FABRICATION AND ASSEMBLING ERRORS INFLUENCE ON THE FATIGUE STRENGTH OF THE SPHERICAL TANKS SHELL – CALCULUS SIMULATIONS

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Abstract. One analyses, for a 1000 mc spherical tank, equatorial supported by 10 cylindrical pillars (which stores propylene at 2.1 MPa), the influence of the different type fabrication and assembling errors on the spherical shell stress concentration factors. First there are presented the calculus results for the welds flaws influences on the stress concentration factor. Afterwards, with the aid of an original MathCAD program, there are calculated the influence of the fabrication errors (local losses of wall thickness, flattening, assembling eccentricities of the supporting system) on the spherical shell state of stress, taking into account a complex loading (working/storage pressure/hydraulic test pressure, wind, hydrostatic pressure, own weight, deformation restraint (due to horizontal forces determined by the temperature difference between working conditions and assembling conditions and also by the seismic loadings) exerted by the supporting system on the spherical shell and loadings due to the supporting system discontinuities). The influences of the two types of errors are then superposed. Finally is calculated the fatigue number of cycles in the presence of all types of errors.

Keywords: spherical tank shell, stress concentration factor, manufacturing errors, assembling errors, fatigue

1. SPHERICAL SHELL STRESS CONCENTRATION FACTOR EVALUATION DUE TO MANUFACTURING AND ASSEMBLING ERRORS

Usually the engineering calculus approach does not presume the worst loading situation, thinking optimistically, that more than one bad luck cannot happen. The latest events showed the world that the superposing of bad lucks is possible and it is better take them into account, as the best method of prevention.

It is analyzed the spherical shell of a 1000 mc tank, for propylene storage (the storage pressure at +50°C is 2.1MPa). The spherical shell is equatorial supported, by 10 tubular pillars of 609 mm diameter (and 11.91 mm thickness), the tube axis being tangent to the median radius of the spherical shell. The calculus method and mathematical models were presented in [1, 2].

The values for the concentration factor due to eccentricity of the weld plates were calculated with the expression presented in the previous paper and the results are presented in Figure 1.

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In Figure 2 there are presented the values of the stress concentration factor function of angular miss-alignment (fixed ends).

In the Figure 3 are presented the influence of the ovalization on the stress concentration factor, the values were calculated for the angle $\theta = 0^{o}$, the most severe case.

It can be seen that the three types of geometrical errors direct to big values for the stress concentration factor, the most important being, as can be seen from Figure 2, for angular miss-alignment (fixed ends).

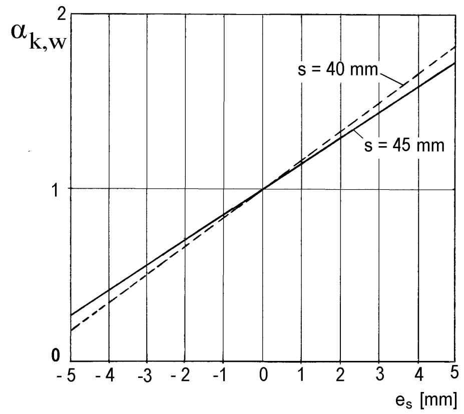


Fig.1. The influence of plates eccentricity on weld stress concentration factor (for two values of the plates' thickness).

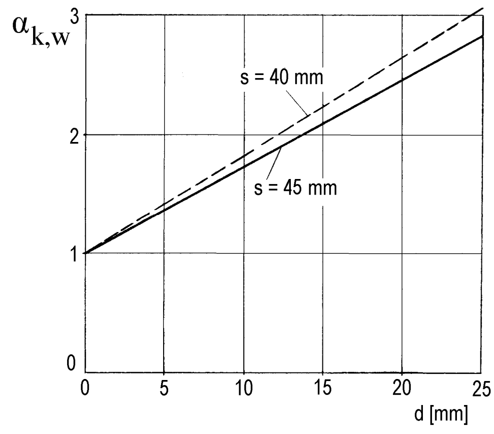


Fig. 2. The influence of weld plates angular miss-alignment (fixed ends).

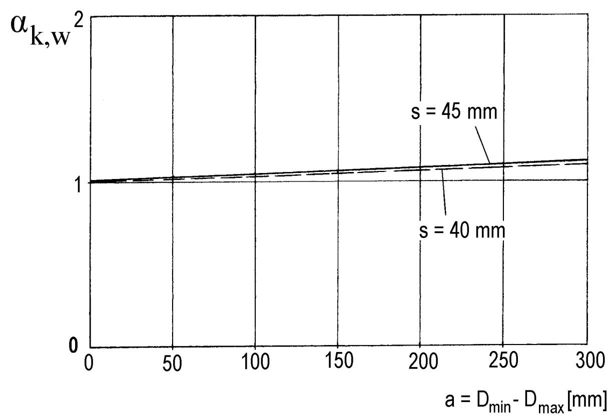


Fig. 3. The influence of spherical shell ovalization (for the angle θ = 0o).

The total stress concentration factor can be calculated with,

$$\alpha_{k,w,t} = \alpha_{k,w,e} \cdot \alpha_{k,w,q} \tag{1}$$

where $\alpha_{k,w,e}$ is the caused by geometrical errors (eccentricities, angular miss-alignment, ovalization etc.); $\alpha_{k,w,q}$ is the caused by quality problems of the weld, like incomplete penetration to the root, weld on only one face, inclusions, pores etc.

The stress concentration coefficient is defined as the ratio between total stress and the membrane stress:

$$C_c = \frac{\sigma_{ech}^t}{\sigma_{ech}^m} = \frac{\sigma_{ech}^m + \sigma_{ech}^c}{\sigma_{ech}^m} \tag{2}$$

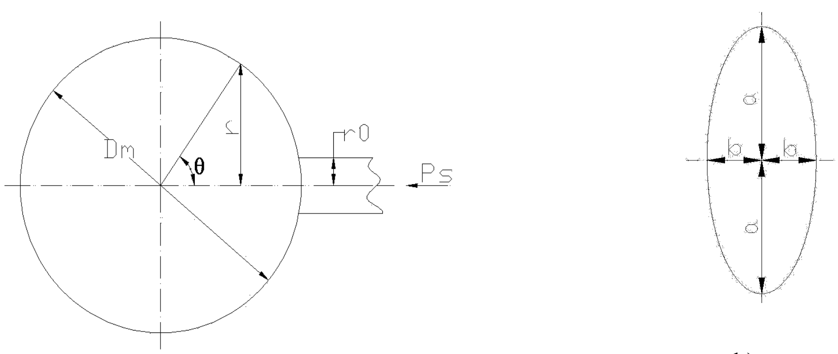
where: σ_{ech}^m is the membrane stress and it was calculated superposing the stresses of the following loadings: inner pressure, shell's own weight, snow' s weight and wind's dynamic pressure; σ_{ech}^c is the supplementary contour stress calculated superposing the stresses of the following loadings: horizontal seismic load manifested like a reaction loading exercised by the supporting pillars, radial deformation restraint imposed by the supporting cylinders (determined by the thermal dilatation, due to the difference between erecting and working temperatures, as well as due to the inner pressure), erecting/assembling eccentricities (permitted deviation from the theoretical position of the pillar, namely the cylinder's axis tangent to the medium shell's radius) and supplementary loading due to the discontinuous supporting.

The influence of the contour loadings is manifested on the l_s distance:

$$l_s = \sqrt[4]{\frac{R_m^2 \cdot s_p^2}{12\left(1-\mu^2\right)}} \tag{3}$$

where: R_m is the medium spherical shell's radius; s_p is the shell's thickness; μ is the Poisson's coefficient.

To calculate the stresses it were used two hypothesis, one supposes that the contact between the cylinder (support) and the spherical shell is an equivalent circle who's radius is determined equalizing the circle's area with the contact spatially curved ellipse's area, resulting $r_{ech} = \sqrt{a \cdot b}$ (*a* and *b* are shown in Figure 4) and the other hypothesis supposes that the contact between the support and the spherical shell is a circle who's radius equals the cylinder support radius (the most severe case).



a) b)

Fig. 4. a) the horizontal loading Ps exercised by the supporting cylindrical pillar on the spherical shell; b) the contact geometry (spatially curved ellipse) between the spherical shell and the supporting cylindrical pillar.

Because the σ_{ech}^{c} have different values at the inner and respectively outer radius, the stress concentration coefficient will have different values too. Also can be defined the safety factor as the ratio between the yield stress and the total, maximal equivalent stress (calculated taking into account all possible loadings).

The calculus principles were presented in papers [1, 2].

The calculus results for the stress concentration coefficient were done with the aid of an original MathCAD program, for exploitation conditions (A) and for hydraulic pressure tests (B) and there are presented in the Figures 5 - 11. The spherical shell thickness is 45 mm but at the end of service life can reach only 40 mm or even less (diminishing because the corrosion, thinning due to technological process, negative tolerance from the thickness and other local thinning). The coefficient which introduces the influence of the seismic intensity for the tank emplacement is k_s = 0.08 corresponding to the degree 8 on the seismic Richter scale. It was considered a non-uniform weight repartition on each pillar (of about $\pm 0.2 \cdot G_1$, where G_1 is the total shell weight (considering also the auxiliary devices mounted on it and the upper part of the supports which are welded on the shell) distributed on a pillar). The erecting eccentricity (permitted deviation from the theoretical position of the pillar, namely the cylinder's axis tangent to the medium shell's radius) usually in the design the eccentricity has the value e = 15 mm. Here for the eccentricity is generated o range of values less or more than that stipulated in the project, having $\pm$ sign, so the eccentricity can give an effort and a momentum of different sign (it will be taken into account those which determines the stress of the same sign as the inner pressure (the worst case)). The number of pillars is fixed – 10 cylindrical supports with the diameter of 609 mm and the thickness of 11.9 mm.

In Figures 5 -10 there are presented the variations of the total equivalent stress values (calculated on the equator both at the inner and the outer diameter of the spherical shell) function of the different types of fabrication errors like flattening, thinning and different values of assembling eccentricities between the support axis and the spherical shell medium radius, for working conditions and hydraulic test conditions, respectively.

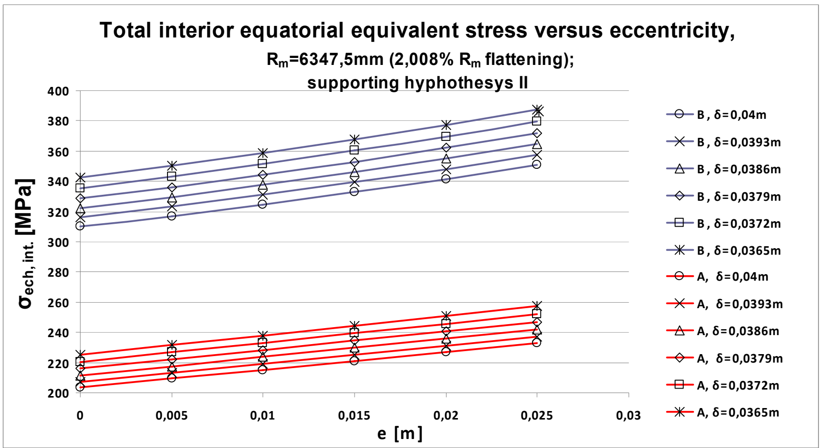


Fig. 5. The superposed influence of flattening and thinning on the shell total equivalent stress at the inner (interior) surface.

In the present paper there were presented only a part of the simulation results.

The stress concentration coefficient variation presented was for the equator (supporting zone), which is the maximum loading zone. It was obtained a slightly non linear variation (power low model) of the total shell equivalent stress function of geometrical errors, in the equatorial supporting zone, in all cases presented in Figures 5 - 10. As one moves away from the equator the stresses diminish, reaching σ_{ech}^{m} at $l > l_s$.

The influence of the butt welds errors can be superposed with the ones of the spherical plates only for the membrane state of stress values because the butt welds are at a distance $l > l_s$.

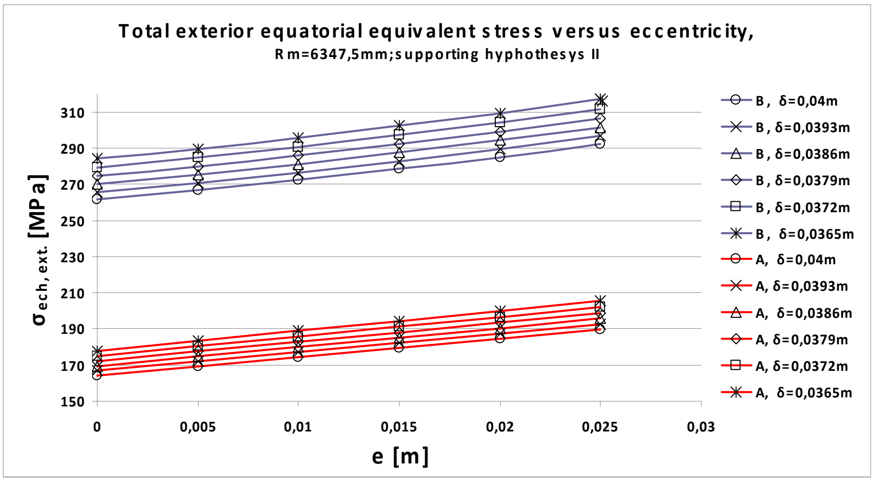


Fig. 6. The superposed influence of flattening, thinning and support assembling eccentricities on the shell total equivalent stress at outer (exterior) surface.

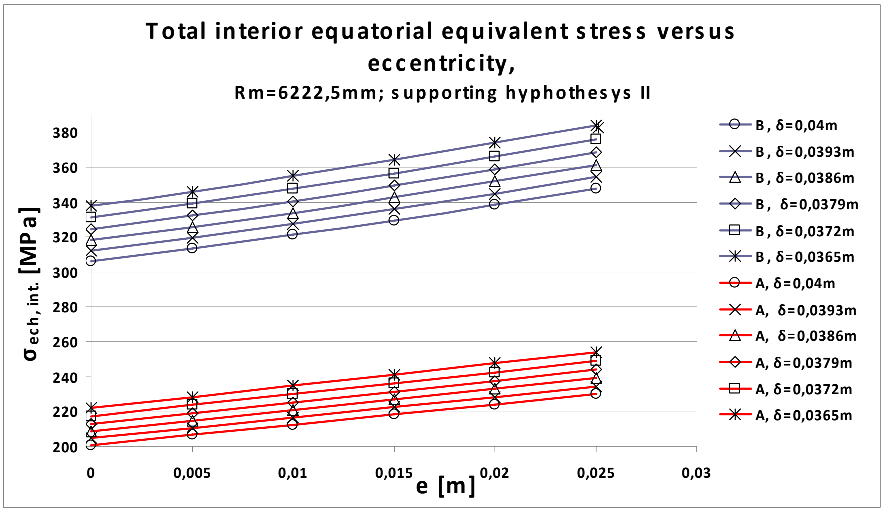


Fig. 7.The superposed influence of thinning and support assembling eccentricities on the shell total equivalent stress at the inner (interior) surface.

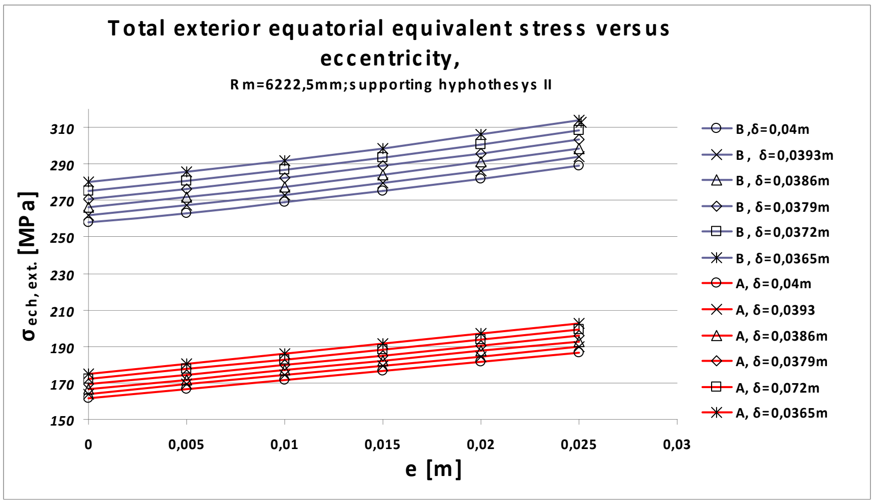


Fig. 8.The superposed influence of thinning and support assembling eccentricities on the shell total equivalent stress at the outer (exterior) surface.

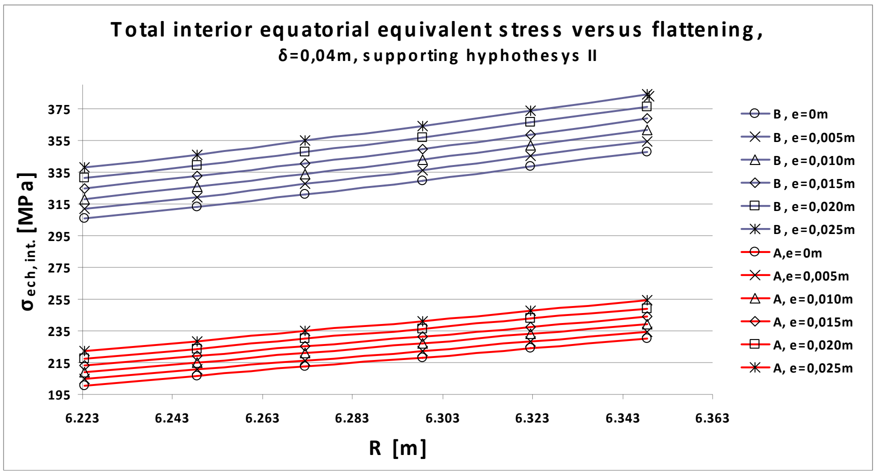


Fig. 9.The superposed influence of flattening and support assembling eccentricities on the shell total equivalent stress at the inner (interior) surface.

From the weld concentration factors calculated and presented in the literature results that the $\alpha_{k,w,t}$ can easily be over 2, which in this case, superposing with the membrane stress influenced by local errors, bigger than allowed ones, overcomes yield stress even in normal exploitation conditions (A), which becomes unacceptable.

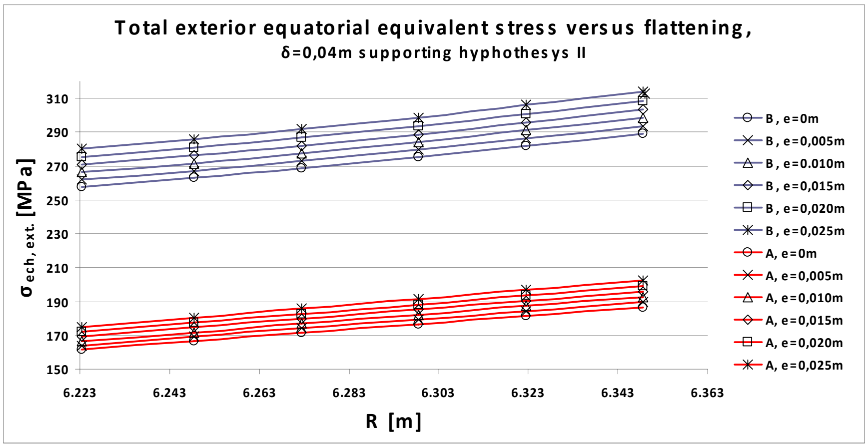


Fig. 10. The superposed influence of flattening and support assembling eccentricities on the shell total equivalent stress at the outer (exterior) surface.

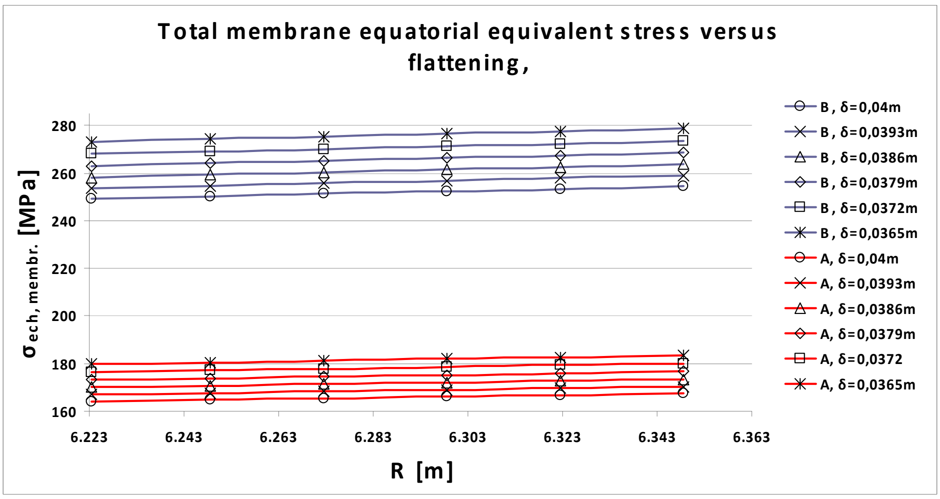


Fig. 11.The superposed influence of flattening and thinning on the shell membrane equivalent stress.

In the last Figure, 11, it is presented the variations of the total membrane equivalent stress due to superposed influence of flattening and thinning.
From the simulations one obtains in the supporting zone, for superposed error influence a stress concentration factor $\alpha_{k,e} = \dfrac{257.359}{200.706} = 1.282$ in normal exploitation conditions (A). Taking into account the membrane stress, in the case of errors presence $\alpha_{k,e} = \dfrac{183.4}{164.1} = 1.118$, with a weld concentration factor $\alpha_{k,w} = 1.1 \cdot 1.5 \cdot 2.2 = 3.63$ and will result a total stress concentration $\alpha_{k,t} = \alpha_{k,e} \cdot \alpha_{k,w} = 4.058$.

With the aid of these values of the spherical shell stress it is possible to calculate the allowable number of fatigue loading cycles or for a maximum admitted fatigue cycles it is possible to calculate the maximum allowable stress value, using the expressions given in the former paper. For pulsating fatigue loading cycle, $\sigma_a = \sigma_m = \dfrac{\sigma_{\max}}{2}$, one ca use the equation deduced from the relation (8) [3,4], given in the first part of the paper,

$$P_T^* = \left(\frac{\sigma_{\max}}{2}\right)^{\alpha+1} \cdot \left[\left(\frac{K_\sigma \cdot c_{-1}}{\varepsilon_d \cdot \gamma_s \cdot \sigma_{-1}}\right)^{\alpha+1} \cdot \left(\frac{N_\sigma}{N_0}\right)^{\frac{\alpha+1}{m}} + \left(\frac{c_s}{\sigma_r}\right)^{\alpha+1}\right] \tag{4}$$

For supporting zone state of stress, using the values P^*=1; $\alpha = 1$; m = 3; K_σ =1.282; ε_d =0.7; γ_s = 0.65; c_{-1} =1.5; $\sigma_{\max} = 205.53$ MPa; N_0=2$\cdot 10^6$; $\sigma_{-1} = 250$ MPa; $\sigma_r = 520$ MPa; c_s = 2.4 from (3) results $N_\sigma = 260253.42$ cycles, much over the number of loading cycles in the spherical tank lifetime. For the zones being in the membrane state of stress, in the weld region, taking into account all the errors, the calculus values are: K_σ =4.058; $\sigma_{\max} = 164.1$ MPa, and the rest is the same, the rest being like above; in this case results $N_\sigma = 18732.67$ cycles, which is over 10.000 cycles, so in a tank lifetime, even extended, the fatigue loading is safe.

2. CONCLUSIONS

As one can see, the calculus simulations can help to know the limits for the fabrication errors and assembling errors from the point of view of the spherical shell's stress values/stress concentration coefficient, safety coefficient, respectively. This is helpful for the designer to decide if can allow larger values for the errors, than those previewed in the project when, happen into the fabrication of spherical tanks, avoiding the disqualifying of spherical plates. The stress concentration coefficient values depends strongly on the supporting geometry, the bigger values are obtained, of course, for the hydraulic test conditions.

The program permits geometric simulations (can generate series of values for the most sensitive errors) and a rapid evaluation of the spherical shell stresses, helping to know if a superposition of different types of errors direct to a non-allowable stress state in working or testing conditions, so it is very useful in industrial/fabrication applications.
It was calculated also the number of fatigue loading cycles, taking into account all the fabrication and assembling errors, using a reliable, simple method.

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