In-Service Stresses in 900 MWe PWR Steam Generator U-Bends

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1 - INTRODUCTION

The alloy 600 steam generator tube bundles of the EDF PWRs are the subject of particular attention by the utility owing to their importance for safety and their potential to affect outage. To ensure safety, the utility seeks to minimize the tube rupture risk and, as regards availability, the utility seeks to minimize the number of reactor shutdowns for primary to secondary leakage.

To attain these objectives at reasonable cost, EDF is optimizing its maintenance policy for this equipement. To retain an acceptable life expectancy for the equipment in service, an alternative has been sought to the policy of systematically plugging any tubes at risk. This has led the utility to make a certain number of studies designed to orient maintenance options. It is thus that it found necessary to obtain greater understanding of in-service stress in small radius bends in tube bundles of certain manufacturing series as some exhibit premature cracking due to stress corrosion, suggesting relatively high overall stress levels. Would it be possible for these bends with apparently high residual manufacturing stresses to envisage corrective action which could delay or remove any risk of corrosion? And this being the case, what process should be implemented and what benefits should be expected?

The preliminary studies had already oriented EDF towards a thermal process for releiving residual stresses, but still left a choice to be made concerning the type of heat treatment: simply relaxing the residual stresses by a "short" treatment or, in addition, structurally improving the alloy by a "long" treatment. Information on the respective distribution of residual manufacturing stresses and of operating stresses should enable a decision to be made. This paper presents the studies designed to determine the stresses and to identify the influence of the various contributing parameters.

2 - EXPERIMENTAL STRESSES EVALUATION

2.1 - Total I.D. stresses

Three U-Bends, bended using an experimental sensitized tube have been submitted to a stress corrosion test in Tetrathionic Solution (pH = 2) with an internal pressure of 100 bar (Argon). Figure 2 shows the Stress versus Time to Cracking curve for this test. In one of the tests, a simulated thermal expansion was applied. The time to cracking of the three U-Bends, between 10 to 11 hours, demonstrated that the maximum stress is at least of about 200 MPa. This stress is a combination of both applied and residual stresses. The former are independent of the mechanical properties of the tube, the latter are not. So, we decided to add

to the 200 MPa a value of 150 MPa corresponding to the Yield Strengh difference between the experimental sensitised Inconel 600 tube and a standard one in the S.G.
The maximum total stress seems so to be at least of about 350 MPa. Other stress corrosion tests in steam allow to define this value more accurately: no leak was observed after 1200 hours in tests where an internal pressure of 200 bars and a simulated thermal expansion were applied. This means that the maximum stress is less than 500 MPa which is the threshold of the steam test.
In conclusion we can admit: 350 < maximum I.D. stress < 500 MPa.

2.2 - Total O.D. stresses

Steam tests allow to quantify O.D. stresses. U-Bends presented O.D cracks in tests with a 100 bar Δp and a simulated thermal expansion. This means that O.D. stresses are higher than 500 MPa (steam test threshold). In the same kind of test, Reverse U-Bends samples, in which stresses can be estimated to 600 MPa, crack in less than 100 hours.
In conclusion, we can admit: 500 < maximum O.D. stress < 600 MPa.

2.3 - Operating I.D. and O.D. stresses

Another test in Tetrathionic Solution has been performed on a stress relieved U-Bend: 3 min. at 1085°C plus 1 hour at 700°C which gives a new sensitisation to the U-Bend (figure 2). An internal pressure (100 bar) and a simulated thermal expansion were both applied during the corrosion test which was performed on I.D. and O.D.
Neither leakage nor crack were observed after 200 hours demonstrating that applied stresses were, in this case, lower than 180 MPa (see figure 2).

3 - ANALYTICAL OPERATING STRESSES EVALUATION

3.1 - General

As a complement to the experimental approach mentionned above, an analytical evaluation has been performed. It takes into account two types of stress:
- the remanent stresses resulting from the conditions and process of assembly,
- the stresses caused by operating loads.

The former, like the residual stresses, are permanent. But the two types are similar in that they cannot be eliminated, or even modified to any significant extent, in service, thus contributing inexorably to the risk of stress corrosion. In addition, they can both be dealt with by means of the same design approach and are therefore subjected to a common analysis here.

Two complementary elastic linear models have been developed for this purpose:
- an analytical model based on simplified beam and shell theory models, giving for the R1 bend of smallest radius, the general stress level and establishing a law for change in stress as a function of bend radius Ri,
- a three-dimensional numerical model using finite element calculations, taking into account factors which an analytical model is unable to allow for in a simple manner, eg multiple non-linearity, heterogenic geometry etc.

3.2 - Analytical models

These assume idealized geometry: constant thickness and out-of-round, as well as simple boundary conditions at the tube support plates. The method proposed in the RCCM code makes it possible to solve in a simplified manner the different loading cases to which the bends are subjected.

3.2.1 - Differential pressure loading

In the absence of out-of-round of the tube, in typical sections, the longitudinal and circumferential stresses are given with adequate accuracy by:

$$\sigma_{L,0} = \Delta P \frac{r_i}{2t} - P_S \qquad \sigma_{C,0} = \Delta P \frac{r_i}{t} - P_S$$

In bent parts, again in the absence of out-of-round, these stresses are modified in the following manner:

$$\sigma_L = \sigma_{L,0} \qquad \sigma_C = 0.5 \frac{2R + r \sin \emptyset}{R + r \sin \emptyset} \cdot \sigma_{C,0}$$

Alone and in the absence of out-of-round, the bend radius parameter scarcely changes the pressure stress as it induces an extra stress σ_C of 10 MPa at the most.

Out-of-round in the bend will induce the local bending in the thickness of the tube. Furthermore, as shall be seen later with the finite element model, the pressure loading will, in this case, tend to open the bend, while its straight legs, possibly maintained by the support plates, will oppose this movement. This opposed tendency to open will create overall bending of which it will only be able to determine the compatibility using a three-dimensional model.

If the compatibility stresses are disregarded, the bending stresses and the thickness can be expressed as follows (defined at the outer wall):

$$\sigma_L^* = \nu . i_3 . P \quad \text{and} \quad \sigma_C^* = i_3 . P \text{ where } i_3 = \frac{r_e}{r_i} . \frac{D}{t} . \frac{1.5\ \varepsilon}{1 + 0.455 . \left(\frac{D}{t}\right)^3 . \frac{P}{E}} \cos 2\emptyset$$

For a maximum out-of-round of 6%, circumferential stress is thus increased by approximately 120 MPa at the inner skin at the flank and at the outer skin at the extrados and intrados of the bend.

It is necessary at this point to allow for the additional stresses of compatibility with the legs. For the R1 bend, these are given by the finite element model. For a bend of radius Ri and free length L, it can be accepted that the variation law for these stresses is of the following type:

$$\sigma(R_i)/\sigma(R_1) = (6k\ R_i^2 - L'^2)/(6k\ R_1^2 - L'^2)$$

This expression is derived from the equation for compatibility movements between two beams, one representing the bent part affected by its own flexibility k_i and the other one of the straight legs of length L' subjected to an imposed displacement (tending to bring the two legs together). The results of the three-dimensional analysis leads to L' = L + d being chosen in which d represents the distance between two support plates.

3.2.2 - Differential expansion loading

At operation under rated conditions, at 100% load, the tube is subjected to a longitudinal gradient ΔT_L of the order of 10°C (average wall temperature) between the hot chamber and the cold water chamber of the steam generator. The average temperature difference causes greater axial displacement on the hot leg at the beginning of the bend, resulting in bending moments in the tube. A simplified model assuming a sliding link between the tube and the tube support plate (permanent contact without rotation being possible) leads, by a displacement compatibility equation, to the following expression:

$$M_\theta = \frac{- 2\ E\ .\ I\ .\ \ell\ .\ a_m\ .\ \Delta T_L}{4\ \ell\ .\ I\ /\ R\ .\ S + k\ \pi\ R^2}\ .\ \mathrm{Cos}\ \theta$$

where M_θ represents the movement in the bend from the point of curved x-axis θ. A finite element beam calculation simulating the clearances and contacts at the intersections with the tube support plates has indicated that the preceding model is too rigid and requires correction by a factor of approximately 0.5.

The numerical application of this model indicates that, for this loading, it is the circumferential stress which is greater (at the inner skin at the flank and at the transition between the bend and the cold leg).

3.2.3 - Heat gradient through thickness

Of the order of 10°C, it induces bending stresses given by:

$$\sigma_L^* = \sigma_C^* = \frac{E\ .\ a_m\ .\ \Delta T}{1 - \nu} \quad \text{(at the outer skin)}$$

3.3 - Finite element models

Bend R1 has been the subject of a three-dimensional finite element calculation combining thick shell elements in the upper part of the bend with beam elements in the straight parts ; special elements link these two meshes and allow transmission of forces. The mesh, which simulates the U-tube over its entire length consists of 216 shell meshes with 8 nodes and 166 beam meshes, was processed with the SYSTUS computer code. A special procedure enables simulation of the conditions of contact with clearances at the intersections with the tube support plates. The following geometrical hypotheses were adopted:

- nominal geometry: constant thickness and zero out-of-round,
- additional geometry including constant non-zero out-of-round, variable out-of-round and variable thickness.

The following assembly parameters were examined:

- free straight length L above top of last tube support plate,
- clearance conditions at tube support plates,
- erection error hypotheses (no staggered rolling, misalignment of plates, etc.).

These different geometrical configurations have been assessed for the loading cases adopted in the analytical models. Analysis has been carried out in terms of the main stresses, the Tresca equivalent stresses and the elementary stresses σ_L and σ_C. An example of a graphical analysis is given in figure 3 for rated operation at 100% of a bend of generalized 6% out-of-round under nominal assembly conditions.

3.4 - Summary of calculation results

Collectively, the above calculations have made it possible to draw the following conclusions concerning operating stresses:

- Of all the geometrical parameters, it is out-of-round which changes stress the most ; at 100 bars a variation of the order of 90 MPa is forseeable. The other parameters concerning geometry (thickness) or assembly (tube to support plate clearance, and ultimate free straight height) only change the stress levels by about 10 to 20 MPa.

- At the points of greatest loading, in the absence of any local geometrical faults, the stress in operation at 100% load reaches a value of the order of 260 MPa in the bends with the greatest out-of-round (6%). This value relates in all cases to the bend transition on the cold leg side, either at the inner skin at the flank or at the outer skin at the intrados of the bend. This consists of three pratically equal parts: the pressure stress in the absence of out-of-round, the extra pressure stress due to 6% out-of-round, and the hot leg/cold leg differential expansion stress. The figure 4 gives the curves of the results for the U-Bends R1 to R6.

4 - CONCLUSION

The preceding studies made it possible to estimate the total in-service stresses of the row 1 and following U-Bends, showing the maximum levels to be about 500 MPa. The operating stresses alone have been estimated to about 250 MPa. So, a stress relief heat treatment able to eliminate residual stresses could then be recommended, at least for the Steam Generators in which the operation remained financially worthwhile. Stress in the U-Bend being reduced to a low value (operating stresses alone), the opportunity of a "long heat treatment" improving the material resistance to Stress Corrosion Cracking has not then been found to be indispensable.

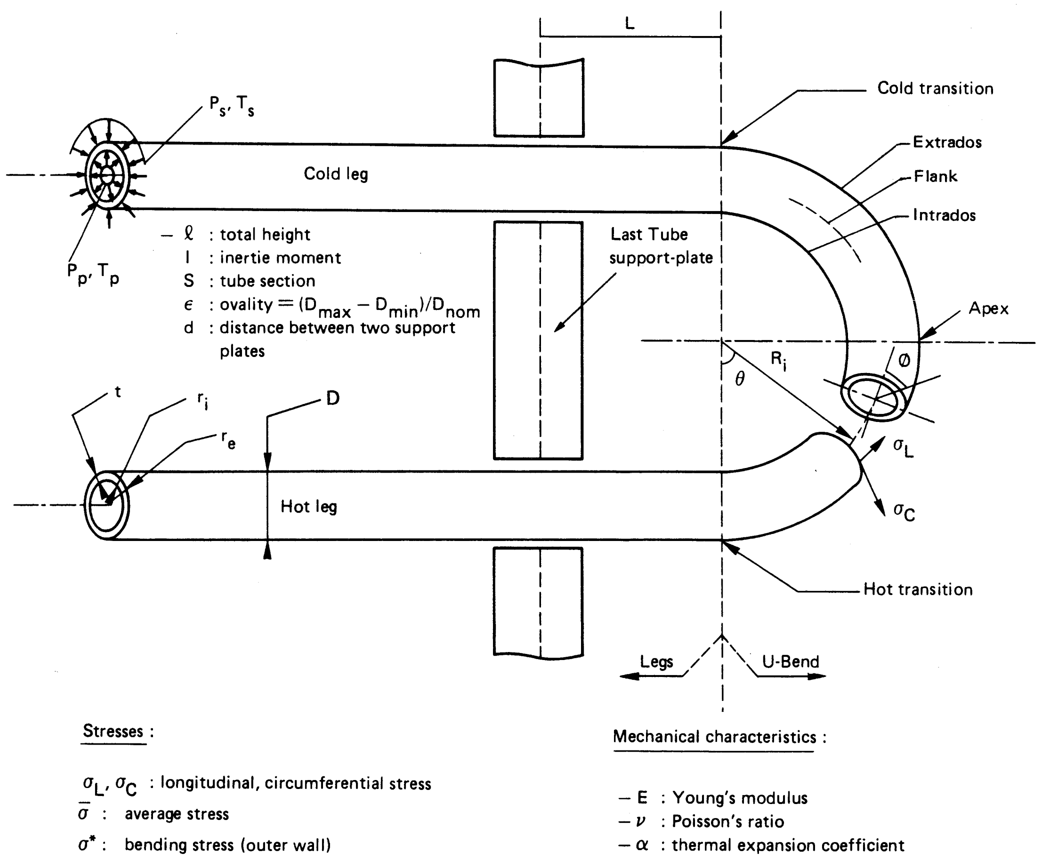

Figure 1 – U-Bends parameters.

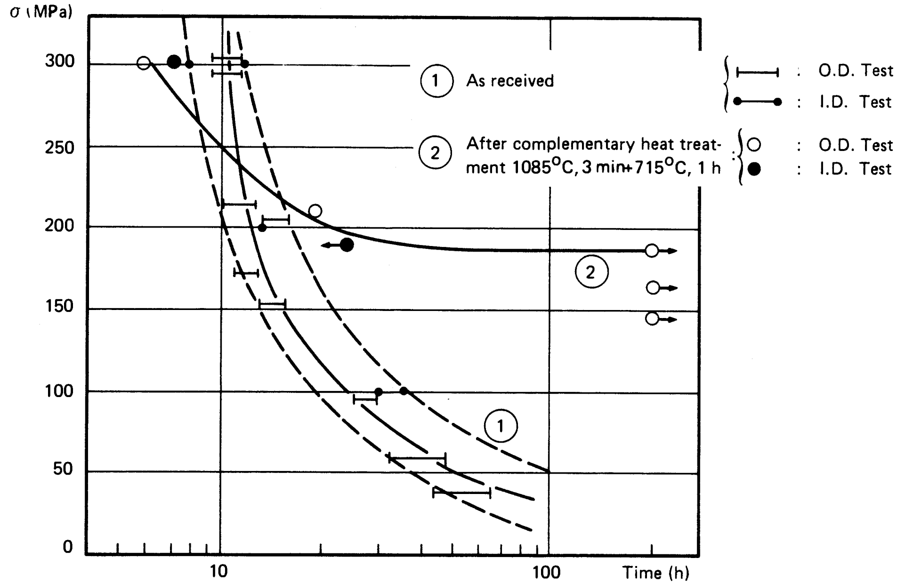

Figure 2 — Stress versus Time to Cracking curve obtained on the tube for stress corrosion test in Tetrathionic Solution.

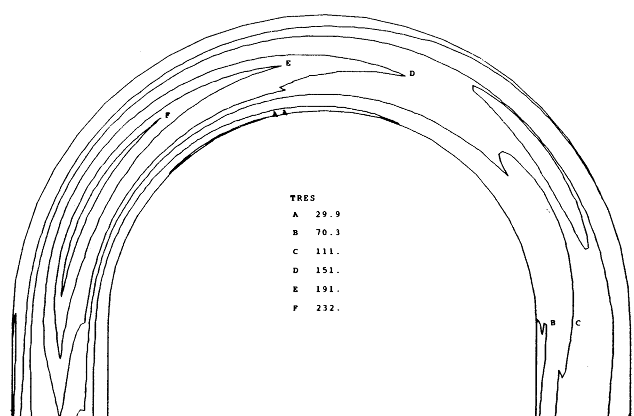

Figure 3 — Rated operation at 100 %. Tresca equivalent stresses in R1 (ovality = 6 %).

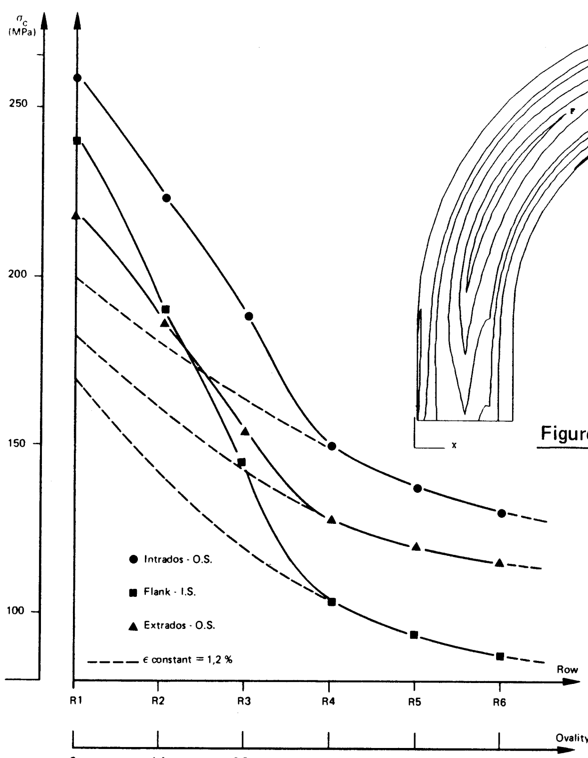

Figure 4 — Rated operation at 100 %. Hoop stresses at cold transition from R1 to R6.