THE ANNALS OF "DUNAREA DE JOS" UNIVERSITY OF GALATI
FASCICLE XI – SHIPBUILDING. ISSN 1221-4620
2016

GENERAL CONSIDERATIONS ABOUT FATIGUE FOR A DOUBLE BOTTOM UNIT

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ABSTRACT

For several decades fatigue cracks and damages have been a problem for the ship designers. Detail design was initially the obvious remedy to improve and solve a part of the fatigue problem given by stainless steels with higher tensile strength used in ship construction parts. The hull and ship deck and also bottom structure are improved with the aim of increasing the strength of hull girder. In the following paper, the fatigue tools results of ANSYS Workbench 16.2 are represented for a double bottom unit model of a Deep Water Installation Vessel.

Keywords: fatigue, deep water installation vessel, ANSYS Workbench 16.2.

1. INTRODUCTION

It is known that fatigue is damage (usually failure) caused by oscillating stress below the fracture stress [1]. The study uses as an example a double bottom unit of a Deep Water Installation Vessel, designed to satisfy the general requirements and new demands of the offshore industry, according to the D.N.V. regulations.

Usually engineers use a type of steels with higher value of tensile to design the structure of the ship deck, hull and bottom also to increase the strength of the structure. Ship designers try to improve the hull design of the ship to avoid the damages made by fatigue cracks using such stainless steels with higher value of tensile stress.

Since 1966 the engineers use a specific coefficient factor f1 with the aim of increasing the allowable stress of the material.

The ship design engineers are reconsidering the importance of the fatigue problems given by stainless steels like higher tensile steels, in the light of the last reports regarding the damage of the structure in ship tank from fatigue cracks.

It is very important for design engineers to take the full control of the fatigue, and to design the ship structure with proper fatigue life [3].

To resolve the project of fatigue analysis for a double bottom unit, it will be required:

- general description of the ship together with the equipments and systems on;
- the fatigue of ship structures evaluation according D.N.V. regulations;
- study upon the pressure loads and moments that are applied on the double bottom unit ;
- the calculations of the experiment in ANSYS Workbench 16.2;
- result report.

2. APLICATION OF SIMPLIFIED CALCULATION METHOD

The experiment refers to a double bottom unit of a Deep Water Installation Vessel, with the main dimensions [2] :

- Length overall approx. 263m
- Length between p.p. approx. 112m
- Breath moulded approx 40m
- Depth main deck approx. 24m
- Design draft approx. 7m
- Max. draft approx. 7.50m
- Dead weight at $d = 6.5\text{m}$ (open moon pool) approx. 3700T
- Dead weight at $d = 7.5\text{m}$ (open moon pool) approx. 6000T
- Gross tonnage international (1969) approx.
- Block coefficient $C_B = 0.70$
- Moment of inertia of hull cross-section about transverse neutral axis $I_N = 458.0\text{m}^4$
- Neutral axis above keel $n_0 = 10.39\text{m}$
- Moment of inertia of hull cross-section about vertical neutral axis $I_C = 1273\text{m}^4$
- Trial speed

Vessel trial speed at ballast draft to be approx 16.5 knots in calm weather and clean hull. On the trial trip exact speed measurement to be performed. Economic speed: $12 - 14$ knots

According to indications of *D.N.V. regulations* [2,3], the loading conditions that have been used for this vessel calculations are:

- $p_n = 0.65$: part of design life value in fully load condition;
- $p_n = 0.20$: part of design life value in ballast load condition.

For each load condition [2], the ship specification gives the coefficients in Table 1 and in Table 2. of the calculated centre of gravity and the centre of free surface for ballast tanks.

The stress concentration coefficient is very important parameter in fatigue analysis. This coefficient describes the relationship of increasing the value of the notch stress, and the dependence with the weld geometry. First the K-factor must compute, because this value is decided for details. Like an example, if considering a triangular bracket on top of a stiffener for the weld at the end and computing the values of the K-factor, [3], we find:

- $K_{g_{axial}} = 1.4$ for axial loading;
- $K_{g_{bending}} = 1.6$ for bending;
- $K_{n2} = 1.52$

Table 1. Dates regarding burden conditions

	Fully loaded	Ballast
Stillwater bending moment	-3874950kNm	-3874950kNm
Draught	$T_f = 13\text{m}$	$T_b = 9\text{m}$
Metacentre height	$GM = 1.6\text{m}$	$GM = 1.6\text{m}$
Roll radius of gyration	$KR = 15.0\text{m}$	$KR = 15.0\text{m}$
Part of time in load condition	$p_n = 0.65$	$p_n = 0.20$
Density	$\rho_{water} = 1025\text{kg/m}^3$	

Table 2. Free surface and centre of gravity for ballast tank

Ballast tank	Distance from A.P. [m]	Distance from C.L. [m]	Distance from B.L. [m]
Centre of gravity	138.5	17.896	7.174
Centre of surface	138.5	18.94	17.624

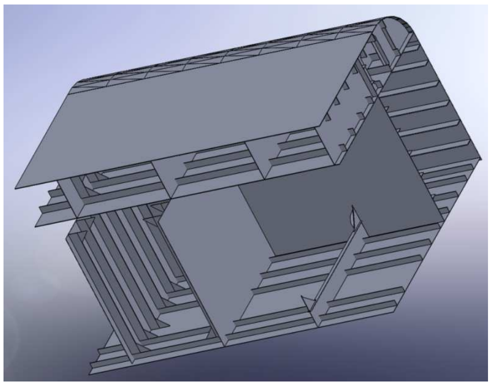

Fig.1 SolidWorks 2015 double bottom model.

To visualize the fatigue and stress that are concentrated in a double bottom unit, that part of the analysis was made in ANSYS Workbench 16.2 with a model created in SolidWorks 2015, Fig. 1.

After the model was ready, it was exported as a STEP file and imported in ANSYS Workbench16.2.

After applying the Earth's gravity to our model we need to put a force that is generated by all the other parts of the ship that are above our module. The value for this force is calculated with a formula.

The next part applies all the pressures and bending moments that are generated by all the other parts of the ship and the sea, with the next formulas and numerical values.

$p_e = r_p p_d$: external sea pressure

where

$$p_{dp} = \max \begin{cases} p_{dp} = p_1 + 135\dfrac{|y|}{B+75} - 1{,}2(T_{act} - z_w) \\ p_{dr} = 10\left[|y|\dfrac{\Phi}{2} + C_B\dfrac{|y + k_f|}{16}\left(0{,}7 + 2\dfrac{z_w}{T_{act}}\right)\right] \end{cases}$$

$$r_p = \begin{cases} 1 & \text{for } z < T_{act} - z_{wl} \\ \dfrac{T_{act} + z_{wl} - Z}{2z_{wl}} & \text{for } T_{act} - z_{wl} < z < T_{act} + z_{wl} \\ 0 & \text{for } T_{act} + z_{wl} < z \end{cases}$$

$\Phi = \dfrac{50c}{B+75}$: maximum roll angle

$c = (1{,}25 - 0{,}025T_R)k \le 30s$: period of roll

$T_R = 2k_r\sqrt{GM}$: roll period

$z_{wl} = \dfrac{3}{4}\dfrac{p_{dT}}{\rho_g}$: distance form actual water line

$F = \gamma V$: Archimedes law

$M_{w0,h} = -0{,}11 f_r k_{wm} C_w L^2 B(C_B + 0{,}7)$: sagging moment

$f_r = 0{,}5^{\frac{1}{h_0}}$: factor to transform the load from 10^{-8} to 10^{-4} probability level

$h_0 = 2{,}21 - 0{,}54\log(L)$: longitudinal term Weibull shape parameter

$$C_w = 10{,}75 - \left[\frac{(300 - L)}{100}\right]^{3/2}$$

$$\Delta\sigma_1 = K_{axial}(M_{w0,h} - M_{w0,s})10^{-3}\frac{z - n_0}{I_N}$$

lateral global stress

$$\Delta\sigma_{hg} = 2K_{axial}M_H 10^{-3}\frac{|y|}{I_C}$$

horizontal global stress

$$M_H = 0{,}22 \cdot f_r L^{\frac{9}{4}}(T_{act} + 0{,}30B)C_B\left(1 - \cos\left(\frac{2\pi x}{L}\right)\right)$$

horizontal wave bending moment

$y = B/2 - h$

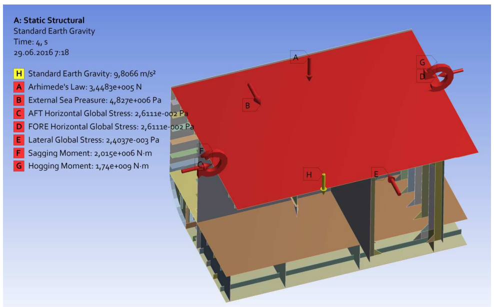

Fig.2 Pressure loads and moments that act to the double bottom unit

3. ANSYS WORKBENCH 16.2 REPORT

There are many forces acting on a ship. The types of forces that occur in waves are the same for every ship but the magnitudes and points of action depend on the profile of the ship below the waterline. The pattern of forces on a ship [7] is very complicated and largely depends on the following parameters:

- The weight of the empty ship
- The weight of the cargo, fuel, ballast, provisions, etc.
- Ice
- Hydrostatic pressure on the hull applied by the water
- Hydrodynamic forces resulting from the movement of the ship in the waves
- Vibrations caused by engines, propeller, pitching

- Incident forces caused by docking collisions

These and other forces cause the ship to be deflected. When the force stops acting, the ship will regain its original shape. If, however, the forces exceed a certain limit [7], the deformation can be permanent.

In the figures 3 to 8, there are shown the graphics for steps force pressure that act among the unit in time, like in table 3 to 8. It is known that all forces and moments are instant applied but, for a better seen of the deformations they wore made in time [6].

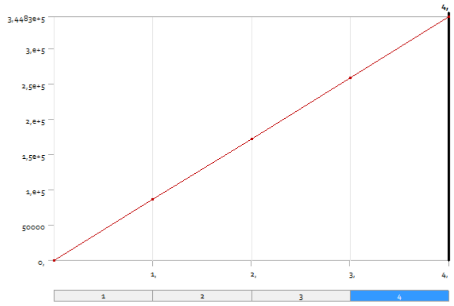

Fig. 3. Archimedes's law variation

Table 3 Archimedes's law values.

Step	Time [s]	Force [N]
1	0	0
	1	86210
2	2	$1.724 \cdot 10^5$
3	3	$2.586 \cdot 10^5$
4	4	$3.4483 \cdot 10^5$

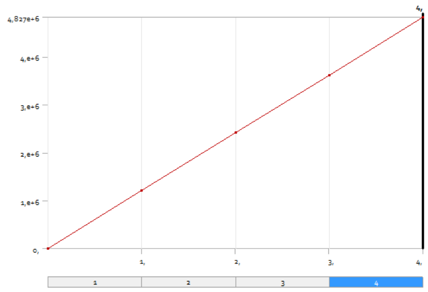

Fig. 4. External sea pressure.

Table 4 External sea pressure.

Step	Time [s]	Force [N]
1	0	0
	1	86210
2	2	$1.724 \cdot 10^5$
3	3	$2.586 \cdot 10^5$
4	4	$3.4483 \cdot 10^5$

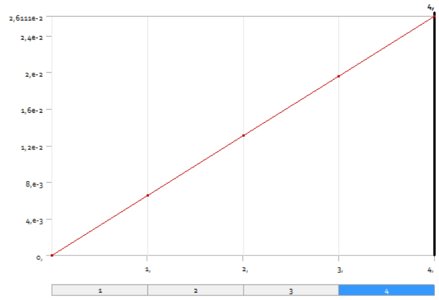

Fig. 5. AFT Horizontal global stress

Table 5. AFT Horizontal global stress

Step	Time [s]	Force [N]
1	0	0
	1	$6.528 \cdot 10^3$
2	2	$1.306 \cdot 10^2$
3	3	$1.958 \cdot 10^2$
4	4	$2.6111 \cdot 10^2$

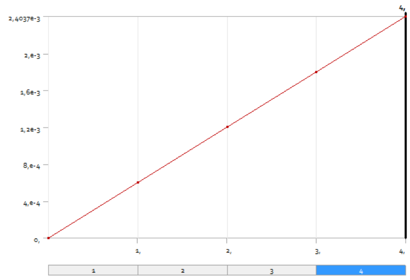

Fig. 6. Lateral global stress

Table 6. Lateral global stress

Step	Time [s]	Force [N]
1	0	0
	1	$6.009 \cdot 10^4$
2	2	$1.202 \cdot 10^3$
3	3	$1.803 \cdot 10^3$
4	4	$2.4037 \cdot 10^3$

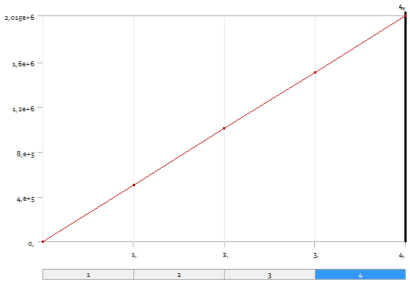

Fig 7. Sagging moment variation

Table 7. Sagging moment values

Step	Time [s]	Force [N]
1	0	0
	1	$5.038 \cdot 10^5$
2	2	$1.008 \cdot 10^6$
3	3	$1.511 \cdot 10^6$
4	4	$2.015 \cdot 10^6$

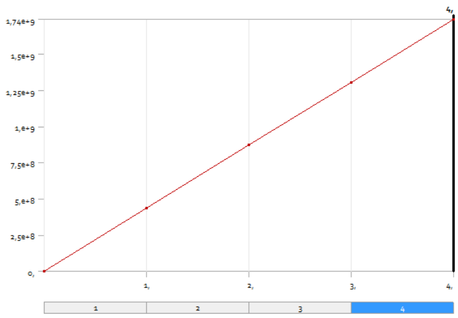

Fig. 8. Hogging moment variation

Table 8. Hogging moment values

Step	Time [s]	Force [N]
1	0	0
	1	86210
2	2	$1.724 \cdot 10^5$
3	3	$2.586 \cdot 10^5$
4	4	$3.4483 \cdot 10^5$

3.1. Total deformation report

The total deformation in mechanics is assimilated to a transformation of a body from a reference configuration to a current configuration [9]. In figure 9 and table 9 there are represented the graphic solution of the total deformation in time.

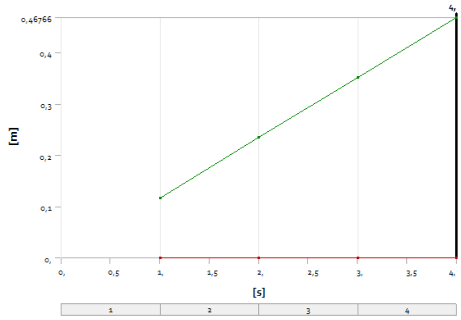

Fig. 9. Total deformation variation

Table 9. Total deformation values

Time [s]	Displacement [m]
0	0
1	0.11694
2	0.23388
3	0.35073
4	0.46766

Fig 10 shows the hot spots of the total deformations. It can be seen the red area is the most affected, and in time it will crack.

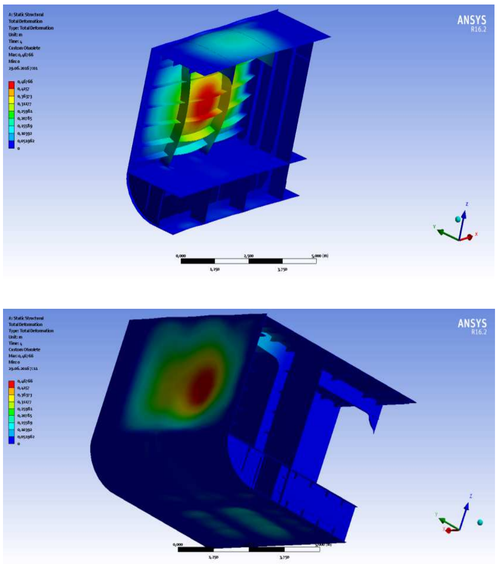

Fig. 10 Double bottom unit total deformation spots.

3.2. Equivalent stress report

The von Mises equivalent stress yield criterion referees to the yielding of materials bandings when the second deviatory stress invariant reaches a critical value [4]. In figure 11, figure 12 and table 11 the values for the double bottom unit example are represented.

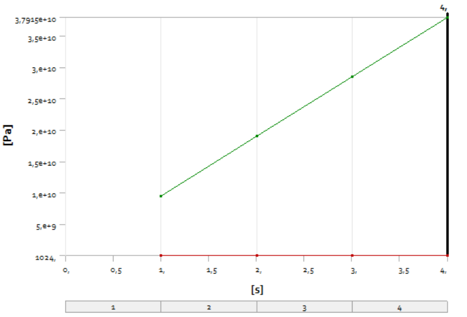

Fig. 11. Equivalent stress variation

Table 11. Equivalent stress values

Time [s]	Min [Pa]	Max [Pa1]
1	1024	$9.4864 \cdot 10^{9}$
2	1692.2	$1.897 \cdot 10^{10}$
3	2596.1	$2.8442 \cdot 10^{10}$
4	3564.8	$3.7915 \cdot 10^{10}$

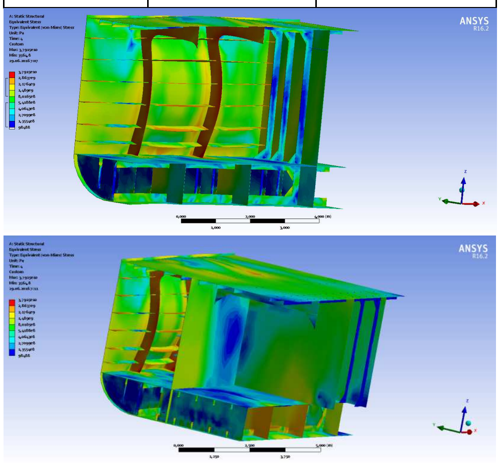

Fig. 12 Double bottom unit equivalents stress

3.3. Fatigue tools report

Figure 13 represents the constant amplitude load for the example of double bottom unit. Classically, constant amplitude and proportional loading was made using computation method namely "back of the envelope", that can be described whether the load has a constant maximum value or continually varies with time. Loading is of constant amplitude, like shown and described in the example, because only one set of F.E. stress results along with a loading ration. In this case, the result is a common type of constant amplitude loading fully reversed (apply a load, then apply a equal and opposite load, a load ration of approximately -1) and zero-based (apply a load then remove it, a load ratio of 0). Because loading is proportional, looking at single set of F.E. results in identifying the critical fatigue locations [4].

Figure 14 represents the mean stress corrections in several empirical options including Gerber, Goodman and Shoderberg theories, which use static material properties along with S-N data to account for any mean stress.

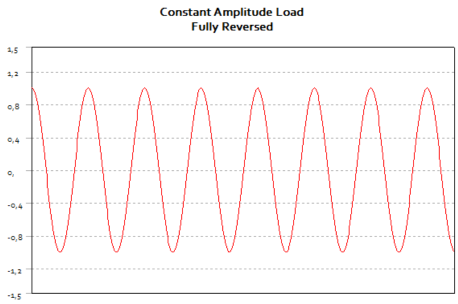

Fig. 13. Constant amplitude loading fully reversed.

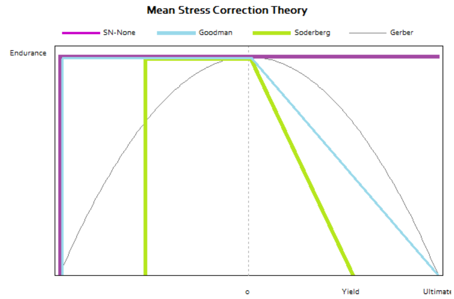

Fig. 14 Graphical representation of the Goodman, Shoderberg, Gerber mean stress along with the S-N data.

The fatigue life has a result over the whole model as seen in figure 15.

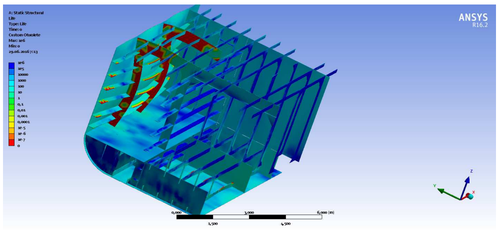

Fig. 15. Fatigue life for the double bottom model

The fatigue damage from figure 16 is a contour plot of the fatigue damage at a design life of 25 years, so the design life divided by the available life defines the fatigue damage. The values greater than 1 indicate failure before the design life is reached.

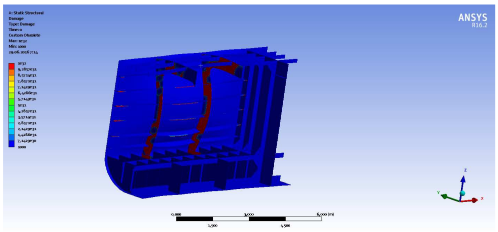

Fig. 16. Fatigue damage for the double bottom model

4. CONCLUSIONS

Design engineers must have the full control of the fatigue phenomenon, to ensure that all designed parts of the ship structure have proper fatigue life.

Computing the fatigue lives and comparing all data of fatigue damages obtained from relevant reports may give to design engineers important information like the axis for the structural design, and also they can make assessment of the structure during fabrication with a proper inspection procedure.

In the first part of the research, some general data about fatigue are presented, in accordance with the D.N.V regulations, from April 2014, that is in force.[3]

- hydrodynamic loads (simplified calculations);
- stress response
  - simplified calculations
  - finite element analysis
- combination of stress components
- long term stress distribution
- fatigue damage calculation

The third part prepares the values of the pressure forces and moments data necessary to introduce in ANSYS Workbench 16.2 , for the experiment of the fatigue analysis.

The last part presents the damages for the double bottom unit with the pressure loads and moments calculated in part three in accordance with D.N.V regulations. The forces applied on the double bottom unit are concluded in this chapter by the total deformation, equivalent stress and by the fatigue tools from ANSYS Workbench 16.2.

In conclusion, for a 25 years life it is presented the fatigue damage. The stiffeners and girders from the shell, also the shell stiffeners are the most damaged parts of the double bottom unit.

Acknowledgements

The present paper has been developed with the support of Constanta Maritime University.

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Paper received on November 25th, 2016