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NUMERICAL ANALYSIS OF THE BEHAVIOR OF PRESSURE VESSEL SUBJECT TO VARIABLE REQUESTS USING THE FINITE ELEMENT METHOD

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Abstract: The behaviour of a metallic material subjected to the action of a variable stress (which generates mechanical stresses in the material with variable intensities over time) differs significantly from that corresponding to the action of constant or monotonically increasing stresses which do not change in intensity over time or which increase continuously in intensity as time passes), its rupture may occur even if the intensity of the mechanical stresses generated during the stress σ is less than the tensile strength of the material Rm. The article studied the static and dynamic behaviour (fatigue) of an equipment consisting of a jacket and two semi-helical lids.
Key words: finite element, fatigue, pressure, equipment, tension, FEM

1. INTRODUCTION

If we analyse the operating regime of the equipment (static devices) that make up the petrochemical and refining installations (and other similar technological installations, in the chemical, energy industry, etc.) it can be seen that there are premises for most of them to be included. in the category of metal constructions with variable stresses in operation, which degrades due to the phenomenon of fatigue. A statistic compiled by the European Pressure Equipment Research Council - EPERC, the conclusions of which were summarized in the histogram in Figure 1, shows the share of equipment (metal constructions such as pressure vessels, tanks, pipes, platforms, etc.) subject to variable in-use demands (in total used equipment) in various industrial fields (installations).

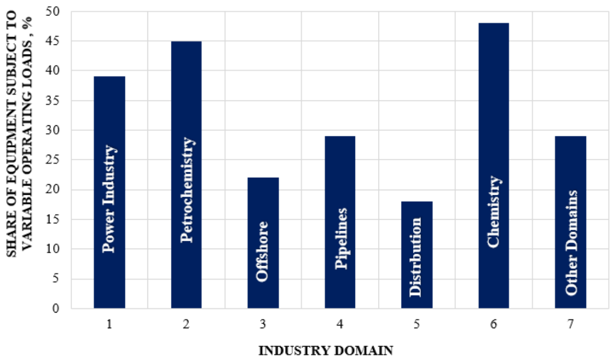

Figure 1 Share of equipment subject to variable operating loads (total equipment used) in various industrial areas (installations)

Proper in-service behaviour of stable metal Pressure vessel that make up petrochemical and refining technological installations is ensured if the following mechanical actions are taken into account when designing this equipment [1,2,3,4]:

- Internal and / or external pressure of fluids or working environments;
- Own weight of Pressure vessel;
- The maximum weight of the contents of the Pressure vessel under the operating conditions;
- Weight of water or liquid used in mechanical strength tests of Pressure vessel;
- Wind and loads of snow, frost or ice;
- Stress due to earth movements (earthquakes);
- Other mechanical loads or loads borne by Pressure vessel, including those acting on them during transport or installation.

Given the multitude of mechanical actions (charges, loads) to which pressure vessels are subjected, various cases of mechanical stress must be considered in their design, which can be classified as follows [5,6,7,8]:

- Cases of normal loading, operation, in which the charges and loads acting during the operation of pressure vessels under normal (normal) conditions are considered, including during the periods of shutdown and / or start of the technological installation of which these vessels are part;
- Cases of exceptional loading, which consider mechanical stresses that could act during events (with low probability of occurrence), such as earthquakes, explosions inside Pressure vessel, etc., which require rapid unloading of Pressure

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vessel and rigorous verification of their technical condition before re-commissioning.

- Cases of loading to verifications, in which the loads and loads acting during the tests to which the Pressure vessel are subjected after manufacture and / or after carrying out corrective maintenance work are considered.

2. STATIC ANALYSIS OF AN EQUIPMENT USING FINITE ELEMENTS METHOD

The stress analysis method is based on the behaviour of pressure vessels considered as thin-walled coatings, in which case the classification of stresses into membrane stresses and bending stresses, as they are identified in theory, is easy to do. Difficulties arise when the primary analysis, in which the pressure vessel is considered as a thin coating, is not used, and the finite element analysis of the Equipment uses axially symmetrical or three-dimensional elements.

In a first step, a statically pressurized Equipment was analysed. The equipment (Figure 2) consists of a jacket (diameter Ø406 mm, 5 mm thick and 1000 mm long), 2 blind flanges (DN60 and DN 50), 2 semi-ellipsoidal bottoms 8 mm thick and 2supports.

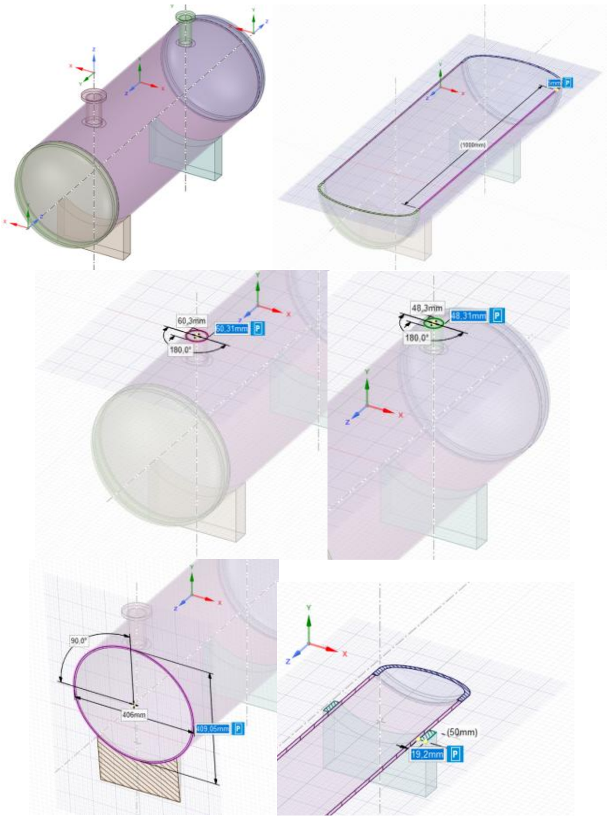

Figure 2 Equipment under analysis

Numerical analyses were performed taking into account that the pressure inside the Equipment is applied in steps of 10 bar up to 50 bar (figure 3).

	Steps	Time [s]	☑ Pressure [MPa]
1	1	0,	0,
2	1	1,	1,
3	2	2,	2,
4	3	3,	3,
5	4	4,	4,
6	5	5,	5,
*			

Figure 3 Successive application of pressure

The results of the static analysis are presented in the sequence of images in Figures 4 ... 6.

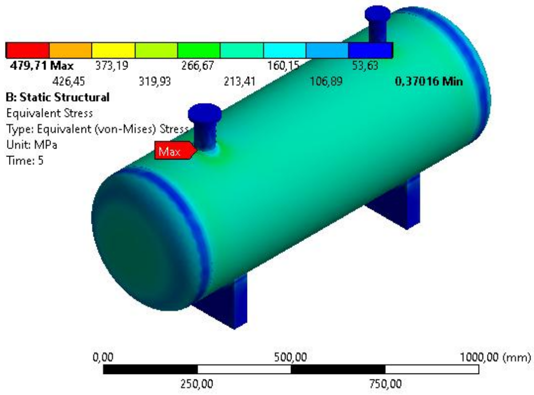

Figure 4 The state of tension in the equipment

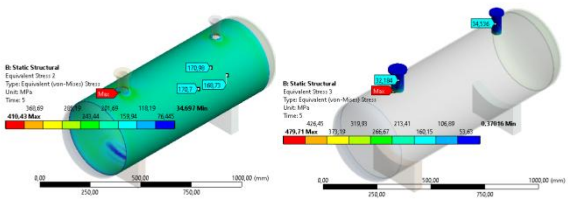

Figure 5 The state of tension in the jacket and connections

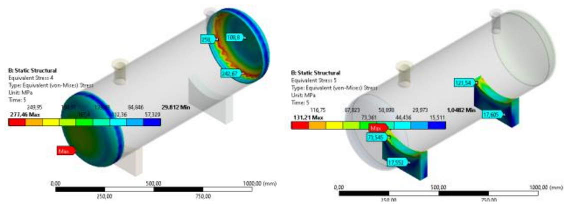

Figure 6 The state of tension in the jacket and connections

The mechanical stresses that are taken into account when evaluating the static behaviour of a pressure vessel are determined (calculated) by a stress analysis on the vessel considered to be free from defects.

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3. ANALYSIS OF FATIGUE BEHAVIOR OF AEQUIPMENT USING FEM

When evaluating the fatigue behaviour of pressure vessel, special attention shall be paid to determining the states of mechanical stress and specific deformations which are generated during each of the stresses to which they are subjected. When establishing them, the method of stress classification recommended in modern standards (EN13445, BS 5500 etc.) must be applied. This method, also known as "stress analysis", applies to pressure vessels of all categories and consists of determining the states of mechanical stress generated by the elastic domain at all points of a vessel and verifying their admissibility on the basis of appropriate evaluation criteria. Assessment of fatigue behaviour can be done using the finite element method - FEM [5].

The material characteristics used in the fatigue analysis are shown in Figure 7, the data used for the analysis being implemented in the theoretical diagrams presented below.

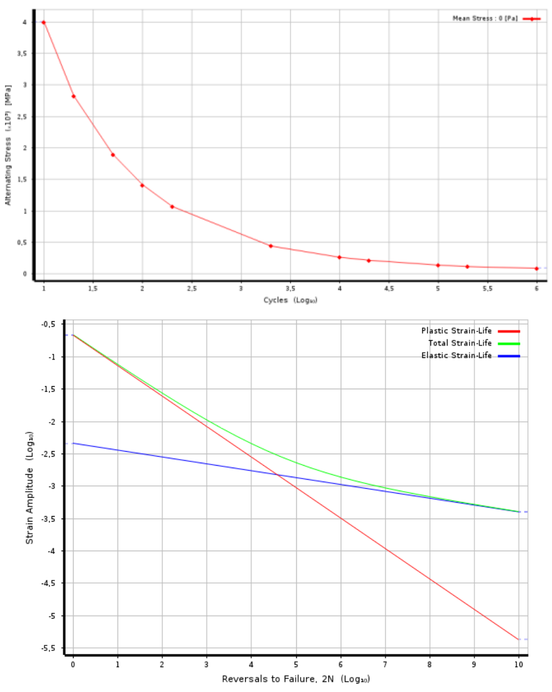


Figure 6 Wöhler theoretical diagram used in numerical analysis

The research was carried out taking into account an alternating stress at which $\sigma_{min}=-\sigma_{max}$ ($R=-1$), an alternating-symmetric stress and an undulating stress at which $\sigma_{min}= 0$ ($R= 0$), as presented in figure 7.

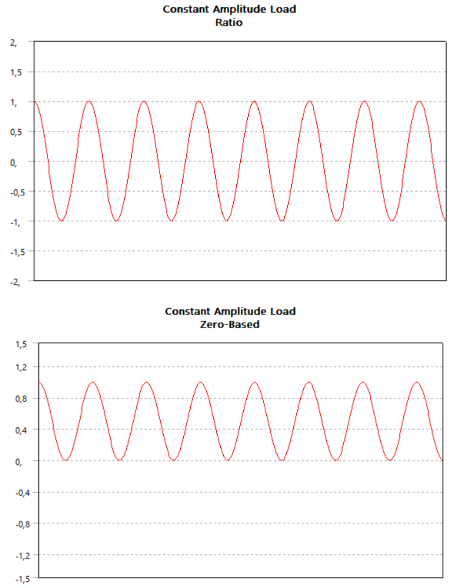


Figure 7 Loaded fatigue stress for Equipment

The analysis of the fatigue phenomenon was performed taking into account the Goodman tension amplitude correction method (see figure 8).

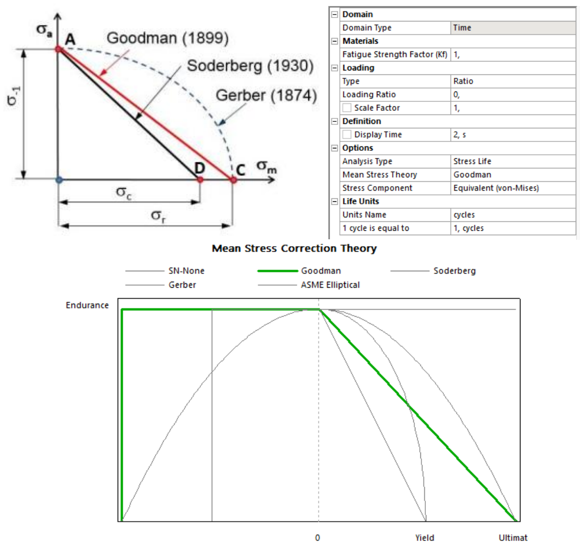
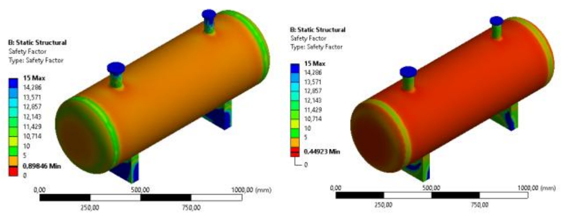


Figure 8 Loaded fatigue stress for Equipment



10 bars 20 bars

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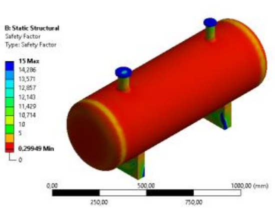
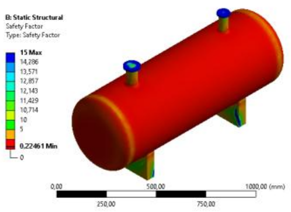
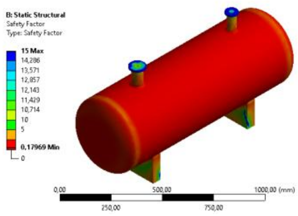


30 bars



40 bars



50 bars

Figure 9 Safety factor for R = -1

The results of the numerical analysis of the fatigue phenomenon (minimum number of cycles to failure, safety factor and diagram of stress cycles) to which the Equipment is subjected, required with internal pressure in 5 steps of 10 bar up to a maximum of 50 bar, are shown suggestively in the sequence of figure 9.

4. CONCLUSIONS

The analysis of the results obtained in the case of static stresses can be used for the optimal choice in terms of load-bearing capacity of the Equipment given that, at a pressure of 50 bar, the maximum stress has a value of ≈ 410 MPa for the jacket in the concentrator area. tension represented by welded joints and ≈ 200 MPa in the body of the jacket. The state of tensions in the bottoms / covers and supports are below the level of the tensions in the jacket.

The phenomenon of fatigue, associated with a variation in pressure (steps of 10 bar, maximum 50 bar), analysed numerically using FEM led to the following general conclusions:

- In the case of a pressure of 10 bar, the number of cycles up to a possible failure is $5{,}7 \times 10^5$ and with a safety factor of ≈ 0,9, the Equipment being able to be operated under operating conditions;

- At a pressure of 20 bar, the number of cycles up to a possible failure varies between 31947 for $R = -1$ and 1.94×10^5 for $R = 0$, and with a safety factor between 0.45 for $R = -1$ and 0.75 for $R = 0$;

- At a pressure of 30 bar, the number of cycles up to a possible fault varies between 7927 for $R = -1$ and 32272 for $R = 0$, and with a safety factor between 0.3 for $R = -1$ and 0, 5 for $R = 0$;

- At a pressure of 40 bar, the number of cycles up to a possible fault varies between 3346 for $R = -1$ and 8005 for $R = 0$, and with a safety factor between 0,2 for $R = -1$ and 0, 4 for $R = 0$;

- At a pressure of 50 bar, the number of cycles up to a possible fault varies between 1735 for $R = -1$ and 2667 for $R = 0$, and with a safety factor between 0.18 for $R = -1$ and 0, 3 for $R = 0$.

Analysing the results obtained, it can be concluded that, for the analysed pressure vessel, the alternating-symmetrical stress type stresses are more dangerous than the undulating stresses.

The numerical results obtained will be verified by making an experimental stand to determine the influence of the fatigue phenomenon on the load-bearing capacity of the respective Equipment.

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