convective heat transfer coefficients for perforated holes, head and nozzle are calculated using Gnielinski correlation [33]. The convective heat transfer coefficient on the head-side and shell-side of the tubesheets are calculated following the procedure described by Ando et al. [16]. The convective heat transfer coefficients for the steam drum and downcomer are set to 1000 W/m² K according to Dzierwa and Taler [34].

Figure 5. Heat transfer boundary conditions: a) Tubesheets; b) Head-nozzle junctions; c) Steam drum-downcomer junctions.

Figure 6 shows the main inputs for the heat transfer and stress analyses. These inputs have been adapted based on SG start-up data reported in [12]. At the beginning of the SG start-up, the initial temperature of the heat exchangers is 290°C, which matches with the cold salt temperature. The initial steam drum pressure is 7.4 MPa. To initialize the SG start-up, a mixture of cold and hot salt is sent to the SG. In a first step, the salt mixture circulates to the EV and PH to establish the minimum saturated steam rate. Secondly, the salt mixture and the saturated steam are sent to the SH and RH. Then, a temperature change rate of 150°C/h of the salt is set to heat-up the SG. Consequently, temperature change rates of 150°C/h are also obtained in the superheated and reheated steam outlet. The superheated and reheated steam outlet temperatures are increased to 550°C while the salt inlet temperature is increased to 565°C. When the turbine synchronization is completed, the steam drum pressure is increased from 7.4 to 12.8 MPa whereas the RH pressure is increased from 0.4 to 3.5 MPa. At the same time, the salt and steam mass flow rates are increased as well. From an initial temperature of 290 °C, the temperature achieves the nominal conditions at 6240 seconds. The full-load conditions are achieved at 7000 seconds. At this point, the temperature difference between the salt and the steam in the SH and RH upper tubesheets is only 15 °C, while in the SH lower tubesheet is 120 °C. Further details of the start-up procedure can be found in [12].

Figure 6. Main inputs for the heat transfer and stress analyses: a) Temperature; b) Pressure.

3.3. Creep and fatigue models for high temperature devices

The assessment of the devices subjected to cyclic operation combined with high temperatures can be carried out using the following standards: ASME Section III Division 1 Subsection NH [14], the British R5 code [15] and UNE EN 12952-4 [35]. All of them consider fatigue and creep damage mechanisms. In this work, ASME Section III approach is used for the SG creep-fatigue analysis.

- **Creep damage calculation**

ASME Section III [14] follows the temporal fraction model for the creep damage calculation. A schematic of the calculation procedure is shown in Figure 7. The creep damage (D_c) is calculated using Equation (2), where t_d is the hold time and t_R is the time to rupture expressed as a function of the stress level and temperature. Normally, the time to rupture is expressed as a negative power-law relationship for uniaxial tests [36]. In this work, the time to rupture for 321 stainless steel is obtained from the European Creep Collaborative Committee (ECCC) [37] and the hold time is assumed equal to the full-load operating hours. Another important creep effect is the stress relaxation, which can be important depending on the material and/or the working temperatures. The stress relaxation mainly depends on the hold time and the temperature [26]. If this effect is considered in the creep analysis, the time to rupture increases and then the lifetime estimated for the component increases as well. This leads to a lower margin of safety, for this reason the stress relaxation effect is not considered in this work as a conservative approach. In the case of multiaxial state, von Mises equivalent stress (Equation 3) should be used [13,38]. A less conservative approach can be followed by using correction factors depending on the degree of multiaxiality [14]. In this work, it is chosen the first approach, that is, the creep damage is calculated using the von Mises equivalent stress.

$$D_c = \int_0^{t_d} \frac{dt}{t_R(\sigma, T)} \quad (2)$$

$$\sigma_{eq} = \sqrt{\frac{(\sigma_1 - \sigma_2)^2 + (\sigma_1 - \sigma_3)^2 + (\sigma_2 - \sigma_3)^2}{2}} \quad (3)$$

- **Fatigue damage calculation**

ASME Section III [14] follows the local strain approach to assess the resistance of a device against cyclic loading. The basic idea of the local strain approach was formulated by Coffin [39]. This approach considers that the number of cycles for a specific device part is taken from a strain-life curve obtained from a strain-controlled test data of smooth specimens. The main limitation of the local strain approach is that the details of the geometry of the devices must be previously defined to be modeled with enough accuracy [40]. The total fatigue damage (D_f) is calculated by means of Equation 4.

$$D_f = \sum_{j=1}^M \frac{N_j}{N_{a,j}} \quad (4)$$

The number of allowable cycles N_a is calculated by a Manson-Coffin curve:

$$\Delta \varepsilon_j = \frac{\sigma'_f}{E'} (2N_{a,j})^b + \varepsilon'_f (2N_{a,j})^c \quad (5)$$

where E' is the dynamic elastic modulus, σ'_f is fatigue strength coefficient, ε'_f is the fatigue ductility, and b and c are coefficients obtained experimentally. All these parameters were calculated experimentally for the 321 stainless steel in Ref. [13]. In the case of multiaxial state, the equivalent strain range is calculated considering the principal strain range variations and the corrected Poisson's ratio using Equation (6).

$$\Delta \varepsilon_{eq,j} = \frac{\sqrt{2}}{2(1+\nu^*)} \left\{ \left[\Delta(\varepsilon_1 - \varepsilon_2)_j \right]^2 + \left[\Delta(\varepsilon_1 - \varepsilon_3)_j \right]^2 + \left[\Delta(\varepsilon_2 - \varepsilon_3)_j \right]^2 \right\}^{1/2} \quad (6)$$

3.4. Fatigue model for low temperature devices

The assessment of the devices subjected to cyclic operation under low temperature conditions is considered when the temperature is low enough to neglect the creep damage ($D_c = 0$). Note that creep becomes important above of 380 °C for carbon steels (steam drum and PH tubesheet) and above of 540 °C for austenitic steels (SH, RH and EV) [19]. In this work, ASME Section VIII Division 2 [29] approach is used to calculate the fatigue damage of low temperature heat exchangers.

ASME Section VIII Division 2 [29] follows the S - N curves approach to calculate the allowable number of cycles. This means that the number of allowable cycles must be calculated as

function of the effective alternative stress, $S_{alt,j}$. In the case of an elastic-plastic analysis, the effective alternative stress is calculated as follows:

$$S_{alt,j} = \frac{E_{yf} \cdot \Delta \varepsilon_{eq,j}}{2} \quad (7)$$

where E_{yf} is the modulus of elasticity used to establish the design fatigue curve defined in Annex 3.F of ASME Section VIII Division 2 [29]. Then, the allowable number of allowable cycles is calculated as follows:

$$N_{a,j} = 10^{X_j} \quad (8)$$

where

$$X_j = \frac{c_1 + c_3 Y + c_5 Y^2 + c_7 Y^3 + c_9 Y^4 + c_{11} Y^5}{1 + c_2 Y + c_4 Y^2 + c_6 Y^3 + c_8 Y^4 + c_{10} Y^5} \quad (9)$$

$$Y = \left(\frac{S_{alt,j}}{C_{us}} \right) \left(\frac{E_{yf}}{E_T} \right)$$

and $C_i (i=1...11)$ are equation constants, C_{us} is the unit conversion factor and E_T is the modulus of elasticity evaluated at the average temperature of the cycle. Finally, the fatigue damage is calculated using Equation 4.

3.5. Lifetime calculation

The lifetime assessment of heat exchangers at high temperature typically combines creep and fatigue damage mechanisms. The failure criterion is set by the allowable limit of damage, which is estimated by means of the interaction diagram. It should be noted that the life of the device is reduced considerably when the creep and fatigue present a similar degree of damage. The sum of creep and fatigue damages must be lower than the maximum allowable limit of damage D_L (Equation 10). The interaction diagram may change depending on the material and/or standard used. British R5 code [15] uses a linear interaction model where the allowable limit of damage has a constant value of $D_L = 1$. ASME Section III [14] considers a bilinear interaction model for austenitic steels as stainless steel 304 and 316 (see Figure 7). With this approach, D_L values depend on the level of creep and fatigue damages and they have to be fitted to meet the bilinear rule.

$$D_f + D_c \leq D_L \quad (10)$$

Defining the fatigue and creep damages in one year as d_f and d_c , respectively, the lifetime of a pressure device can be calculated as follows:

$$n_{year} d_f + n_{year} d_c = D_L \rightarrow n_{year} = \frac{D_L}{d_f + d_c} \quad (11)$$

In the case of devices working at low temperature, the lifetime is calculated also using Equation 11 with a limit of damage of $D_L = 1$ and a creep damage $d_c = 0$.

Figure 7. Creep-fatigue calculation procedure.

4. Simulation of SG working conditions

In this work ABAQUS/CAE [24] software is used to carry out the stress analysis of the critical parts of the SG. ABAQUS/CAE [24] software offers different type of multi-physics analysis. Heat transfer, elastic and elastic-plastic stress analyses are conducted to investigate the structural integrity of the SG.

Based on the stress analysis of the SG of a previous work [12] and the working temperatures, the critical parts from a creep-fatigue point of view are: SH tubesheet, RH tubesheet, SH head-

nozzle junction and RH head-nozzle junction. The steam drum has been included in the SG lifetime analysis. Although its working temperatures are low enough to assure that the creep damage is negligible, the fatigue could be important due to the high stress variations during the start-up operation [12]. Finally, EV and PH are excluded of this analysis due to the low stress variations and the negligible creep damage.

4.1 Heat transfer and elastic stress analyses of SG start-up

To identify the time points at the minimum and maximum stress, the heat transfer and the elastic stress analyses are performed. The pressures, temperatures and temperature change rates described in section 3.2 are used to obtain the working conditions of the SG devices during the start-up.

Figure 8 illustrates the variation of the maximum normal stress ($\Delta\sigma_1$) for the critical SG points during the start-up. As it can be seen, all the components have the minimum stress point (i.e. the point with the maximum compressive load during the start-up) at around 6240 seconds. This is because at this time, the steam pressure starts to rise and the negative thermal stress is balanced with the positive pressure stress. The maximum stress point (i.e. the point with the maximum tensile load during the start-up) is obtained at steady state conditions for all components except the SH lower tubesheet, which obtains the maximum stress point at the beginning of the start-up. It is worth to mention that the high stress experienced by the SH lower tubesheet is due to the high temperature difference between the head and shell sides (see Figure 6).

Figure 8. Results of the elastic stress analysis.

The heat transfer simulations are made with linear type elements (DC3D8: 8-node linear heat transfer brick). The tubesheets are simulated considering only a sector of 45 or 30 degrees depending of the tube pitch layout. This is a widely used practice in tubesheets with axial-

symmetry in loads and geometry [16,41–43]. The details of the geometrical dimensions of the SG parts can be found in [25].

The results of the heat transfer simulations are illustrated in Figure 9. The results show the temperature field at the minimum stress point, i. e. when the maximum temperature differences in the metal walls are obtained (at 6240 seconds after the beginning of the SG start-up). The results show relative high temperature gradients in the outermost holes of the tubesheets (Fig. 9-a/c/e). This explains the high thermal stresses, which are typically obtained in this zone during transient operations [16,43]. Although, the outermost holes could be considered as a potential critical point, it is necessary consider the effect of the pressure stress. In the case of the T-junctions (Fig. 9-b/d/f), the higher temperature gradients are obtained at the inner wall surfaces of the T-junctions.

a) SH upper tubesheet

b) SH head-nozzle junction

c) RH tubesheet

d) RH head-nozzle junction

e) SH lower tubesheet

f) Drum-downcomer junction

Figure 9. Results of the heat transfer analysis.

4.2 Elastic-plastic stress analysis

The aim of the elastic stress analysis consisted of identifying the time of the maximum and minimum stress points (i.e. the maximum and minimum stress points are referred to the points at the maximum tensile load and maximum compressive load, respectively) during the start-up.

Once these points were determined, the elastic-plastic analysis is performed separately from the heat transfer analysis. Although both simulations can be performed simultaneously, ABAQUS/CAE [24] software allows decoupling the previously mentioned tasks in order to reduce the computational cost. In a first step, the temperature field at the minimum stress point is loaded considering zero stress state. In a second step, the temperature field at maximum stress point is loaded together with the pressure range. This is made to simulate the stabilized hysteresis path from the minimum to the maximum stress point, and then, the results are referred to the equivalent stress/strain ranges.

The results of the elastic-plastic analysis are shown in Figure 10. The elastic-plastic stress simulation is made using linear type elements (C3D8: 8-node linear brick). The boundary conditions for the elastic-plastic stress analyses of tubesheets are: i) fixed displacement on the normal direction of the planes: xy and xy rotated 45 degrees for the SH tubesheets and 30 degrees for the RH tubesheet. ii) fixed displacement on the normal direction of the end cap of the head. The boundary conditions used for the elastic-plastic stress analyses of the T-junctions are: i) xy symmetry (the displacement and rotation are fixed on the xy plane, and the x and y directions, respectively), ii) yz plane symmetry condition, iii) fixed displacement on the normal direction of the xz plane on the lower cut surface and iv) pressure load on z direction due to the end cap.

Figure 10-a/c/e shows the results of the elastic-plastic stress analysis of the SH and the RH tubesheets. As it can be seen, for the SH upper tubesheet (Fig. 10-a), the maximum equivalent stress range is obtained at the outermost holes on the head-side of the tubesheet. Other critical point is the head-tubesheet junction showing slightly lower stress ranges. In the case of RH tubesheet (Fig. 10-c) the maximum equivalent stress range is obtained at the middle zone holes. Other critical point of the RH tubesheet could be the outermost zone of the RH tubesheet-head joint due to the high stress ranges obtained. However, this zone has a negligible stress level at full-load and therefore it is expected a lower creep-fatigue damage than the middle zone holes. Lastly, the SH lower tubesheet (Fig. 10-e) obtains the maximum stress range in the outermost holes on the shell-side of the tubesheet.

Figure 10-b/d/f shows the result of the elastic-plastic stress analysis of the T-junctions. As it can be seen, the SH head-nozzle junction (Fig. 10-b), the RH head-nozzle junction (Fig. 10-d) and the drum-downcomer junction (Fig. 10-f) obtain the maximum stress ranges at P_1 . These results are in concordance with the results of a previous work [12], showing that P_1 obtains the higher stress range variations.

a) SH upper tubesheet

b) SH head-nozzle junction

c) RH tubesheet

d) RH head-nozzle junction

e) SH lower tubesheet

f) Drum-downcomer junction

Figure 10. Results of the elastic-plastic stress analysis.

The results presented in Figure 10 are valid to obtain the strain range required for the fatigue damage calculations. However, the calculation of the stress level at full-load conditions is required to estimate the creep damage. In this case, the cyclic stress-strain curve must be used following the procedure described in Figure 7. The results of the elastic-plastic stress analysis are summarized in Table 3. In addition to the stress range analysis, the equivalent stress results at full-load conditions are also shown.

Table 3. Results of the elastic-plastic analysis.

	$\Delta\sigma_{eq}$ (MPa)	$\Delta\varepsilon_{eq}$ ($\mu\varepsilon$)	σ_{eq} (full-load) (MPa)
SH upper tubesheet	261.3	2733.4	76.2
SH lower tubesheet	293.6	2730.7	185.8
SH head-nozzle junction	237.0	2368.4	142.1
RH tubesheet	166.9	1379.2	25.6
RH head-nozzle junction	144.1	1148.4	117.4
Drum-downcomer junction	269.7	1506.7	328.9

A mesh sensitivity analysis was conducted modifying the local mesh size at the critical points. For that purpose, the mesh size was adjusted from + 50% to -75% of the mesh size selected for the creep-fatigue analysis (Ref. case). The results showed that the SH head-nozzle junction and the steam drum-downcomer junction present deviations lower than 1% comparing the Ref. case, with the maximum mesh size reduction case (-75%). This shows the mesh size independence on the results at these points. However, the SH tubesheet obtained a maximum deviation of 3.8% comparing the Ref. case and the maximum mesh size reduction case (-75%). Still bearing in mind this deviation, the conservative approach used in this work for creep-fatigue damage calculations assures that the results are on the safety side (creep damage calculations without stress relaxation and without stress reduction factors).

5. Application to the lifetime analysis of the SG of a solar tower plant

The SG heat exchangers were designed using TEMA standards [44] and ASME code Sections II [30] and VIII Division 1 [45]. This is a common practice in the heat exchanger design [19]. However, this normative assumes continuous nominal operation and thus, it is unclear whether the cyclic operation of the SG can fulfill the lifetime design requirement of 25 years.

Two different cases are analyzed to investigate the lifetime of the SG:

- 1) Intermittent operation with a daily start-up and shutdown. The case of intermittent operation can be considered similar to Crescent Dunes STP [46]. The following assumptions are made for this case. On the one hand, 300 start-ups and shutdowns per year are considered [47]. On the other hand, 4550 hours at full-load conditions per year are assumed [25].
- 2) Baseload operation. The baseload case with continuous operation can be considered similar to 20 MWe STP Gemasolar, which operates 24/7 at full-load in the high solar radiation months and part-load in the winter [2]. An annual capacity factor of 70% can be achieved with this operation mode. Currently, there are not commercial STP of 100 MWe using this operation mode, however this case is plausible for a STP with high solar multiple and storage, or using a hybridization system [2,48]. Based on the aforementioned data, the annual operating hours was estimated as:

Annual Operating Hours = $8760 \times \text{CapacityFactor}$, obtaining 6100 hours at full-load conditions per year. On the other hand, one start-up/shutdown per year was assumed.

Table 4 shows the results of the lifetime estimation for the intermittent operation case. As it can be seen, the SH head-nozzle junction is the most critical point with an estimated lifetime of 20 years. On the one hand, the SH lower tubesheet presents higher lifetime than the SH head-nozzle junction despite that the first has a higher stress range and stress at full-load. This is due to the lower working temperature of the SH lower tubesheet, which is only 450°C, against 550°C of the SH head-nozzle junction. At 450°C the creep damage is practically negligible.

On the other hand, the strain range of the SH upper tubesheet is higher than the SH head-nozzle junction, and the results show a lower lifetime for the SH head-nozzle junction. This is mainly because the stress at full-load conditions is higher for the SH head-nozzle junction, and then, the combination of creep and fatigue damage gives a lower lifetime.

The lifetime analysis shows that the RH largely satisfies the lifetime condition of 25 years. This is mainly due to the lower working pressure which leads to lower stress levels.

Table 4. Lifetime results for the intermittent operation case.

	$D_f (-)$	$D_c (-)$	$D_L (-)$	Lifetime (years)
SH upper tubesheet	0.9797	0.0087	0.9884	29.0
SH lower tubesheet	1.0000	0	1.0000	29.8
SH head-nozzle junction	0.3568	0.2757	0.6324	20.0
RH tubesheet	0.9133	0.0372	0.9504	679.1
RH head-nozzle junction	0.0976	0.7724	0.8699	188.7
Drum-downcomer junction	1	0	1	47.8

As stated above, the SH head-nozzle junction does not fulfil the lifetime design condition of 25 years with its current SH head thickness (130 mm). For this reason, The SH head thickness is analyzed to increase the SH head-nozzle junction lifetime. Figure 11 shows the results of the analysis of the SH head-nozzle lifetime for the intermittent operation case. The analysis is performed using the elastic models for the mechanical and thermal stresses according to the EN 12952-3 European standard [49]. In addition, correction factors are used to meet the results of the elastic-plastic finite element analysis (FEA). As it can be seen, the combination of creep and fatigue damage leads to a trade-off on the SH head-nozzle junction lifetime. The lifetime decreases for low thicknesses due to the higher stresses at full-load producing higher creep damage. On the other hand, high thicknesses reduce the stress at full-load but increase the thermal stresses. This leads to higher stress ranges and higher fatigue damage. As a result, a maximum lifetime of 27.5 years can be obtained on SH head-nozzle junction using a thickness of 110 mm for the SH head.

The optimization analysis is made keeping constant a temperature change rate of 150°C/h. In the case of faster temperature change rates, the optimum could be displaced to lower thicknesses. Nevertheless, the SH head-nozzle junction could not achieve the lifetime of 25 years for faster temperature change rates. This is because for a 150°C/h there is a low margin of safety, as it can be seen in Figure 11. This suggests that the material (321 stainless steel) operates on its mechanical limit. Therefore, other material or SH head-nozzle junction design should be considered to increase the margin of safety. Possible material candidates would be the stainless steels: type 347 or Alloy 800H.

Figure 11. SH head-nozzle junction lifetime analysis for the intermittent operation case.

It is worth to mention that daily thermal stress cycles can be reduced using a large storage system. In this way, the daily thermal stress cycles can be eliminated operating at baseload conditions. This practically eliminates the fatigue damage on the SG, however, the creep damage grows because the operating hours at full-load condition increase.

Table 5 shows the results of the lifetime estimation for the baseload case. Due to the low start-up cycles assumed for the baseload case, it is not clear which approach should be used: monotonic or cyclic stress-strain curve. The monotonic curve may tend to higher deformations and higher fatigue damages. In contrast, cyclic curve may produce higher stress levels and higher creep damages. For this reason, the creep-fatigue analysis was performed using both approaches. Both analyses were conducted, showing that the use of the cyclic stress-strain curve obtains more conservative results. Therefore, the data showed in Table 5 refers to the results obtained using the cyclic stress-strain curve.

As in the intermittent operation case, the baseload case shows that the SH head-nozzle junction is the most critical point with a lifetime of 65.1 years. The results show a general increase of the lifetime for the baseload case. The creep damage is dominant against the fatigue damage. This means that the total allowable damage may tend to D_L values near to 1.

Table 5. Lifetime results for the baseload case.

	$D_f(-)$	$D_c(-)$	$D_L(-)$	Lifetime (years)
SH upper tubesheet	0.1866	0.5647	0.7513	1662
SH lower tubesheet	0.9998	7.34e-05	0.9999	8950
SH head-nozzle junction	0.0039	0.9910	0.9948	65.1
RH tubesheet	0.0443	0.8966	0.9409	9882
RH head-nozzle junction	3.79e-04	0.9991	0.9995	220
Drum-downcomer junction	1	0	1	14344

A second SH head-nozzle junction lifetime optimization is conducted for the baseload case, Fig. 12. As it can be seen, the lifetime increases for larger SH head thicknesses following an exponential growth. This means that the fatigue damage is negligible against the creep damage during the 25 years of operation. The results show that the optimum SH head thickness is obtained around 96 mm. Note that the optimum is considered for the thickness that ensures the minimum lifetime while minimizing the thickness of the SH head.

It is worth to mention that the SH head thickness obtained using the ASME VIII Division 1 [45] formulation and the allowable stress S is 86 mm. Despite the high safety factor considered in the allowable stress S (around 0.3 times the ultimate tensile strength), the SH head-nozzle junction does not achieve the minimum lifetime requirement. This is because the ASME VIII Division 1 [45] formulation considers only the membrane stress and does not take into account the concentration stress points, such as in the SH head-nozzle junction.

Figure 12. SH head-nozzle junction lifetime analysis for the baseload case.

6. Conclusions

A creep-fatigue analysis of the steam generator of a solar power plant is performed according to ASME Section III [14]. Two plant operation regimes are investigated: i) intermittent operation with daily start-ups and shutdowns; ii) baseload mode with continuous operation. Heat transfer and elastic-plastic simulations are carried out using ABAQUS/CAE software [24]. The heads and tubesheets of the superheater and reheater are selected as critical creep-fatigue points of the steam generator based on the results of a previous work [12].

The results of the creep-fatigue analysis show that a steam generator based on shell-tube heat exchangers can operate with 300 start-ups per year during 25 years with temperature change rates up to 150°C/h. The highest damage is obtained at the superheater head-nozzle junction whereas the reheater presents a lifetime one order of magnitude higher than the superheater.

A lifetime optimization of the superheater head-nozzle junction is performed for both operation modes considered. In the case of intermittent operation mode, a compromise between the fatigue damage and creep damage is obtained on the superheater head-nozzle junction. The results show that higher temperature change rates of 150°C/h would require a different design of the superheater head, and/or a material with better creep-fatigue properties than the properties of 321 stainless steel. In the case of baseload operation mode, creep is the dominant mechanism of damage and the lifetime of the superheater head-nozzle junction grows exponentially when the superheater head thickness increases.

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