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STRESS-STRAIN ANALYSIS OF THE EFZ–22 SAND FILTER PRESSURE VESSEL

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Abstract: *Non-heat able pressure vessels appear on a large scale in engineering practice. In the field of water supply, i.e. in water treatment plants, this is one of the basic characteristics for a closed filter container. It is therefore obvious that there is a need to devote more attention to this discipline in terms of construction, calculations of individual shapes, attachments, penetrations and the establishment of these structural devices. The company Envites (www.envites.cz) is engaged in the market from designing, calculating up to the in-house production of individual products that serve as technologies for water and sludge treatment. One of the main aspects is the separate calculation of the pressure part of the vessel. In many cases, it is a pressurized, insoluble container of larger volumes and weights. In this post, we will deal with the calculation for three pressure vessels and show the calculation procedure on one of these vessels. These two containers are designed as sand filters for industrial water filtration, which is used in industrial facilities.*

Keywords: Envites, pressure vessel, shell, numerical calculation, EFZ–22 sand filter.

1. Introduction

One of the essential raw materials for industry is water. In the field of industry, however, it is not water that is obtained only from wells or rivers. It is necessary to adjust this water so that it meets the specified input parameters for possible further use in engineering. It is thus the removal of unwanted elements, minerals and microbes from the water, which under the given conditions lead to damage or the subsequent disposal of machinery. Another possible source of water is from waste parts of the operation. The water obtained in this way needs to be technologically processed so that its parameters correspond to the input requirements for the given technology. One of the basic and suitable treatments is pressure sand filtration. The efficiency of vertical pressure filters is determined by the grain size of the sand used as filling. The grain size ranges from 0.6 mm to 1.2 mm. The average capture of impurities in the given grain size range is up to 40 μm. In order for the equipment designed in this way to work flawlessly, it is necessary to design the steel pressure vessel correctly. The basic dimensions of the pressure vessel are determined by the input parameters, which are the filter area, the working pressure and the required volume of the filtered medium. These parameters will affect both the size and the thickness of such a container.

Another influence on the pressure vessel is given by the technical location of the vessel, connection holes (necks), weight, etc. All the required input parameters were taken when the design was entered.

2. Methods

The design of the steel part of the pressure filter was carried out in two design steps. The ČSN EN 13445-3 (2018, 2021) standard was used for the design and analytical calculation, where the rough design of the pressure part of the vessel, which is the cylindrical shell and toro spheric bottoms, was carried out using empirical relationships (1–10).

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Calculation of the nominal wall thickness of the pressure vessel:

$$e = \frac{p \cdot D_i}{2 \cdot f \cdot z - p} \tag{1}$$

For the geometry given p_{max}:

$$p_{max} = \frac{2 \cdot f \cdot z \cdot e_a}{D_m} \tag{2}$$

$$e_a = (e_{min} - c) \tag{3}$$

$$e_{min} = (e_{ex} + c + e) \tag{4}$$

Toro spherical bottom:

$$e_s = \frac{p \cdot R}{2 \cdot f \cdot z - 0{,}5 \cdot p} \tag{5}$$

$$e_y = \frac{\beta \cdot p \cdot (0{,}75 \cdot R + 0{,}2 \cdot D_i}{f} \tag{6}$$

$$e_b = (0{,}75 \cdot R + 0{,}2 \cdot D_i).\left[\frac{p}{111 \cdot f_b} \cdot \left(\frac{D_i}{r}\right)^{\left(\frac{1}{1{,}5}\right)}\right] \tag{7}$$

Pressure loading of the toro spherical bottom:

$$p_s = \frac{2 \cdot f \cdot z \cdot e_a}{R + 0{,}5 \cdot e_a} \tag{8}$$

$$p_y = \frac{f \cdot e_a}{\beta \cdot (0{,}75 \cdot R + 0{,}2 \cdot D_i)} \tag{9}$$

$$p_b = 111 \cdot f_b \cdot \left(\frac{e_a}{0{,}75 \cdot R + 0{,}2 \cdot D_i}\right)^{1{,}5} \cdot \left(\frac{r}{D_i}\right)^{0{,}825} \tag{10}$$

The geometry designed in this way needs to be verified by numerical calculation. An analysis of critical or risk points on the vessel's pressure jacket was performed using the finite element method. The identified locations were the junction of the cylindrical shell with the toro spheric bottoms and the large flanges that were inserted into the pressure vessel shell and into the toro spheric bottom. These openings can affect the stiffness of the designed geometry and a numerical calculation must be performed to determine whether it is necessary to propose a local increase in the thickness of the pressure vessel in the given areas, see Fig. 1. Discretization of the container was carried out, see Fig. 2. The results of the stress-strain analysis (Ondracek, 2006) were evaluated and entered in Tab. 1. Further, the individual results of stress and deformation of critical points on the pressure vessel are shown in Figs. 3–9.

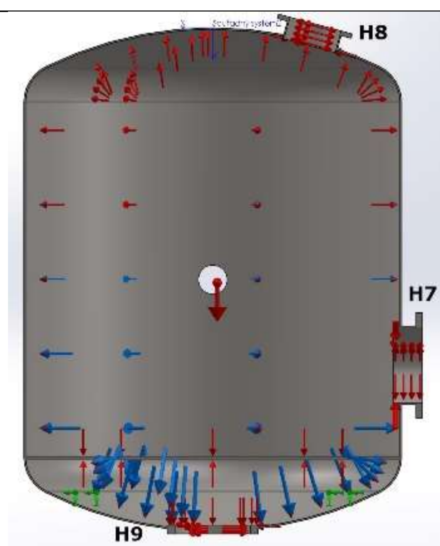

Fig. 1: Defined boundary conditions.

Fig. 2: Container discretization.

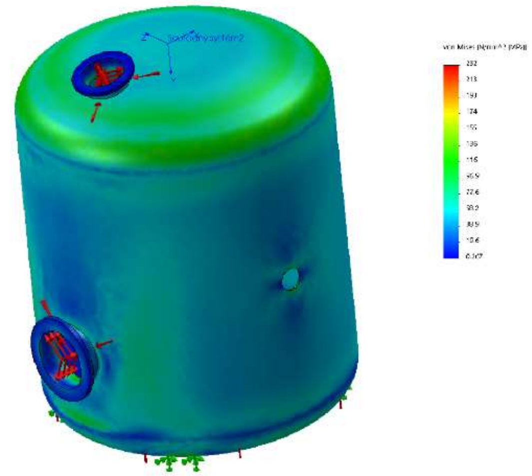

Fig. 3: Von Mises stress on the filter shell – overpressure 0.5 MPa.

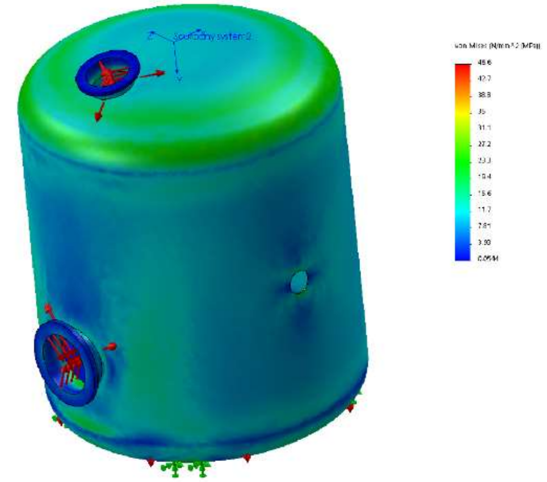

Fig. 4: Von Mises stress on the filter shell – vacuum 0.1 MPa.

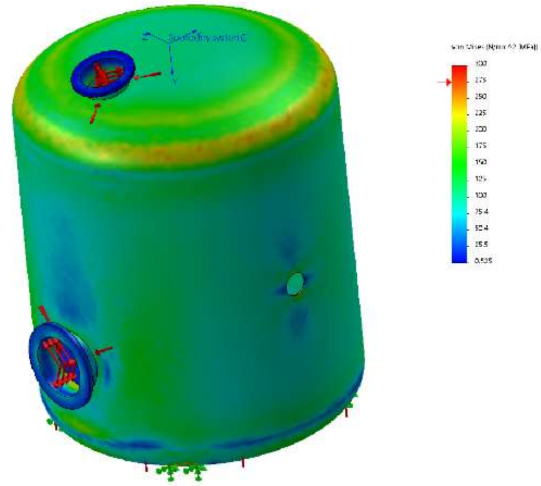

Fig. 5: Von Mises stress on the filter shell – overpressure 0.95 MPa.

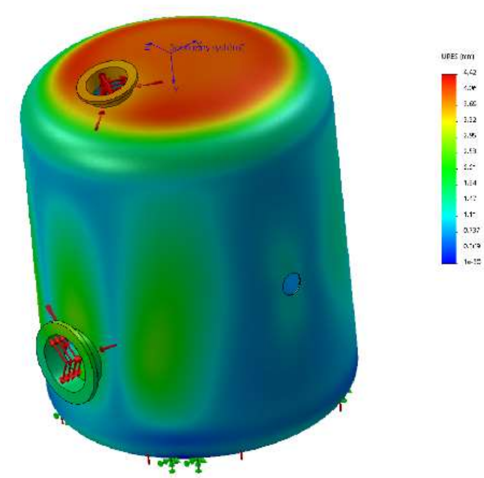

Fig. 6: Total deformation on the filter shell – overpressure 0.95 MPa.

Fig. 7: Von Mises stress around the circumference of the H7 flange – overpressure 0.95 MPa

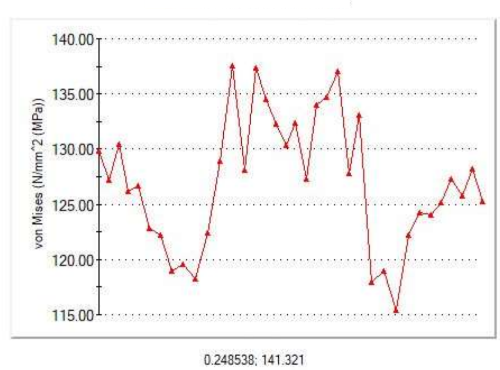

Fig. 8: Von Mises stress around the circumference of the H8 flange – overpressure 0.95 MPa.

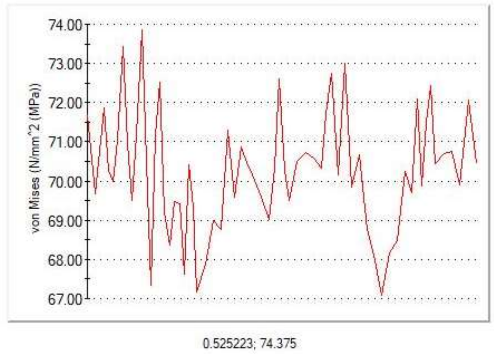

Fig. 9: Von Mises stress around the circumference of the H9 flange – overpressure 0.95 MPa.

Load states	Flange H7	Flange H8	Flange H9
	[MPa]	[MPa]	[MPa]
Overpressure 0.5 MPa	120	120	38.9
Vacuum 0.1 MPa	15.8	7.76	14.5
Test pressure 0.95 MPa	150	135.5	73.8
Test pressure 0.95 MPa + hydro. pressure + gravity	152	137	74.1

Tab. 1: Maximum von Mises stress values on flanges H7, H8, H9 (see Figs. 3–5).

3. Conclusions

From the obtained results of the numerical simulation evaluated in Tab. 1, it can be seen that at an operating overpressure of 0.5 MPa, in the critical places of the necks H7, H8 are below the determined yield limit of the material by 36 %, and at the neck H9, the value of the stress relative to the yield is 12 %. The value of the yield strength of the material used according to the documented material sheets is 340 MPa. The percentage difference is due to the construction of the H9 neck, which is reinforced with a strong ring. At full load (test pressure 0.95 MPa + hydro. pressure + own weight), the stress on the H7, H8 Throat increased by a percentage of 17 %. As can be seen from the numerical simulation results, the stress values are far below the yield point. During the design, a weld joint coefficient of 0.7 was assumed, which is the value when the weld does not pass NDT, and thus the main weld joints on the pressure vessel move in the safe area. At maximum load, the total deformation on the pressure part of the vessel is 4.27 mm, see Fig. 6. The calculated nominal thickness of the pressure part of the container was determined according to the normative results to a value of 5 mm.

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