

STRUCTURE DESIGN OF CARGO TANKS IN RIVER LIQUEFIED GAS CARRIERS

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1. Introduction

The growth of gas sea- and river- transport is a result of an increase in energy consumption. Gas is transported in liquefied state due to reduction of volume, by means of special ships called Liquefied Gas Carriers. Depending on the cargo type, Liquefied Natural Gas Carriers (LNG) and Liquefied Petroleum Gas Carriers (LPG) are distinguished.

Liquid natural gas is insulated at cryogenic temperature and slightly pressurised above atmospheric pressure. Liquid petroleum gases are transported in one of three conditions, [Safe Havens 2003]:

- fully refrigerated at slightly above atmospheric pressure,
- refrigerated, semi-pressurised below ambient temperature and above atmospheric pressure,
- fully pressurised at ambient temperature.

For liquefied gas transport different cargo tanks are used: integral tanks, membrane tanks, semi-membrane tanks and independent tanks. The design vapour pressure for the integral, membrane and semi-membrane tanks is limited to 0.25 bar. If the scantlings are increased then the pressure may be increased accordingly up to 0.7 bar.

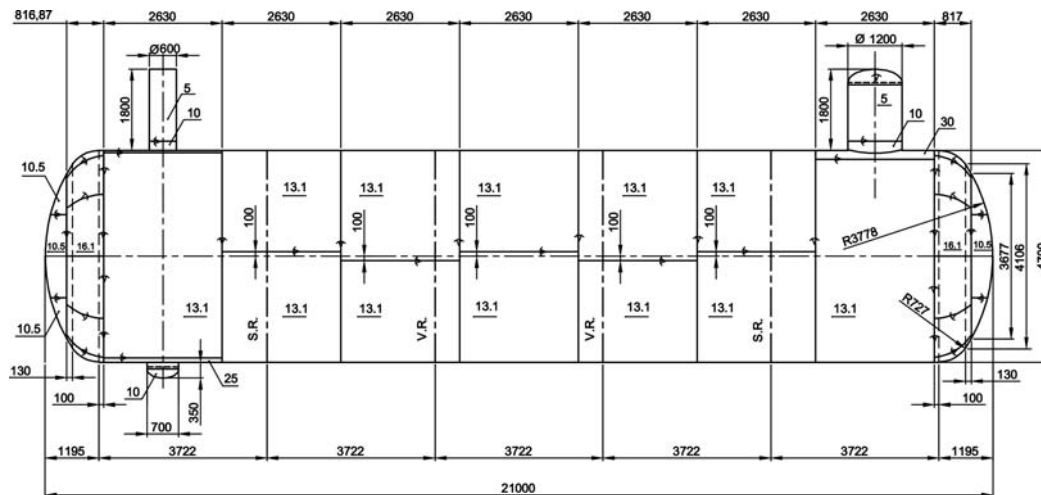


Figure 1. Cylindrical tank, side view

Independent cargo tanks are self-supported structures and do not participate in the ship's strength. They are further subdivided into A, B and C type. The first two tank categories are usually plane structures and the design vapour pressure is up to 0.7 bar. The type C independent tanks are shell structures meeting the vessel criteria. They operate up to the design vapour pressure of 20 bar. Since the transport of gas is hazardous due to potential danger as fire, toxicity, corrosiveness, reactivity, low temperature and high pressure, it is regulated by the International Code for Construction and Equipment of Ships Carrying Liquefied Gases in Bulk (IGC Code) [Resolution MSC.5(48) 1993, Böckenbauer 1981]. The Unified Requirements G1 and G2 of the IGC Code are specified by the International Association of Classification Societies (IACS), and each of IACS members has incorporated this document into their own classification rules, [Rules GL 2000]. This paper deals with the methodology of structure design of the type C independent cargo tanks, also referred to as pressure vessels, built in river ships. The design of tanks for river ships is much easier than for the seagoing ships. Only the design vapour pressure without liquid pressure caused by ship's motion in waves is taken into account. Thus, swash bulkheads are not required for tanks in river ships. The tank supports have to be designed to withstand longitudinal acceleration of 0.3 *g* in case of collision. The volume of one tank is limited to 380 m³.

2. Tank particulars

The design procedure is illustrated in case of a river ship which is equipped with eight cylindrical tanks in two rows. The tanks particulars are the following, Figs. 1 and 2:

outer diameter	4730 mm
total length	21 000 mm
design pressure	15.8 bar
test pressure	237 bar
max. outer overpressure	0.6 bar (vacuum)
temperature	-10°C up to +40°C
material	S 690 QL1

Essential equipment reads:

one manhole, diameter	700 mm
one dome for gauge, diameter	500 mm
one valve dome, diameter	1200 mm
one nozzle for deep well pump, diameter	600 mm
one sump for deep well pump, diameter	700 mm
ten nozzles, diameter	50 – 125 mm
three thermowells, four sample-lines	
inside upper spray line, diameter	100 mm
inside lower spray line, diameter	125 mm
ladder	

The tanks are covered with epoxy coating of 300 micron.

The tanks are designed according to the Rules of Germanischer Lloyd (GL) and the rules for the transport of dangerous goods on the river Rhine (ADNR). The applicable GL Rules are [Rules GL 2000]:

- Pressure Vessels for Inland Waterway Vessels, (general requirements),
- Steam Boilers (calculation formulae applicable to pressure vessels),
- Materials (steel for cargo tanks),
- Welding of pressure vessels.

ADNR Rules are written in French and German. However, the ADNR technical rules, Part 9, are nearly identical to the ADN technical rules, Part 9. The ADN are the recommendations for the transport of dangerous goods on inland waterways from the United Nations Economic Commission for Europe [Recommendations...].

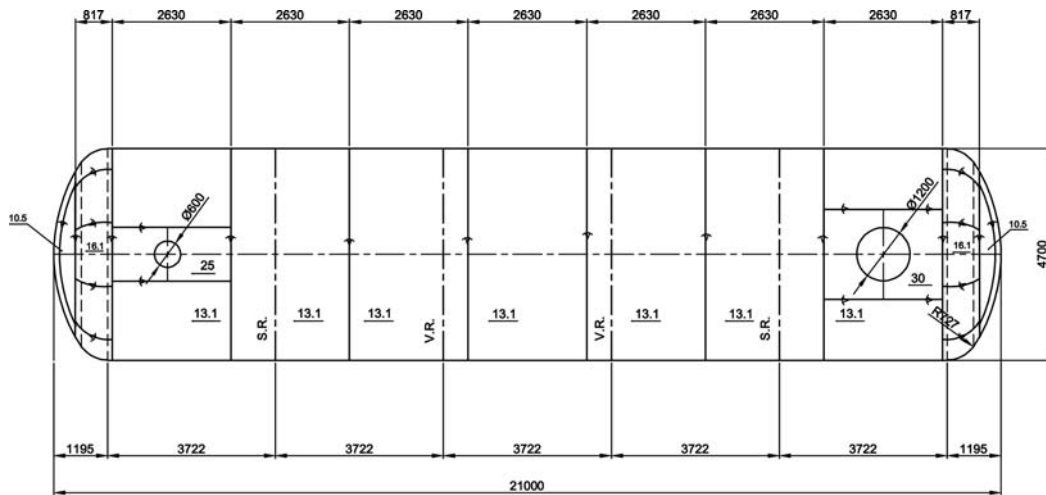


Figure 2. Cylindrical tank, top view

Material for tank production is S 690 QL1 with the following characteristics:

Young's modulus,	$E = 2.06 \cdot 10^5 \text{ N/mm}^2$
Poisson's ratio,	$\nu = 0.3$
yield stress,	$R_e = 690 \text{ N/mm}^2$
tensile strength,	$R_m = 770 \text{ N/mm}^2$

The allowable stress according to GL yields:

allowable membrane stress,	$\sigma_{am} = R_m / 2.7 = 285 \text{ N/mm}^2$
allowable total stress,	$\sigma_{at} = 0.57 R_m = 439 \text{ N/mm}^2$

3. Shell thickness

The cargo tanks structure consists of cylindrical shells and spherical dished ends with a toroidal connection for the reduction of stress concentration. In a similar way, the tank domes and sumps are constructed.

Thickness of the constitutive shell types is determined by using the GL Rules formulae for pressure vessel and steam boilers, which are based on the membrane theory and allowable stresses, as given below [Rules GL 2000].

Cylindrical shell

$$t = \frac{D_a p_c}{20 \sigma_{am} v + p_c}$$

Spherical shell

$$t = \frac{D_a p_c}{40 \sigma_{am} v + p_c}$$

Toroidal shell

$$t = \frac{D_a p_c \beta}{40 \sigma_{am} v}$$

where:

- t - thickness, mm
- D_a - outside diameter, mm
- p_c - design pressure, bar
- β - design coefficient for dished ends
- v - weakening factor
- σ_{am} - allowable membrane stress

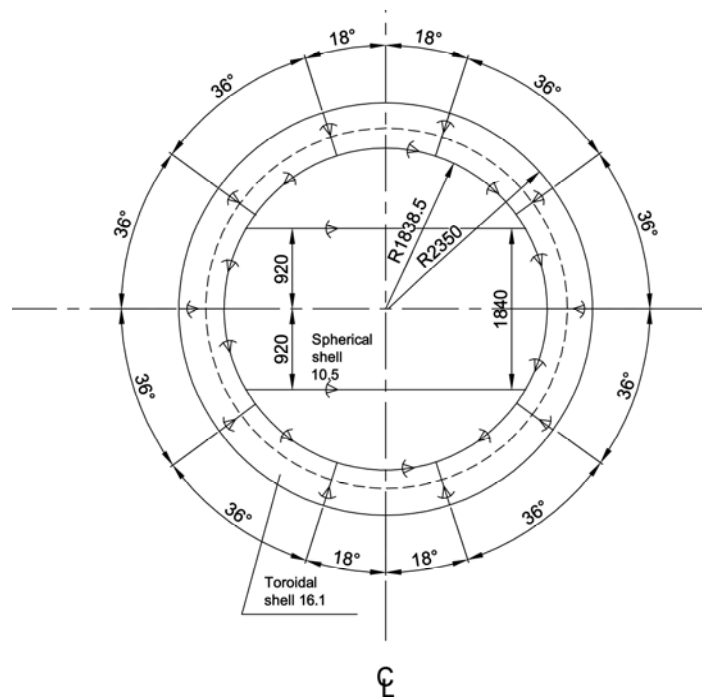


Figure 3. Head of cylindrical tank

In the considered case the value of the outer diameter of the cylindrical and toroidal shell reads $D_a = 4730$ mm, while diameter of the spherical shell is $D_a = 7580$ mm. Furthermore, the value of the weakening factor $\nu = 1$ is taken into account. Based on the above data, the thickness of the cylindrical and spherical shells yields 13.1 mm and 10.5 mm respectively, Fig. 3. The value of the design coefficient $\beta = 2.45$ for the toroidal shell is determined from the corresponding diagram in [Rules GL 2000] based on the ratio H/D_a and t/D_a , where H is the height of dished ends.

The thickness of the dome and sump is determined in a similar way according to the GL Rules and AD Merkblatt [AD Merkblatt 1990]. The weakening effect in the tank shell due to the dome and sump existence is calculated by taking into account the compensation for the open areas. The tank shell is shown in Figures 1, 2 and 3.

4. Shell Buckling

4.1 Cylindrical shell

In the case of external pressure a segment of the cylindrical tank shell between the two vacuum rings may lose stability. According to the GL Rules the critical pressure for elastic buckling of a cylindrical shell is determined by the following formula [Rules GL 2000]:

$$p_{cr} = 20 \frac{E_t}{S_k} \left\{ \frac{\frac{t-c}{D_a}}{(n^2-1) \left[1 + \left(\frac{n}{z} \right)^2 \right]^2} + \frac{\left(\frac{t-c}{D_a} \right)^3}{3(1-\nu^2)} \left[n^2 - 1 + \frac{2n^2 - 1 - \nu}{1 + \left(\frac{n}{z} \right)^2} \right] \right\} \quad (1)$$

where

$$z = \frac{\pi D_a}{2l}, \quad S_k = 3 + \frac{0.002}{\frac{t-c}{R}}, \quad R = D_a/2 \quad (2)$$

and further

- p_{cr} - critical pressure, bar
- t - shell thickness, mm
- D_a - outside diameter, mm
- l - length of shell, mm
- c - allowance for corrosion and wear, mm
- E_t - modulus of elasticity at design temperature, N/mm²
- ν - Poisson's ratio
- S_k - safety factor against elastic buckling
- n - number of buckled folds occurring round the periphery in the event of failure, which gives minimum p_{cr} value.

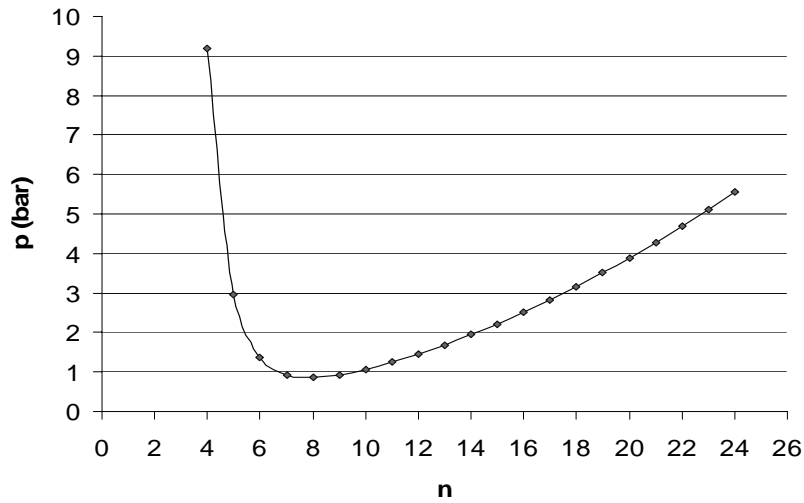


Figure 4. Critical pressure of cylindrical segment

In the considered case $l = 3\,722$ mm and $c = 0$ so that $z = 1.995$ and $S_k = 3.36$. By varying the number of buckled folds n in (1) value of critical pressure is determined as shown in Figure 4. The minimum pressure value $p_{cr} = 0.856$ bar is obtained for $n = 8$, that is higher than the given external value $p_e = 0.6$ bar. Therefore, the cylindrical shell reinforced by the vacuum rings at distance $l = 3\,722$ mm is stable.

4.2 Spherical shell

The tank dished ends are designed as segments of the spherical shell. Their stability may be checked by determining the buckling pressure of the complete sphere. According to the GL Rules, buckling pressure is given by the formula [Rules GL 2000]:

$$p_{cr} = 3.66 \frac{E_t}{S_k} \left(\frac{t-c}{R} \right)^2 \quad (3)$$

Since $S_k = 3.72$ one finds $p_{cr} = 1.565$ bar that is higher than the external pressure $p_e = 0.6$ bar. Thus, the spherical shell is stable.

5. Vacuum rings

The vacuum rings are used to ensure the stability of the tank shell in the event of the internal pressure being reduced below the atmospheric value. The design pressure difference, called external pressure, is $p_e = 0.6$ bar.

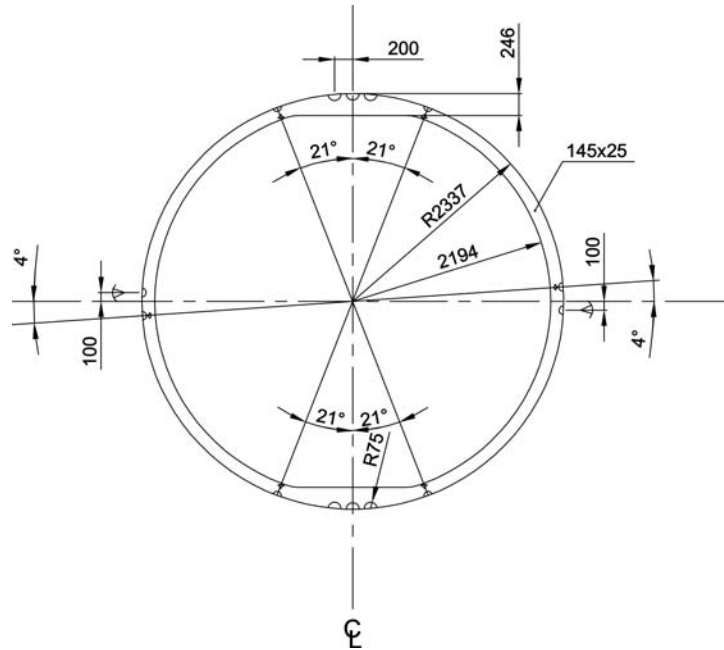


Figure 5. Vacuum ring

The vacuum ring is shown in Figure 5. Its cross-section 145×25 mm, together with the effective breadth of the cylindrical shell

$$b_m = 1.56\sqrt{Rt} = 274 \text{ mm} \quad (4)$$

forms a T-profile with the moment of inertia $I = 0.168 \cdot 10^{-4} \text{ m}^4$.

The span between the vacuum rings, $l = 3722$ mm, is determined in the buckling analysis of the cylindrical shell. Critical pressure of a ring is given by formula [Handbook of Structural Stability 1971]:

$$p_{cr} = \frac{3EI}{R^3l} \quad (5)$$

Thus, one finds $p_{cr} = 2.15$ bar, so that the safety factor takes an acceptable value $S = p_{cr}/p_e = 3.58$.

6. Stiffening rings

6.1 Ring load

Steel mass is approximately 34.4 t. Tank volume is 351.4 m^3 and in case of transport of vinyl chloride monomer, where mass density is $\rho = 0.97 \text{ t/m}^3$, the cargo mass reads 341 t. Thus, the total tank weight takes the value of 3682 kN. The tank support is loaded by one half of that amount, i.e. $Q = 1841$ kN. The stiffening ring is exposed to the action of circumferential shear load due to tank bending as a thin

wall structure between two supports. Thus, the shear distribution is presented with the formula, Figure 6a

$$q_t = q_{t_0} \sin \varphi, \quad 0 \leq \varphi \leq 2\pi \quad (6)$$

where q_{t_0} is obtained from equilibrium condition of vertical forces

$$Q = 2 \int_0^{\pi} q_t \sin \varphi R d\varphi \quad (7)$$

By substituting (6) into (7) one finds

$$q_{t_0} = \frac{Q}{R\pi} = 248 \text{ kN/m} \quad (8)$$

On the other side, the reaction of the saddle support is distributed as shown in Figure 6b, i.e.

$$q_s = q_{s_0} \cos \varphi, \quad \frac{\pi}{2} \leq \varphi \leq \frac{3}{2}\pi \quad (9)$$

where based on the equilibrium condition of vertical forces

$$Q = 2 \int_{\pi/2}^{\pi} q_s \cos \varphi R d\varphi \quad (10)$$

Thus, by substituting (9) into (10) yields

$$q_{s_0} = \frac{2Q}{R\pi} \quad (11)$$

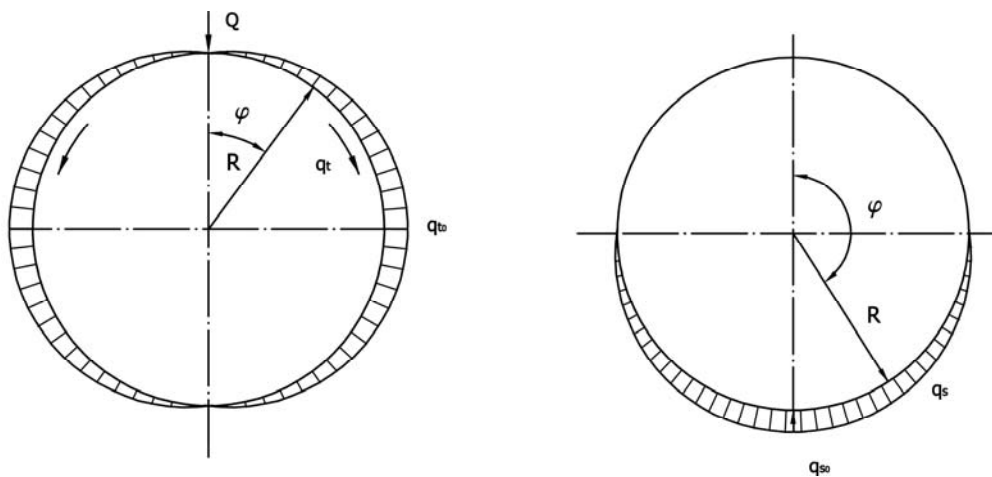


Figure 6. Load of stiffening ring: a – shear load, b – reaction load

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MODEL: T2-1  
1: LINEAR ANALYSIS  
3DAL DISPLACEALL  
4X = .313E-4  
IN = 0  
SCALE FACTOR = .254E5
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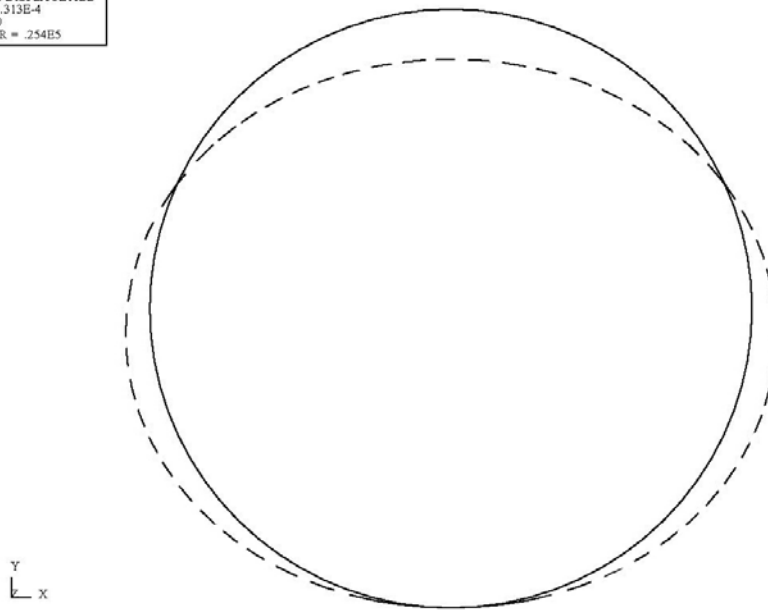


Figure 7. Deformation of stiffening ring
(the bottom point is fixed)

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MODEL: T2-1  
1: LINEAR ANALYSIS  
GAUSS G-FORCE NXX  
MAX = -.319  
MIN = -.96
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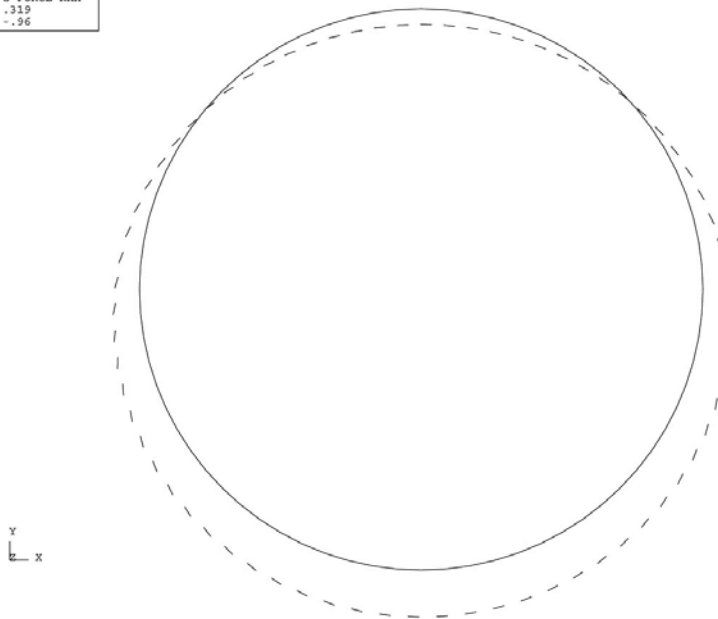


Figure 8. Axial force of stiffening ring N_{xx}
(maximum value at the ring bottom, $\varphi = 180^\circ$)

MODEL: T2-1
 1: LINEAR ANALYSIS
 GAUSS G-FORCE N1Z
 MAX = -.217
 MIN = -.217

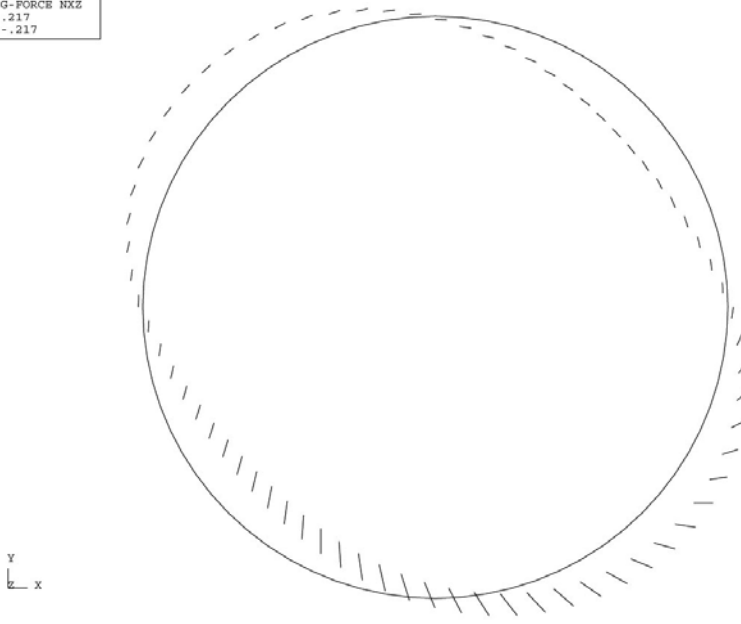


Figure 9. Shear force of stiffening ring N_{1z}
 (extreme values in the points $\varphi = 45^\circ, 135^\circ, 225^\circ$ and 315°)

MODEL: T2-1
 1: LINEAR ANALYSIS
 GAUSS G-FORCE M1
 MAX = .405
 MIN = -.378

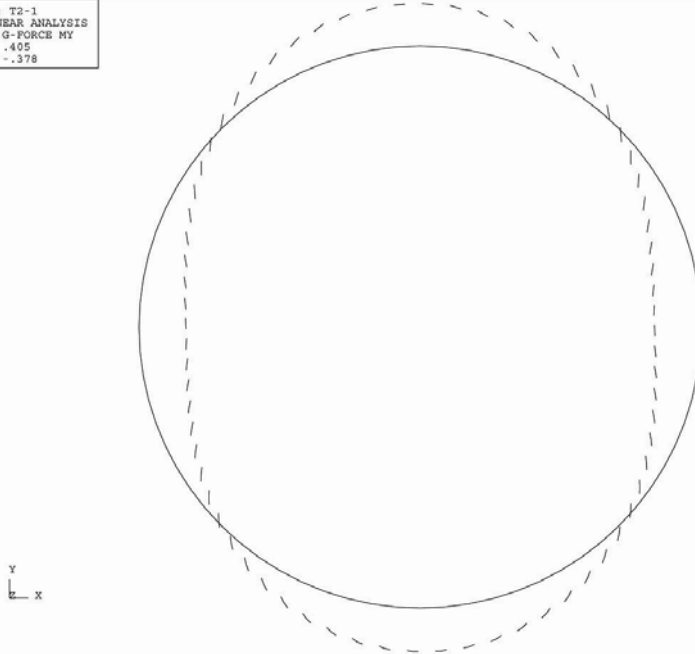


Figure 10. Bending moment of stiffening ring M_1
 (extreme values in the points $\varphi = 0^\circ, 90^\circ, 180^\circ$ and 270°)

6.2 Ring forces

The ring sectional forces due to the relative circumferential shear load and normal support reaction, i.e. taking $Q = 1$ kN into account, are determined by the finite element method, using the program [SESAM 1999]. The T-profile of the ring cross-section is assumed as follows: effective breadth of the cylindrical shell

$$b_m = 1.56\sqrt{Rt_c} = 274 \text{ mm}$$

web 500×20 and flange 300×25 mm.

Thus, the resulting cross-section parameters are:

- cross-section area, $A = 0.021 \text{ m}^2$
- shear area, $A_w = 0.01 \text{ m}^2$
- moment of inertia, $I = 0.919 \cdot 10^{-3} \text{ m}^4$.

The ring circumference is divided into 72 arch elements by step of the central angle of 5° , starting from the ring top. The rigid body motion of the ring model is suspended.

The obtained results of the FEM analysis are shown in Figures 7, 8, 9 and 10.

The actual sectional forces for each stiffening ring are obtained by multiplying their relative values calculated for the unit tank shear force with the corresponding value of the support reaction.

6.3 Ring stresses

The stresses caused by the actual sectional forces are calculated at five positions of the ring cross-section in two Gaussian points of each arch element of the ring FEM model. The stress positions are chosen in the symmetry line of the cross-section, at the level of neutral axis, at the ends of the web and at the outer side of the flange and tank shell, Figure 11. Furthermore, the equivalent stresses at the same positions and points are determined using the von Mises formula

$$\sigma_e = \sqrt{\sigma_x^2 + \sigma_y^2 - \sigma_x \sigma_y + 3\tau_{xy}^2} \tag{12}$$

$$\sigma_x = \sigma_n + \sigma_b$$

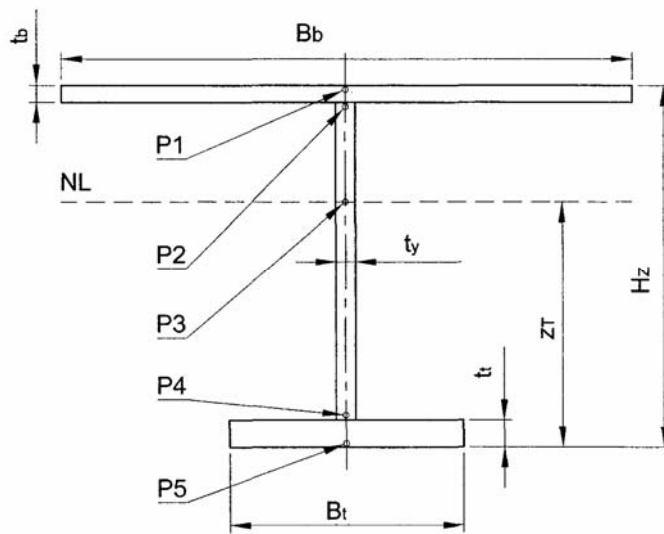


Figure 11. Cross-section of stiffening ring

where σ_x and σ_y are normal stresses in the x and y direction respectively, τ_{xy} is shear stress in the xy plane, σ_n and σ_b are normal stresses due to the axial force and the bending moment respectively. The final dimensions of the stiffening rings at different cross-sections are determined by varying the initial scantlings until meeting the stress criteria. Based on the difference between the equivalent and allowable stresses, the flange and web thickness and the web height are changed.

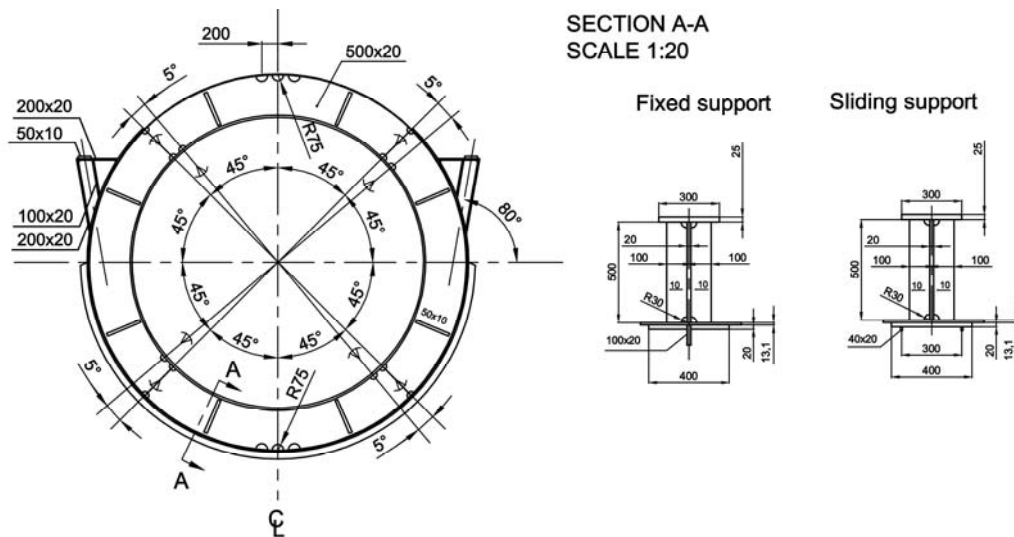


Figure 12. Stiffening ring

The final scantlings of the ring are shown in Figure 12. The web is reinforced with bars against buckling. Distribution of the equivalent von Mises stress at five points of the ring cross-section is presented in Figure 13. Clockwise numbering of the arch finite elements starts from the tank top. Maximum stress occurs in the cylindrical shell at the ring bottom very close to the allowable value $\sigma_{at} = 439 \text{ N/mm}^2$.

7. Conclusion

Liquefied Gas Carriers are special and sophisticated ships. Due to high pressure and low temperature, the design of their cargo tanks requires special attention [Senjanović et al. 2002]. Therefore, the Classification Rules are the implementation of the IMO Code requirements.

The design of cargo tanks in river ships is somewhat simpler than that for sea going ships since hydrodynamic load due to ship motion in waves is not actual. Only longitudinal acceleration of 0.3 g in case of ship collision is taken into account for the design of tank supports.

This paper deals with the design methodology and procedure of the C type cargo tanks in river ships. The integral procedure includes the way of

- determination of shell thickness according to the Classification Rules,
- checking of stability of structural elements (cylindrical and spherical shells and vacuum rings) according to the buckling theory,
- scantlings of stiffening rings based on the theory of thin-walled girders.

The procedure may be used as a design standard since it is in accordance with the requirements of Classification Societies. The presented example of tank design illustrates the application of the procedure that may be useful in design of similar tanks.

Besides the challenge of the tank design, some problems during tank manufacturing may be of special interest as for instance the remedy of tank misalignment, [Senjanović et al. 2005].

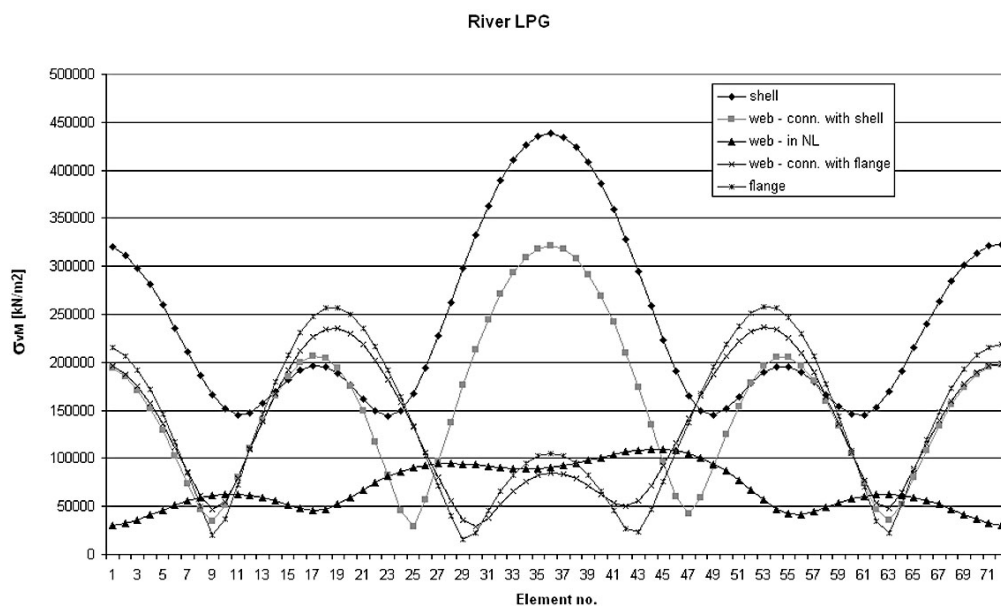


Figure 13. Von Mises stresses in stiffening ring

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