On the fatigue stress range calculations with on-line monitoring systems in nuclear power plants

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Abstract. Nuclear power plants are generally designed and inspected according to the ASME Code. This code indicates the stress intensity (S_{INT}) as the parameter to be used in the stress analysis of components. One of the particularities of S_{INT} is that it always takes positive values, independently of the sign of the stress (tensile or compressive). This circumstance is relevant in the Fatigue Monitoring Systems used in nuclear power plants, due to the manner the different variable stresses are combined in order to obtain the final total stress range. This paper describes some situations derived from the application of the ASME Code, shows different ways of dealing with them and illustrates their influence in the evaluation of the fatigue usage through the application to a practical example.

1. INTRODUCTION

Monitoring Systems (MS) constitute an alternative for the fatigue assessment of components and structures. These systems allow performing such type of assessments in an automated real time way and, for this purpose, require the record of all those parameters affecting the stress state of the component being assessed.

When using MS, the coupling of the different stresses acting in the assessed component is a key factor. Commonly, the different stress states (coming from the different loads) are added using the so called Stress Intensity (S_{INT}) parameter, as indicated by the ASME Code [1]. This practice can lead to errors in the calculation of the total stress state, as shown below. Therefore, despite this methodology simplifies the analysis, it sometimes provides results that are not always optimised.

The main objective of this work is to determine the effect of this simplification by comparison with an assessment in which the appropriate stress coupling is performed, and its corresponding effect in the fatigue evaluation of components.

2. STRESS INTENSITY CONCEPT

In conventional fatigue MS the different loads are analysed independently. Therefore, the stress analysis requires the combination of the corresponding stresses. The direct addition of the different stress components can lead to errors in the determination of the total stress of certain transients, given that such addition is usually performed through the S_{INT} parameter, as indicated by the ASME Code [1]. The S_{INT} is defined as the maximum absolute value of the differences between the principal stresses (Tresca Criterion). This parameter is always positive, so it is not possible to distinguish between tensile and compressive stresses:

$$S_{INT} = \max(|S_1-S_2|, |S_2-S_3|, |S_3-S_1|) \quad (1)$$

S_1, S_2 y S_3 are the principal stresses.

If the sign of the stresses is not known, errors in the coupling process (and then in the total stress calculation) can be produced.

3. STRESS COUPLING

An adequate stress coupling methodology must be applied when using MS due to their specific stress evaluation methodology, which consists in the independent analysis of the different loads acting in the component. In a conventional assessment, stresses are obtained through the analysis of all the loads acting simultaneously, without considering any coupling effect. This methodology can not be applied when using MS, given that they are based on transfer functions that are particular for each given load and location. On one hand, assessments using MS methodology allow the calculation of the stress state in a component subjected to different loads acting (or not) simultaneously, and independently of the variations in the different loads. On the other hand, it requires performing the stress coupling in order to determine the effect of the combination of the different loads acting at the same time.

Furthermore, stress calculation using MS is performed through their separation in components, depending on their respective origin, (i.e, pressure stresses, thermal stresses, differential pressure stresses, thermal stratification stresses…).

The appropriate and precise methodology here proposed, relies in the calculation of the six stress components of the different loads (S_{xi}, S_{yi}, S_{zi}, S_{xyi}, S_{yzi} and S_{xzi}). These stress components have their corresponding sign and a fixed direction. Therefore, the stress component of each load (*i*) in a given direction (i.e. S_{xi}) can be coupled as scalar quantities, and then, the different components of the total stress (i.e., S_{xTotal}, S_{yTotal}, etc) can be derived. With such six total stresses (S_{xTotal}, S_{yTotal}, S_{zTotal}, $S_{xyTotal}$, $S_{yzTotal}$ and $S_{xzTotal}$), the principal stress components (S_1, S_2 and S_3) corresponding to the combined effect of all loads acting simultaneously, can be obtained. Finally, this principal stresses allow calculating the actual stress intensity (S_{INT}). This methodology will be defined here as "*actual coupling*".

Nowadays, the most extended calculation methodology is based on the addition of the stress intensities (S_{INT}) from different loads. This procedure simplifies the analysis, but can lead to overconservative results and therefore, noticeable reductions in the fatigue life of the components being assessed. The low accuracy of this methodology, called here "*positive coupling*", comes from the addition of the stress intensities (S_{INT}), a parameter derived from the difference between principal stresses. The principal stresses generated by the different loads have not the same directions, and that is the reason why a direct sum can produce significant errors. The "*positive coupling*" only provides suitable results when the principal stresses from the different loads have the same direction and sign, but that is not probable in any case.

Therefore, a correct stress coupling methodology is required for a consistent realistic fatigue evaluation. In this work, "*positive coupling*" and "*actual coupling*" will be compared.

4. CASE STUDY: APPLICATION TO A NUCLEAR POWER PLANT COMPONENT

The methodology outlined above for stress evaluation of components using MS is the procedure here proposed instead of the positive stress coupling. Both methodologies are compared by means of a stress evaluation in the critical location of the feedwater nozzle of nuclear power plant. The feedwater system provides warm-up water to the vessel, being the water pressure the same to that existing in the vessel. The vessel has two feedwater loops, with two nozzles in each one.

4.1 Geometry and materials

Figure 1 shows a scheme of the nozzle geometry. The Safe End New (SA 508 CL1) and the Safe End Old (A-105 Gr. II) are made of a carbon steel [2]. The vessel wall and the nozzle base material is a low alloy steel (ASTM A 336 cc1332 [3]). The primary thermal sleeve (SB-167) is an Ni-based alloy [4]. Finally, the outer thermal sleeve (XM-19) and the cladding (ER308) are made of a stainless steel [4]. All temperature dependent material properties at an average temperature of 325°F are shown in Table 1

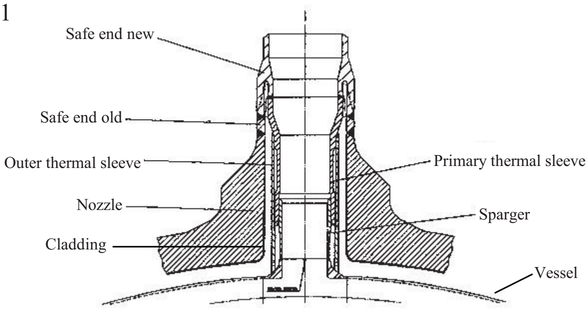


Figure 1. Nozzle scheme and materials at different locations

Table 1. Material properties at 325 ºF

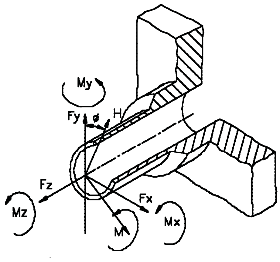
	Safe Ends	*Nozzle Forging & Vessel Wall*	*Cladding Stainless*	*Primary Thermal Sleeve*	*Outer Thermal Sleeve*
Elastic Modulus (psi)	27.95E6	26.55E6	26.875E6	26.875E6	29.725E6
Coefficient Of Thermal Expansion (in/in/°F)	7.4E-6	7.4E-6	9.9E-6	9.15E-6	8.0E-6
Thermal Conductivity, (Btu/hr-ft-°F)(1)	31.95	23.35	9.95	7.85	9.7
Specific Heat (Btu/lb- °F)	0.119	0.121	0.126	0.126	0.123
Density (lb/in^3)	0.283	0.283	0.283	0.283	0.283
Poisson's (4) Ratio	0.3	0.3	0.3	0.3	0.3

4.2 Loads

Among all the applied loads produced by the plant transients in the component, only variable loads are considered in a fatigue assessment. Then, the fatigue analysis in these locations will be performed taking into account the following loads:

- *Pressure loads*: derived from the pressure in the vessel.
- *Mechanical loads*: interaction with the rest of the pipes of the feedwater system. Mechanical loads are shown in Figure 2 and defined in [5] and [6]. These loads vary linearly between zero (shutdown) and the values shown in Figure 2 (normal operation).
- *Thermal loads*: derived from the thermal changes during the different transients.

Differential pressure load will not be considered in this assessment, given that its low value has negligible influence in the total stress. Seismic loads are not considered in fatigue analysis; in case of earthquake a particular evaluation must be done.



Loading	H (Kips)	M (Inch Kips)
Design Mechanical	0.0	408
Dead Weight	5.31	147
Seismic Primary	11.4	206
Seismic Restrained Free End	11.4	206
Thermal Restrained Free End	24.5	613

Notes:
1. Loads are specified at the safe-end-to-pipe weld location.
2. The X, Y and Z axes are orthogonal coordinates. The Z is radially outward from the vessel centerline (in a horizontal plane). The Y direction is upward. An orientation angle for H and M (in X-Y plane) shall be selected which will produce the greatest stress intensity at the location under consideration when all other loads on the nozzle are considered.

Figure 2. Mechanical loads in the nozzle

4.3 Critical location

In fatigue analysis, the critical locations are considered those points in the components where the fatigue usage is the highest. Stress analysis of the feedwater nozzle has been performed using a finite element model [7] with the aim of detecting this point. The critical point has been indentified in the safe-end, in the location shown as point 39 in Figure 3.

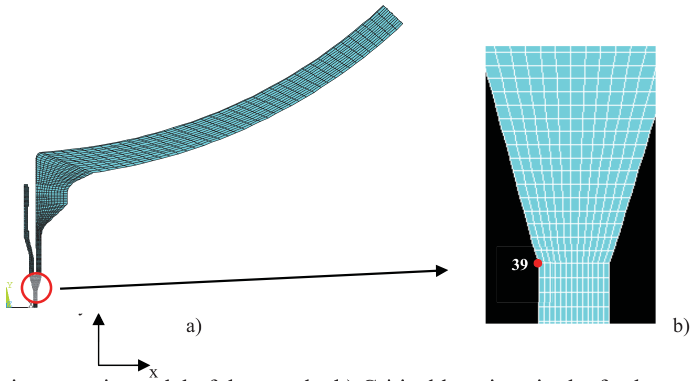


Figure 3. a) 2D axisymmetric model of the nozzle. b) Critical locations in the feedwater nozzle

4.4 Stress analysis

The stresses produced by the different loads are analysed in the Safe end of feedwater nozzle (Figure 3).

4.4.1 Pressure stress

The resulting stresses (psi) in the critical locations caused by the pressure loads are gathered in Table 2 (axes following Figure 3). A constant pressure of 1000 psi (easy scaled up or down to account for the different pressures occuring during the different transients) was applied along the inside surface of the feedwater nozzle and the reactor vessel wall. S_x, S_y, S_z, S_{xy}, S_{yz} and S_{xz} are the six stress components referred to the mentioned coordinate system, and S_{INT} the corresponding stress intensity (1).

Table 2. Stresses (in psi) caused by pressure load

Sx	*Sy*	*Sz*	*Sxy*	*Syz*	*Sxz*	*S_{INT}*
7187	4510	-501	-101	-379	-98	*7723*

4.4.2 Mechanical stress

In order to be conservative enough, the *Φ* angle defined in Figure 2 has to be selected as that one that produces the highest S_{INT} in the critical locations [6]. Table 3 shows the stresses (psi) obtained for the critical locations corresponding to different *Φ* values. It can be seen that the maximum stress is obtained when *Φ*=0.

Table 3. Mechanical stresses (psi) for different *Φ* values

H (ft-lbs)	*Φ*	*Fx* (lbs)	*Fy (Fz in the model)(lbs)*	*S_{INT} (psi)* Safe end
24500	0	-24500	0	4802
24500	22.5	-22635	9375	4550
24500	45	-17324	17324	3840
24500	67.5	-9375	22635	3704
24500	90	0	24500	3624

The stresses caused by the mechanical loads in the critical locations are gathered in Table 4.

Table 4. Stresses (in psi) caused by mechanical loads

Sx	*Sy*	*Sz*	*Sxy*	*Syz*	*Sxz*	*S_{INT}*
72	267	8	2398	1	-49	*4802*

In order to demonstrate the conservatism of the "*positive coupling*", it was calculated by means of finite element analysis the stresses caused by the combination of mechanical and pressure loads. The stress results (Table 5) reveals how mechanical and pressure loads add their effects in the Safe End location (positive stress coupling).

Table 5. Stresses (in psi) caused by the joint action of pressure and mechanical loads

Sx	*Sy*	*Sz*	*Sxy*	*Syz*	*Sxz*	*S_{INT}*
7259	4778	-493	2297	378	-147	*9162*

On the other hand, the conservatism of stress coupling is very noticeable in the Safe end location:

$$S_{PRESS.+MECH.} < S_{PRESS.}+S_{MECH.} \rightarrow 9162 \text{ psi} < 7723+4802 \text{ psi} \rightarrow 9162 \text{ psi} < 12525 \text{ psi}$$

4.4.3 Thermal stress

The stress response (Green's Functions) to a one degree temperature change (unit step change) is used to determine the thermal stress for any temperature change occurring during transients.

Applying the Green's Function in the convolution integral is possible to obtain the thermal stress in any condition. A thermal shock from 182.78°C (361°F) to 132.78°C (271°F) was applied to the feedwater nozzle model, given that these values cover almost all the temperatures that occur during most of the transients. The most representative transients are: start-up, shutdown, scram and power reduction.

The stress history caused by thermal loads (thermal shock from 182.78°C to 132.78°C) is shown in Figure 4.

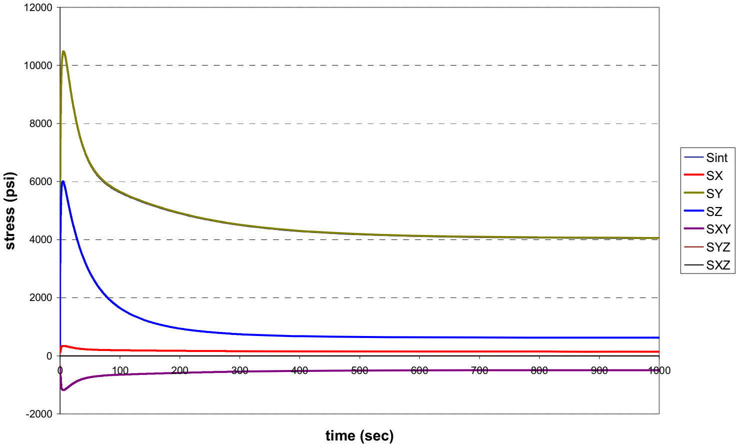


Figure 4. Thermal stresses in the Safe End conditions (ΔT = -90 ºF)

5. FATIGUE MONITORING SYSTEM APPLICATION

5.1 Fatigue evaluation in MS

The maximum allowable number of cycles for the component being analysed can be obtained using the fatigue curves of the design Code [1] once the values of the applied stresses are known. When the component is subjected to stress cycles of different amplitudes (due to the different transients occurring in the plant), the fatigue damage produced by each amplitude level (u_i) is calculated as the relation between the number of cycles actually applied (n_i) and the number of maximum cycles allowed (N_i) at such amplitude.

$$u_i = \frac{n_i}{N_i} \tag{2}$$

The failure criterion is given by the Miner´s law (3):

$$\sum_1^m \frac{n_i}{N_i} = \frac{n_1}{N_1} + \frac{n_2}{N_2} + \frac{n_3}{N_3} + \ldots + \frac{n_m}{N_m} = 1 \tag{3}$$

The total stress in a component is given by the stresses caused by the different loads occurring during the transients. In the feedwater nozzle here analysed, the different types of stresses considered and the corresponding methodologies used for their evaluation are briefly described in Table 6. All the parameters (temperature, pressure, flow…) needed for the stress calculations are obtained by means of the different instrumentation installed in plant.

Table 6. Stresses considered. Expressions and some of their characteristics.

Load / Stress	*Origin*	*Type load*	*Usual way of calculation*	*General expression*	*Particular expression*
Pressure stress	Pressure in vessel	I	FEM	Sp= f(pressure)	**Sp = A* Press** -Sp = pressure stress intensity S_{INT} -A = constant calculated from stress report -Press = reactor pressure
Mechanical stress	Interaction with the rest of the pipes of the feedwater system	I	FEM	Sm = f(local temperature)	$Sm = [(T - T_0) / (T_{100} - T_0)] \cdot \sigma_m$ -S_m = mechanical stress intensity (S_{INT}) -σ_m= mechanical stress intensity at temperature T_{100} -T = temperature in the location of interest -T_0= reference temperature (mechanical stress is null (0%) -T_{100}=temperature that produces the highest mechanical stress (100%)
Thermal stress	Thermal load due to variations of the water temperature	II	FEM	$S_t(t)=\int_0^{\tau} x_\theta(t-\tau)\Delta T(\tau)d\tau$	$S_T(t)=\int_0^{\tau} Gr_t(\tau)\cdot\Delta T(\tau)d\tau$
				- St(t) = thermal stress intensity (S_{INT}) along time - $\Delta T(\tau)$ = temperature change in the interest location - $x_\theta(t-\tau)$ = $Gr_t(\tau)$ = response to a unit step thermal change (Green Function)	

As seen in Table 6, the methodology used for the stress calculation varies depending on the type of the stress: Type I or Type II. Type I stresses are calculated by means of transfer functions and the stress intensity is linearly proportional to the applied load. Type II stresses needs the calculation of the Green's Function, that is, the response to a unitary change of the load (i.e. temperature in thermal stresses). Using the Green's Function, and applying the Duhamel (convolution) integral, the stress associated to any other load is obtained.

5.2 Stress and fatigue damage assessment.

The stress assessment and its corresponding fatigue damage analysis (Miner´s law) in the critical location is here evaluated by using two types of stress coupling:

a) *Positive* stress coupling (most common)

b) *Actual* stress coupling (considering the coupling between the stress components).

The fatigue damage is calculated for Start up + Shutdown transients, given that they are the most representative and critical in terms of fatigue damage. The stresses caused by these transients in the Safe end are shown in Figure 6. The fatigue damage result is shown in Table 7. The stress intensity calculated by means of positive coupling is higher than the actual stress, and therefore it produces more fatigue damage (12-13 times higher), as shown in Table 7.

*Table 7***.** Fatigue damage, U

	Positive coupling	*Actual coupling*
Start up + Shut down	6,573 E-4	5,191 E-5

As shown in the table, the damage associated to positive coupling is more than ten times the damage associated to the actual coupling.

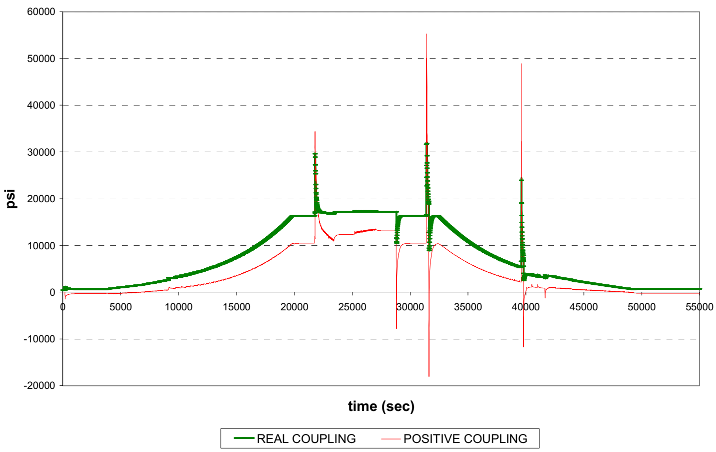


Figure 6. Total stresses applying different procedures of calculation

7. CONCLUSIONS

"Positive coupling", which directly adds the stress intensity associated to the different loads, is the most extended methodology for the stress assessment in MS of nuclear power plants. Its application is a conservative method for the calculation of stresses and can lead to noticeable reductions in the fatigue life estimation of the components of the nuclear power plant being assessed.

The *"actual coupling"* is the methodology here proposed for the stress assessment in the evaluation of fatigue damage. This methodology is based on the calculation of the stress components of the different loads, their addition, and the subsequent obtainment of the actual stress intensity.

The application of both coupling options to a real case has demonstrated that the conservatism associated to "*positive coupling*" can be noticeable.

The effort in the initial implementation of the "*actual coupling*" is bigger than the "*positive coupling*". However, after that, the stress and fatigue damage calculations are analysed with computers and the computation time differences between "actual" and "positive" coupling is negligible.

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