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Prediction of the strength and fracture of the fuel storage tank

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1. Introduction

Despite considerable advance of science and technologies in recent years, oil storage tanks are still among the most dangerous objects. The reason of this is high combustibility of oil products, complicated control of weld length, irregular shell joints in assembling as well as soil sedimentation, complex loading of the structure at the point of shell to bottom joint where the control of the weld is difficult. The analysis carried out in Russia showed that wear of vertical tanks amounts to 60-80% [1]. The statistical analysis of the recent 30 years showed that the main cause of tanks accidents is brittle cracking (63.1%) followed by explosions and fire (12.4%). Normally, a crack in the structure occurs in the place of stress concentration where stress level is the highest [1-4]. The analysis of scientific literature shows that location of maximal stresses occurs in the tank's shell, at the bottom, where hydrostatic pressure is the highest [5-7]. In Lithuania, Subaciaus fuel base, tanks over 40-45 years old are still in operation.

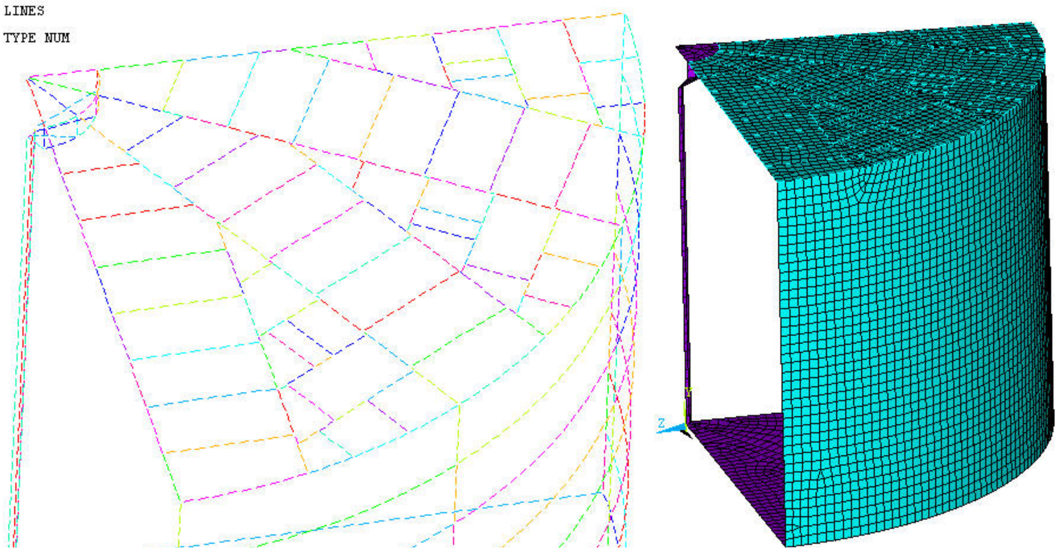
The aim of this work is modelling of stress inside the diesel fuel storage tank structure and determination of critical crack size for stability of postulated defect. In addition, with the development of computer modelling and corresponding numerical methods in modern engineering and science, the finite element method (FEM) was also employed in the structural analysis of storage tank.

2. Numerical model of the tank

Diesel fuel tanks used in this fuel base are classified as fixed vertical cylindrical low inside pressure tanks in the roof of which pressure/vacuum check valves are installed to provide the release of saturated vapour formed in the product kept inside, into the atmosphere or the air inlet if the vapour inside is condensed and pressure is reduced. Since tanks are designed for long storage of the product and it is but rarely loaded, such maintenance creates favorable conditions for intensive corrosion of inner lower part of the walls and the bottom as well as causes danger due to possible occurrence of micro fractures in higher loading zones.

The roof of the tank consists of a steel shell with thickness of $d = 7$ mm which is welded to the steel framework of the roof by intermittent chain (Fig. 1, a). The framework of the roof structure consists of bearing girders located in radial direction, which are welded together by cross joists. The roof frame in the centre of the tank is supported by a column.

The bottom of the tank consists of a ring plate of $t = 10$ mm thickness to which the first steel band cylindrical wall (shell) of $t_1 = 10$ mm thickness and the bottom plate of the tank are welded. The side walls of the tank shell are welded from separate steel sheets arranged in regular quadrangular form. The sheets are welded together and placed in horizontal bands the thickness of which decreases in upwards direction (Fig. 2).



a b

Fig. 1 Roof structure (a) and the numerical model (b) of the diesel fuel storage tank

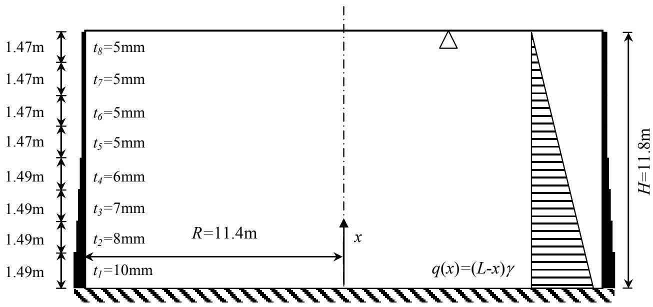


Fig. 2 Stepped wall of a tank under water pressure

Table 1

Main technical data of diesel storage tank

Parameter	Value
Inside diameter, m	22.8
Height of cylindrical part, m	11.8
Height of conical roof, m	0,63
Thickness of cylindrical shell t, mm	from 10 to 5
Thickness of roof shell, mm	7
Thickness of bottom shell, mm	10
Volume, m^3 (for diesel fuel surface height $h = 11.4$ m)	5000

The wall thickness from t_1 to t_8 of the 1st-8th shell courses from bottom to top are 10, 8, 7, 6, 5, 5, 5, 5 mm, respectively. The course heights are expressed as $l = l_1 = l_2 = l_3 = l_4 = 1.49$ m and $l_5 = l_6 = l_7 = l_8 = 1.47$ m. The main technical data of storage tank is provided in Table 1.

The storage tank is loaded during operation by:

- self-weight of the structure,
- hydrostatic pressure of the contained liquid and by pressure (or under pressure) above the liquid surface in accordance with working conditions defined by operating manuals,
- climatic loading (snow);
- temperature;
- ground settlement.

All these loads were evaluated in the numerical model of the tank. It is necessary to note that during operation of the storage tank the pressure of 2.03 kPa produced by steam above the liquid surface and -1.013 kPa for vacuum in winter time is ensured by the control system of the tank. The calculated snow load value in Panevezys region is 1.2 kN/m^2 [8]. The side shell of the tank is made of St.3 steel the mechanical properties of which are represented in Table 2.

Table 2

Mechanical properties of St.3 steel

Parameter	Value
Young's modulus E, GPa	200
Poisson's ratio ν	0.35
Yield stress, σ_Y, MPa	260
Ultimate strength σ_U, MPa	380
Fracture toughness K_{IC}, MPa×m$^{3/2}$	50

The fracture toughness values K_{mat} for steels were accepted according to literature [9] data.

All finite element results presented here were obtained using the general-purpose finite element program ANSYS [10]. Quarter of the tank has been modeled due to the symmetry (see Fig. 1, a and b). The cylindrical shell, bottom, roof and the supporting column of the tank were modeled by using SHELL63 type FE and assigning relevant thickness of the wall to these structural elements. Roof framework was modeled by using BEAM44 FE where each beam was assigned a beam of relevant profile [10]. In the model, ground settlement was taken into consideration by assuming that the tank rests on very stiff foundation the foundation stiffness of which is 147 kN/m^3 [11]. The ground settlement was taken into account because it can be affect the strength of fuel storage tank.

3. Stress analysis of the cylindrical shell

The highest probability is that crack will appear in the side cylindrical wall of the tank, i.e. in the area of highest loading. So, the shell of cylindrical form was chosen. Uniform gas pressure and hydrostatic liquid pressure are commonly considered in the design process [12, 13]. The stresses can be calculated by membrane theory. Due to boundary effects, geometric and any other discontinuities, which are often present in reality, one must resort to the bending theory, which results in a fourth-order differential governing equation.

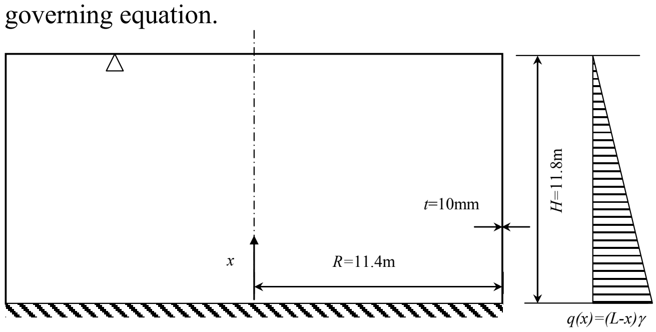


Fig. 3 Clamped semiinfinite cylinder under water pressure

The tank should experience a water-filling test before going into operation. The shell water-filling height is $h = 11.8$ m. For large storage tanks, the edge effect is usually restricted within the first and second courses. For the upper shell course, the bending moment and shear force are really negligible and thus assumed as zero [5]. Therefore, in this work were considered the 1st-8th course which has the different height as shown in Fig. 2. The compatibility conditions at the nodes with different wall thickness are that the upper and lower courses have the same bending moment and shear force. The boundary condition at the top of the 8th course is that its bending moment and shear force are zero. But firstly a cylindrical shell with uniform wall thickness was considered as shown in Fig. 3. After that a cylindrical shell with stepped wall thickness was considered as shown in Fig. 2.

The governing equation is well know equation which defines the variation of pressure loading due to liquid [5-7]

$$D\frac{d^4w}{dx^4} + 4D\beta^4 w = -\gamma g(h-x) \tag{1}$$

where γ is density of the liquid, h is height of the storage tank, β is the constant which can be calculated $\beta = \left(\frac{3(1-\nu^2)}{R^2 t^2} \right)^{1/4}$; t is uniform wall thickness; R is radius of cylindrical storage tank; ν is the Poisson ratio; D is flextural rigidity and can be calculated $D = \frac{Eh^3}{12(1-\nu^2)}$; E is Young's modulus.

The particular integral part of the solution is $-\gamma g(h-x)/4D\beta^4$ and the complete solution is

$$w = exp(\beta x)(C_1 \cos \beta x + C_2 \sin \beta x) + exp(-\beta x)(C_3 \cos \beta x + C_4 \sin \beta x) - \frac{\gamma g(h-x)}{4D\beta^4} \quad (2)$$

The boundary conditions are as follows.

1. The height of the cylinder can be regarded as 'infinite', so that $M \to 0$ and $w \to 0$, giving $C_1 = C_2 = 0$.

2. At $x = 0$, $w = 0$ and $dw/dx = 0$; hence $C_3 = \frac{\gamma g h}{4D\beta^4}$ and $C_4 = \frac{\gamma g}{4D\beta^4}\left(h - \frac{1}{\beta}\right)$.

Putting these values in Eq. (2) gives the deflection curve

$$w(x) = \frac{\gamma g h}{4D\beta^4}\left[1 - \frac{x}{h} - exp^{-\beta x} \cos(\beta x) - exp^{-\beta x}\left(1 - \frac{1}{\beta h}\right) \sin(\beta x)\right] \quad (3)$$

But $M = -D(d^2 w/dx^2)$, so that differentiating Eq. (3) twice and substituting gives

$$M(x) = \frac{\rho g h}{2\beta^2}\left[-exp^{-\beta x} \sin(\beta x) + \left(1 - \frac{1}{\beta h}\right) exp^{-\beta x} \cos(\beta x)\right] \quad (4)$$

The resultant axial stresses can now be calculated

$$\sigma_{axial}(x) = \frac{6M(x)}{t^2} \quad (5)$$

The bending of the strip also produces circumferential stresses: (a) since the strip displacement, *w*, varies only in the axial direction, the strip is loaded in cylindrical bending giving circumferential stresses of $\pm 6wM/t^2$; (b) stresses due to shortening of the circumference of $-Ew/R$. The resultant hoop stresses can be calculated

$$\sigma_{hoop}(x) = \frac{Q_0(x) exp^{-\beta \cdot x}}{\beta t^2}\left[6w \sin(\beta x) - \frac{12(1-\nu^2)}{2\beta^2 t R} \cos(\beta x)\right] + \frac{\gamma g(h-x)R}{t} \quad (6)$$

where $Q_0(x)$ is the shear force per unit length and can be calculated

$$Q_0(x) = \frac{\gamma g(h-x)}{8\beta} \quad (7)$$

The results of stresses distribution analysis in the shell are presented in Figs. 4 and 5. As mentioned above, distribution of hoop and axial stresses in the tank wall with uniform thickness was analyzed first as the tank was under water hydrostatic pressure (Fig. 3). It was assumed that water level in the tank reaches 11.8 m. The results of analytical calculations according to formulas (1)-(7) are presented in Fig. 4. For the comparison of results, numerical model of the tank's shell was created. The results (curves *3* and *4*) of numerical modeling are also presented in Fig. 4. We can see that the results of numerical and analytical calculations are different only in the place of the shell junction to the bottom. Further, stress distribution in the side cylindrical wall with stepped thickness was analyzed (the corresponding model of the tank is presented in Fig. 2). For this, in the numerical model, such wall thickness was assumed in the shell course which should correspond to the wall thickness distribution of the diesel fuel tank with the capacity of 5000 m^3. The results are presented in Fig. 5. In comparing the distribution of hoop and axial stresses in the models with uniform and stepped thickness of the wall, we can see that relevant stresses in the zone of joint shell to the bottom fixing wall coincide. The results show that the wall thickness makes no influence on the axial stresses, and for hoop stresses, the wall thickness is of utmost importance, because in the stepped wall these stresses are considerably higher. Here, we can make a conclusion that edge effect in the cylindrical shell with stepped thickness manifests itself at the distance of 0.59 m from bottom, where $\sigma_{axial} = 0$.

After estimation of the stress analysis of the cylindrical shell the further modelling of the storage tank using the finite elements of ANSYS program was fulfilled evaluating all the loads mentioned in Chapter 2. The storage tank was modelled for winter time in order to imitate the worst environmental situation and weather conditions. It has been assumed that vacuum $p = -1.03$ kPa was inside the storage tank, and the outside temperature was -

20°C. The storage tank was filled with diesel fuel by 80%, i.e. up to 11.4 m level.

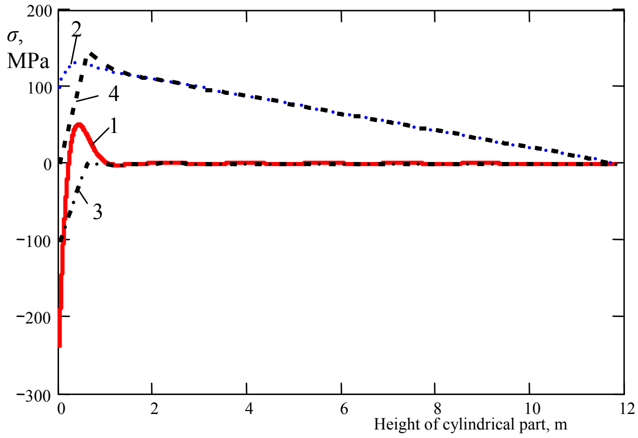


Fig. 4 Distribution of hoop σ_{hoop} (*1, 3* - curves) and axial σ_{axial} (*2, 4* - curves) stresses in the tank's wall with uniform wall thickness: *1, 2* - stresses are calculated according to analytical formulas; *3, 4* - stresses are calculated FE

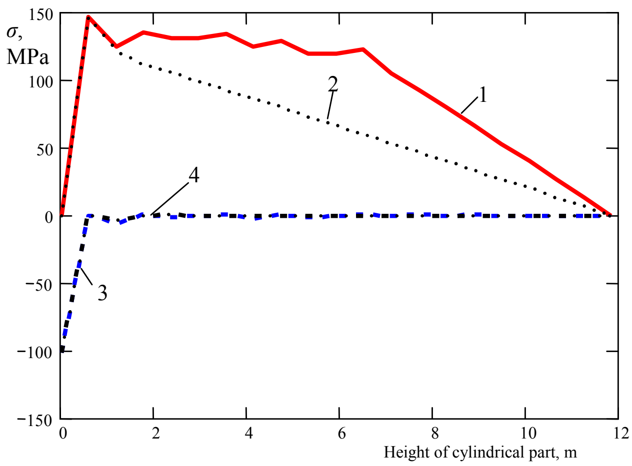


Fig. 5 Distribution of hoop σ_{hoop} (*1, 2* - curves) and axial σ_{axial} (*3, 4* - curves) stresses in the tank's wall: *1, 3* - curves for the tank with stepped wall thickness, *2, 4* - curves for the tank with uniform wall thickness

The stress distribution in the diesel fuel storage tank has been also analyzed. The calculations showed that the maximum equivalent stresses $\sigma_{ekv} = 141$ MPa occur in the area where roof frameworks connected with the column and in the first shell course. Thus the safety factor is rather great because it equals to 1.84.

The distribution of the hoop stresses σ_{hoop} and axial stresses σ_{axial} in the shell wall of the storage tank has been also analyzed. These results are given in Fig. 6. They show that maximum stresses $\sigma_{hoop} = 141$ MPa occur in the first shell course at approximately 0.59 m distance from the area where the wall is connected with the bottom. The maximum stresses $\sigma_{axial} = 40$ MPa also occur in the first shell course at the area where the shell to the bottom is connected (Fig. 6), at approximately 0.59 m distance they become about equal to 0. Thus the periphery (circumference) effect occurs at this distance.

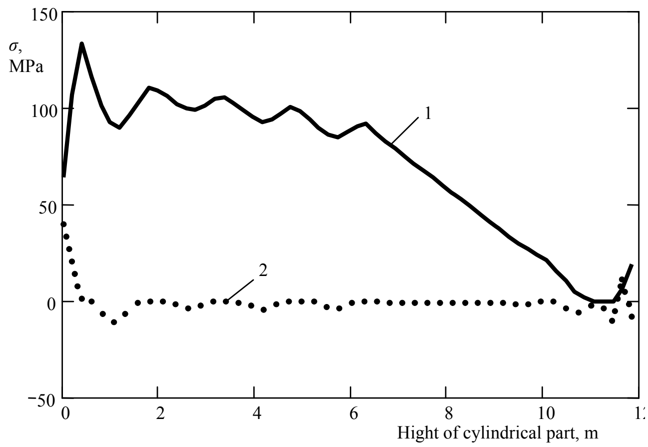


Fig. 6 Distribution of the hoop σ_{hoop} (curve *1*) and axial σ_{axial} (curve *2*) stresses in the side wall of the diesel fuel storage tank. ($t = -20$ °C, $p = -1.03$ kPa and the tank is filled up with diesel fuel)

4. Crack modelling and calculations in the cylindrical shell

One of the factors determining the brittle fracture of the tank wall is the temperature. Low environmental temperature has an impact on the metal strength. It is known that when a metal with no incisions and cracks is tested in low temperature its strength does not minimize, and when the metal is tested with stress concentrator in low temperature it may be brittle and breakable. But as the tank exploitation practice shows the low temperature and the increased probability of the brittle fracture is not the prerequisite for the structure collapse [1].

Considering the results of the stress analysis it has been concluded that the crack in the side wall of the tank may occur in axial direction in the first shell course where the greatest hoop stresses occur.

Thus the various lengths leaky tank crack through the wall thickness has been modeled. In order to calculate the critical crack length the stress intensity factor K_I at the crack top should be estimated first of all. The critical crack length was defined using the world-wide adopted R6 methodology [14].

The tank side surface has great curvature beam if compared with the wall thickness ($R = 11.4$ m, $t = 10$ mm) thus when a thin strip with the crack has been cut from the cylinder surface it can be further modeled as the plate. Such simplification has been also accepted because FE in program ANSYS shell type elements do not calculate the stress intensity factor. If the solid type elements were used in the model, parameter K_I can be calculated, but the calculations become very long, as lots of computer time will be used. The plate has been modeled using PLANE 82 type FE that has the possibility to calculate the stress intensity factor K_I [10].

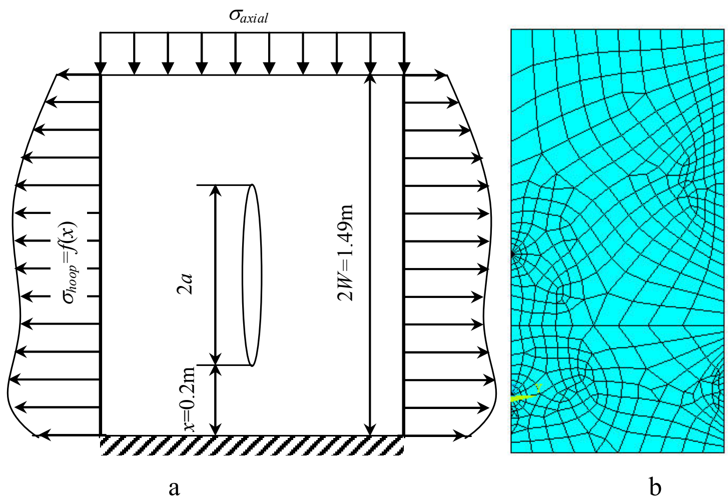


Fig. 7 Model of the plate with a crack (a) and the finite element model (b)

After such geometrical simplification the fracture mechanics parameters can be calculated and employing the classical model of the finite length of the plate with the central crack [14-16] according to the formula

$$K_I = \sqrt{sec\frac{\pi a}{2W}} x\sigma_{hoop}\sqrt{\pi a} \tag{8}$$

where a is half crack length, $2W$ is the plate length, in this case it coincides with the height of the circular course, σ_{hoop} is the mean value uniform hoop stress.

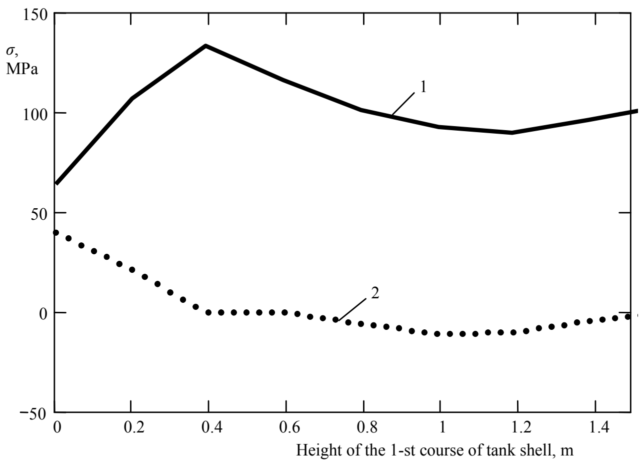


Fig. 8 Hoop σ_{hoop} (curve *1*) and axial σ_{axial} (curve *2*) stresses distribution in the first course of tank shell

The crack in the plate has been oriented in axial direction from the point of view of the storage tank. One side of the plate has been rigidly restrained thus modeling the cylindrical shell connection with the bottom (see Fig. 7, a), and the hoop stress $\sigma_{hoop} = f(x)$ has been added in the direction perpendicular to the crack (see Fig. 8). According to the analysis results this distribution corresponds to the distribution of the tank hoop stresses in the first shell course. Half of the plate with one side rigidly restrained has been modeled due to the symmetry (see Fig. 7, b), and this corresponds to the tank side wall connection with the bottom. In linear elastic problems, it is known that the displacements near the crack tip (or crack front) vary as $\sqrt{r}$ where r is the distance from the crack tip. The stresses and strains are singular at the crack tip, varying as $1/\sqrt{r}$. To resolve the singularity in strain, the singular elements [10] were used at the crack tip with the midside nodes placed at the quarter points (see Fig. 7, b). The element size closest to the crack tip was 0.001 of the crack length, the radius ratio was 1.5 and number of elements in the circumferential direction was 8.

As it has been mentioned above the critical crack length has been determined using limiting R6 Option 1 method curve $f(L_r)$ [2]

$$f(L_r) = \frac{0.3 + 0.7\,exp\left(-0.6L_r^6\right)}{\sqrt{1+0.5L_r^2}} \tag{9}$$

where $f(L_r) = K_r$ is the proximity to fracture; L_r is the proximity to plastic collapse, defined for the ideally plastic material (without hardening).

Proximity to plastic collapse L_r can be calculated

$$L_r = \frac{\sigma_{ref}}{\sigma_Y} \tag{10}$$

where σ_Y is the yield stress; σ_{ref} is the reference stress.

The reference stress is a single quantity which characterizes the geometry and loading of the section. The reference stress for centre-cracked panel can be calculated [2]

$$\sigma_{ref} = \frac{\sigma_{hoop}}{1 - a/W} \tag{11}$$

The maximum value of the proximity to plastic collapse L_r can be calculated

$$L_r^{max} = \frac{\sigma_f}{\sigma_Y} \tag{12}$$

where $\sigma_f = \dfrac{\sigma_Y + \sigma_U}{2}$ is flow stress according R6 methodology [2].

Proximity to fracture can be K_r calculated according this formula [2, 5]

$$K_r = \frac{K_1}{K_{IC}} \tag{13}$$

where K_{IC} is fracture toughness.

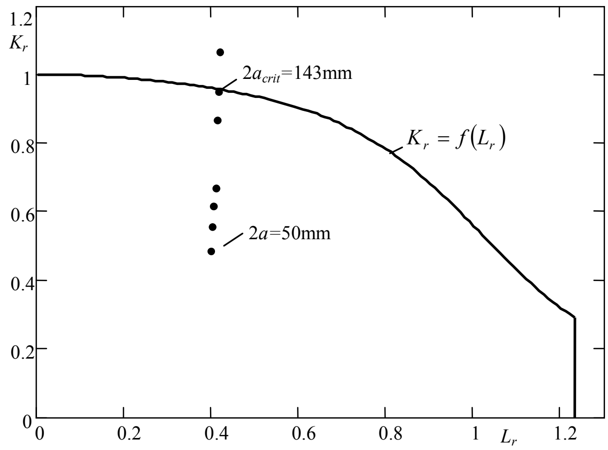


Fig. 9 Determination of critical crack size according to methodology R6

The results of the critical crack length determination according to methodology R6 are shown in Fig. 9. It is obvious that the critical crack length is equal to $2a_{crit} = 143$ mm, i.e., when the crack growth curve intersects the limit curve R6. When the crack length exceeds the critical length the catastrophic crack growth occurs and the tank will collapse.

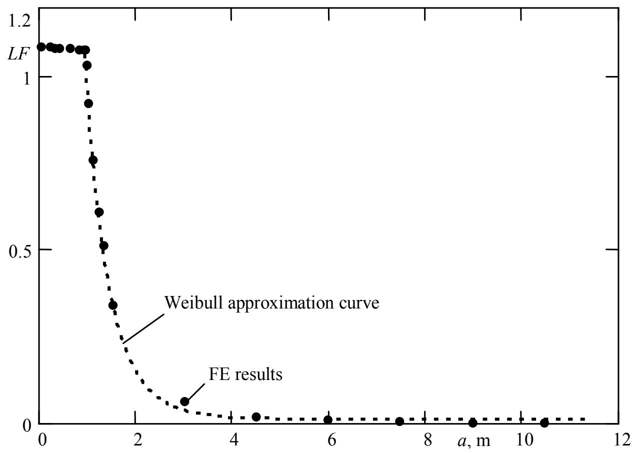


Fig. 10 Determination of critical crack size according to stability analysis which leads to tank collapse

Besides this work it had to be shown whether the increase of the crack length can be lead to the loose of the tank stability before the crack length reaches its critical length calculated by the method R6. For this purpose the numerical model of the tank with the axial crack has been created. Model has estimated all the loads, and as a result the load factor has been determined. If the load factor is $LF \geq 1.0$, it means that the tank sustains the stability, and if is $LF < 1.0$, then the tank looses its stability. Calculation results are given in Fig.10 (shown as dots). Results were obtained using Table Curve 2D software. It has been determined that the loss of the tank local stability subject to crack length in the cylindrical shell corresponds to the Weibull Cumulative distribution, and correlation coefficient is equal to $r^2 = 0.9996$

$$LF = a_1 + b\left[1 - exp\left[-\left(\frac{a + d\,ln\,2^{1/e} - c}{d}\right)^e\right]\right] \tag{14}$$

where $a_1 = 1.0825$, $b = -1.0705$, $c = 1.2583$, $d = 0.4828$ and $e = 0.9139$ are coefficients.

Calculations made with the ANSYS software showed that when the crack length is $2a < 1000$ mm the tank global stability would be lost in the roof structure first of all, and when the crack length is $2a \geq 1000$ mm the local stability is lost at the crack. In summary it is necessary remark that the critical crack length in the tank wall is 143 mm, from which the catastrophic crack growth begins and the tank will loss of the local stability when the crack length achieves 1000 mm.

5. Conclusions

1. The comparison of the circular and axial stress distributions in the models with constant and stepped wall thickness shows that the wall thickness has no impact on the axial stresses, but it has significant impact on the hoop stresses as these stresses are much greater in the stepped wall.

2. Circumferential effect in the cylindrical shell of the stepped thickness occurs approximately at the distance of 0.59 m from the area where the tank bottom is joined to the cylindrical shell. At this distance the axial stress is $\sigma_{axial} = 0$.

3. Estimation of the strength analysis of the whole tank revealed that the maximum equivalent stresses $\sigma_{ekv} = 141$ MPa occur in the area where roof girders connect with the column and the first shell course of the side wall, and the safety margin is 1.84.

4. Utilization of the method R6 disclosed that the critical crack length in the tank wall is 143 mm, from which the catastrophic crack growth begins.

5. Stability analysis showed that when the crack reaches the length of up to 1000 mm the local stability of the tank is loosed.

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KURO TALPYKLOS STIPRUMO IR IRIMO PROGNOZAVIMAS

R e z i u m ė

Darbas skirtas naudotų naftos produktų talpyklų mechaninei būklei įvertinti. Išanalizuotas talpyklos kevalo cilindrinės dalies įtempių būvis, nustatyti kritiniai menamo defekto matmenys. Nustatyti ir analitiniame modelyje įvertinti faktiniai talpyklų parametrai. Naudojant analitinį bei skaitinį metodus atlikta talpyklos sienelės šalia dugno įtempių būvio analizė. Toliau buvo modeliuojamas kiaurinis plyšys sienelėje. Keičiant plyšio ilgį talpyklos šoninėje sienelėje, skaitiniu bei R6 metodu apskaičiuoti kritiniai postuluojamo plyšio parametrai. Nustatyta, kad kritinis plyšio ilgis 10 mm storio sienelei yra 143 mm. Atlikus stabilumo analizę, apskaičiuota, kad talpykla vietomis praranda stabilumą esant 1000 mm plyšiui.

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PREDICTION OF THE STRENGTH AND FRACTURE OF THE FUEL STORAGE TANK

S u m m a r y

The work is designed to evaluate mechanical condition of used petroleum product (diesel) tank by analyzing stress state of the tank shell lower courses and defining critical crack sizes for postulated defect. The actual parameters of used tank are determined and adopted in the analytical model. By using analytical and numerical calculation stresses state of the shell near the bottom has been determined. By varying through wall crack length in the shell wall the critical crack sizes were defined by using R6 method. Calculations have shown that the critical crack size is 143 mm for the 10 mm shell thickness. Stability analysis showed that when the crack reaches the length of up to 1000 mm the local stability of the tank is loosed.

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ПРОГНОЗИРОВАНИЕ ПРОЧНОСТИ И РАЗРУШЕНИЯ РЕЗЕРВУАРА ХРАНЕНИЯ ТОПЛИВА

Р е з ю м е

Работа предназначена для оценки механического ресурса эксплуатируемых резервуаров нефтяных продуктов (дизельного топлива). Проведен аналитический и численный анализ напряженного состояния в цилиндрической части стенки резервуара с целью определения критических параметров дефекта. При аналитическом моделировании и вычислениях определены и использованы фактические параметры резервуаров. На основе аналитических и численных решений и выполнен анализ напряженного состояния и при изменении длины сквозной трещины в стенке резервуара по методу R6 определены критические размеры трещины. Установлено, что при 10 мм номинальной толщине стенки критический размер трещины составляет 143 мм. Выполненный анализ по потере стабильности показал, что при длине трещины 1000 мм резервуар локально теряет свою стабильность.

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