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SIMULATION AND ANALYSIS OF THE MECHANICAL BEHAVIOUR OF GAS STORAGE TANKS

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ABSTRACT

Pressure tanks are closed containers in which fluids are kept at a pressure higher than atmospheric pressure or under vacuum at different temperatures. They are used for storing various fluids or gases, transporting them or for various technological purposes. Based on the simulation results, the influence of the container thickness and internal pressure on the distributions of the maximum values of total deformations and equivalent stresses were analyzed. For the range of input parameters analyzed (internal gas pressure and container shell thickness), the construction and design solution that meets the requirements of the design (maximum gas pressure of 19.6 bar, testing pressure (with water) 29.6 bar, internal volume 16850 litres, nominal diameter of 2000 mm and length of 5627 mm) is that with a container having a shell thickness of 16 mm to meet the safety criteria provided in SR EN 13445-3:2014 with subsequent amendments.

Keywords: pressure tanks, simulation

1. INTRODUCTION

Pressure tanks are closed vessels in which fluids are kept at a pressure higher than atmospheric pressure or under vacuum at different temperatures. Vessels are used for the storage of various fluids, for their transport or for various technological purposes.

The construction elements of this tank are, as follows in Fig. 1: container cover for position 1, container bottoms for 2, the manhole for 3, container resealing system for 4, lifting system for 5 and connections for position 6.

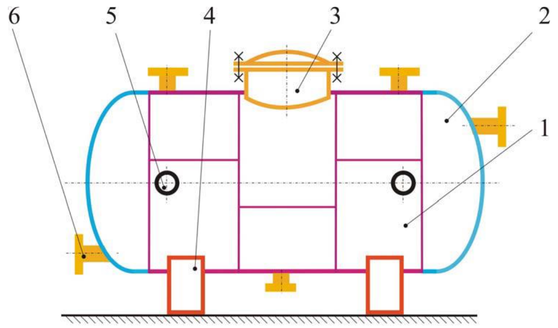


Fig. 1. Horizontal container [1]

2. CHARACTERISTICS OF THE TANK ANALYSED

The original destination of the tank assembly was to transport various types of gases by road transport. With the launch of the order to relieve and repair the tank assemblies, the beneficiary has changed their use: they will be used for gas storage in a gas bottling line to be built. A tank assembly consists of a metal support frame inside which a tank with a capacity of 16850 litres is fixed, see Fig. 2.

The metal frame supporting the tank is a welded construction of square pipe elements. The tank is horizontal, with ellipsoidal bottoms, constructed by welding a 16 mm thick steel sheet. At one end, in the middle, the tank is fitted with a manhole cover on which a mechanical level indicator is placed.

Fig. 2. Side view of a horizontal tank assembly [Criomec SA, Galati]

3. RECIPIENT MODEL

The finite element method (FEM) is far from perfect, but it is the best method currently available for a wide variety of types of calculations in all areas of engineering. The method, and software based on it, have become fundamental components of computer-aided design (CAD) systems and are indispensable in all situations where competitive engineering is required. The problems involved in practical work are often complex in terms of the physical and geometric composition of the parts, loading conditions, boundary conditions, etc., so the integration of differential equations is difficult or even impossible.

The tank is analyzed for a simplified structure, where bolt-flange systems are not represented.

3.1 MODEL OVERVIEW

A tank with a nominal diameter of Ø2000 mm and a length of 5627 mm has been designed in Ansys [2].

The technology line where the tank is included asked for a maximum gas pressure of 19.6 bar, resulting in a testing pressure (with water) of 29.6 bar, internal volume of 16850 litres).

The tank is not analyzed with all the openings (only the inlet and outlet connections have been modelled); in reality, the tank has another manhole, connections for measuring equipment, etc. The supports are mounted directly on the container, see Fig. 3. In reality the supports are mounted on originally circular reinforcements The model is isothermal; in reality, the gas inside can have temperatures between -50 °C and +50 °C.

The discretization network presented in Fig. 4 is fine, the size of the elements is 5 cm, resulting in 15713 elements with 15658 nodes.

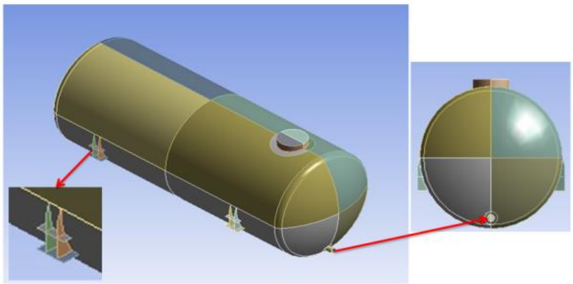


Fig. 3. The model made with details on a support and loading port

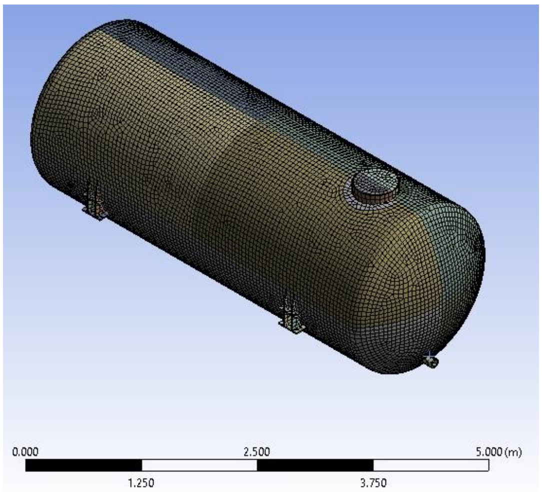


Fig. 4. The discretization of the model

Figure 5 shows the loadings used (constant internal pressure, 1.96 MPa, gravitational forces for the tank shell). The tank was embedded in the lower base of each support.

Table 1 shows the material properties for the vessel model, using a bilinear hardening model, characteristic of a pressure vessel steel (16Mo3, SR EN 10028-2) [3]-[7], graphically presented in Fig. 6.

This an analysis in static state, load conditions being the internal pressure and the gravitational forces of the tank mass, all considered as constant). The lower surfaces of the support are considered fixed (in the frame tank, not represented here). No failure criterion was introduced as the objective of the analyses was to determine the thickness that is necessary for maximum stress under the allowable value given by yield stress divided with the safety coefficient in static load.

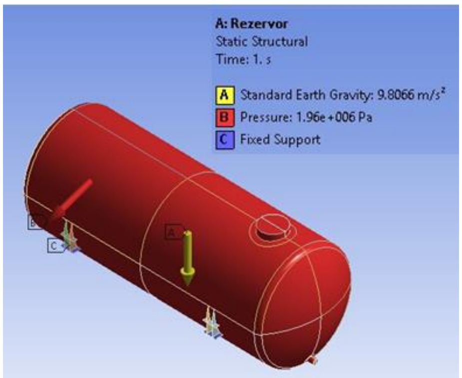


Fig. 5. The load conditions

Table 1. Material properties

Property	Value
Density, kg/m^3	7850
Tensile yield point, MPa	250
Compressive yield point, MPa	250
Tensile ultimate strength, MPa	460
Compressive ultimate strength, MPa	460
Young modulus, MPa	2×10^5
Poisson coefficient	0.3
Shear modulus, MPa	0.769×10^5
Tangent modulus, MPa	7600

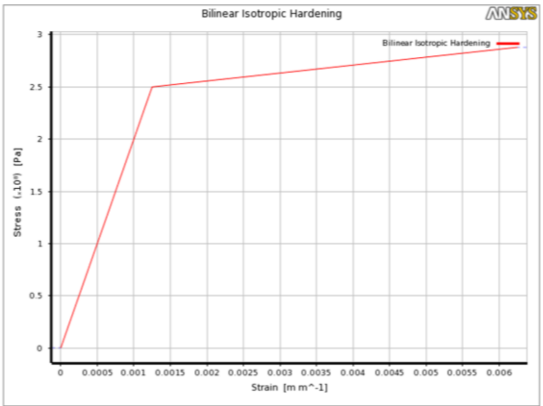


Fig. 6. Material properties involved

There are simplifying assumptions for this model:

- welds are not taken into account when assessing deformations and stresses, meaning all components of the tank are made of the same material,
- the vessel is not analyzed with all the openings (only the inlet and outlet connections have been modelled), in reality, the vessel has another manhole, connections for measuring equipment etc.,
- the supports are mounted directly on the container. In reality, the supports are mounted on originally circular reinforcements,
- the model is isothermal; in reality, the gas inside can have temperatures between -50 and +50 °C,

- the stresses are statically acting (constant in time).

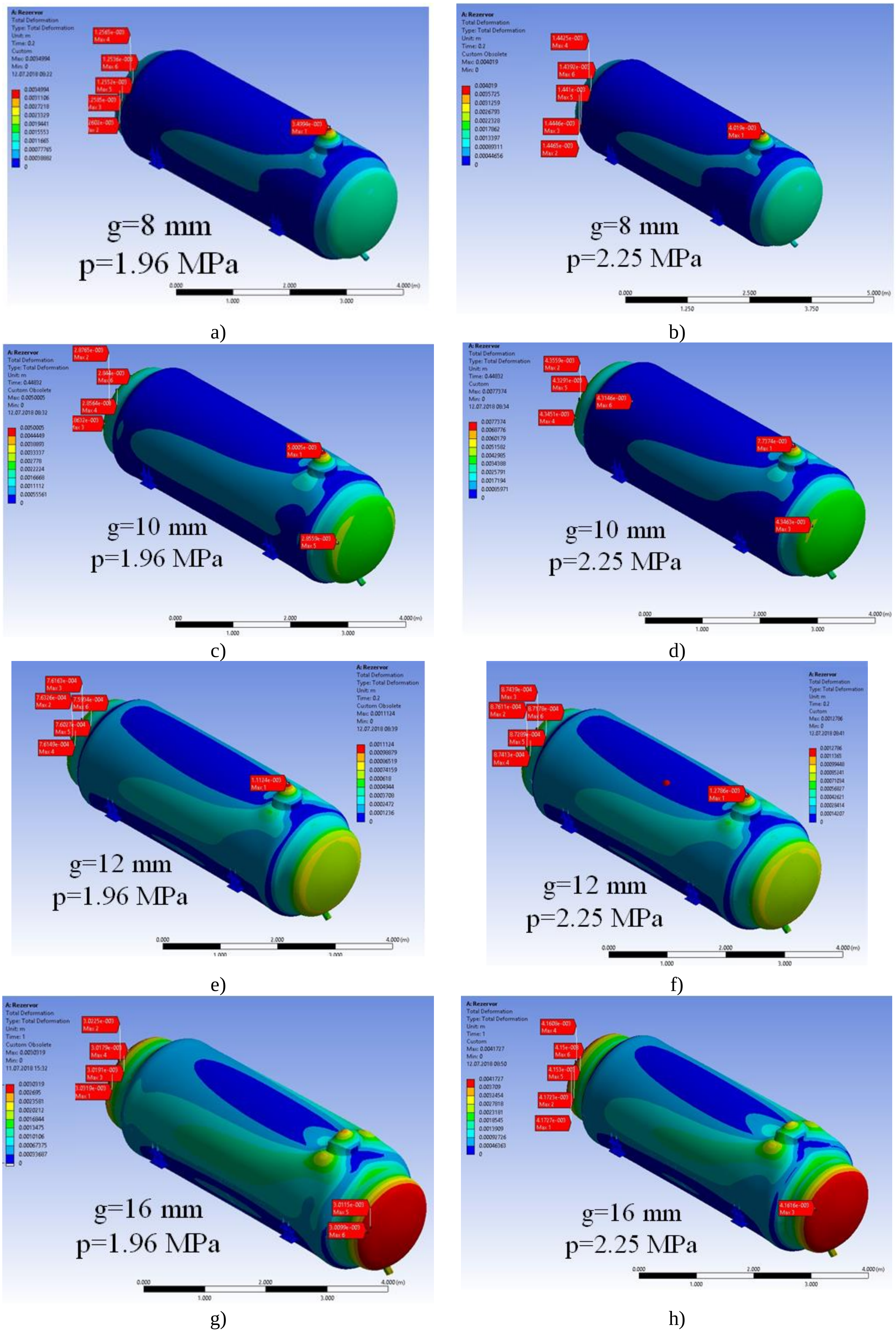
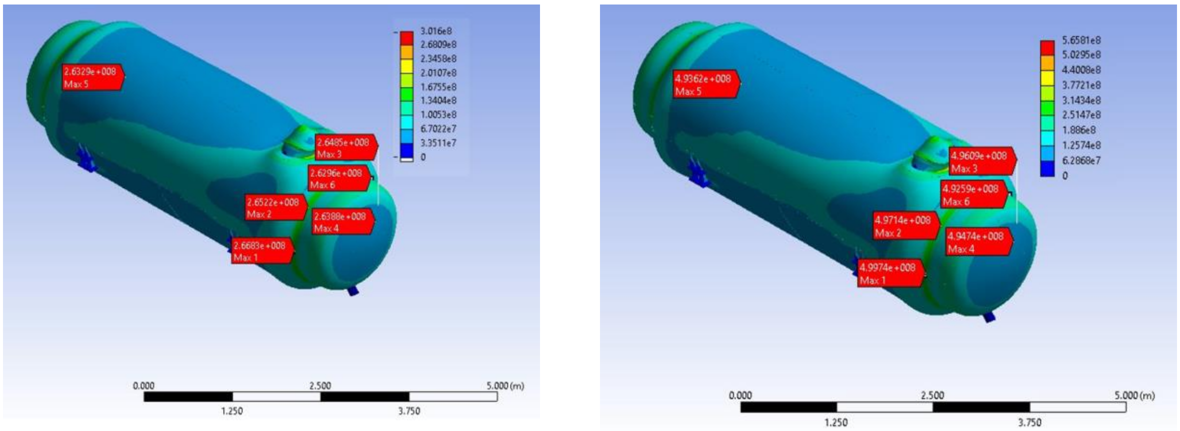


Fig. 7. Total deformation distributions for four cases



a) 1.2 MPa b) 2.25 MPa

Fig. 8. Von Mises stress distribution for a tank of 8 mm thickness (Scale deformation is greater than real).

4. RESULTS

In order to analyze the container, we performed runs with 4 values for shell thicknesses (8 mm, 10 mm, 12 mm and 16 mm, respectively) and each case was analyzed with 4 internal pressures (1.2 MPa, 1.96 MPa, 2.25 MPa and 3MPa).

In the following images for stress and deformation distributions, each image has its colour scale.

Analyzing the total deformation distributions in Fig. 7, one may notice that for steel plate thickness of 8 mm. the higher values are obtained on the input pipe zone and in the region of the opposite bottom and their position are similar for both represented pressure, 1.96 MPa and 2.25 MPa. When the thickness is 10 mm, high values of the total deformation appear on both bottom regions and for the internal pressure of 2.25 MPa, a high value is obtained even on the tank cylinder. Increasing the thickness to 12 mm, the distribution of high values for total deformation is on the input pipe and the opposite bottom. For 16 mm thickness, high values of total deformations appear on both bottom zones, but they are smaller than those recorded for other thicknesses.

Regarding the deformations (Fig. 7), it can be seen that the maximum deformation value increases with pressure, but between 1.96 MPa and 2.25 MPa gas pressure, the maximum deformation values evolve in a small range. The higher values of von Mises stress are in the neighbourhood of the input pipe, towards the right bottom in Fig. 8. Attention has to be paid to the colour scale, as

The shape of the graph is also preserved for the maximum values of the equivalent von Mises strain (see Figs. 8 and 9).

Analyzing the images for deformation distributions, one may notice that the localization of the maximum value is changing from one run to another. This is an advantage of the FE analysis because it could point out both qualitative and quantitative aspects of the loaded system.

From Figure 9, it can be seen that the maximum values of the equivalent von Mises stress are inversely proportional to the thickness of the coating, but not in a direct relationship. For thicknesses of 12 mm and 16 mm the maximum values of equivalent stresses are close, but the designer has to take into account the corrosion addition for the thickness of such a vessel and will therefore opt for the 16 mm thickness.

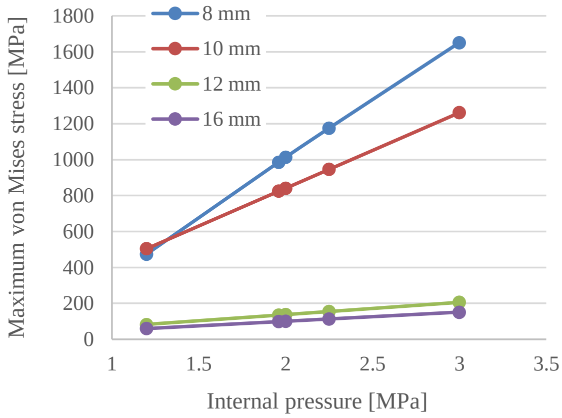


Fig. 9. Maximum values of the equivalent von Mises stress as a function of tank shell thickness (bilinear material model without failure criterion)

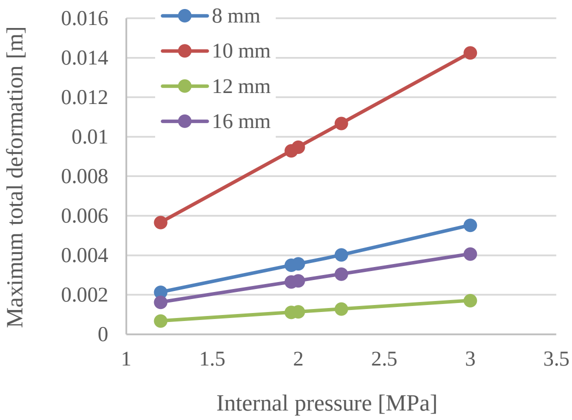


Fig. 10. Maximum total deformation for several thicknesses of the tank shell

It is obvious that, in reality, shells with 8 or 10 mm will fail as the maximum von Mises values overpass the ultimate strength of 460 MPa as supposed for both traction and compression. But for thicknesses of 12 mm and 16 mm, the maximum values for von Mises stress are under the yield point, offering even a good safety coefficient for the static load.

The maximum total deformations (see Fig. 7) show that their values and locations are different, depending on the thickness of the coating, with lower values obtained for larger thicknesses (Fig. 10).

From this analysis, the nominal thickness of 16 mm would be recommended for the tank shell, as lower tolerance could reduce the effective thickness by several hundred o microns, as in Table 2, from EN 10029:2010 [12] and the designer should specify the tolerance class. For class C, the simulation already run is taken into account but a finer analysis could be done with an effective minimum thickness of other selected classes (A, B or D).

Table 2. Tolerances on the nominal thickness

Nominal thickness	Class A		Class B		Class C		Class D	
	L	U	L	U	L	U	L	U
[mm]								
16	-0.6	+1.0	-0.3	+1.0	0	1.6	-0.8	+0.8

** L- lower, U - upper

5. CONCLUSION

A FEM study of the mechanical behaviour of a gas storage tank was carried out.

A model of the tank was created, and 16 cases were run with different shell thicknesses and internal pressures.

Based on the simulation results, the influence of the thickness of the container shell and internal pressure on the distribution of the maximum values of the total deformations and equivalent stresses was analyzed.

Considering the results of this study, the use of the finite element method for the analysis of metallic structures is useful and with its help, it is possible to find optimized solutions for shape, materials, and operating conditions (total allowable deformation, allowable stresses, etc.).

For the range of input parameters analyzed (internal gas pressure and container shell thickness), the construction and design solution that meets the requirements of the design basis (maximum gas pressure of 19.6 bar, testing pressure (with water) 29.6 bar, internal volume 16850 litres) is that with a container having a shell thickness of 16 mm to meet the safety criteria provided in SR EN 13445-3:2014 with subsequent amendments [8]-[11].

REFERENCES

[1] Palade V, Ştefănescu I., Recipiente și aparate tubulare (in Romanian), http://www.om.ugal.ro/om/biblioteca/Recipiente_si_aparate_tubulare.pdf (accessed 12.07.2018)

[2] *** 2021 ANSYS Mechanical User's Guide (Southpointe USA: ANSYS, Inc)

[3] *** SR EN 10028-1:2017 Flat products made of steels for pressure purposes - Part 1: General requirements

[4] *** SR EN 10028-2:2017 Flat products made of steels for pressure purposes - Part 2: Non-alloy and alloy steels with specified elevated temperature properties

[5] Moller M., Swedish Radiation Safety Authority, Research, Report number: 2017, 05 ISSN: 2000-0456, Available at www.stralsakerhetsmyndigheten.se, Robust structural verification of pressurized nuclear components subjected to ratcheting, part two – stress-strain curves for cyclic elastoplastic analysis, (accessed at 20.06.2018)

[6] Chen D., Tong C., Liu K., The method of tangent modulus factor for the design of external pressure vessels, Advanced Materials Research Online: 2012, Vols. 479-481, pp. 1578-1584, doi:10.4028/www.scientific.net/AMR.479-481.1578 (accessed 20.06.2018)

[7] Marcal P. V., Elastic-plastic analysis of pressure vessel components, Division of engineering Brown University Providence, Department of defense, Advanced Research Projects Agency, Contract SD-86 material Research program, January 1969, www.dtic.mil/dtic/tr/fulltext/u2/682482.pdf (accessed 20.06.2018)

[8] *** SR EN 13445-3:2014 Unfired pressure vessels - Part 3: Design

[9] *** SR EN 13445-3:2014/A1:2015. Unfired pressure vessels - Part 3: Design

[10] *** SR EN 13445-3:2014/A2:2017 Unfired pressure vessels - Part 3: Design

[11] *** SR EN 13445-3:2014/A3:2018 Unfired pressure vessels - Part 3: Design

[12] *** EN 10029:2010 Hot-rolled steel plates 3 mm thick or above - Tolerances on dimensions and shape