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STRESS STATE ANALYSIS OF PROPANE BUTANE GAS SPHERICAL STORAGE TANKS

ANALIZA NAPONSKOG STANJA SFERNIH REZERVOARA ZA SKLADIŠTENJE PROPAN BUTAN GASA

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- spherical tank

Ključne reči
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- eksperimentalno ispitivanje
- metoda konačnih elemenata (MKE)
- sferni rezervoar

Abstract

The paper presents the design of a spherical tank using a combination of analytical procedure with FEM analysis and experimental testing in order to minimize design time and verify the design strength. The analytical procedure for calculating the tank strength in the initial stages of the design process is briefly presented. Based on analytical results, the tank is designed and by created FEM model. FEM analysis is used to identify areas of high stress concentration. FEM results show that the equivalent value of stress at the points of spherical tank support exceeds the values of yield stress, but not significantly and in a very small area, so the overall design is deemed worthy. Experimental measurements verify FEM results that show it is not necessary to reinforce the spherical tank at the points of support. After 8 months, experiments are repeated giving the same results as the original measurements, thus justifying the decision not to reinforce tank supports.

Izvod

U radu je prikazano projektovanje sfernog rezervoara kombinacijom analitičke procedure sa MKE analizom i eksperimentalnim ispitivanjem, kako bi se minimiziralo vreme projektovanja i proverila nosivost. Ukratko je prikazan analitički postupak za proračun čvrstoće rezervoara u početnim fazama procesa projektovanja. Na osnovu analitičkih rezultata rezervoar je dimenzionisan uz kreiranje MKE modela. MKE analiza se koristi za identifikaciju oblasti sa velikom koncentracijom napona. Rezultati MKE pokazuju da ekvivalentna vrednost napona na tačkama oslonca sfernog rezervoara premašuje napon tečenja, ali ovo prekoračenje nije značajno, na veoma maloj površini, pa je celokupno projektovanje ocenjeno kao korektno. Eksperimentalna merenja potvrđuju MKE rezultate da nije potrebno ojačati sferni rezervoar na tačkama oslonca. Posle 8 meseci eksperimenti su ponovljeni dajući iste rezultate kao i originalna merenja, čime se pravda odluka da se ne pojačaju nosači rezervoara.

INTRODUCTION

Spherical tanks are among the most critical structures due to high stresses and typically stored dangerous gases, /1-4/. In Fig. 1 a spherical tank of volume $V = 600$ m^3 for storage of propane-butane gas is shown, indicating basic geometric measures, discretization of finite elements, and the tank support in two ways.

In the first case, the tank has 8 symmetrical supports, so that only one-eighth of the tank can be analysed. An angle $\alpha = 90°$ is adopted here, and fixed points of support are indicated by reinforced lines (between nodes 38 and 47). An overpressure of $p_g = 16.7$ bar, hydrostatic pressure-height of liquid $H_1 = R$ and self-weight are taken into account, /5/.

In the second case, the support is performed along a circular line (nodes 38 and 46). Figure 2 shows displacements of some characteristic cross-sections, as follows:

- cross section *x*-*z*,
- cross section *y*-*z*, and
- cross section *x*-*y*.

Solid lines show displacements in the case of bearing along the equator line, and dashed lines in the case of bearing in points. From Fig. 2 it can be seen that we have an increase in radius everywhere (except in the points and on the bearing line), and that due to hydrostatic pressure, this increase is slightly higher in the lower than in the upper half (in the vertices they are in the case of leaning along the line, displacements of 8.85 and 9.76 mm) /6, 7/.

In Fig. 2 a,b and c, deformations are shown in radial directions for individual cross-section planes, which shows that deformations in case of bearings along the equator line (38-46) are slightly larger than those in bearing at individual points behind the *x*-*z* and *y*-*z* planes. This is explained by the fact that all points on the equator line, due to the support, cannot have radial expansion, and this must be transferred to all other points where this movement is allowed. In the case of bearing at points, the number of points at which the radial propagation is limited to is smaller compared to the bearing along the line, and hence, the radial propagation, distributed over a larger number of points, has a lower value. (Fig. 2 a, b- line *a*).

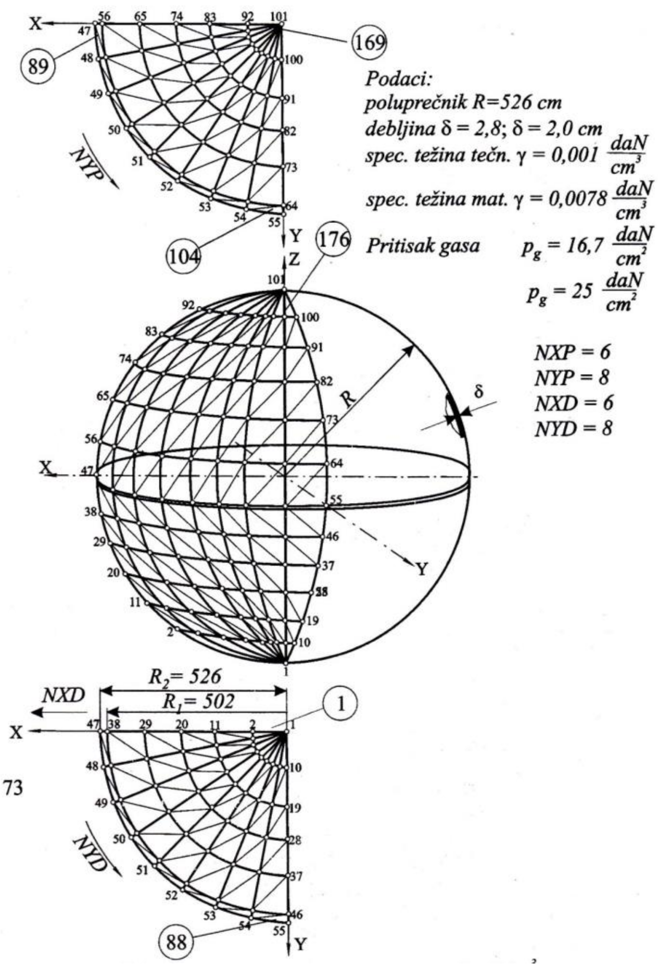

Figure 1. Spherical tank volume $V = 600$ m^3 (fine mesh).

As for the plane *x-y* (Fig. 2c), a larger radial spread is observed in the case of support at points for nodes 49 and 53. If we keep in mind that the support is in nodes 47, 51 and 55, and that nodes 49 and 53 are located in the middle of the lines connecting these nodes, the only possible maximal expansion is in nodes 49 and 53.

Distribution of the first main stress σ_1 for the vertical circle passing through the centres of gravity of the elements closest to the vertical plane *x-z* (in the case of bearing at points and in the case of bearing along the equator) is given in Fig. 3a, and for the elements closest to the horizontal plane from the lower side in Fig. 3b.

For the main stress σ_1 (Fig. 3a, b), an increase is also noted in case of line support due to obstructed displacement on line 38-46.

There is also a more even distribution of stress in the case of bearing along the line, as opposed to bearing at points, when we have a reduction in the main stress.

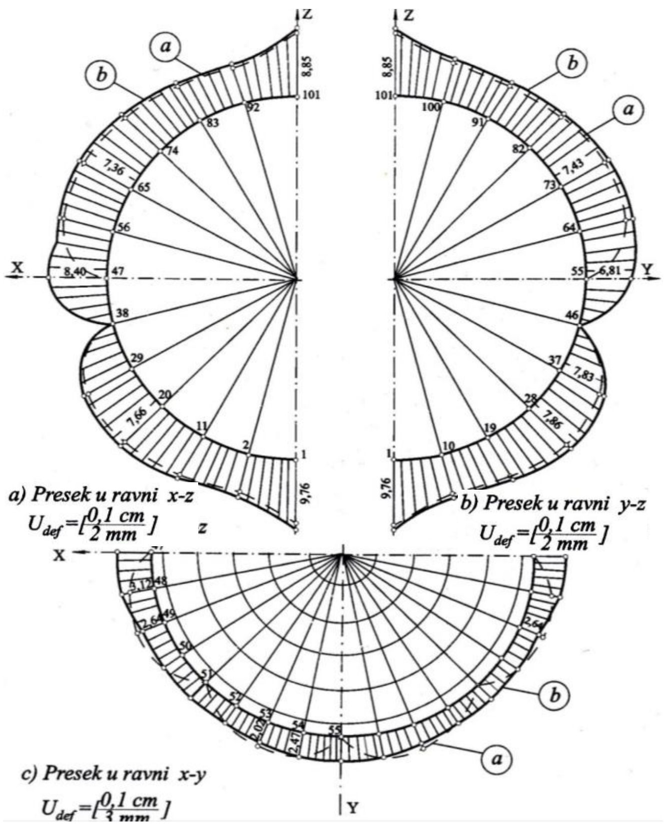

Figure 2. Deformations in radial directions (in mm) *a*-support at points; *b*- support along the equatorial line.

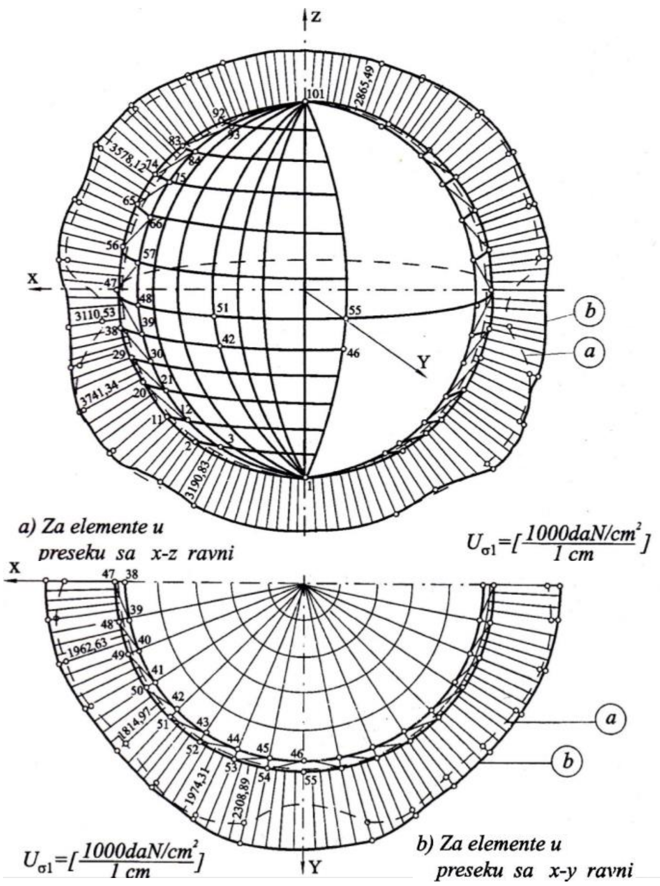

Figure 3. Main stress diagram (daN/cm^2); *a*-support at points; *b*-support along the equatorial line.

The uniformity of the distribution of main stress points leads us to the conclusion that by leaning on the line we approach the membrane stress state and that only the restriction on line 38-46 prevents us from doing so, /8, 9/.

Displacements for some characteristic cross-sections are shown in Figs. 4 and 5, as well as the distribution of the first main stress in case of a change in wall thickness from 2.8 to 2.0 cm under the same load conditions, /10, 11/. The influence of thickness change on deformations and stress fields is shown in Figs. 4 and 5. The wall thickness is reduced from 2.8 to 2.0 cm (i.e. about 30 %) and the displacements are 30 % higher.

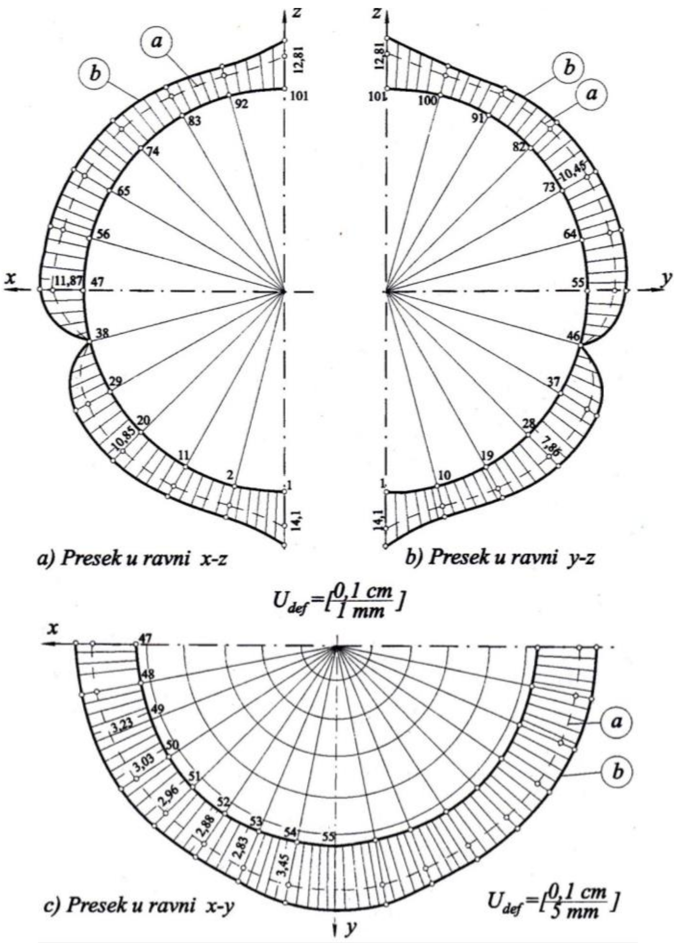

Figure 4. Deformations in radial direction (mm); *a*-shell thickness 2.8 mm; *b*- shell thickness 2.0 mm.

The main stress σ_1 at characteristic points has increased 2.78 times. Here, we would like to point out that even a small change in wall thickness significantly affects the amount of stress and for these reasons, when designing such tanks, care should be taken to add to the corrosion /12, 13/.

The method of contact elements covers the influence of the support, so the increase in stress in relation to the analytical solution originates from the bending stress which manifests further from the place of support. In the specific case for support along line 38-46 and wall thickness 2.8 cm (Fig. 3 - line *b*), the main stress magnitude is 311 MPa, while the analytical solution is σ_φ = 164.5 MPa. It should be noted that the analytical solution takes into account the disturbance of the membrane stress state only from the action of hydrostatic pressure at the point of support, and that the disturbance of the membrane stress state from the action of gas pressure is not included.

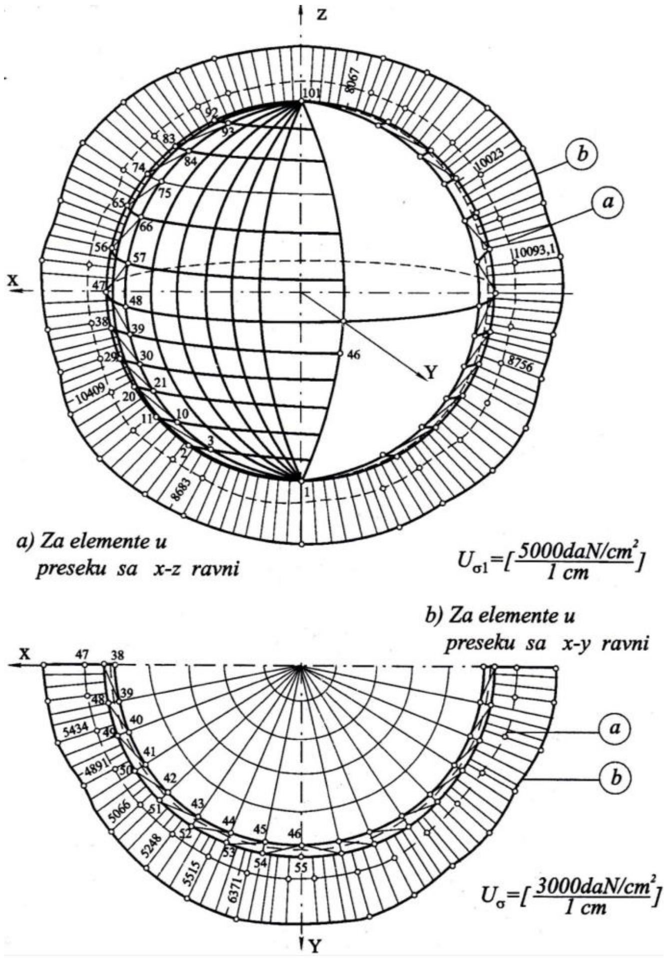

Figure 5. Main stress diagram (daN/cm^2); *a*-shell thickness 2.8 mm; *b*- shell thickness 2.0 mm.

The disturbance of the membrane stress state due to the action of gas pressure at the point of support cannot be captured by accurate analytical methods, so approximate solutions or numerical methods are used. The finite element method offers great possibilities in this regard.

CONCLUSION

With the developed methodology, problems of strength of the bearing structure of the tank can be successfully solved for a general case of loading. By determining the stress and displacement of tank walls as a geometrically nonlinear problem, stability of walls can be examined. The procedure for automatically generating data on geometry and loads provides great opportunities in everyday practice. The approach presented here is applicable to all typical tank structures, and generally cylindrical and spherical pressure vessels. Designers are given the opportunity to choose the optimal solution from the point of view of material use and construction itself. The applied procedure can also take into account the impact of design inaccuracies on tank stability.

It should be especially emphasized that it is not possible to cover all the influences of load and disturbance of the membrane stress state by analytical methods, which is possible in the case of FEM application. The results obtained by FEM are very close to the experimental ones.

Bearing in mind that the developed procedure can provide a complete picture of stress and deformation, the results obtained in each case can be used to create an experimental program for testing the structure of the tank.

Results obtained here allow research to be continued in several areas. It is also possible to analyse thermal stresses and their influence on welding parameters. In the field of process technology, the influence of vacuum on the stability of the tank is of special importance, so further research in this field would be useful.

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