THE STRESSES AND STRAINS EVALUATIONS IN THE HOT WATER BOILERS PARTS

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Abstract: *Cylindrical shells forming pressure vessels are realized by heat drawing. In many cases after this process occurs deviation of planarity. These departures are muffled by shell forcing to the given shares, by clamping with fixing rods and welding of smoke piping angered at specifically shares. This technology instills supplementary stresses in the cylindrical shells and smoke piping. Therefore, the clamping rods influences the stresses and strains state in the adjacent areas of clamping, recommending the avoiding of the high values of obtained stresses. For the removal the clamping rods and this technology, it proposed that the smoke piping to be angered at size highest projected dimensions and to be worked at technological final size after the welding in the back cylindrical shall.*

Key words: *strains, stresses, cylindrical shells, clamping forces, strains, and gauges strains.*

1. INTRODUCTION

The hot water boilers (ex. hot water boiler CIMAF 5 Gcal/h from TERMOROM company Cluj Napoca) are block tubular boilers with focus tub and pipes fume, for producing hot water for heating in industrial applications. The pressure body was made have a boiler casing, and tubular plates in front and behind, and focus tube and fume pipes. The tubular plates are realized by heat cupping an in many cases will occur flatness deviation. These differences will disappear by taking forcing plates to the described dimensions. This technology induces additional stresses in the tubular plates and in fume pipes, and the thermal stresses cumulating with those will reduce the life work of boiler.

2. THE STUDY OF TUBULAR PLATES

Assimilating the tubular behind plate with a circular plate fitted with a center hole, plate that is fixed on the boundary and central loaded as in figure 1, the relations for maximum displacement and maximum stress σ_{max} are [2]:

$$w_o = c_1 \frac{F \cdot a^2}{E \cdot h^2} \quad \text{and} \quad \sigma_{max} = c_2 \frac{F}{h^2}$$

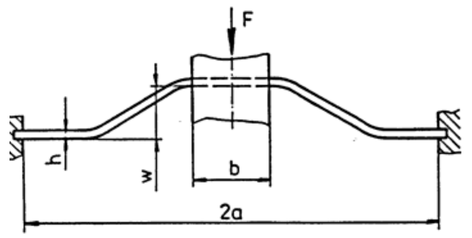

Fig. 1. Schematization for the tubular behind plate calculus

coefficients c_1 and c_2 taking them function of ratio a/b. For a/b = 1,5 result c_1 = 0,0064 and c_2 = 0,220. For made simplifies and function of constructive characteristics of the used rods, in table 1 was tabulated maximum clamping forces and maximum obtained stresses (absolute values).

Table 1

Specification	Clamping forces, [kN]		Obtained stresses [N/mm^2]		Working stress, σ_a [N/mm^2] - [1]
	Approx. calculus	Experimental	Approx. calculus	Experimental	
Tubular behind plate	96,6	215,2	189,94	228,5	136,6

Due to especially tubular shape plate studied, analytical calculus was combined with an experimental proceeding: method of strain gauge measurement. In figure 2 is plotted an assembly view of a tubular behind plate with indication of strain gauge displacement (TER type 10H120-INCERC Bucharest) radial and hoop, as well as the strain gauges - type RC120-Mikrotechna, a detail regarding its placement and a photo of a tubular plate, ready for measurement. The strain gauges was fixed on the established locations by means of Dinox 010 expoxidic resin, finally was realized the set up by means of electronic tensometer with six channels type N 2302. These were balanced when tubular behind plate, mechanical unstressed, has plane deviations. By helping of metal rods of ring cross-section, (D_{ext} = 38 mm and D_{int} = 30 mm, of 4900 mm length, having square screw thread on 100 mm length) mounted in some tubular plate apertures was applied clamping forces. In this case was take of plane deviations and de strains was retained.

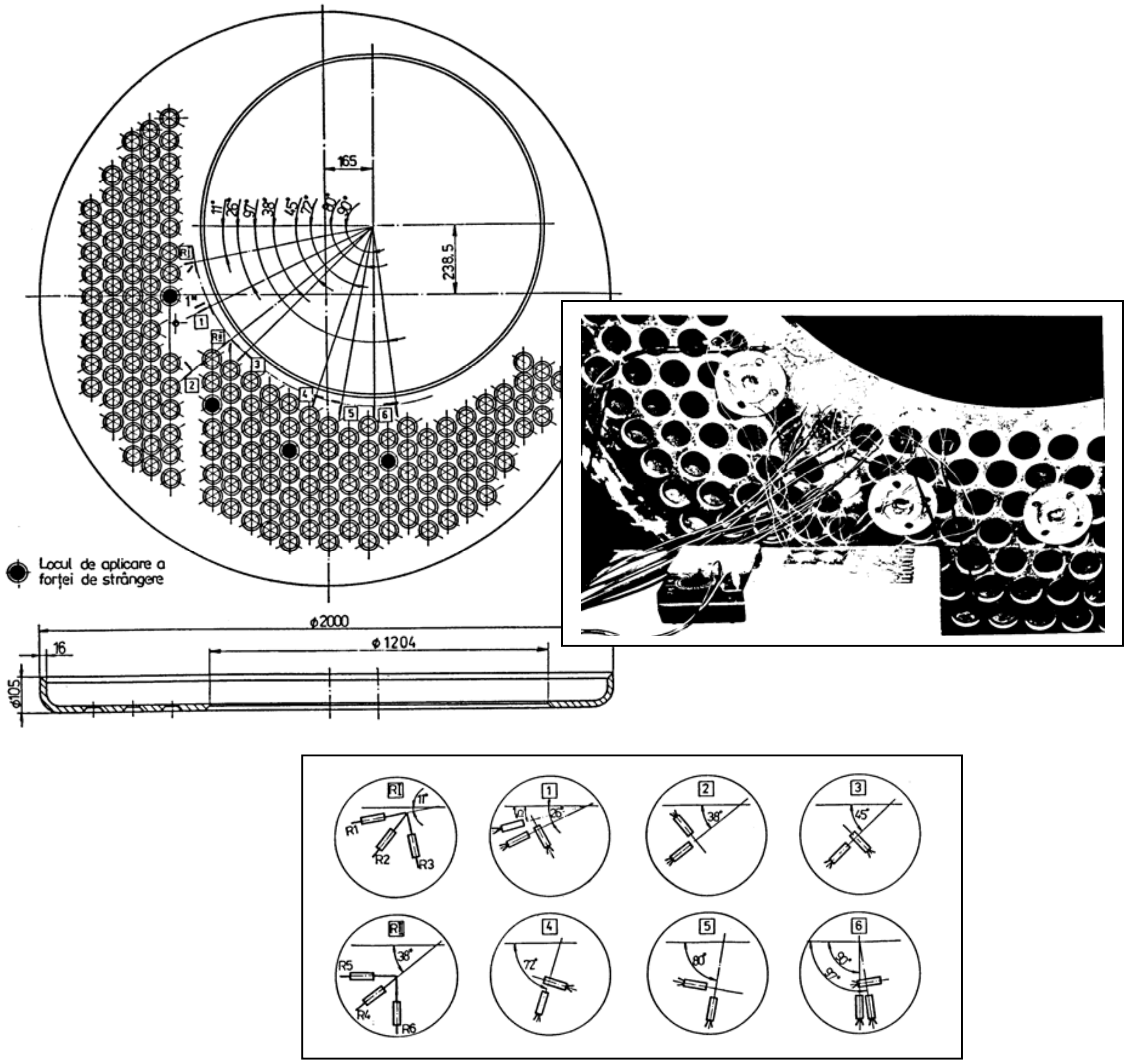

Fig. 2. An assembly view of a tubular behind plate, with strain gauges placement
and the photo ready for measurements

3. EXPERIMENTAL RESULTS

Some more series measures were made, with increasing and decreasing of clamping force. The linear values of deformations, ε measured with electronic tensometers, and finally was processed by the method of the least squares [3], using relations:

$$y = c \cdot x \qquad \text{where} \qquad c = \frac{\sum x \cdot y_i}{\sum m \cdot x^2} \qquad \text{and} \qquad m - \text{number of measures.}$$

Normal stresses σ was find with relation: σ = E·ε, for TER (strain gauge) fixed by radial and hoop directions, and for strain gauges set up with relations:

$$\sigma_{1,2} = \frac{E}{2}\left[\frac{\varepsilon_a + \varepsilon_c}{1-\nu} \pm \frac{1}{1+\nu}\sqrt{(\varepsilon_a - \varepsilon_c)^2 + (2\varepsilon_b - \varepsilon_a - \varepsilon_c)^2}\right]$$

where $E = 2{,}1 \cdot 10^5$ N/mm^2 and $\upsilon = 0{,}3$ (Poisson's ratio), and ε_a, ε_b and ε_c are strains given by TER.

For TER 6 and strain gauges set up R3, in table 2 was tabulated obtained strains, those find by means of –c - parameter and result stresses, function of turning numbers of binding nuts.

In the graphs the stresses values was represented in multi-stage that drag on the origin. Figure 3 presents the stresses variation made by strains gauges fixed as in figure 2, and for extreme values was plotted the experimental determined stresses, directly from read strains.

Table 2

Number TER	Specification	Rotates Number for clamping/Strains measured [mm]						
		0	0,5	1	1,5	2	2,5	3
		11,1	9,0	7,0	4,5	2,5	1,0	0
6 Hoop TER	Read strains, $\varepsilon \cdot 10^{-6}$	-55	-225 -195	-400 -340	-600	-780 -750	-920	-1050 -970
	Parameter, c	-362,9						
	Corrected strains, $\varepsilon \cdot 10^{-6}$	0	-181,5	-362,9	-544,3	-725,8	-907,2	-1088
	Calculated stresses, σ [N/mm^2]	0	-38,1	-76,2	-114,3	-152,4	-190,5	-228,5
R3 Strain gauges	Read strains, $\varepsilon \cdot 10^{-6}$	-92 -106 -101	-250 -220 -280	-360 -380	-530 -510	-640 -600	-790 -790 -700	-820 -820 -825
	Parameter, c	-295,8						
	Corrected strains, $\varepsilon \cdot 10^{-6}$	0	-147,9	-295,8	-443,7	-591,6	-739,5	-887,4
	Calculated stresses, σ [N/mm^2]	0	-31,1	-62,1	-93,2	-124,2	-155,3	-186,3

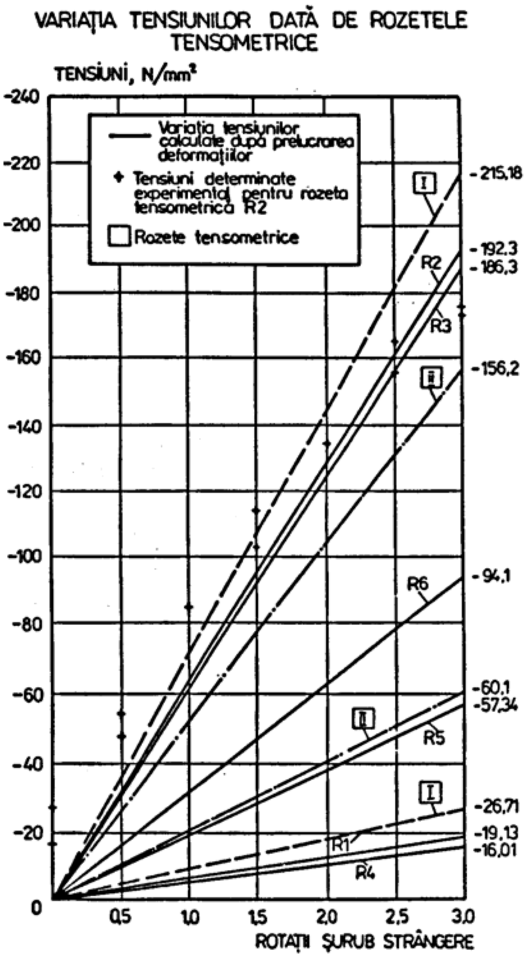

Fig. 3 Variația tensiunilor dată de rozetele tensometrice

4. CONCLUSIONS

a) The constructive shape of tubular behind plates does not permit usage some known analytical relations existing in literature. For determining more precise of stresses and strains is imposed experimental determinations by strain gauges measurement method and post processing obtained results;

b) The realized experimental determinations indicated that the most great stresses appear in case of R1 (-215,18 N/mm^2) and TER 6 strain gauges on hoop direction (-228,5 N/mm^2), with 16-20 % greater than those found by numerical calculus. For K 41 C.2b material - STAS 2883/3-88, calculus breviary indicates, σ_a = 136,6 N/mm^2 ($\sigma_a = \min \{R_m/2,4;\ R^{150}_{p0,2}/1,5\}$). By experiments resulted stresses till 40 % grater than those obtained by calculus and with 35 % greater than those calculated by approximation;

c) So, is considered that the clamping rods used for realizing the plane correlation of the tubular plates influence the state of stresses and strains in the zones around its fixing place;

d) It is recommended avoid the great values of the obtained stresses. Next is important as the pipes fume to be cut out at the dimensions grater than those designed, for respecting the plane correlations. For removing the pipes fume and a plane correlation technology of tubular plates, the pipes must be manufactured at the dimensions only after its welding on the tubular plate.

BIBLIOGRAPHY

1. *** Breviar de calcul de rezistență. Cazan ignitubular de apă fierbinte CIMAF 5 Gcal/h, A 147 BCR, SC SIETA SA Cluj Napoca.

2. *** Manualul inginerului mecanic, vol.2. Bucureşti, Editura tehnică, 1966.

3. Theocaris, P.S., ş.a., Analiza experimentală a tensiunilor, vol.1 şi vol. 2. Bucureşti, Editura tehnică, 1976 şi 1977.