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Technical note

SELECTED ASPECTS OF CRYOGENIC TANK FATIGUE CALCULATIONS FOR OFFSHORE APPLICATION

J. SKRZYPACZ* and P. JASZAK
Wroclaw University of Science and Technology
Faculty of Mechanical and Power Engineering
Wybrzeże Wyspiańskiego Street 27, 50-370 Wrocław, POLAND
E-mails: janusz.skrzypacz@pwr.edu.pl
przemysław.jaszak@pwr.edu.pl

The paper presents the way of the fatigue life calculation of a cryogenic tank dedicated for the carriers ship application. The independent tank type C was taken into consideration. The calculation took into account a vast range of the load spectrum resulting in the ship accelerations. The stress at the most critical point of the tank was determined by means of the finite element method. The computation methods and codes used in the design of the LNG tank were presented. The number of fatigue cycles was determined by means of S-N curve. The cumulated linear damage theory was used to determine life factor.

Key words: LNG, tank, fatigue, ship.

1. Introduction

Liquefied natural gas (LNG) becomes more and more popular as an alternative source of energy. In offshore applications it can be used as a fuel to ship engines. In accordance with [1-2], tanks for LNG carriers can be divided onto four categories: integral tanks, membrane tanks, semi-mebrane and independent tanks. It is worth mentioning that depending on the storage pressure, three types of independent tanks can be distinguished: A, B and C. A and B are dedicated for storage pressure under or equal to *0.7* bar and C for pressure over *0.7* up to *20* bars. Due to a very fast vaporing of the LNG in normal conditions, the medium is stored in a special construction tank similar to a thermos. A typical design of such tank consists of two shells: external and internal one, connected together with supports. There is vacuum between the shells that helps to keep natural gas in liquid form. The computation of such tank elements is very complex and vast as well as a large type of loads must be taken into account that correspond to: ship's motions, collision, thrust acting on the tanks in case of ship flooding and many others [3]. In the standards [4-5], there are adequate calculation codes enabling proper design structures but in case of most complex shapes the best tool to design is computational analysis by means of the finite element method. In [6] comprehensive calculations of the C type tank shell structure were made both in accordance with the calculating code and by FE analysis. Besides the shell structures, one of the crucial element is an internal support. There is some technical contradiction in the design of such a structure. On the one hand, the internal support must be durable enough (large cross-sections) but on the other hand, heat fluxes should be minimalized (small cross-sections). So in the design of internal supports a compromise between the mentioned demands must be found. A very complex analysis of internal tanks support was presented in [7-11]. Different types of material such as polyamide, PTFE where taken into consideration. Based on the literature review there is no procedure of the calculation of the LNG tanks fatigue life. Therefore the main aim of this paper is to show a fatigue life calculation procedure in the design of LNG tank structures.

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* To whom correspondence should be addressed

2. Tank loads determination

Tanks installed on the ship's board have to fulfill rigorous strength requirements. Examples of the load cases required by DNV are presented in Fig.1. Nevertheless critical design points have to be checked by additional fatigue calculations.

Ultimate limit state condition	LC1 Yield check	Cylindrical shell	Pt.4 Ch.7 Sec.4 C200	— Tank system self weight — Static and dynamic pressure due to *cargo (ellipsoid)* — Internal vapour pressure
		Spherical shell	Pt.4 Ch.7 Sec.4 C300	
		Dished ends	Pt.4 Ch.7 Sec.4 C500	
		Shell in way of support	See [3.6]	
		Openings and rein-forcements	Pt.4 Ch.7 Sec.4 D300	
		Supports	See [3.7]	
	LC2 Yield check	Swash bulkhead and ring stiffener if relevant	Pt.3 Ch.1 Sec.4 C300	— Sloshing pressure — Internal vapour pressure
	LC3 Yield and buckling check	Tank	As for LC 1	— Tank system self weight — Static cargo pressure heeled 30 deg. — Internal vapour pressure
		Supports	As for LC 1	
	LC4 Buckling check	Cylindrical shell	Pt.5 Ch.5 Sec.5 I700	— Design external pressure (— Partial filling)
		Spherical shell	Pt.5 Ch.5 Sec.5 I800	
		Dished ends	Pt.5 Ch.5 Sec.5 I900	
Accident limit state condition	LC5 Forward collision	Supports	Pt.5 Ch.5 Sec.5 A1100	— Tank system self weight — Static cargo pressure — Longitudinal dynamic cargo pressure (0.5 g) in forward direction
	LC6 Aft collision	Supports	Pt.5 Ch.5 Sec.5 A1100	— Tank system self weight — Static cargo pressure — Longitudinal dynamic cargo pressure (0.25 g) in aft direction
	LC7 Flooding condition	Flotation supports (tanks located below waterline)	Pt.5 Ch.5 Sec.5 A1100	— Empty tank — External liquid height in cargo hold up to design water line
Tank test	LC8 Tank test	Tank and support	Pt.5 Ch.5 Sec.5 N300	— Full tank filled with fresh water

Fig.1. Load cases applied to a tank design for a maritime application in accordance with DNV rules [1].

Fatigue loads are a result of ship motions on the waves. The vessel hull is subjected to three linear accelerations and three rotation (angular accelerations). The values of the accelerations depend on the hull's length and shape, location of the center of gravity and center of buoyancy, cursing speed and waves.

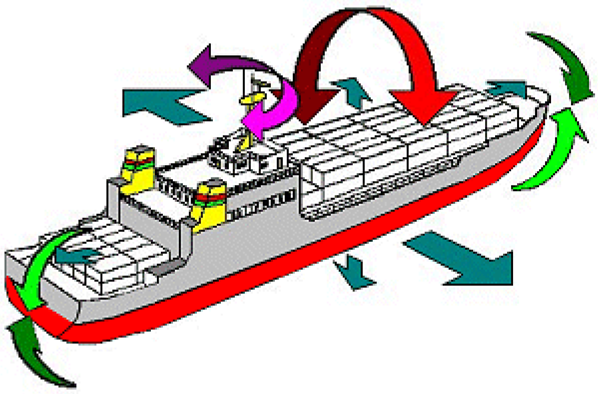

Fig.2. Accelerations acting on the ship due to movement.

For independent tanks type C, design accelerations can be calculated as dimensionless in accordance with the following formulas [1]

vertical
$$a_z = \pm a_0 \sqrt{1 + \left(5.3 - \frac{45}{L}\right)^2 \left(\frac{x}{L} + 0.05\right)^2 \left(\frac{0.6}{C_B}\right)^{\frac{3}{2}}}\,, \tag{2.1}$$

transverse
$$a_y = \pm a_0 \sqrt{0.6 + 2.5\left(\frac{x}{L} + 0.05\right)^2 + \kappa\left(1 + 0.6\frac{\kappa z}{B}\right)^2}\,, \tag{2.2}$$

longitudinal
$$a_x = \pm a_0 \sqrt{0.06 + A^2 - 0.25A}\,, \tag{2.3}$$

total
$$a_t = \left(a_x\hat{}2 + a_y\hat{}2 + a_z\hat{}2\right)\hat{}\,0.5\,. \tag{2.4}$$

It is worth mentioning that a_z does not include the component of static mass, a_y includes the static components in the transverse direction due to rolling, a_x includes the static components in the longitudinal direction due to pitching and a_t is the total acceleration that corresponds to the load P_0 on probability level $Q=10^{-8}$.

In accordance with calculated accelerations, the load spectrum can be determined as the most probable largest load spectrum which can occur once per $1 \cdot 10^8$ waves encountered on the North Atlantic. The load spectrum can be expressed by a number of 8 fatigue loads with a certain number of cycles and probability. A graphical representation of the load spectrum is presented in Fig.3, in pursuance of the following formulas [12]

$$P_i = [(17\text{-}2i)/16]P_o \tag{2.5}$$

$$n_i = 0.9 \cdot 10^i \tag{2.6}$$

where

$i = 1, 2, 3, 4, 5, 6, 7, 8$

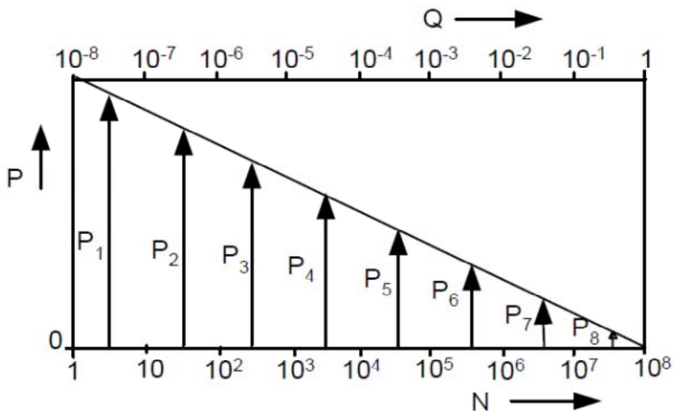

Fig.3. Long term wave-induced load spectrum [2].

3. Fatigue calculations

The fatigue calculations should be made for 6 load cases:

- LC1 – due to a_y+g (in positive transverse direction)
- LC2 – due to -a_y+g (in negative transverse direction)
- LC3 – due to a_x+g (in positive longitudinal direction)
- LC4 – due to -a_x+g (in negative longitudinal direction)
- LC5 – due to a_z+g (in positive vertical direction)
- LC6 – due to -a_z+g (in negative vertical direction)

According to the results of the calculations, the amplitude stresses in every direction can be determined [4]:

- fully reversed cycle

$$\Delta\sigma_{long} = abs\left(\sigma_{long,pos}\right) + abs\left(\sigma_{long,neg}\right),$$

$$\Delta\sigma_{trv} = abs\left(\sigma_{trvg,pos}\right) + abs\left(\sigma_{trv,neg}\right),$$

- one side cycle

$$\Delta\sigma_{vert} = abs\left(\sigma_{vert,pos}\right) - abs\left(\sigma_{vert,neg}\right).$$

The combined stress range was calculated, based on the formula

$$\Delta\sigma_{comb} = \sqrt{\Delta\sigma_{long}^2 + \Delta\sigma_{trv}^2 + \Delta\sigma_{vert}^2}\ . \tag{3.1}$$

The fatigue life can be calculated based on the S-N fatigue approach, under the assumption of linear cumulative damage. The accumulated fatigue damage may be determined by means of formula [4, 5]

$$D = \sum_{i=1}^{k} \frac{n_i}{N_i} = \frac{1}{a}\sum_{i=1}^{k} n_i \left(\Delta\sigma_i\right)^m \le \eta\,. \tag{3.2}$$

Different tank filling ratio may be taken into account during calculations. In this case, the total stress can be expressed as [5]

$$\sigma_{total,FR} = FR^{*}\sigma_{1,\,100\%} + \sigma_{\mathrm{int}\,eria} \tag{3.3}$$

where

$\sigma_{\mathrm{int}\,eria}$ - stress due to inertia forces (weight of the tank), determined by FE,

$\sigma_{1,\,100\%}$ - stress due to weight of the *100*% level liquid

$$\sigma_{1,\,100\%} = \sigma_{total,\,100\%} - \sigma_{\mathrm{int}\,eria}\,.$$

The total accumulated fatigue damage D_{total} taking into consideration changeable fluid level inside the tank, may be calculated according to the formula

$$D_{total}=f_{op}*(f_{FR90}*D_{FR90}+f_{FR70}*D_{FR70}+f_{FR50}*D_{FR50}+f_{FR20}*D_{FR20}). \tag{3.4}$$

4. Example of the calculations

The calculation against fatigue was performed for the LNG tank type C, installed on a ferry board. The tank consists of two vessels: internal where LNG is stored and external that is a vacuum jacket. The structure of the tank is presented in Fig.4. The most critical place is the weld connection between the internal support elements and internal tank.

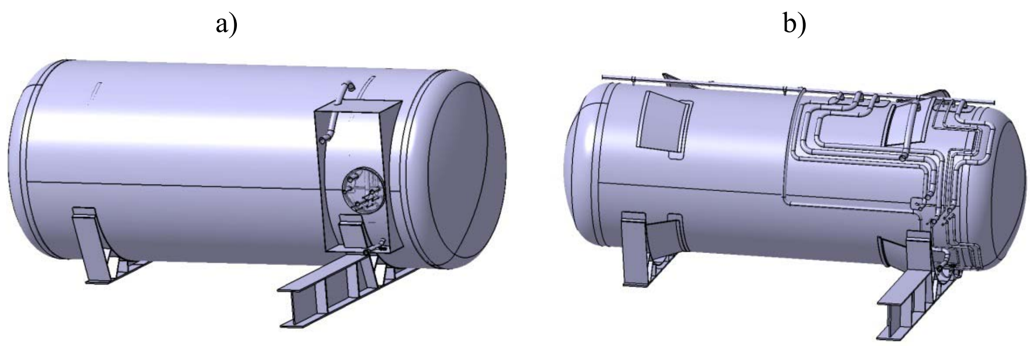

Fig.4 Design of the LNG tank: a) general view, b) view without outer jacket.

The calculations of fatigue life of the inner support elements will be made based on the S-N fatigue approach under the assumption of linear cumulative damage [13]. The assumptions for the calculations are as follows:

1. S-N curve D in air was selected, where: $m_1=3$, $\log a_1=12.164$ ($N<10^7$ cycles), $m_2=5$, $\log a_2=15.606$ ($N>10^7$ cycles) – Fig.5.
2. Applied loads are due to accelerations only combined with gravity.
3. Acceleration components were defined as separate load cases.
4. Dynamic loads and number of the cycles due to wave will be estimated according to formulas (2.5)-(2.6)

$$P_i=P_0*(17-2i)/16, \qquad n_i=0.9*10i, \qquad \text{where} \qquad i=1,2,3,4,5,6,7,8.$$

5. Load P_0 (load on probability level $Q=10^{-8}$) will be estimated as: $\boldsymbol{P_0 = w*a_0}$
   where

   $w = 29000\ kg$ – weight of the liquid and pressure vessel (weight carried by internal supports);
   $a_t = (a_x\verb|^|2+a_y\verb|^|2+a_z\verb|^|2)\verb|^|0.5$ - total design acceleration $a_x=3.6\ m/s^2$, $a_y=12.5\ m/s^2$, $a_z=5.67\ m/s^2$.

6. Accelerations for P_i load were determined under the following assumptions

   $a_i=P_i/w.$

7. Stress for every load will be determined by FE.
8. The utilization factor was estimated as 0.1, what corresponds to DFF = 10 and design life = 20 years. The value of this factor was determined under the assumption of a low probability level of a superficial break – Fig.6.

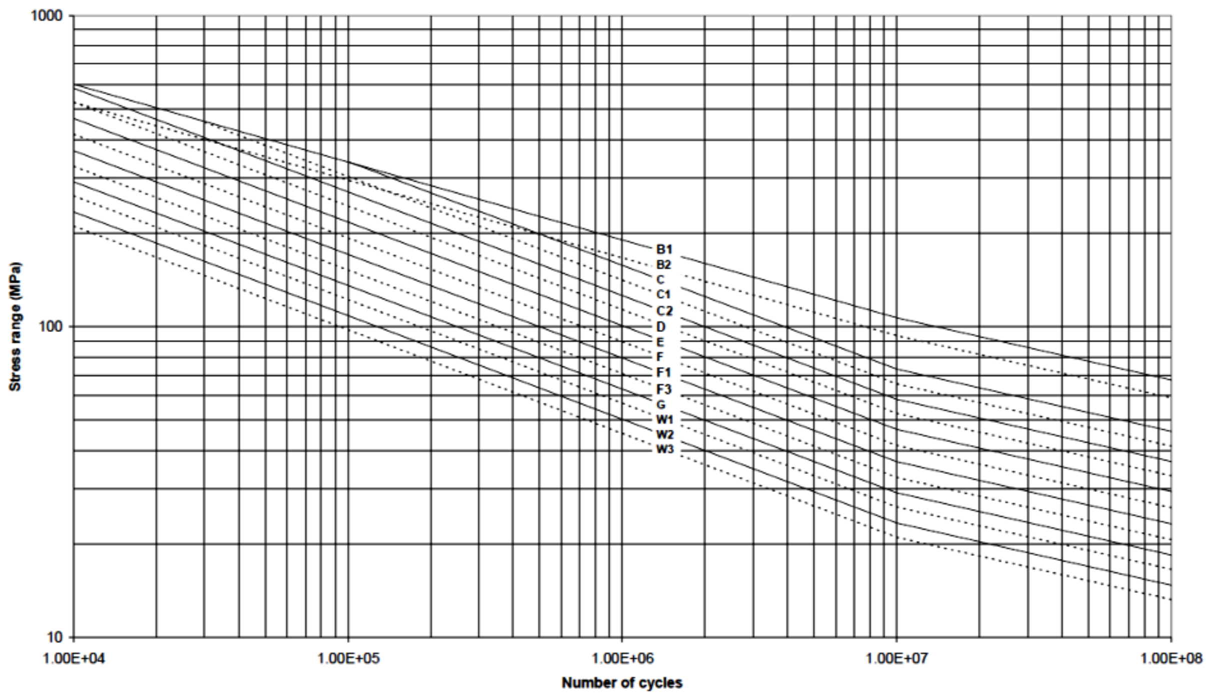

Fig.5. S-N curves in air [13].

DFF	Design life in years						
	5	10	15	20	25	30	50
1	4.0	2.0	1.33	1.00	0.80	0.67	0.40
2	2.0	1.0	0.67	0.50	0.40	0.33	0.20
3	1.33	0.67	0.44	0.33	0.27	0.22	0.13
5	0.80	0.40	0.27	0.20	0.16	0.13	0.08
10	0.40	0.20	0.13	0.10	0.08	0.07	0.04

Fig.6. Utilization factors η as a function of Design Fatigue Factor DFF and design life [13].

The examples of numerical calculations were presented in Figs 7-8. Similar calculations were made for the other load cases and tank filling ratio. A summarization of the calculations for *100*% filling ratio (FR) is shown in Tabs 1-2.

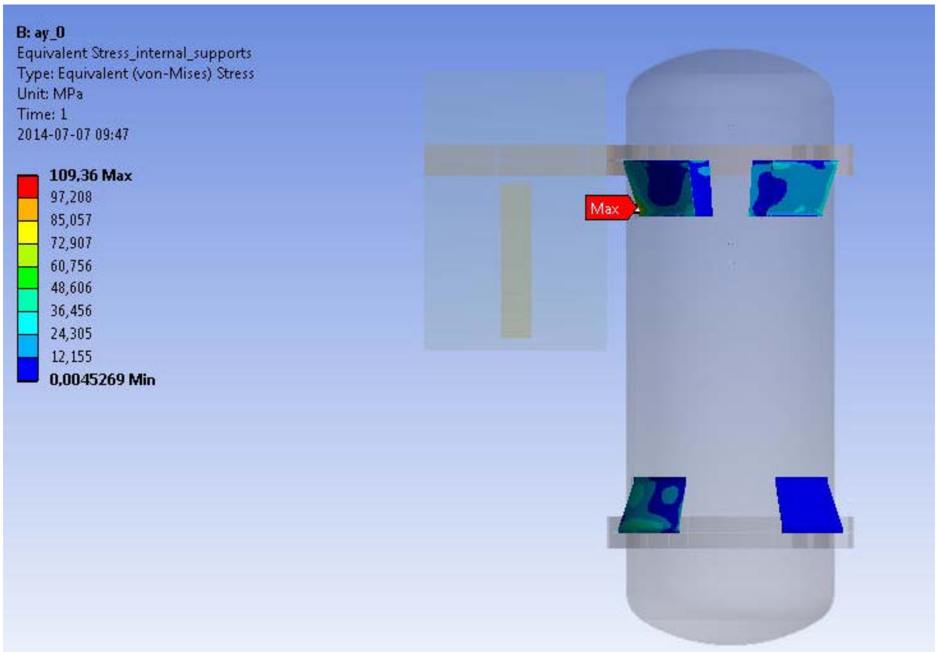

Fig.7. General view on the stress distribution at the internal support wings for LC1 (FR *100*%).

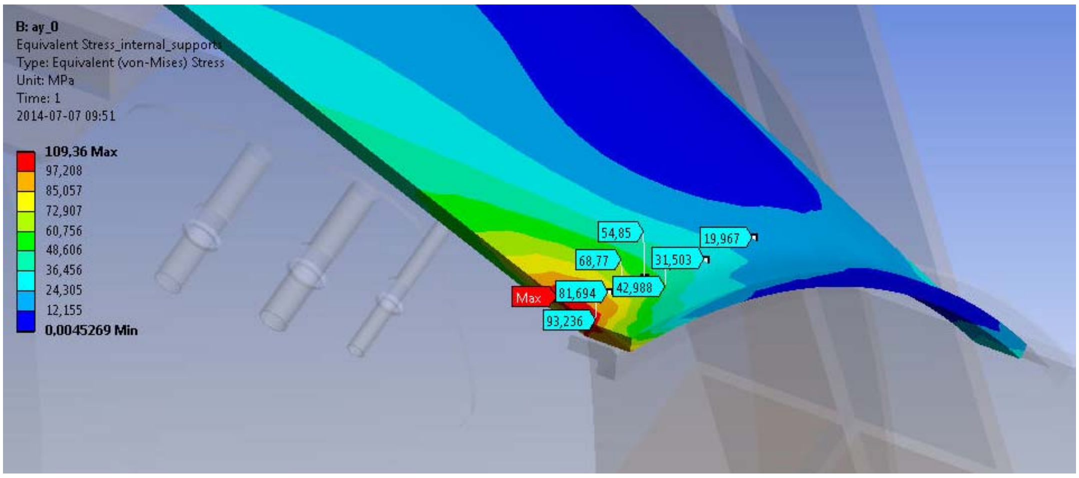

Fig.8. Area of maximal stress concentration on the left wing for LC1 (Fr *100*%).

Table 1. Summary of the stress calculations for *100*% tank filling.

	LC1 - LC2	LC3 - LC4	LC5 - LC6
σ_{pos} [MPa]	*109*	*3.,6*	*57.5*
σ_{neg} [MPa]	*35*	*39.8*	*16.1*
$\Delta\sigma$ [MPa]	*144*	*74.4*	*41.4*
$\Delta\sigma_{total}$ [MPa]	***167.3165***		

Table 2. Accumulated fatigue damage D for *100*% tank filling.

Block	a [m/s^2]	i	ni	σ [MPa]	logN	Ni	ni/Ni
a0	*14.03*	*0*	*1*	*167.31*	*5.494*	*3.116E+05*	*3.209E-06*
a1	*13.15*	*1*	*9*	*156.83*	*5.578*	*3.782E+05*	*2.380E-05*
a2	*11.40*	*2*	*90*	*135.92*	*5.764*	*5.809E+05*	*1.549E-04*
a3	*9.64*	*3*	*900*	*115.01*	*5.982*	*9.589E+05*	*9.385E-04*
a4	*7.89*	*4*	*9000*	*94.099*	*6.243*	*1.751E+06*	*5.141E-03*
a5	*6.14*	*5*	*90000*	*73.188*	*6.571*	*3.721E+06*	*2.419E-02*
a6	*4.38*	*6*	*900000*	*52.277*	*7.014*	*1.034E+07*	*8.706E-02*
a7	*2.63*	*7*	*9000000*	*31.366*	*8.124*	*1.329E+08*	*6.770E-02*
a8	*0.88*	*8*	*90000000*	*10.455*	*10.509*	*3.231E+10*	*2.786E-03*
SUMA							***0.188***

The total accumulated fatigue damage D was calculated for different filling ratios in accordance with formula (3.4). The result was *0.055* what is lower than the allowable value of *0.1.*

$$D_{total} = 0.85*(0.25*0.14+0.25*0.074+0.25*0.036+0.25*0.01)=0.055.$$

5. Summary

The article presents a procedure of the fatigue calculations of LNG tanks in the offshore application, based on DNV recommendations. The critical weld connections located especially in an inaccessible area must be cheeked extremely carefully with the assumption of a low level of probability of the coating break (large value of DFF). It is possible to take into account a changeable fluid level inside a tank. The best method for stress determination is the finite element method. As an example, the calculations of a real tank were presented.

Nomenclature

a – intercept of the design S-N curve with the log N axis
a_0 – nominal acceleration in m/s^2
a_x – combined dynamic horizontal longitudinal acceleration in m/s^2
a_y – combined dynamic horizontal traversal acceleration in m/s^2
a_z – combined dynamic vertical acceleration in m/s^2
a_t – total acceleration in m/s^2
B – breadth of the hull
C_B – block coefficient
D – accumulated fatigue damage
FR – considered filling ratio
k – number of stress blocks
L – total length of the ship
m – negative inverse slop of the S-N curve
n_i – number of stress cycles in stress block I
N_i – number of cycles to failure at constant stress range $\Delta\sigma_I$
P_o – load on probability level $Q=10e\text{-}8$
η – utilization factor = 1/DFF (Design Fatigue Factor)
κ_z – coefficient that depends on metacentric height

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