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Fatigue analysis of the steam generator of a parabolic trough solar power plant

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Abstract

This work proposes a methodology to perform the fatigue analysis of the steam generator of a parabolic trough power plant. The following methodology assumes cycling duty scenario (300 start-ups and shut downs per year). Two start-up operations of the steam generator are analyzed: evaporator temperature ramp and heat transfer fluid temperature ramp. Furthermore, a part load operation is studied.

The results show that the most compromised parts of the steam generator are the reheater tubesheet for the TEMA CFU design, the steam drum-downcomer junction and the superheater nozzle. Besides, former reheater design (TEMA CFU) does not fulfil the 25-year lifetime condition. To overcome this problem, two alternative designs for the reheater are proposed: U-shell and two TEMA CFU in series. Finally, the lifetimes of TEMA NXU and kettle evaporator designs are compared. Our results show that TEMA NXU allows temperature ramps up to 9°C/min without putting the steam generator lifetime at risk, whereas kettle temperature ramps greater than 5°C/min do not fulfill the designed lifetime of 25 years.

Respect to the load changes, the evaporator drum is the most stressed element. A 50% load change produces a damage equivalent to 60% of the damage produced by a daily start-up/shut-down cycle.

Key words: *Parabolic trough plant; Steam generator; Stress analysis; Fatigue analysis.*

Nomenclature

Abbreviations

CCPs : combined-cycle plants

CSP : concentrating solar plants.

EV : evaporator.

EVts : evaporator tubesheet.

HCF : high cycle fatigue.

35 *HRSG* : heat recovery steam generator.

36 *HTF* : heat transfer fluid.

37 *LCF* : low cycle fatigue.

38 *PH* : preheater.

39 *PHts* : preheater tubesheet.

40 *PTPP* : parabolic trough power plant

41 *RH* : reheater.

42 *RHts* : reheater tubesheet.

43 *SG* : steam generator.

44 *SH* : superheater.

45 *SHNo* : superheater nozzle outlet.

46 *SHts* : superheater tubesheet.

47 *UTS* : ultimate tensile strength.

48

49 ***Symbols***

50 *D* : fatigue damage (-).

51 *E* : modulus of elasticity (MPa).

52 *F* : peak stress (MPa).

53 *K_e* : fatigue penalty factor (-).

54 *K_f* : fatigue strength reduction (-).

55 *K_{v,k}* : Poisson's correction factor (-).

56 *N_k* : number of allowable cycles (-).

57 *P_L* : primary local membrane stress (MPa).

58 *P_m* : primary general membrane stress (MPa).

59 *P_b* : primary bending stress (MPa).

60 *Q* : secondary stress (MPa).

- 61 S : maximum allowable stress defined in ASME Section II (MPa).
- 62 S_a : allowable stress amplitude for fatigue analysis (MPa).
- 63 S_{PS} : maximum allowable stress for ratcheting analysis (MPa).
- 64 S_y : yield stress limit (MPa).
- 65 X : exponent used to compute the allowable number of cycles (-).
- 66 h : thickness (m).
- 67 t : time (minutes).
- 68 y : vertical coordinate (m).
- 69
- 70 ***Greek Symbols***
- 71 ΔT : temperature difference (°C).
- 72 $\Delta S_{alt,k}$: effective alternative stress (-).
- 73 $\Delta S_{n,k}$: equivalent alternative stress (-).
- 74 $\Delta S_{LT,k}$: equivalent thermal stress (-).
- 75 $\Delta S_{P,k}$: equivalent peak stress (-).
- 76 ν : Poisson's ratio (-).
- 77 $\bar{\nu}_{T,EV}$: average temperature ramp of the evaporator (°C/min)
- 78 σ : stress (MPa).
- 79 σ_b : Bending stress defined according to the ASME linearization method (MPa).
- 80 σ_m : Membrane stress defined according to the ASME linearization method (MPa).
- 81 σ_p : Peak stress defined according to the ASME linearization method (MPa).
- 82
- 83 **1. Introduction**

Current electricity markets present a significant contribution of renewable energies such wind or photovoltaics. However, these energy sources are considered as non-dispatchables and, therefore, they may lead to unbalanced grid problems [1]. For this reason, dispatchable power plants are pushed to increase its flexibility in order to meet the electricity demand and compensate the unpredictable fluctuations of these renewable non-dispatchable energy technologies [2].

Concentrating solar power (CSP) plants with thermal storage have been positioned as a potential renewable alternative to conventional plants. These plants can be considered as dispatchable when integrating with cost-effective thermal energy storage, hybridization with fossil fuels and steam turbines [3]. The flexibility of CSP plants is a critical issue to increase their competitiveness. In this way, fast start-ups and load changes have become an important feature. Furthermore, fast start-ups are especially important for CSP plants because of the increase in the annual electricity production [4]. However, fast transient operations lead to high stresses on thick-walled components. Thus, a good plant management requires a proper estimation of the lifetime reduction/increment driving to the increment/reduction of the flexibility of the plant.

The steam generator (SG) is a critical component of conventional and CSP plants that limits the plant load changes. The SG is formed by four heat exchangers: preheater (PH), evaporator (EV), superheater (SH) and reheater (RH). These thick-walled components suffer thermal stresses due to temperature change rates [5]. Thus, a dynamic simulation of the SG is an essential previous step in a fatigue analysis in order to identify the most critical points.

Several researchers have investigated the critical stress points of combined cycle power plants (CCPPs) [6–9]. Dynamic models are usually developed to calculate the thermodynamic quantities (pressure, temperature, mass flow, etc). Most of these works estimate the mechanical and thermal stress using European Standards [6–8]. Only few accomplish a fatigue analysis. For example, Benato et al. [6] performed a fatigue analysis of a heat recovery steam generator (HRSG). They developed a dynamic simulator tool to estimate the trends of the main thermodynamic parameters (flow rates, temperatures and pressures). To validate this model, they also developed a HRSG model based on Modelica language [10]. A comparison study of both models for single pressure level HRSG can be found in [11]. They carried out a fatigue analysis using the European Standard EN 13445 [12]. They identified the most critical stress points, which are the high pressure steam drum and the SH collectors. The fatigue results show that the highest lifetime reduction is obtained in the SH collector for a load change of 50%. However, details of the geometric locations of the stress critical points and their respective stress concentration factors or even the number of allowable cycles are not shown. Angerer et al. [7] performed a fatigue analysis of the SH header using European Standard EN 12952-3 [13]. They studied a novel buffer storage system for the thermal decoupling of the gas turbine from the HRSG. The results show that this innovative system reduces around a 90% the fatigue damage in the SH header. Tonti et al. [8] presented some guidelines for calculation of in-service creep-fatigue damage of steam generators following European Standard EN 12952-4 [14]. Furthermore, an example of fatigue analysis of the SH outlet header is also presented.

Most of the literature focused in steam generation, which combines dynamic simulation and stress analysis, use European standards. Nevertheless, some works can be found using ASME code. For example, Mirandola et al. [15] analyzed the residual lifetime of a SH from a 320 MW coal-fired steam power plant. They studied the creep-fatigue damage on the SH pipes using ASME Section III [16]. The structural analysis to obtain deformations and stresses is performed using ANSYS software [17]. Two operation management cases are also compared economically. Furthermore, heat exchangers and/or pressure vessels have been also studied for

fatigue analysis. The ASME VIII Div 2 approach is used in [18,19]. Shen et al. [18] presented a fatigue analysis of a tubesheet subjected to thermal-shock. The results show that the most critical point is the tubesheet junction on the shell-side. Dong et al. [19] studied the lifetime of pressure vessel focusing in welded points. They also performed a comparison study of the fatigue calculations using different standards.

Commercial CSP plants typically use an indirect steam generator system based on conventional shell-and-tube heat exchangers [20–22]. The most common methodology for the design of shell-and-tube heat exchangers is based on TEMA standards [23] and ASME Section VIII Div 1 [24] considering nominal operation. However, the intermittent operation of the SG with daily start-ups, shutdowns and load changes, makes necessary the use of alternative standards that consider fatigue assessment, such as ASME Section VIII Div 2 [25] or ASME Section III Subsection NH [16].

This work analyzes if the SG of a parabolic trough power plant (PTPP) based on conventional shell-and-tube heat exchangers is able to operate 25 years with daily start-ups and shutdowns. For that purpose, a fatigue analysis of the SG is carried out. Several critical points of the SG are considered: tubesheets, tubesheet junctions, head-nozzle junctions and drum-downcomer junction. Two transient operations of the SG start-up are studied for the fatigue analysis. The first considers five different evaporator temperature ramps whereas the second considers five inlet-temperature ramps. In addition, a SG load change operation is also analyzed. The stress calculations are made using the models presented in González-Gómez [26], which have been validated using finite element analysis. The fatigue calculations are performed following ASME Section VIII Div 2 [25].

2 Steam generator description

The case selected for the fatigue analysis is a SG of a 50 MWe PTPP with 7.5 storage hours and a solar multiple of 2 (i.e. the ratio between thermal power of the solar field power and thermal power of the power block). The energy collected in the solar field is used to heat up the heat transfer fluid (HTF), which enters to the SG at 393°C. In the SG the heat is transferred to high and low pressure steams until the HTF cools down to 293°C returning to the solar field at this temperature. The SG is divided into two parallel trains each one formed by a superheater, reheater, evaporator and preheater. Figure 1 illustrates one train of heat exchangers of the SG. The high pressure water/steam is placed on tube-side whereas the HTF is placed on the shell-side for SH, RH and PH. The fluids are placed on the opposite way for the EV. The SG heat exchangers are designed following TEMA [23] and ASME standards [24]. The SH and PH consist of TEMA DHU heat exchangers, which provides very low pressure drop on the shell-side. The RH consists of TEMA CFU heat exchanger to provide high thermal efficiency. The EV consists of two TEMA NXU heat exchangers arranged in parallel with a steam drum. This design based on natural circulation system avoids the power pump consumption and reduces the potential fouling on the shell-side. In addition, this design with three separated units, two heat exchangers and one steam drum, reduces the shell diameter, and hence the metal wall thickness. In this way, the behavior against thermal stress is improved. The material selected for the shell, head and tubesheet of the heat exchangers is SA-266-Gr. 2. The material selected for the tubes is SA-210-Gr. A1 and for the steam drum is SA-516-Gr. 70. Further details about whole solar plant and SG design parameters can be found in González-Gómez et al. [27]. Table 1 shows the mechanical properties of the materials for different temperatures according to ASME Section II [28].

Table 1. Properties of the heat exchangers materials at different temperatures.

Material	T (°C)	S (MPa)	S_y (MPa)	S_{PS} (MPa)	UTS (MPa)	E (GPa)	ν (-)
SA-266-Gr.2	240	136	204	408	483	189	0.3
	380	101	171	342	476	171	0.3
SA-516-Gr.70	240	138	216	432	483	189	0.3
	380	101	181	362	476	171	0.3

In order to study the fatigue damage of the SG, the start-up operation is analyzed. To that end, all heat exchangers are assumed to be at 240°C, which is the saturation temperature of the water of the steam drum at the beginning of the start-up. In a first step, the HTF is sent to the EV at 260°C and it is also circulated to the PH. The HTF mass flow is fixed to 4%. The feed water temperature is assumed equal to 100°C and it is heated using drum water circulation to 220°C in order to avoid high thermal stress in the PH. This operation is maintained until the steam generation is established to around 4% of the nominal steam mass flow. Then, the saturated steam is circulated to the SH. At the same time, the HTF is sent to the SH and the RH. The SH steam outlet is bypassed to the RH, where the attemperation system keeps the RH steam inlet in a suitable temperature range. The HTF inlet temperature is increased by a constant change rate until 393°C. When the SH and the RH outlet steams achieve 320°C, they are sent to the turbine in order to be synchronized. After around 8 minutes, the turbine synchronization is completed. Then, the HTF mass flow rate is increased by a ramp until the 100%. As a result, the steam drum pressure and the steam mass flow rate increase until nominal conditions. After around 45 minutes, the SG start-up is completed. More details of the SG start-up procedure and the models used for the SG simulation can be found in [26].

The critical points of the SG for the fatigue analysis are obtained from the results of the SG stress analysis [26]. These points are shown in Figure 1. As it can be seen, both the tubesheet at no-tube-lane zone and the nozzle-head junction are considered for the SH. The critical point for the RH is the tubesheet at no-tube-lane zone point. The RH nozzle-head junction is not considered because the thin metal wall produce low thermal stress at this point. In the case of the EV, the tubesheet at junction at no-tube-lane zone and the drum-downcomer junction are considered. Finally, the most limiting point of the PH is the tubesheet at the rim zone.

SHNo	SH head-nozzle junction	SHts	SH Tubesheet	RHts	RH Tubesheet

EVts	EV Tubesheet	Drum	Drum-downcomer junction	PHts	PH Tubesheet

Figure 1. Critical SG points for the fatigue analysis.

3 Methodology for the fatigue analysis: Lifetime estimation according ASME Section VIII Div 2 [25].

Fatigue is defined as the damage of a structural component due to cyclic loads that involve cycles of stresses and strains. The fatigue failure is produced as the accumulation of the damage at localized zones, where the nucleation of cracks is extended until material fracture. There are two types of fatigue mechanisms: high cycle fatigue (HCF) and low cycle fatigue (LCF) [29]. In LCF, the peak stresses are above the yield tensile strength, obtaining plastic strains. The allowable number of cycles are typically suited below of 10^4 or $5 \cdot 10^4$. In HCF, the stresses are below the yield tensile strength and hence the strains produced are in the elastic region.

Creep is the anelastic deformation mechanism produced when a constant stress works at relatively high temperatures. The creep damage is mainly function of the stress, time and

temperature. Normally, the creep damage becomes significant for temperatures above the 40% of the melting temperature. In the case of low alloy steels this temperature is around 380 °C whereas for austenitic steels is around 540 °C [29]. Creep and fatigue damage can be combined using Palmgren-Miner equation [30], where it can be seen that the damage is much higher if creep and fatigue take place simultaneously.

Creep and fatigue analysis are normally accomplished by using design standards. Some of the most important are: ASME Section III [16] and VIII [25], UNE EN 13445-3 [12], UNE EN 12952-3 [13]. In this work, the fatigue analysis is performed using ASME Section VIII Div 2 [25] due to the work temperatures are below of the 40% of the melting temperature. Furthermore, 300 start-ups and shutdowns per year have been considered.

The assessment under cycling operation of the heat exchangers for standard industrial applications at low temperatures (<427°C) can be made following ASME Section VIII Div 2 [25]. In the case of higher temperatures, the creep damage should be considered and then other standards as ASME Section III [16] should be used. Two assessments should be considered for the protection against cycling operation: ratcheting and fatigue. The equivalent stresses are calculated considering operating load cycles, instead of design loads.

Figure 2 illustrates this stress classification according to ASME Section VIII Div 2 [25]. The structural stresses can be classified in three categories. The first category is formed by the primary stresses, which are: general membrane (P_m), local membrane (P_L) and primary bending (P_b). They are calculated considering only mechanical loads and excluding local stress discontinuities. The second category is formed by secondary membrane and bending (Q). They are calculated considering mechanical loads and/or thermal loads that produce thermal differential expansion. The local stress concentration factors are excluded. The third category is the peak stresses (F). These stresses are calculated considering mechanical loads and/or thermal loads that do not produce distortion of the vessel shape and also considering local stress concentrations.

Typically the first step in the heat exchanger design consists of the estimation of the thicknesses of the main parts (shell, tubes, head, tubesheet, nozzles, etc) using ASME Section II [28] and Section VIII Div 1 [24] considering design loads, primary membrane and bending stresses, and the maximum allowable stress S defined.

Operating loads instead of design loads must be used for the cycling assessment. In addition, the stress due to thermal expansion and local discontinuities must be considered. For example, for ratcheting assessment membrane, bending and secondary stresses must be considered ($P_L + P_b + Q$). Moreover, the maximum allowable stress S_{PS} is calculated as the highest value between: three times the allowable stress limit S or two times the yield stress of the material S_y . The results of the ratcheting assessment of the SG can be found in González-Gómez [26].

In case of fatigue assessment the peak stress must be included (F), and hence, the local stress concentration factors must be used. The allowable stress amplitude for fatigue analysis S_a is calculated according the $S-N$ curves from ASME Section VIII Div 2 [25]. In the case of the steam drum (carbon steel), the maximum allowable stress amplitude S_a for 10^4 and $5 \cdot 10^3$ cycles are 242 and 301 MPa, respectively.

Figure 2. Stress classification according to ASME Section VIII Div 2 [25].

According to the ASME linearization method, the stress can be classified mainly in three types: membrane, bending and peak, which are defined analytically as follows [19]:

$$\begin{aligned}
 \sigma_m &= \frac{1}{h} \int_0^h \sigma_x(y) dy \\
 \sigma_b &= \frac{6}{h^2} \int_0^h \sigma_x(y) \cdot \left(\frac{h}{2} - y \right) dy \\
 \sigma_p &= \sigma_x(y) - \sigma_m - \sigma_b \cdot \left(1 - \frac{2y}{h} \right)
 \end{aligned} \tag{1}$$

where the $\sigma_x(y)$ is the normal stress (normal to fissure propagation plane) obtained from shell theory neglecting the local stresses due to singularities.

The fatigue analysis is normally accomplished by using time series of load cycles, also called Remaining Sequence of Extremes (RSE) [8]. There are different possibilities to estimate the number of cycles in a fatigue analysis based on RSE. The main idea of these methods consists of counting the maximum and minimum relative extremes. There are different ways to consider the load cycles by means of different patterns to link an extreme with the other. Hence, the fatigue results can vary around 25 % depending of the selected method [8]. For the sake of simplicity, in this work the maximum stress range is considered as one cycle per day (Figure 3).

Figure 3. Example of stress range calculation for the fatigue analysis for one cycle.

The elastic-fatigue analysis of ASME Section VIII Div 2 [25] consists of the following steps:

1. Determine the load history: it is calculated according to the results of the SG simulations where the stresses of the main components are calculated for each time step. The critical points of the SG are chosen according to the results of the González-Gómez [26].
2. Calculation of the stress range $\Delta\sigma_{ij}$ for the principal stress directions considering their respective signs.
3. Use the counting cycle method to the stress range $\Delta\sigma_{ij}$ to estimate the maximum relative stress $^m\sigma_{ij,k}$ and minimum relative stress $^n\sigma_{ij,k}$ of each k cycle. The total M fatigue cycles ($k=1,...,M$) and the actual number of repetitions n_k of each k cycle are also calculated.
4. Calculation of the equivalent stress range considering normal and shear stresses:

$$\Delta\sigma_{ij,k} = ^m\sigma_{ij,k} - ^n\sigma_{ij,k}$$

$$\Delta S_k = \frac{1}{\sqrt{2}} \left[\frac{(\Delta\sigma_{11,k} - \Delta\sigma_{22,k})^2 + (\Delta\sigma_{11,k} - \Delta\sigma_{33,k})^2 + (\Delta\sigma_{22,k} - \Delta\sigma_{33,k})^2 + 6(\Delta\sigma_{12,k}^2 + \Delta\sigma_{13,k}^2 + \Delta\sigma_{23,k}^2)}{2} \right]^{0.5} \quad (2)$$

5. The effective alternating stress amplitude is calculated as follows:

$$\Delta S_{alt,k} = \frac{K_f K_e (\Delta S_{P,k} - \Delta S_{LT,k}) + K_{v,k} \Delta S_{LT,k}}{2} \quad (3)$$

where the equivalent stress range, $\Delta S_{P,k}$, is calculated considering primary, secondary and peak stresses. The equivalent thermal stress range, $\Delta S_{LT,k}$, is calculated considering secondary and peak stresses produced by thermal loads. The fatigue strength reduction, K_f , should be included to take into account local notch or weld effects. The fatigue penalty factor, K_e , is calculated as a function of the material and the equivalent alternative stresses, $\Delta S_{n,k}$, which is calculated considering only primary and secondary

stresses. Lastly, the Poisson correction factor due to thermal loads, $K_{v,k}$, is computed as function of the elastic and the equivalent plastic Poisson's ratios.

6. The allowable number of cycles is calculated using $S-N$ curves according to ASME Section VIII Div 2 [25]:

$$N_k = 10^X \quad (4)$$

where the exponent to compute the allowable number of cycles, X , is calculated as a function of the modulus of elasticity, E , the equivalent stress range, ΔS_{alt} , and the ultimate tensile strength, UTS .

7. Finally, the total fatigue damage for k cycles is calculated as follows:

$$D = \sum_{k=1}^M \frac{n_k}{N_k} \leq 1 \quad (5)$$

Further details about the elastic-fatigue calculations can be found in Sowoniski et al. [31]. In addition, they presented a wide variety of design calculations examples following ASME Section VIII Div 2 [25].

4. Results of the fatigue analysis

The fatigue analysis is performed for the SG start-up and shutdown with different ramp rates of the evaporator temperature and the HTF inlet temperature. In addition, the fatigue damage of the 50% load change operation is also studied. A reference transient case ("Ref.") is used for comparison. This transient case was previously analyzed in [26]. In addition, four transient cases are considered: two faster than the reference case, "25% Faster" and "50% Faster", and two slower than the reference case, "25% Slower" and "50% Slower". Table 2 shows the temperature ramps of each transient case.

Table 2
Transient cases for SG fatigue analysis.

Transient case	50% Slower	25% Slower	Ref.	25% Faster	50% Faster
Average evaporator temperature ramp (°C/min)	3.5	5.2	7	8.7	10.5
HTF inlet temperature ramp (°C/min)	4	6	8	10	12

The fatigue analysis and the consequent lifetime estimation are computed in accordance to ASME VIII Div 2 [25]. The lifetime is calculated by dividing the number of allowable cycles into 300 start-ups and shutdowns per year. It is worth to point out that, from the operation point of view, this analysis can be considered conservative. This is because it has been considered relatively low initial temperatures of the heat exchangers (240°C).

To carry out the fatigue analysis on the SG numerous dynamic and stress models are needed. To estimate the steam drum stresses 19 different temperature profiles along the drum wall radius are calculated. The heat exchangers are discretized along the longitudinal coordinate into 26 cells each one in order to obtain enough accurate results. For the tubesheet temperature field

model, an axis-symmetric discretization mesh model of 25x25 is made for radial and longitudinal coordinates. More details of these models can be found in [26].

4.1 Fatigue analysis for the evaporator temperature ramp operation during the SG start-up.

Figure 4 shows the start-up results for different evaporator temperature ramps. In order to accomplish such operation, the HTF mass flow rate is increased to obtain the selected average temperature ramp in the evaporator, which is calculated according Equation 6. A significant increment of the drum pressure must be accomplished during the start-up from 34 bar to 106 bar (Figure 4-a), whereas the temperature varies from 240°C to 314°C (Figure 4-b). Figure 4-c/d show the HTF inlet conditions to SG during the SG start-up.

$$\bar{v}_{T,Ev} = \frac{T_{nom} - T_{start,ramp}}{t_{nom} - t_{start,ramp}} \quad (6)$$

Figure 4. Simulation results for different evaporator temperature ramps. a) Drum pressure; b) Drum temperature; c) HTF mass flow; d) HTF inlet temperature.

According to the results shown in [26], the critical points of the SG for the fatigue analysis are: tubesheet ligaments, tubesheet junctions, SH nozzle-head junction and steam drum-downcomer

junctions (see Figure 1). Among these points, the most sensitive to the evaporator temperature ramps are the SH nozzle and drum junctions. Figure 5 shows the stress evolution of these points for the different ramps considered. Note that two points are analyzed in the T-junctions, as was explained in the stress model shown in Figure 1.

Figure 5. Stress results for different evaporator temperature ramps: a) Point P₁ of the drum-downcomer junction; b) Point P₂ of the drum-downcomer junction; c) Point P₁ of the SH inlet nozzle-head junction; d) Point P₂ of the SH inlet nozzle-head junction.

Figure 6 shows the fatigue results. As it is expected, the steam drum of TEMA NXU evaporator presents a high sensitivity against the evaporator temperature ramp operation. This means that the transient thermal stress has a high impact on the total stress. The results show that TEMA NXU evaporator satisfies the 25 years lifetime condition even for the "50% faster" transient case.

Another critical point of the SG train can be found in the reheater, due to the high temperature difference across the no-tube-lane zone of the tubesheet. Note that the RH stress is not sensitive to the evaporator temperature ramps since the maximum stress is obtained at nominal conditions, with a value of 509.5 MPa [26]. The results show that the lifetime obtained is around 18 years, violating the design condition of 25 years in spite of the satisfactory results

obtained in the stress analysis performed in [26]. Note that the lifetime is calculated considering 300 start-ups and shutdowns per year.

For the SH nozzle-head junction, a slight lifetime variation is observed for the transient cases studied. In a first step, a similar lifetime behavior of the drum-downcomer junction is expected. since both parts show similar stress values for point P_1 (Figure 5-a and Figure 5-c). However, a significant difference is obtained in the stress of P_2 (Figure 5-b/d), which is more sensitive for the drum-downcomer junction (Figure 5-b), and hence, this leads to higher variations of the lifetime calculations (Figure 6).

Figure 6. Fatigue results for different evaporator temperature ramps.

In case of EV, SH and PH tubesheets, the calculations show some variations in the lifetime for different evaporator temperature ramps. Their lifetimes are greater than 25 years, with enough safety margin, especially for PH. It is worth to mention that the lifetime does not have an increment for the slower evaporator temperature ramps in the case of SH tubesheet. This is because the HTF temperature ramp is kept constant (Figure 4-d), and then, for a slower evaporator temperature ramp, the SH tubesheet temperature difference increases (Figure 7-a/b). Hence the stress increases as well (Figure 7-c). As a result, the lifetime decreases with a slower evaporator temperature ramp. A similar behavior is also obtained for the EV (Figure 7- d/e/f). However, for the PH tubesheet the stress presents an opposite behavior, i. e. the stress increases for higher evaporator temperature ramps (see Figure 7-g/h/i). This is due to the delay of the PH water inlet temperature control system.

Figure 7. Simulation results for SH, EV and PH tubesheets: a) SHts steam temperatures; b) SHts steam temperature difference; c) SHts stress; d) EVts HTF temperatures; e) SHts HTF temperature difference; f) EVts stress; g) PHts water temperatures; h) PHts water temperature difference; i) PHts stress.

4.2 Fatigue analysis for the HTF inlet temperature ramp operation during the SG start-up

Figure 8 shows the start-up simulation results for the different HTF inlet temperature ramps. The HTF temperature ranges from 260 °C to around 393 °C. A HTF temperature ramp around 8 °C/min is considered as “Ref” case. This means that the fast cases are around 10 and 12 °C/min, whereas the slow cases are around 6 and 4 °C/min. It should be noted that HTF temperature rates are normally below of 8 °C/min according to data reported in [32,33]. This is due to the high thermal inertia of the solar field, combined with the lower solar radiation in the early morning. However, higher HTF temperatures of 8 °C/min could be possible for PTPPs with HTF gas or electrical heaters.

Figure 8. Simulation results for different HTF inlet temperature ramps: a) HTF inlet temperature; b) SH outlet steam temperature.

Figure 9 shows the stress values of the most sensitive points, which are P_1 and P_2 of the SH nozzle-head junction. Although the temperature step is similar for HTF and steam sides, the lower thickness of the SH nozzle on shell-side compared to the nozzle of head-side leads to a significant thermal stress reduction. According to Figure 9, the point P_1 shows a significant stress variation compared to point P_2 .

Figure 9. Stress results for different HTF inlet temperature ramps: a) Point P_1 of the SH outlet nozzle-head junction; b) Point P_2 of the SH outlet nozzle-head junction.

The results of the fatigue analysis for the different HTF temperature ramps are shown in Figure 10. Most of the critical points of the SG are insensitive for the HTF temperature ramps, except the SH nozzle-head junction. The results show that the HTF temperature rates equal and lower to $8^\circ\text{C}/\text{min}$ obtain lifetimes higher than 25 years for the SH nozzle-head junction.

Figure 10. Fatigue results for different HTF temperature ramps.

4.3 Fatigue analysis for a 50% load change operation of the SG

Current electricity markets push CSP plants to increase their flexibility. In this scenario, load change operations might be frequent to meet the electricity market requirements at peak demand periods and for grid stabilization. For this reason, a 50% load change operation is selected to analyze the fatigue damage of the SG.

Figure 11 illustrates the load change simulation results for different evaporator temperature ramps. The part load performance of the turbine is modeled as sliding pressure operation mode. Then, to obtain a 50% load change, a pressure around 50 bar (Figure 11-a) is necessary to decrease 50% the steam mass flow. The evaporator temperature ramps studied are the same that the start-up case (Table 2). The HTF mass flow is calculated to meet the SG and turbine part load performance (Figure 11-c). It should be noted that the steam turbine inlet temperatures are kept constant during the load change operation, and then HFT inlet temperature to the SG is also kept constant (Figure 11-d).

Figure 11. Simulation results for a 50% load change: a) Drum pressure; b) Drum temperature; c) HTF mass flow; d) HTF inlet temperature.

Figure 12 shows the most sensitive stress points for the load change operation: steam drum (Figure 12-a/b) and the SH nozzle (Figure 12-c/d) junctions. According to the results obtained in the finite element analysis of a T-junction in [26], P_1 stress values are more sensitive to pressure changes, whereas the stress values of P_2 are more sensitive to the transient thermal stresses. This can be checked analyzing the sign of the stresses: pressure stresses are always in traction (positive) whereas the thermal stresses are in traction (positive) for cooling processes and in compression (negative) for heating processes.

Figure 12. Stress results for 50% load change: a) Point 1 of the drum-downcomer junction; b) Point 2 of the drum-downcomer junction; c) Point 1 of the SH inlet nozzle-head junction; d) Point 2 of the SH inlet nozzle-head junction.

Fatigue results of the 50% load change are shown in Figure 13. The results are expressed as the fatigue damage equivalent, D_{eq} . In the case of $D_{eq,j} = 1$, the fatigue damage of the component j is the same as the fatigue damage produced by the start-up reference case. Figure 13 shows that the highest damage is obtained for both the SH nozzle and the steam drum junctions. This makes sense because the load change operation produces a steam drum pressure variation, which involves an evaporator temperature ramp. On the other hand, the damage on the SH, RH, EV and PH tubesheets present low sensitivity for load changes. This is because the tubesheet temperature difference variations are higher for the start-up than the 50% load change operation, and then the thermal stress variations are higher as well. It should be noted that the damage in tubesheets are dominated by the thermal stresses at no-tube-lane of the tubesheet. As a result, the tubesheet fatigue damages during the start-up are one or two orders of magnitude higher than at 50% load change case.

Figure 13. Fatigue results for a 50% load change transient case.

4.4. Fatigue analysis for TEMA NXU and kettle evaporators

Pioneering PTPPs such as Luz Solar Electric Generating Stations installed a kettle evaporator type obtaining successful operation results [34]. For this reason, this approach was also selected for more recent PTPPs [21,35]. The kettle evaporator provides a relatively low capital cost solution because this design integrates the heat exchanger and the steam drum in one shell. However, it leads to higher shell diameters and then to higher metal wall thicknesses than other designs, such as the TEMA NXU evaporator. Due to that, the dynamic behavior of the kettle evaporator was investigated for a SG start-up operation, obtaining a stress 150% higher than the TEMA NXU evaporator [26]. Because it indicates a higher risk of premature fatigue failure, the lifetime of the kettle evaporator is also investigated.

The critical stress points of kettle and TEMA NXU evaporators are illustrated in Figure 14. These points are the same for both designs: the tubesheet junction at no-tube-lane and the drum-downcomer junction. Details of the kettle design parameters used for this analysis are shown in [27] and details about the stress analysis of the kettle evaporator can be found in [26].

Figure 14. Critical points of TEMA NXU and kettle evaporators.

Figure 15 shows the fatigue results of the evaporator designs. The transient operation selected for the fatigue analysis is the evaporator temperature ramp, due to the higher sensitivity to the stress compared to the HTF temperature ramp operation. As it can be seen, the kettle design does not achieve the minimum lifetime condition of 25 years for the reference case in the drum-downcomer junction. This is mainly due to the high metal wall thickness, which also leads to a high thermal stress. For the "25% slower" transient case (5.2 °C/min) the kettle design has enough safety margin for the lifetime design condition. This means an increment of the start-up time around 5 minutes compared to the reference case (7 °C/min) and around 7 minutes for "25% faster" case. It should be taken into account that the reference case is the minimum value of the range of evaporator ramps to optimize the plant electricity production [4]. Hence, it can be concluded that kettle evaporator might not be an appropriate design for a PTPP with a fast daily start-up operation.

On the other hand, the EV tubesheet results obtain a very similar lifetime for the kettle and TEMA NXU evaporators. In these components, the stress is dominated by the temperature difference across the no-tube-lane of the tubesheet, which is the same for both designs.

Figure 15. Fatigue results of the comparison with TEMA NXU and kettle evaporators.

4.5 Fatigue analysis for the alternative reheater designs

As stated above, a high potential risk of fatigue failure is obtained for the RH design with a single TEMA CFU shell heat exchanger. In order to find a RH design with enough lifetime safety margin, two alternative design options are proposed (see Figure 16): i) dividing the RH into two TEMA CFU shell heat exchangers in series or ii) using a U-shell/U-tube heat exchanger design to avoid the high temperature difference in the no-tube-lane zone.

To carry out the fatigue analysis of the proposed designs, first it is necessary calculate the main design parameters. For that purpose, it has been followed the methodology proposed by González-Gómez et al. [27], where the main steps can be resumed as: i) selection of preliminary design parameters, ii) thermal and hydraulic calculations, iii) mechanical design following ASME Section VIII Div 1 [24], iv) cost analysis and optimization using genetic algorithm.

The results of the proposed RH designs are shown in Table 3. As it can be seen, the U-shell RH obtains slightly lower capital cost compared to two TEMA CFU RH. Both designs include double-segmental baffle in order to reduce the pressure drop on the shell-side. Nevertheless, the U-shell RH obtains a pressure drop on the shell-side around two times lower, because of the high number of tube passes of the two TEMA CFU RH. On the other hand, the TEMA CFU design provides advantages for the cleaning operations on the shell-side because removable heads can be used.

URHts	RH head-Tubesheet junction	divRHts	RH Tubesheet

Figure 16. Critical stress points of the proposed designs for RH.

Table 3
Design parameters for the proposed designs for Reheater.

Parameter	U-shell RH	Two TEMA CFU RH
Shell diameter (mm)	810	1114
Baffle cut (%)	20.4	18.5
Baffle spacing (mm)	203	226
Tubes ext. diameter (mm)	12.7	22.2
Tubes int. diameter (mm)	10.2	18
Tube pitch (mm)	15.9	27.8
Tube layout (°)	90	90
Tube passes (-)	1	2
Tubes number (-)	1856	564
Tube length (m)	18	8.71
Shell thickness (mm)	12.7	15.9
Tubesheet thickness (mm)	55.1	75.7
Mass flow reheat steam the tube-side (kg/s)	24.8	24.8
Mass flow thermal oil on the shell-side (kg/s)	35.1	35.1
Flow velocity on the tube-side (m/s)	23.3	24.7
Flow velocity on the shell-side (m/s)	0.33	0.47
Convective heat transfer coefficient on the tube-side ($W/m^2 \text{ } ^\circ C$)	989	932
Convective heat transfer coefficient on the shell-side ($W/m^2 \text{ } ^\circ C$)	1457	1503
Fouling resistance on the tube-side ($^\circ C \text{ m}^2/W$)	1.765e-4	1.765e-4
Fouling resistance on the shell-side ($^\circ C \text{ m}^2/W$)	1.765e-4	1.765e-4
Overall heat transfer coefficient ($W/m^2 \text{ } ^\circ C$)	421	408
Heat exchange area (per shell) (m^2)	1333	686
Pressure drop on the shell-side (kPa)	45.77	132.3
Pressure drop on the tube-side (kPa)	84.6	113.4
Baffle type	Double-segmental	Double-segmental
Total number of shells (-)	2	4
Total investment cost (k€)	616	671

Once the heat exchanger design is performed, the following step is the stress analysis. The critical stress points for both designs are shown in Figure 16. The two TEMA CFU RH in series shows the same critical point as the single RH design: the no-tube-lane zone of the tubesheet. In constrast, the U-shell RH design avoids the high thermal gradients in the no-tube-lane, obtaining the head-tubesheet junction as the critical stress point [36].

Figure 17 illustrates the fatigue results for the proposed RH designs. Since the temperature ramps in the evaporator and in the HTF present low influence on the RH stress, only the reference transient case is considered. As mentioned before, the single TEMA CFU RH does not fulfill the lifetime design condition of 25 years. In constrast, both alternative RH designs satisfy this condition. It is worth to mention that the U-shell RH design provides a lifetime of one order of magnitude higher than the two TEMA CFU in series, since the first design avoids the undesirable thermal stresses at no-tube-lane of the tubesheet.

Figure 17. Fatigue results of the proposed designs for RH.

5. Conclusions

A fatigue analysis of the steam generator of a parabolic trough power plant is carried out using ASME Section VIII Div 2 [25]. Two transient operations of steam generator start-up are investigated: the evaporator temperature ramp and the heat transfer fluid temperature ramp. In addition, a 50% load change of the steam generator is also studied.

The results of the fatigue analysis for the steam generator start-up and shutdown show significant variations on the lifetime of the heat exchangers when different evaporator temperature ramps are used. An extremely short lifetime (18 years considering 300 start-ups per year) is obtained for the TEMA CFU reheater due to the high thermal stress obtained on the tubesheet. For this reason, two alternative reheater designs are investigated to solve this issue: U-shell and Two TEMA CFU in series. The results show that both designs satisfy the lifetime design condition of 25 years.

As a result of the fatigue analysis, it is shown that the TEMA NXU evaporator provides a very good alternative against kettle evaporator. The TEMA NXU evaporator allows the reduction of the steam generator start-up time with temperature ramps up to 9 °C/min obtaining enough safety margin to prevent premature SG failure.

The results show that the load change operation produces the highest fatigue damage on SH nozzle and steam drum junctions. In addition, it is observed low fatigue damage on heat exchanger tubesheets during the load change operation.

Finally, it can be concluded that the assumption of continuous operation conditions used in widely used standards for heat exchanger design such TEMA [23] or ASME VIII Div1[24] are not suitable for the de steam generator design of a parabolic trough power plant. Therefore, the correct steam generator design should consider the trade-off exiting between lifetime, complexity and cost (capital, operation and maintenance).

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