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MODERN METHODS OF TESTING AND EVALUATION IN SCIENCE

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PREFACE

The volume of proceedings includes the selected papers and abstracts presented at the 2nd International Conference "Modern Methods of Testing and Evaluation in Science" NANT 2015. The Conference takes place every year and this year was held on 14-15th December 2015. at Faculty of Mechanical Engineering, University of Belgrade.

The main aim of this Conference is to provide a Forum for researchers and experts from different country to exchange their ideas and achieved results, but also to include young people and students in scientific research and acquaint them closer with the methods of testing and evaluation in science. Having that in mind, we put additional emphasis on active participation of students and young researchers, so the idea is that all papers will be presented by students who previously contributed to these papers with their older colleagues.

The Conference brought together the participants from institutes and universities from different countries: Croatia, Romania, USA, Bosnia and Herzegovina, Macedonia, Sweden, Libya, Iraq, Spain, Montenegro, Ukraine, Belarus, Poland, Slovenia, India and others.

The aim of the conference is, also, to connect different fields of science, because we can find many common points between different research areas, and by doing that, to open possibilities of developing new technologies or improving the old ones. Therefore, the Conference covers various topics from the following fields: mechanical science, transport and traffic engineering, material science, metallurgy, electrical engineering and other engineering areas, but all other sciences as well, including for example medical science, which uses different techniques of experimental examination and testing.

The program of the Second Conference consists of keynote lectures, oral and poster presentations.
Co-organizer of the Conference is Innovation Center of Faculty of Mechanical Engineering in Belgrade and main sponsor is BAS from Belgrade. We would like to kindly thank them for their help.

We would like to thank all authors who have contributed to this volume and also to the Scientific Committee, Organizing Committee, reviewers, speakers, chairpersons, and all the conference participants for their support for a successful scientific meeting.

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TABLE OF CONTENTS

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COMPARATIVE COMPUTATION OF CYLINDRICAL SHELLS LOADED BY EXTERNAL PRESSURE ACCORDING TO SRPS AND ASME STANDARDS

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Abstract: *This paper give area of application of SRPS M. E2.254:1991 and ASME standards for calculating thickness of cylindrical shell subjected to external pressure. It is shown the design of a vertical cylindrical pressure vessel filled with non-aggressive liquid made of austenitic steel, where it was found that by applying the aforementioned standards, approximately identical value of thickness of the cylindrical shell of the pressure vessel has been obtained.*

Key words: *pressure vessel, cylindrical shell, external pressure*

1. INTRODUCTION

Calculation of the wall thickness of pressure vessel parts are carried out in order to ensure reliable operation of the vessel, processing appliances and boilers throughout their lifecycle. The calculation of the appropriate wall thicknesses is prescribed by the relevant standards. Within each standard it is showed the area of its application, data (yield strength, tensile strength, etc., depending on the temperature) of the most used materials in the industry for manufacturing of pressure vessels, calculation procedures with the limits within they can be applied, as well as limitations of the limitations in relationships with geometrical characteristics of vessels such as length/width, etc.

Some countries have over the years developed their own standards for the calculation of the pressure vessels, whereas other countries have adopted and bought licenses of the other countries standards that apply for their own needs. The most commonly used standards in the world for the calculation of the pressure equipment are: DIN-Germany, ASME-America, GOST-Russia, BS-United Kingdom while in recent times in use EN standards for calculation thickness of pressure vessels. In our country SRPS standards are applied which practicaly represent DIN's standards, and quite often for calculations aforementioned standards is appled too, mostly for pressure equipment imported from abroad. Bearing in mind the above mentioned facts and the aim of the this article will be the implementation of the computation of cylindrical shell loaded by extrnal pressure according to SRPS standard and according to ASME standard and determining deviations which occurs in this occasion.

2. STANDARD SRPS M.E2.254 /1991

Standard SRPS M.E2.254 /1991 is applied for calculating of the thickness of cylindrical shells under external pressure. This standard has emerged by revision of JUS M.E2.254 /1981 standard. In the following text, scope of the standard and area of application for which this standard can be applied will be given, as well as values of degree of certainty for materials that are commonly used in engineering practice[1].

2.1. Subject of standards and application area of standards

This standard sets out the conditions and the method of strength calculation of straight cylindrical shells which are components of pressure vessels and pipes, which are on their surface exposed to external pressure. Standard applies to straight cylindrical shells and pipes exposed to external pressure in which the ratio of outer and inner diameter is $D_o / D_i \leq 1,2$. For pipes with $D_o \leq 200$ mm this standard applies to the ratio $D_o / D_i = 1,7$.

The cylindrical shells under internal pressure is treated as cylindrical shells exposed to the influence of external pressure. SRPS standard applies to cylindrical shells with or without reinforcement. Also this standard is applied with mandatory use of the regulations of the JUS M.E2.250 / 1981 standard.

For tanks under pressure of cast iron it is sufficient computation to internal pressure by applying the appropriate degree of certainty from Table 1. In determining the value of internal over pressure, the power and influence of external overpressure is applied too [1].

2.2 Degree of certainty

Table 1 gives values of safety coefficients S, which are used to calculate the resistance of cylindrical shell of plastic deformation. Degree of certainty at the test pressure (S') is determined by standard SRPS (JUS) M.E2.250 /1981. The safety coefficients when calculating the resistance of the elastic indentation (S_k) regardless of the material is S_k =3 and is valid for $u \leq 1,5\%$. For $u \geq 1,5\%$, $S_k = 2,25 + 0,5u$. If required pressure tests is higher then 1,3p then value S_k' must be at least $2,2 \cdot S_k / 3$.

Table 1

Material of cylindrical shell strength	The safety factor (S) relative to the yield
1. Rolled and forged steel	1,6
2. Cast steel	2
3. Nodular cast	
3.1 NL70	5
3.2 NL60	-
3.3 NL50	4
3.4 NL42	3,5
3.5 NL38	2,4
4. Aluminium and aluminum alloys	-
Material for crushing	1,6
The safety factor (S) relative to the yield strength	
5. Cast iron	6
6. Copper and its alloys including cast bronze	4

2.3. Impairment caused by cutouts

Cylindrical shells weakened by cutouts are checked according to JUS M.E2.256 / 1981 standard, whereby the pressure p is defined as internal. If necessary, the edges of the cutouts are reinforced. Cutouts in double shell when they have reinforcement on both sides are taken into account when calculating the wall thickness.

2.4. Design of cylindrical shell

Resistance to elastic bulge is calculated according to t.7.2 in relation to plastic deformation according to t.7.3 of this standard. Smaller calculated value is valid for p.

2.4.1 Calculation of resistance according to elastic bulge

The pressure at the elastic bulge is determined from the expresion:

$$p = \frac{E}{S_k}\left\{ \frac{20\cdot(s_e - C_1 - C_2)}{(n^2-1)\left[1+\left(\frac{n^2}{z}\right)^2\right]^2}\cdot\frac{1}{D_o} + \frac{80}{12\cdot(1-\nu^2)}\left[n^2 - 1 + \frac{2\cdot n^2 - 1 - \nu}{1+\left(\frac{n^2}{z}\right)}\right]\cdot\left[\frac{s_e - C_1 - C_2}{D_o}\right]^2 \right\} \qquad (1)$$

where is:

- l, mm, length of elastic bulge,
- q, mm, flattening,
- u, %, deviation from the circular shape (oval, flatness),
- D_o, mm, outer diameter of the cylindrical shell,
- D_i, mm, inner diameter of the cylindrical shell,
- S_k, degree of certainty, in relation with elastic indentation,
- n, number of bulge wave around the shell,
- ν, Poisson's ratio.

and

$$z = 0,5\cdot\pi\cdot D_o / l \qquad (2)$$

Value for n is selected so as to achieve the smallest value for the pressure p and to satisfy the requirements that n is integer, $n \ge 2$, and $n \ge z$.
The number n can be determined from the form:

$$n = 1,63\cdot\sqrt[4]{\frac{D_o^3}{l^2\cdot(s_e - C_1 - C_2)}} \qquad (3)$$

An easier way to determine the required thickness of the shell wall s when there is elastic indentation is trough using diagrams 1. This figure applies to materials whose Poisson's ratio is $\nu = 0,3$. In the case of others, significantly different values of Poisson's ratio, pressure p is calculated according to the form given in t.7.2.1 of this SRPS standard. The values of figure 1 is used when are:

$$D_o / l \le 5 \text{ and } D_o / (100(s_e - C_1 - C_2)) < 10 \qquad (4)$$

For pipes, the pressure at the elastic indentation is checked using calculation in the following form [2],[3]:

$$p = 20 \cdot E \cdot (s_e - C_1 - C_2)^3 / (S_k \cdot (1 - \nu^2) \cdot D_o^3) \tag{5}$$

2.4.2. Calculation of resistance to plastic deformation

Pressure for plastic bulge for relation $D_o / l \leq 5$ is determend from the form:

$$p = \frac{20 \cdot K}{S} \cdot \frac{s_e - C_1 - C_2}{D_o} \cdot \frac{1}{1 + \dfrac{1{,}5 \cdot u \cdot \left(1 - 0{,}2 \cdot \dfrac{D_o}{l}\right) \cdot D_o}{100 \cdot (\mathrm{s}_e - \mathrm{C}_1 - \mathrm{C}_2)}} \tag{6}$$

The wall thickness *s* can be directly determined using Figure 2 for the usual dimensions and with *u*=1,5%.

For case when $D_o / l > 5$ it is valid greater pressure for determining the allowable working pressure calculated in accordance with forms:

$$p = (20 \cdot \mathrm{K/S})(\mathrm{s}_e - \mathrm{C}_1 - \mathrm{C}_2) / \mathrm{D}_o \tag{7}$$

$$p =^2 (30 \cdot K / S)(\mathrm{s}_e - \mathrm{C}_1 - \mathrm{C}_2)^2 / l^2 \tag{8}$$

For a deviation from circular form (*u*,%), following forms are valid:
In the case of ovality,

$$u = 2 \cdot \frac{D_{i\max} - D_{i\min}}{D_{i\max} D_{i\min}} \cdot 100,\ \% \tag{9}$$

In the case of flatness of pressure vessels (see Figure 3) [1],[2]

$$u = 4 \cdot q \cdot 100 / D_o \tag{10}$$

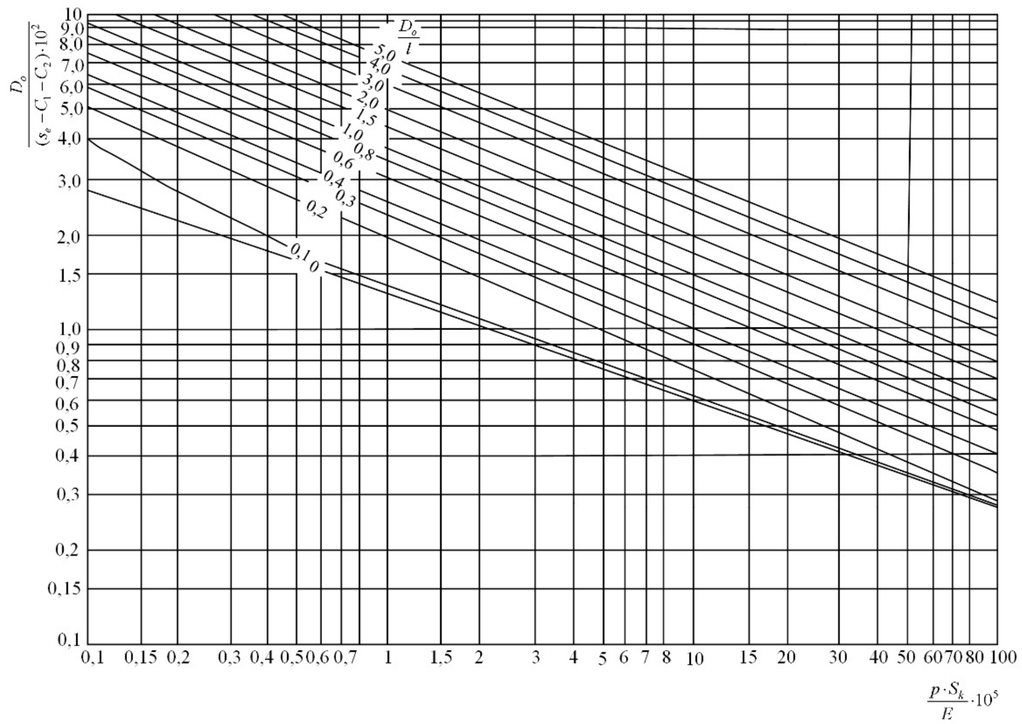

Figure 1: Diagram for determination wall thickness at elastic indentation

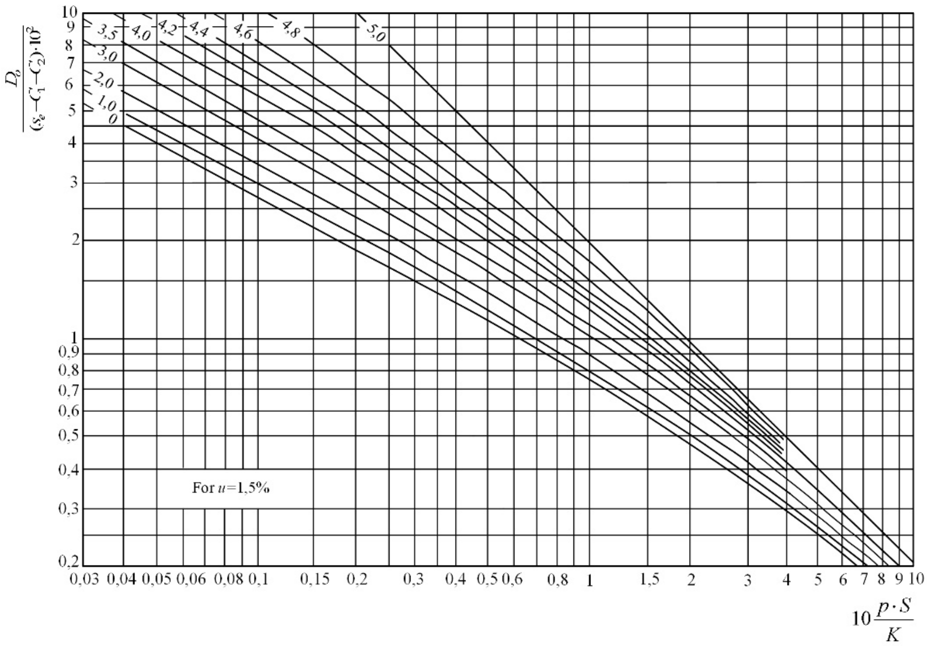

Figure 2: Diagram for determination wall thickness at plastic indentation

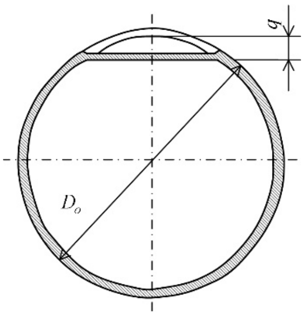

Figure 3: Flattening of pressure vessel

3. ASME (American Society Of Mechanical Engineers)

In the industry there is different devices operating under the vacuum, such as for example, vacuum condensers, destillation columns, crystallizer etc. Such vessels operate under external pressure from the atmosphere. Jacketed vessels that are heated by condensing vapor under pressure in the jacket also produce an external pressure on the vessel.Substitution of internal pressue by external pressure induces a change in the stress sign, and in the shells operating under external pressure, compression stresses arise instead of tensile.

A cylindrical shell under external pressure has an induced circumferential compressive stress equal to twice the longitudinal compressive stress. Under such a condition the shell is apt to collapse because of elastic instability caused by circumferential compressive stress.

If a thin walled shell [at $(s-C)/\mathrm{d} \leq 0,04$] is under external pressure it is possible for the wall to collapse into the shell. The shell thickness needed to prevent this is determined from the condition of strength in compression:

$$s = p \cdot d / (2 \cdot \sigma'_{all} \cdot \psi) \qquad (11)$$

where:

- p, N/m^2 the external pressure acting on the cylindrical shell,
- d, m external diameter of apparatus,

- ψ coefficient of welded joint (joint efficiency),
- σ'_{all}, N/m^2 allowed stress

$$\sigma'_{all} = \sigma'_y / 4,\ \text{N/m}^2 \tag{12}$$

Long, thin skills will buckle at stresses below the yield point of the material. The corresponding critical pressure at which buckling is a function only s/d and modulus of elsaticity, *E*. Based on strength consideration, the design thickness of a cylindrical shell, under external pressure, satisfying the following condition:

$$0{,}4 \geq p_e / \sigma_{eT} \geq 0{,}2 \tag{13}$$

where is: p_e, N/m^2 computational pressure and σ_{eT}, N/m^2, yield strenght
The needed thickness of the shell wall is calculated using the following relations:

$$s_e = 0{,}5\left(\sqrt{\frac{\sigma_{eT}\cdot\psi}{\sigma_{eT}\cdot\psi - 1{,}73\cdot p_e}} - 1\right) + C \tag{14}$$

where is:
$C = C_1 + C_2 + C_3$ - constructive supplement

- C_1, m is taken depending on the internal and external media with regard to corrosion from both side of the wall,
- C_2, m , addition to the dimensions of the material deviation,
- C_3, m , addition that takes into account small deviations from the theoretical shape

For the most cases when calculating the external pressure it is necessary that some conditions are fulfilled:

$$1 \leq l/d \leq 8 \text{ and } (p_e \cdot l)^{0,4} / (E \cdot d)^{0,4} \leq 0{,}523 \tag{15}$$

where is: *l*, m, lenght of indentation [4].
Determination of the thickness of the cylindrical shell s_e for materials with Poisson's ratio $\mu = 0{,}3$ (equivalent to $\nu = 0{,}3$ in JUS) using the following relationships:

$$s_e = 1{,}25 \cdot d \cdot (p_e \cdot l)^{0,4} / (E \cdot d)^{0,4} + C \tag{16}$$

Where the calculated length *l*, of the cylindrical part of the vessel under pressure. Calculated length of cylindrical shell is:

- the distance between the two flanges,
- length cilindric shell plus one-third eliptical part, when that shell ends with elliptical tip,
- the length of the cylindrical shell to the bottom of the vessel,
- the distance between the centers when on the shell are stiffening rings.

For shells made of a material having Poisson's ratio $\mu \approx 0{,}3$ in the case when the condition 3.6 is not fullfiled, except for layers composed of any type of plastic, it will be:

$$0{,}1 < l/d < 10 \tag{17}$$

and nominal value of wall thickness of the shell (without addition) is determined using the diagram 4 [4],[5]. Diagram 4 is used for calculation of cilindrical shell of carbon steel for temperature of shell wall greater then $t_w = 20^\circ C$ material ($t_w = t_{wall}$ temperature of shell, corresponds to calculated teperature in SRPS standard) and it is a function of *l* and critical pressure wich is calculated using relation 18:

$$p_{cr} = p_e \cdot n_s \tag{18}$$

Where is: n_s - stabilty factor , $n_s = 4$ for vertical shells and $n_s = 5$ for horizontal shell.

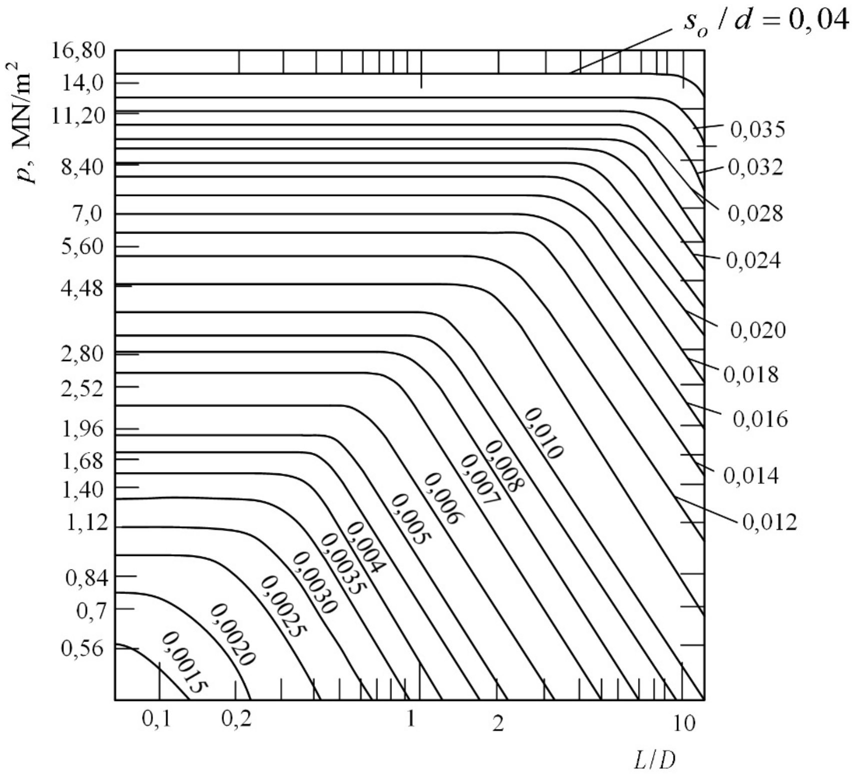

Figure 4: Lines for determination of thicknessof cilindrical shells made of carbon steel, exposed to external pressure

Figure 4 can be used for shells while $t_w > 20$ °C for different materials. Thickness s is read using values p'_{cr} wich is for steepness in the diagram is equal to:

$$p'_{cr} = p_{cr} \cdot 210 \cdot 10^9 / E, \text{ N/m}^2 \tag{19}$$

while for horizontal line parts is equal to:

$$p'_{cr} = p_{cr} \cdot 182 \cdot 10^6 / \sigma_{yt,c}, \text{ N/m}^2 \tag{20}$$

Where is $\sigma_{yt,c}$, yield strenght at the calculated temperature (yield point for compression).

Determination of values can be checked using maximal allowed pressure $\left(p_{e,dop} \geq p_e\right)$ from:

$$p_{e,all} = \frac{2 \cdot E \cdot s_e}{d \cdot n_s (z^2 - 1) \cdot \left[1 + 0,4 \cdot z^2 (l/d)^2\right]^2} + \frac{0,67 \cdot E \cdot s_e}{d^3 \cdot n_s (1 - \mu^2)} \left[(z^2 - 1) + \frac{2 \cdot z^2 - 1 - \mu}{1 + 0,4 \cdot z^2 \cdot (l/d^2)}\right], \text{ N/m}^2 \tag{21}$$

Where is: z, number of waves to fracture (correspond to n in SRPS standard).
Number of waves to fracture coresponds minimal value of allowed pressure $p_{e,all}$ which cause vesell bulge, it can be determened from diagram 5. If cross section is beetween lines s_e / d and l / d of a diagram $p_{e,all}$ can be determened using following relation 21, for greater values z were $p_{e,all}$ is always lower. In designing cilindrical shells loaded by externall pressure it is nessesery that ratio l / d choose as small as possible. For strong shells with ratio of l / d a number of waves does not be checked. For shells with standard design of ticknesses t_0 it can be determined

using diagram 5 and checked by relation 21. For determination of true wall tickness of shell next relation is used:

$$s_e = s_0 + C \tag{22}$$

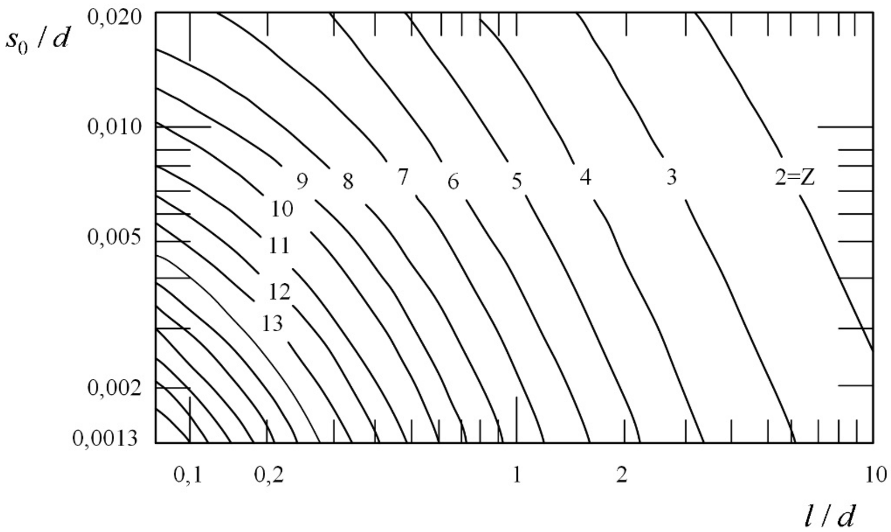

Figure 5: Lines for defining number of waves to fracture, in accordance to relation 21

Chosen wall thickness of cilindrical shell acording to condition:

$$l > 2,2\sqrt{d \cdot s_e}, \text{ m} \tag{23}$$

should be cheched for on combined stress σ, from bending and compresio or (tensile) from the next formulas:

For vertical apparatus:

$$\sigma = \frac{(d+s_e)\cdot p_e}{1,1\cdot(s_e - C)} + \frac{M}{W} + \left[\frac{(d+2\cdot s_e)\cdot p_e}{4\cdot(s_e - C)} - \frac{P \pm G}{\pi(d+s_e)(s_e - C)}\right] < \sigma_{all}, \text{ N/m}^2 \tag{24}$$

For horizontal apparatus:

$$\sigma = \frac{(d+s_e)\cdot p_e}{1,1\cdot(s_e - C)} + \frac{M}{W} + \frac{(d+2\cdot s_e)\cdot p_e}{4\cdot(s_e - C)} < \sigma_{all}, \text{ N/m}^2 \tag{25}$$

where:

- *P*, N, axial stress,
- *G*, N, gravitational force of pressure vessel filled with water,
- *M*, Nm, external momentum of flexsion,
- *W*, m^3, resistance momentum.

In the relations 24 and 25 the third member occurs only in case of operation of the external pressure on the shell, otherwise it is equal to 0. The fourth member in relation 24 when he was awarded the + sign is the case, when the force of gravity and axial stresses are in the same direction, and it is assigned character - when they are in opposite direction. Duplicators with high pressure in the liner appliances of apparatus are checked to the combined load with the relations 24 and 25 with condition 23, and also are checked for stress in the wall while hydrostatic testing with pressure p_{et}.

For vertical apparatus:

$$\sigma = \frac{(d+s_e)\cdot p_{et}}{1,1\cdot(s_e - C)} + \frac{M}{W}\left[\frac{(d+2\cdot s_e)\cdot p_e}{4\cdot(s_e - C)} + \frac{P + G}{\pi(d+s_e)\cdot(s_e - C)}\right] < \frac{\sigma_y}{1,2}, \text{ N/m}^2 \tag{26}$$

For horizontal apparatus:

$$\sigma = \frac{(d+s_e)\cdot p_{et}}{1{,}1\cdot(s_e - C)} + \frac{M}{W} + \frac{(d+2\cdot s_e)\cdot p_e}{4\cdot(s_e - C)} < \frac{\sigma_y}{1{,}2},\ \mathrm{N/m^2} \tag{27}$$

Designation in relations 24 and 25 and are valid for relations 26 and 27.
If conditiones 24 to 27 are not fullfiled it is nessesery to biger wall thickness of shell [4],[6].

4. EXAMPLE OF DEISGN

Determine wall thickness of part of apparatus (Figure 6) loaded by external pressure. Material of cilindricall shell is austenitic steel of next caracteristcs: $E_{(150\ °C)} = 185\cdot 10^9\ \mathrm{N/m^2}$, $\sigma_{y(150\ °C)} = 210\cdot 10^6\ \mathrm{N/m^2}$, $\sigma_{y(20\ °C)} = 220\cdot 10^6\ \mathrm{N/m^2}$. Medium is nonaggresive luiqid, internal medium is gas with under pressure of $p = 10\ \mathrm{N/m^2}$, external pressure is, $p_e = 0{,}6\ \mathrm{MN/m^2}$ calculated tepperatre is $t = 150\ ^\circ\mathrm{C}$. Diametar of apparatus is $d = 0{,}8\ \mathrm{m}$, height of cilindrical shell is H=2,4 m. Gravitational force of emty vessel is G=10000 N, while gravitational force of filled vessel with water is G_1=20000 N. Shell has not cutouts.

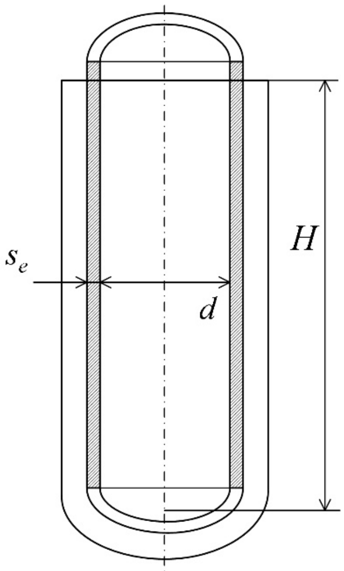

Figure 6 Vertical pressure vessel loading by external pressure

4.1 Solution according to SRPS:
Determination of wall tickness of shell due to externall pressure (JUS M.E2.254)
According to standard, weld coefficiens is not taken into account, and since shell does not have cutouts then is $\psi = 1$

Computation of elastic bulge

First anticipated value of wall tickness is $s_e = 12\ \mathrm{mm}$. Additional values are:

$z = 0{,}5\cdot\pi\cdot 800/2400 = 0{,}52$

$n = 1{,}63\cdot\sqrt[4]{800^3/(2400^2(12-0{,}5-1))} = 2{,}78$

We take $n = 3$, which is orientation value

$$p=\frac{1,85\cdot 10^5}{3}\left\{\frac{20}{(3^2-1)\left[1+\left(\frac{3}{0,52}\right)^2\right]^2}\cdot\frac{12-0,5-1}{800}+\frac{80}{12(1-0,3^2)}\left[3^2-1+\frac{2\cdot 3^2-1-0,3}{\left(\frac{3}{0,52}\right)^2-1}\right]\left[\frac{12-0,5-1}{800}\right]^3\right\}=10,42\text{ bar}$$

We are looking for number of wave indentation which will be caused by lowest pressure:
for $n=2$ then $p=25{,}21\,\text{bar}$
for $n=4$ then $p=16{,}16\,\text{bar}$
It can be seen that talas number $n=3$ coused by lowest pressure of $p=10{,}42\,\text{bar}$ and since that preesure is higer then computation pressure, we can get wall tickness of $s_e=12\,\text{mm}$

<u>Computation of plastic deformations</u>

Since

$$D_o/l=800/2400=0,33\le 5$$

applies next equation

$$p=20\cdot\frac{210}{1{,}6}\cdot\frac{12-0{,}5-1}{800}\cdot\frac{1}{1+\dfrac{1{,}5\cdot 1\cdot\left(1-0{,}2\cdot\dfrac{800}{2400}\right)\cdot 800}{100\cdot(12-0{,}5-1)}}=16{,}64\text{ bar}$$

where is:

$K=210\ \text{N/mm}^2$, calculeted yield strenght at working temperature $t=150\ ^\circ\text{C}$
$u=1\%$, deviation from the circular shape (oval, flattening)
S=1,6, safety factor.
Since calculated pressure for $s_e=12\,\text{mm}$, higer then computed, plastic deformations will not occur also condition for application of standard $D_o/D_i=800/776=1,03\le 1,2$ is satisfied.We take wall thickness as $s_e=12\,\text{mm}$.

3.2 <u>Solution according to ASME</u>

Determination of work external pressure that load outer shell is:
$p_e=p_e+0,1\cdot 10^6=(0,6+0,1)\cdot 10^6=0,7\cdot 10^6\ \text{N/m}^2$
First is checked condition 15 that
$l<H/d=2,4/0,8=3<8$ and $(p_e\cdot H)^{0,4}/(E_{150}/d)^{0,4}=0,0105<0,523$
Since conditions 15 is satisfied calculation of shell is performed according to next procedure:
$s_e=1,25\cdot d\cdot(p_e\cdot \text{H})^{0,4}/(\text{E}_{150}/\text{d})^{0,4}+C=1,25\cdot 0,8\cdot 0,0105+C=10,4\cdot 10^{-3}+C$
where:
$C=C_1+C_2+C_3=(1+0+0{,}8)\cdot 10^{-3}=1{,}8\cdot 10^{-3}\ \text{m}$
$C_1=1\cdot 10^{-3}\ \text{m}$ - addition for corrosion and wear of materials $C_2=0\,\text{m}$ addition to the dimensions of the material deviation,
$C_3=0{,}8\cdot 10^{-3}\ \text{m}$ - addition that takes into account deviations from the theoretical shape
After calculation we get value:
$s_e=10,5\cdot 10^{-3}+1,8\cdot 10^{-3}=12,3\cdot 10^{-3}\ \text{m}$
Value for thickness s_e=12 m and it is 2,5 % lower then calculated value, which is allowed.
Checking of shell on the all stresses by bending and pressure, is:

$l = H = 2,4 \text{ m} > 2,2\sqrt{d \cdot s_e} = 2,2\sqrt{0,8 \cdot 12 \cdot 10^{-3}} = 0,216 \text{ m}$

Shell is not loaded by external forces and momentum of flexsion. Load that derivates from weight of shell acts on the shell in the contra direction from externall pressure Combine stress of shell is calculated as:

$$\sigma = \frac{(0,8+12\cdot10^{-3})\cdot0,7\cdot10^{6}}{1,1\cdot(12-1,8)\cdot10^{-3}} + \left[\frac{(0,8+2\cdot12\cdot10^{-3})\cdot0,7\cdot10^{6}}{4\cdot(12-1,8)\cdot10^{-3}} - \frac{10000}{\pi\cdot(0,8+12\cdot10^{-3})\cdot(12-1,8)\cdot10^{-3}}\right] =$$

$\sigma = 64,20\cdot10^{6} \text{ N/m}^2 < \sigma_{all} = \sigma_{y(150)} / \eta_y = 210\cdot10^{6} / 1,5 = 140 \text{ N/m}^2$

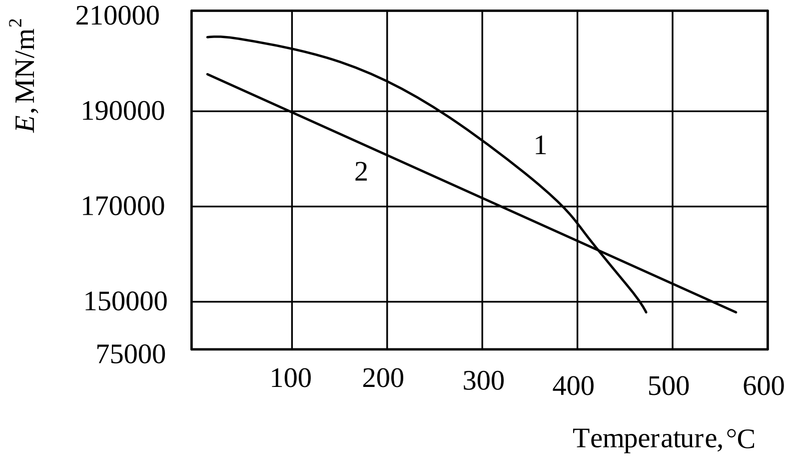

Figure 7: Modul of elasticity of steels: 1-carbon steels, 2-austenitic steels

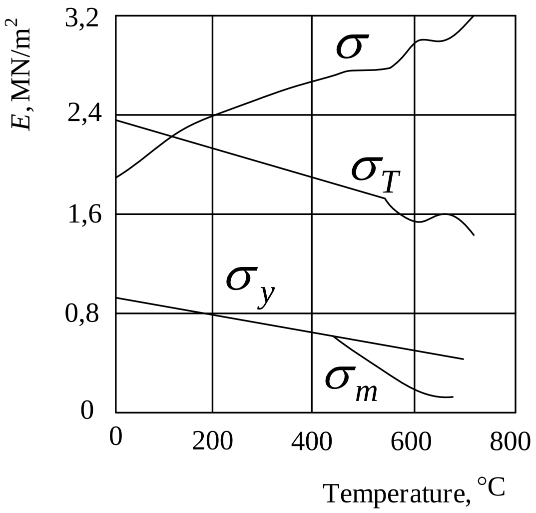

Figure 8: Mechanical properties of austenitic steels depending of temperature

Then shell is checked on the combine stress from flexsion and outer pressure coused by water testing. Hidrostatic pressure has small influence on the all load when computing vesell, and it is:is checked using relations:

$p_{ef} \approx 10\cdot\rho_{ef}\cdot H_{ef} = 10\cdot1\cdot10^{3}\cdot2,4 = 0,024\cdot10^{6} \text{ N/m}^2$

Where are H_{ef}, m height of liquid and ρ_{ef}, kg/m^3 density of liquid.

Determination of test pressure is performed according to relation:

$p_{et} = p_h + p_e = (0,7+0,3+0,024)\cdot10^{6} = 1,024\cdot10^{6} \text{ N/m}^2$

Combine stress in shell with hidrostatic pressure is defined according to 26

$$\sigma = \frac{(0,8+12\cdot 10^{-3})\cdot 1,024\cdot 10^{6}}{1,1\cdot(12-1,8)\cdot 10^{-3}} + \left[\frac{(0,8+2\cdot 12\cdot 10^{-3})\cdot 1,024\cdot 10^{6}}{4\cdot(12-1,8)\cdot 10^{-3}} - \frac{20000}{\pi(0,8+12\cdot 10^{-3})\cdot(12-1,8)\cdot 10^{-3}}\right] =$$

$$\sigma = 93,80\cdot 10^{6}\ \text{N/m}^2 < \sigma_{all} = \sigma_{y(20)}/1,2 = 220\cdot 10^{6}/1,2 = 183\cdot 10^{6}\ \text{N/m}^2$$

Table 3.1: Reviewing of taken values for wall tickness of shell:

Calculation of cilindrical shell according to SRPS standard	Calculation of cilindrical shell according to ASME standard
Wall thickness is $s = 12$ mm	Wall thickness is $t = 12$ mm

5. CONCLUSION

In the paper exposed the effective computation procedure for designing of cylindrical shells under according to ASME and SRPS standard. As can be noted for the calculation are used similar mathematical relations and relations which at appropriate standards have to be fulfilled including primarily ratio of diameter and length of apparatus. The basic procedure for calculation at SRPS is based on iterative methodsin which the assumed wall thickness of cylindrical shell enters in calculation and checks is it possible predicts loading. If it is not fulfilled in calculation is entered with novel assumed value of wall thickness and calculation is conducted while the assumed wall thickness does not satisfy calculation in terms of envisaged pressure of apparatus. On the other hand at ASME standards before of process design are checked geometric conditions of pressure vessels and then according to operating parameters determines stress in material of apparatus. If the design stress greater than allowed it is assumed the novel wall thickness and calculation is conducted while mentioned conditions can not be fulfilled. Such as it can be see value for wall thickness obtained by calculation by both standards are approximately equal which leads to conclusion thast both standards are based on the same postulates of mechanics of materials with the only difference in the way of the solving problems.

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