Once-Through Straight Tube Steam Generators Subject to Severe Thermal Transients Compensation Systems Optimization

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FOREWORD

In the frame of an ANSALDO project devoted to study the feasibility of using classical Once-Through Straight Tube Steam Generators under very severe thermal transient conditions in nuclear applications, the methodology herein described has been developed.
Such a kind of Steam Generators can go to some trouble if required to work under high Tubes vs. Shell-Course temperature differences. The Tubes can be subjected in such a situation to high compressive loads which can significantly overcome the critical one. A mean to face this difficulty is to use Compensation Systems whose stiffness be very small in order to reduce the Shell-Course to Tube Bundle Stiffness Ratio (K_s/K_t). Since the Compensation System has to be positioned in the Shell-Course region it becomes part of the Pressure Boundary; then, it is subject to the requirements of the Section III of the ASME Code (Subsection NB and Code Case N-290-1, for Nuclear Safety Class 1 Components, 1986).

GENERAL DESIGN CONSIDERATIONS

The design of such a Compensation System has to be a suitable compromise among the longitudinal overall flexibility of the shell, the stress level due to pressure, the thermal loadings and the necessity to provide an adequate flexural stiffness in order to avoid dynamic amplifications and/or large lateral shell displacements due to seismic loads.
Since Compensation Systems have some peculiar characteristics (as the intrinsic no-linear behaviour - see, EJMA Standard, 1980) which make the satisfaction of so stringent requirements as those provided by Nuclear Codes not a simple matter, great attention has to be paid in conceiving a shape which can minimize the no-linear features, making thus possible the application of the linear approach for stress verification adopted by the Code (see, ASME Code Case N-290-1, 1986), and assure at the same time the required high flexibility and a high manufacturing reliability (see, Stastny, 1982).
Further considerations relative to the manufacturing capability, the requirement to avoid welds in the highly stressed regions and the satisfaction of possible lay-out limitations on the overall size (both in the longitudinal and in the diametral directions) of the Expansion Joint can also play a fundamental role in the design phase.
A shape like that shown in Fig. 10, corresponding to the "Flanged and Flued Head Expansion Joint" described by Singh (1984), can be seen as a reference solution which joins together the advantage to be approachable by the Code Case N-290-1 (1986) methodology, to be of current industrial practice, and to be flexibile enough for the scope.

DESIGN METHODOLOGY

The following steps are required in order to finalize the design:

(1) The Once-Through Straight Tube Steam Generator preliminary design has to be carried out without considering any Expansion Joint.

(2) Considering the postulated thermal loading conditions, the stress state in the Tubes can be evaluated. Tubes which are stressed above the critical buckling value can be so found.
The evaluation of the Shell-Course equivalent axial stiffness necessary to meet the buckling allowable stress limits in the Tubes can be performed by a parametric study. Figures 1 and 2 show how typically the Tube stress changes with a variation of the K_s/K_t ratio.

(3) Once a $(K_s/K_t)^*$ ratio satisfying the above limits is found, a tentative Expansion Joint design can be attempt, such as $K_b^{-1} = K_s^{*-1} - K_s^{-1}$, where K_s^* indicates the Shell Stiffness and K_b indicates the Expansion Joint Stiffness values which satisfy the above condition (see, Singh, 1984). The design shall define geometry (number of heads, fillet radius, outside radius) by treatment of joint stresses due to pressure acording to the requirements of ASME Code Case N-290-1 (1986), based on which the P_m, determined by mean of the following relation:

$$P_m = (p A^* / A_c) + 0.5 p \qquad (1)$$

shall be less than the allowable stress intensity S_m (see, Fig. 3, for illustration of symbols A^* and A_c, and Fig. 4, for the A^* and A_c dependence on the Expansion Joint outside radius).
The Expansion Joint thickness may be determined by means of the above relation or set equal to the Shell-Course thickness, as done in the empirical design practice in the industry (see, Singh, 1984).
Parametric studies can be usefully run to find the Outside Radius value which the minimal primary membrane stress intensity, P_m, corresponds to. Figure 5 shows that, for a given geometry, the convolute hoop stress decreases as the outside radius of the Joint grows; while Figure 7 shows that the viceversa helds for the convolute longitudinal stress. From Fig. 7 also it clearly appears that the optimal outside radius value is that given by the intersection of the hoop stress and longitudinal stress curves. Fig. 6 shows the effect of the Joint fillet radius on the Joint primary stress intensity P_m. It is observed that the effect of the fillet radius is of lesser importance with respect to the other geometrical features.

(4) The initial Steam Generator Shell-Course and the Expansion Joint are coupled and new calculations are run to find the loading acting on the Tubes and on the Joint itself. If the loads on the Tubes meet the allowables, a detailed structural analysis of the Expansion Joint is performed; otherwise, a further refinement of the Expansion Joint design is necessary.

(5) Finally, in order to completely meet the ASME Code Case N-290-1 (1986) requirements, the complete structural analysis of the Expansion Joint is performed. A Finite Element approach can be very useful for this purpose (Figure 9 provides a sketch of such a possible model): it allows to evaluate accurately the axial stiffness of the Expansion Joint, K_e, as well as the state of stress due to the operating conditions (applied and thermal differential expansion loads). The state of stress so obtained, suitably classified in the primary and secondary categories, as required by the Code, shall be compared to the allowables as well as the Ke stiffness value shall result less than the required value, K_b. The configuration which satisfies the structural Code limits and presents a stiffness value less than K_b is the required one.

COMPUTER CODES

Due to the many iterative processes necessary to reach the final Steam Generator configuration, in order to minimize time and costs, the above

outlined design procedure has been implemented by means of some specific P.C. computer codes. These codes mainly allow to solve the parametric studies at steps (2) and (3) above. The step (2) activity has been performed by a specific code based on the Theory of Plates on Elastic Foundations (see, Singh, 1984 and Timoshenko, 1959) which simulates the cylindrical region, the Tube Bundle and the Tubesheets behaviour. Capabilities to account for fictitious stiffnesses in the secondary shell have been introduced. Step (3) has been carried out by a related program which allows to scan all over a set of geometric values, for a given pressure, computing the parameters of interest (A_c, A^*, P_m, $P_{m,ax}$) and providing the curves set shown by Figures 4 through 7, which lead to optimize the design of the Expansion Joint.
Additionaly, the stress analysis has been carried out by means of a general purpose Finite Element code (Ansys, rev. 4.3, 1987), which, connected to a further specific routine of the P.C. program, allows to get authomatic stress verification according to the ASME Code Case N-290-1 (1986) requirements.

EXAMPLE OF APPLICATION OF THE METHODOLOGY

The methodology above described has been applied to size a Steam Generator whose main characteristics are given in Tables 1 and 2. The anticipated operating and design conditions are given in Table 3.
Figures 1 and 2 show the stresses in the Tubes (at the center and at the outermost position of the bundle) for a set of stiffness ratio (K_s/K_t) values. Since for these Tubes the critical compressive stress has been computed to be 37.00 MPa, the above cited Figures indicate the necesity to reduce the stiffness ratio up to a value of 0.4, which calls for an Expansion Joint whose stiffness be at least 2.0 E+6 N/mm.
A reference Expansion Joint solution has been found having the geometry shown in Figure 8. The stiffness of such an Expansion Joint has been computed (by the Finite Element model shown in Fig. 9) to have a value of 7.0 E+5 N/mm, then completely fulfilling the design requirement (the final stiffness ratio so becomes K_s/K_t = 0.13).
A detailed stress analysis of the final solution provided the stress values on the Expansion Joint given in Table 4 whose variation across the Joint is shown by the Figures 10 through 12. As clearly shown by Table 4, for the example case herein considered the necessity to use at least two Expansion Joints in series arises due to the limits on the Stress Intensity Range.

CONCLUSIONS

To use Once-Through Straight Tube Steam Generators when severe thermal transients are anticipated, Compensation Systems might be required to reduce stresses below the allowable ones in the Tubes.
Due to the high stresses that the Expansion Joint shall withstand, a suitable design requiring a lot of parametric studies is necessary. An authomatized procedure has been implemented to reduce cost and time. This procedure can be also applied in the design of conventional components.

REFERENCES

ANSYS Engineering Analysis System / Rev. 4.3a - Swanson Analysis Systems, Inc.

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Specification, and Installation" 82-PVP-4 - ASME Paper.
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TABLE 1
Steam Generator Main Geometric Features

	Minimum Req. Th.	Selected Thick.
Shell Course	27.40	35/55[1]
Tubes	0.42	0.86
Tubesheets[2]	244.00	300.00
Chan. Head Cyl. Reg.	50.60	100.00
Chan. Head Sph. Reg.	25.30	100.00

Cyl. Reg. In. Radius	R_i	= 1120. mm
Chan. Head Sph. In. Rad.	r_i	= 985. mm
Tubes Outside Diameter	d_o	= 15.875mm
Tubesheet Holes Pitch	p	= 22. mm
Amount of Tubes	N_t	= 6348. mm
Nominal Tube Length	L_t	= 15000.mm
Max. Tube Span Length	l	= 1000. mm

(1) 55 mm close to the Tubesheets only; (2) On the basis of T.E.M.A. rules

TABLE 2
Steam Generator Main Materials

Cylindrical Shell, Head and Tubesheet	SA-508 cl. 3A
Tubes	SB-163 Gr. 600

TABLE 3
Operating and Design Conditions

Normal Operating	Tube-Side Pressure	= 9.0 MPa (g)
	Shell-Side Pressure	= 4.0 MPa (g)
	Tube-Side Temperature	= 300.0 °C
	Shell-Side Temperature	= 300.0 °C
Accidental Operations (Pump Trip)	Tube-Side Pressure	= 9.0 MPa (g)
	Shell-Side Pressure	= 0.0 MPa (g)
	Tube-Side Temperature	= 300.0 °C
	Shell-Side Temperature	= 250.0 °C
Temperature Regulations	Tube-Side Pressure	= 9.0 MPa (g)
	Shell-Side Pressure	= 0.0/4.0 MPa (g)
	(Tube-Side - Shell-Side) Temp.	= -150/+150 °C
Design	Primary Side Pressure	= 10.35 MPa (g)
	Primary Side Temperature	= 315.0 °C
	Secondary Side Pressure	= 5.0 MPa (g)
	Secondary Side Temperat.	= 293.0 °C
	Prim./Sec. Side Dif. Pres.	=-5.0/10.35 MPa(g)
	Prim./Sec. Side Temperature	= 315.0 °C

TABLE 4
S.G. Expansion Joint Structural Analysis Results for Normal Service

Bellows in series	Loading	Primary S.I. $P_l + P_b$	P.Allowab. 1.5 S_m	Secondary S.I. $(P_l+P_b+Q)_{Range}$	S.Allowab. 3.0 S_m
2	Pressure	269.4 MPa	310.3 MPa	-----	-----
	Axial Load	-----	-----	541.9 MPa	620.6 MPa

FIGURE 1

Stresses on the Central Tubes as functions of the Shell vs. Tubes Bundle Temperature Difference for various Stiffness Ratios (K_s/K_t)

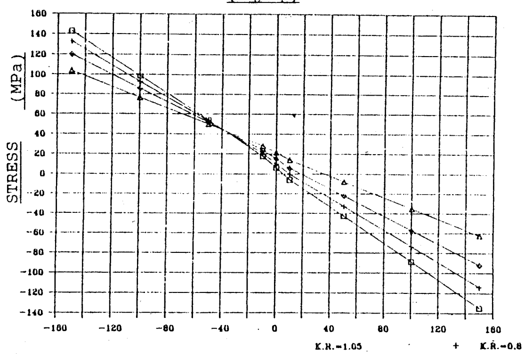

FIGURE 2

Stresses on the External Tubes as functions of the Shell vs. Tubes Bundle Temperature Difference for various Stiffness Ratios (K_s/K_t)

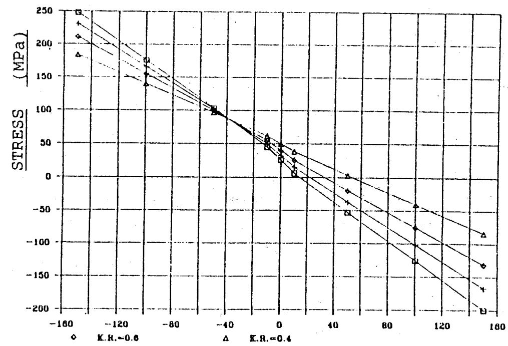

FIGURE 3

Schematic for the Individuation of Areas A_c and A^* for Expansion Joints (from ASME Code Case N-290-1, 1986)

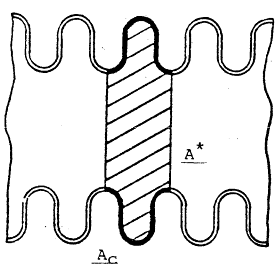

FIGURE 4

Areas A^* and A_c as functions of the Expansion Joint Outside Radius

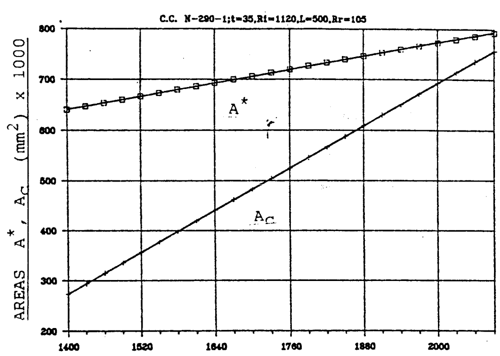

FIGURE 5

Expansion Joints Primary Membrane Stress Intensity, P_m, as a function of the Expansion Joint Outside Radius

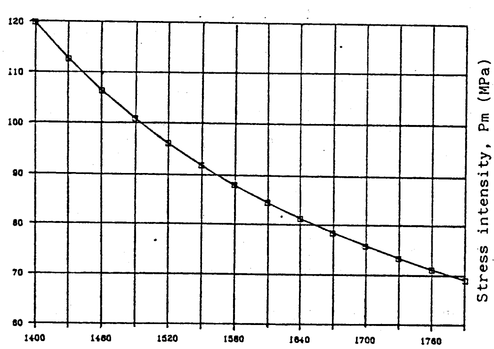

FIGURE 6

Expansion Joints Primary Membrane Stress Intensity, P_m, as a function of the Expansion Joint Outside Fillet Radius

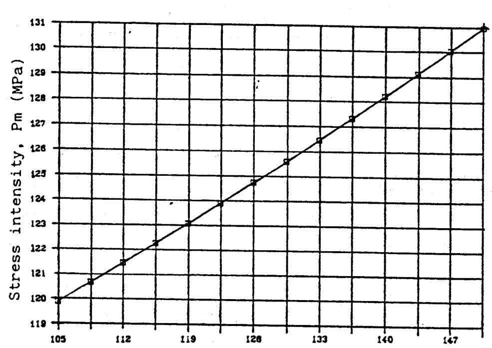

FIGURE 7
Expansion Joints Circumferential and Longitudinal Stress Intensities as functions of the Expansion Joint Outside Radius

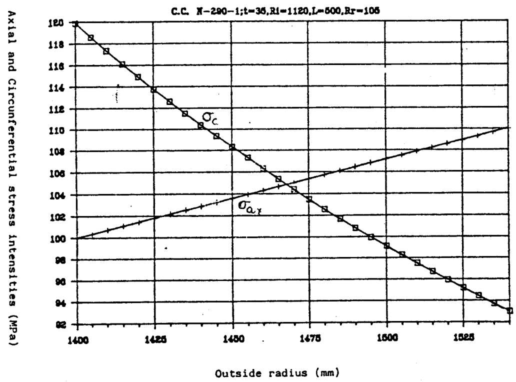

FIGURE 8
Dimensions of the Optimized Expansion Joint

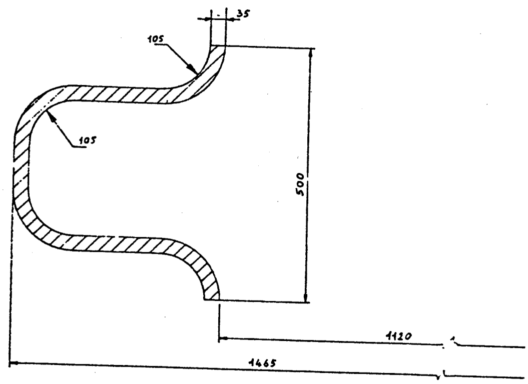

FIGURE 9
Finite Element Model of the Optimized Expansion Joint

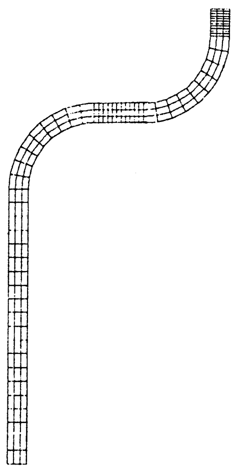

FIGURE 10
Pressure Principal Stresses Distribution across the Inner Side of the Expansion Joint

Inner Side Position

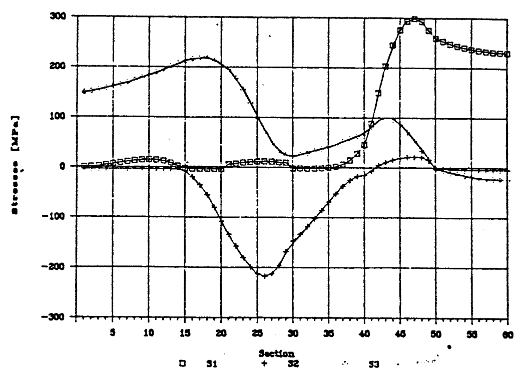

FIGURE 11
Pressure Principal Stresses Distribution across the Outer Side of the Expansion Joint

Outer Side Position

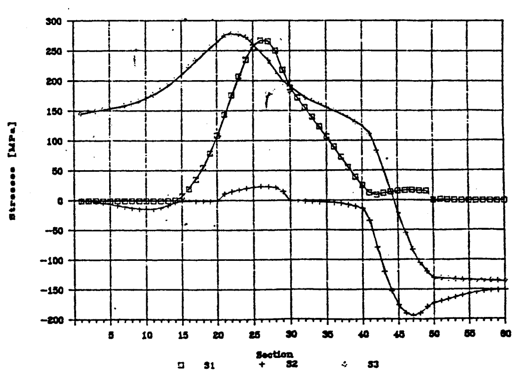

FIGURE 12
Axial Load Principal Stresses Distribution across the Inner Side of the Expansion Joint

Inner Side Position