Journal of Marine Science and Technology

Volume 25 | Issue 4 Article 11

EVALUATION OF DIFFERENT TRADING ROUTES ON FATIGUE DAMAGE FOR A 216K M3 LNG CARRIER

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Journal of Marine Science and Technology, Vol. 25, No. 4, pp. 458-463 (2017)
DOI: 10.6119/JMST-017-0406-1

EVALUATION OF DIFFERENT TRADING ROUTES ON FATIGUE DAMAGE FOR A 216K M^3 LNG CARRIER

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Key words: LNG carrier, cumulate fatigue damage, North Atlantic, wave load, finite element analysis, trading route.

ABSTRACT

This paper aims to address long term loads and fatigue damage accumulation for different trading routes relative to North Atlantic operations based on IACS scatter diagram for a 216,000 m^3 LNG carrier. The evaluation is based on direct wave load analysis using WASIM and scatter diagrams created in accordance with DNV Code 30.5 for the following trades:

1. North Atlantic
2. World Wide
3. Qatar - Boston
4. Qatar - UK
5. Murmansk - Boston
6. Murmansk - Rotterdam
7. St. Petersburg - Canada
8. Sakhalin - Mexico port close to California USA
9. Sakhalin - Shanghai, China

It is found that in general Murmansk - Boston and St. Petersburg - Canada are the most severe routes after North Atlantic trade for the fatigue strength.

I. INTRODUCTION

LNG carrier is recognized as kind of complex vessel with high technology and high appended value. The size of LNG carriers has gradually increased in the past period of time from - 120,000 m^3 to - 150,000 m^3, with 140,000 m^3 often designated as the standard or conventional size. Sizes of 210,000 m^3 and 220,000 m^3 were constructed in 2006. Further the vessels in size of around - 240,000 m^3 have been recently designed. Designers and owners need to have confidence in the structural performance of the latest generation of these large LNG ships in such harsh wave environments.

It is normally assumed that the vessel trades on a worldwide basis, but the vessel may be sailing on a dedicate route, which may be more severe than a general trade on a worldwide basis. A minimum design life of 25 years is assumed.

The design life is used to define the sea state contour in order to identify the critical sea states for ship motion and fatigue life calculations (DNV, 2006).

Fatigue Limit State (FLS) analysis intended for calculation of dynamic loads used for fatigue assessment of critical details of the structure. The assessment of the fatigue life of the vessels' structural detail is strongly associated with the nature of the environmental loads and the scatter of the fatigue resistance capacity of the detail (Lotsberg, 2006).

Fatigue design assessments, particularly for LNG carriers, become more demanding and challenging as ship operations often require long fatigue design life in harsh wave environments, e.g., 40 years fatigue life in North Atlantic wave environments. The fatigue performance of the hull structure is a key component for operational capability and reliability perspective. Fatigue damage is a direct consequence of cyclic loading, construction standards and trading wave environments, with alignment also playing an important part (Fagan et al., 2010).

The applied tools for calculations of wave loads should be based on recognized software. As recognized software is considered all wave load programs that can show results to the satisfaction of Classification Society. Forward speed effects have to be properly taken into account. In addition, non-linear effects have to be accounted for in extreme sea conditions (Vladimir et al., 2016).

II. GLOBAL FINITE ELEMENT MODEL

A 3-dimensional finite element model of the entire hull has been modelled. The model and the different SESAM programs are used (SESAM, 2004). The global model in Fig. 1 extends over the whole length of the vessel. Typically, a FE-model consists of several super elements, which are assembled together to represent the whole hull of the vessel.

A right hand co-ordinate system is being used for the global

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Table 1. Boundary conditions.

Fixation point	Restraints
AP, Centreline, Baseline	Dx, Dy, Dz
AP, Centreline, Main deck	Dy
FP, Centreline, Baseline	Dy, Dz

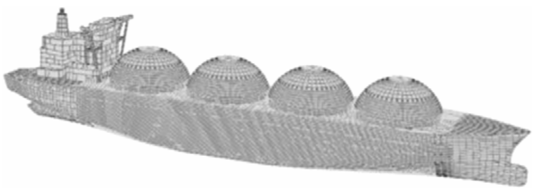

Fig. 1. Global FE model.

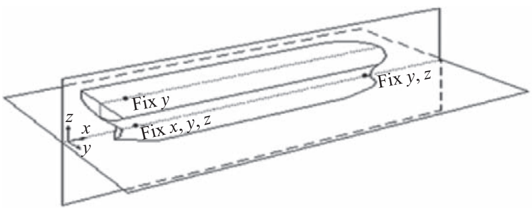

Fig. 2. Boundary conditions applied.

FE-model. The global x-axis is defined as the longitudinal direction of the ship, and the z-axis is defined in the vertical direction (positive upward).

The global model has the boundary conditions as described in Table 1. The boundary conditions chosen should reflect simple supporting to avoid built in stresses. Typically, the fixation points should be located in areas where stresses are not of great importance.

The boundary conditions for the global structural model should reflect simple supports that will avoid built-in stresses. A three-two-one fixation as shown is applied in accordance with DNV Classification Notes CN 34.1 CSA - Direct Analysis of Ship Structures (DNV, 2013). Further, the fixation points should be located away from areas of interest, as the loads transferred from the hydrodynamic load analysis may lead to imbalance in the model. Fixation points are often applied at the centreline close to the aft and the forward ends of the vessel as shown Fig. 2.

III. HYDRODYNAMIC MODEL

There should be adequate correlation between hydrodynamic and structural models such as both models should have equal buoyancy and geometry, equal mass, balance, and center of gravity. The hydrodynamic model and the mass model should be in proper balance, giving still water shear force distribution with zero value at FP and AP. Any imbalance between the mass model and hydrodynamic model should be corrected by modification of the mass model.

The element size of the panels for the 3-D hydrodynamic analysis shall be sufficiently small to avoid numerical inaccuracies. The mesh should provide a good representation of areas with large transitions in shape, hence the bow and aft areas are normally modelled with a higher element density than the parallel ship area. The hydrodynamic model should not include skewed panels. The number of elements near the surface needs to be sufficient in order to represent the change of pressure amplitude and phasing, since the dynamic wave loads increases exponentially towards the surface. This is particularly important when the loads are to be used for fatigue assessment. In order to verify that the number of elements is sufficient, it is recommended to double the number of elements and run head sea analysis for comparison of pressure time series. The number of panels needed to converge differs from code to code.

In this study, the number of nodes and elements used are given as below:

Total number of nodes: 27396
Total number of elements: 32574

30476 linear quadrilateral shell elements and 2098 linear triangular shell elements.

Conventional Shell Element was used because it provides with,

1. Uniformly reduced integration to avoid shear and membrane locking.
2. The element has several hourglass modes that may propagate over the mesh
3. Converges to shear flexible theory for thick shells and classical theory for thin shells.
4. Quadrilateral shell element is a robust, general-purpose element that is suitable for a wide range of applications

Where it is not applicable to use quadrilateral shell elements, triangular shell elements were used.

In hydrodynamic panel model, the global mesh size was set up 600 mm, while the minimum size was 100 mm.

IV. WASIM SIMULATIONS

For the fatigue wave loads 2/3 of the service speed according to CN 30.7 “Fatigue assessment of ship structures” (DNV, 2015) has been applied. Further, the wave headings relative to the ship are assumed to have equal probability during ship operation, i.e., the time the ship is exposed to beam sea is equal to the time it is exposed to head sea.

The WASIM calculations have been carried out for 13 knots (2/3 of 19.5 knots) for the ballast condition and for the full load condition. The calculations are performed for 12 wave-headings (from 0° to 330° with a 30°-step).

In this study the element size of the panels for the 3-D hydrodynamic analysis are sufficiently small to avoid numerical inaccuracies. In addition, the mesh is believed to provide a good representation of areas with large transitions in shape, hence the

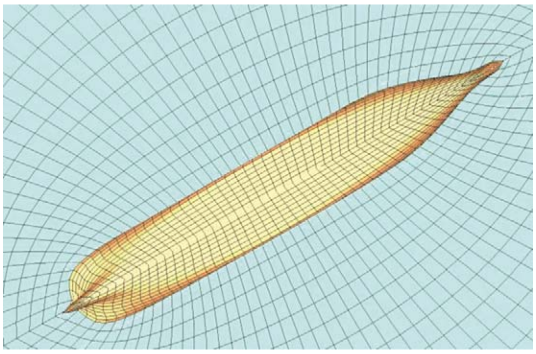

Fig. 3. WASIM hydrodynamic model.

bow and aft areas are normally with a higher element density than the parallel midship area. In this paper, hydrodynamic model does not include skewed panels. The number of elements near the surface is sufficient in order to represent the change of pressure amplitude and phasing, since the dynamic wave loads increases exponentially towards the surface. This is particularly important when the loads are used for fatigue assessment. The panels are vertical oriented as shown in Fig. 3 in order to ease the load transfer.

The time series of the responses calculated by WASIM, 2004 are Fourier transformed to obtain transfer-functions for the responses in the frequency domain. The transfer-functions give response per unit wave height as a function of the wave period for each wave heading. The peak of the transfer-function shows the wave period which gives the largest response for the headings considered.

The linear long-term response for midship vertical bending moment is corrected for non-linear effects by running non-linear WASIM in a design wave.

All calculations are based on direct calculated wave loads using a 3D hydrodynamic program including effect of forward speed. The pressure and inertia loads from the hydrodynamic analysis are transferred to to the structural FE models maintining the phasing definitions.

For fatigue damage calculations full stochastic (spectral) fatigue analysis is used as per DNV Classification Notes CN 34.1 CSA - Direct Analysis of Ship Structures (DNV, 2013), where the structural calculations are performed with linear static analysis.

The transfer-functions for midship vertical bending moment (vbm) show that head sea gives the largest response. The design wave is calculated dividing the long-term response for midship vbm with the peak value of the transfer-function for midship vbm. The WASIM time-domain simulation gives the design sagging and hogging moment. The still water vertical bending is then subtracted to give the wave vertical bending moment.

V. FATIGUE DAMAGE CALCULATIONS

The fatigue calculations have been based on the followings:

(1) Loads (transfer functions) are taken from the hydrodynamic analysis.

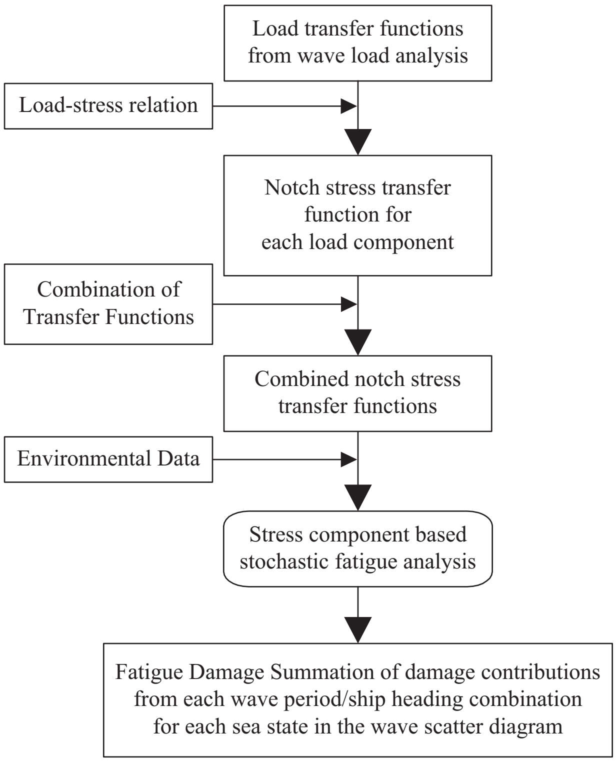

Fig. 4. Fatigue analysis flow chart.

(2) Fatigue damage is calculated on basis of the Palmgrens-Miner rule, assuming linear cumulative damage.
(3) North Atlantic wave data are applied according to (DNV, 2015). Short crested waves with a wave spreading function $\cos^2$, a constant wave directional distribution and Pierson Moskowitz wave spectrum are used.
(4) A vessel speed of 13 knots, corresponding to 2/3 of design speed is used.
(5) 12 headings with 22 periods for each heading has been used.
(6) The target life is set to 25 years (DNV, 2013).
(7) The S-N Curve I (Welded joint, Air or Cathodic protection), is used for the deck.
(8) Both S-N Curve II (Welded joint, Corrosive Environment) and SN Curve I, is used for details in the ballast tanks. Fatigue damage both for coated and uncoated ballast tanks are calculated.
(9) The fraction of time at sea has been taken as 0.85 (0.45 in full load and 0.40 in ballast).
(10) Stress reduction factor, fm = 0.85, due to mean stress effect is included (assumed zero mean stress).

The fatigue assessment is based on the component stochastic fatigue approach according to DNV CN 30.7, 2015. A flow diagram of the procedure is shown in Fig. 4.

The transfer function for the combined stress response, $H_\sigma(\omega)$, is determined by a linear complex summation of stress transfer function as follows:

$$H_\sigma(\omega) = A_{VBM} \cdot H_{VBM}(\omega) + A_{HBM} \cdot H_{HBM}(\omega) + A_P \cdot H_{PE}(\omega) + A_P \cdot H_{PI}(\omega) \tag{1}$$

where,

$H_{VBM}(\omega)$ = Transfer function for vertical bending moment
$H_{HBM}(\omega)$ = Transfer function for horizontal bending moment
$H_{PE}(\omega)$ = Transfer function for external pressure
$H_{PI}(\omega)$ = Transfer function for internal pressure
A_{VBM} = Stress factor per unit vertical bending moment
A_{HBM} = Stress factor per unit horizontal bending moment
A_P = Stress factor per unit pressure for local stiffener bending

For structural details the following transfer function for the combined stress response, $H_\sigma(\omega)$ is used:

$$H_\sigma(\omega) = A_{VBM} \cdot H_{VBM}(\omega) + A_{HBM} \cdot H_{HBM}(\omega) + A_{SP} \cdot H_{PE}(\omega) + A_{SP} \cdot H_{PI}(\omega) \tag{2}$$

where,

$H_{PE}(\omega)$ = Transfer function for external pressure
$H_{PI}(\omega)$ = Transfer function for internal pressure
As_P = Stress factor per unit pressure for lug due to shear

This combination of transfer functions is performed in the SESAM program POSTRESP, which take the phase relations from the contributing transfer functions into account.

When the transfer function of the stress in the hot spot is established, the statistical features in POSTRESP is applied. The transfer function is combined with a wave spectrum and the directional wave energy spreading function, $\cos^2$, to find the short-term stress for specific wave periods. The short-term stress for each wave period is then combined with the relevant scatter diagrams (significant wave height versus wave period) to calculate the fatigue damage contributions from each wave period/ship heading combination for each cell in the scatter diagram.

The fatigue damage calculations are based on the S-N fatigue approach under the assumption of linear cumulative damage (Palmgrens-Miner rule).

VI. LOAD TRANSFER FUNCTIONS

The fatigue calculations are based on the transfer functions determined in hydrodynamic analysis. The linear mode of the wave load program WASIM, 2004 calculates the dynamic pressures below the waterline. In order to include the effect of intermittent wet and dry surfaces, the pressure is reduced above $T_{act} - z_{wl}$, using the factor r_p, see Fig. 5. The external dynamic pressure amplitude, p_e, related to the draught of the load condition considered, is taken as:

$$p_e = r_p \cdot p_d \tag{3}$$

Table 2. S-N parameters curve I.

Cycles	Loga	M
$N \leq 10^7$	12.65	3
$N > 10^7$	16.42	5

Table 3. S-N parameters curve II.

Cycles	Loga	M
All	12.38	3

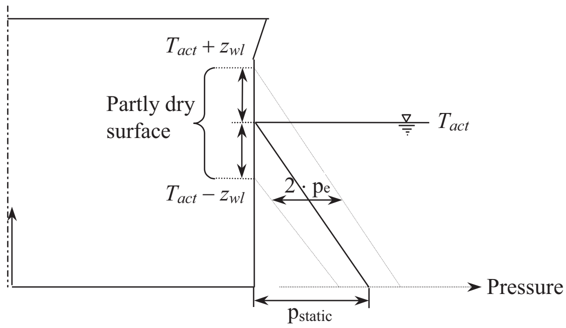

Fig. 5. Reduced pressure range in the surface region.

where,

p_d = dynamic pressure amplitude calculated by WASIM
r_p = reduction and extrapolation of the pressure amplitude in the surface zone
= 1.0 for $z < T_{act} - z_{wl}$
= $\dfrac{T_{act} + z_{wl} - z}{1 z_{wl}}$ for $T_{act} - z_{wl} < z < T_{act} + z_{wl}$
= 0.0 for $T_{act} + z_{wl} < z$
z_{wl} = height of dynamic wave pressure, measured from actual water line
= $\dfrac{3}{4}\dfrac{p_{dT}}{\rho g}$
p_{dT} = dynamic pressure at 10^{-4} probability level (from WASIM) at $z = T_{act}$
T_{act} = the draught in the considered load condition
ρ = density of sea water = 1025 kg/m³
z = distance from B.L. to considered point

VII. S-N CURVES

For deck plating S-N curve I, for welded joints in air is applied in the calculation of the fatigue damage. The S-N parameters are given in Table 2.

The S-N curve II, for welded joints in corrosive environment is applied in the calculation of the fatigue damage for stiffeners. The S-N parameters are given in Table 3.

S-N curves are based on the mean minus two-standard-deviation curves for relevant experimental data, and thus associated with a 97.6% probability of survival.

Table 4. Analyzed loading conditions in FLS.

Loading Condition FLS	Draught AP	Draught FP	Displacement	COG from midship	KG	GM
	[m]	[m]	[Tonnes]	[m]	[m]	[m]
Ballast; Normal ballast cond. Arr.	9.35	9.35	108543.3	-0.18	12.34	14.21
Full load. Cond. Dep.	13.05	13.00	155880.7	-2.71	17.48	3.86

Table 5. Creation of scatter diagrams (DNV, 2012).

Route	Areas: Fraction of time
Qatar - Boston	38:0.0329, 39:0.0395, 50:0.092, 37:0.1382, 27:0.1513, 26:0.1184, 25:0.0921, 16:0.0921, 15:0.0921, 24:0.0921, 23:0.0593
Qatar - UK	38:0.048, 39:0.057, 50:0.133, 37:0.2, 27:0.219, 26:0.172, 25:0.048, 16:0.095, 10:0.029
Murmansk - Boston	1:0.19, 4:0.28, 9:0.16, 8:0.13, 15:0.24
Murmansk - Rotterdam	1:0.52, 4:0.24, 11:0.24
St. Petersburg - Canada	8:0.31, 9:0.31, 11:0.13, 5:0.26
Sakhalin - Mexico Port, California	19:0.11, 20:0.35, 21:0.36, 22:0.18
Sakhalin - Shanghai	28:0.38, 18:0.62

VIII. FE FACTOR AND FATIGUE ROUTE FACTOR

The current study is based on the work reported in DNV, 2012. For fatigue the loading conditions in Table 4 are analyzed. The trading routes are considered as follows:

1. North Atlantic
2. World Wide
3. Qatar - Boston
4. Qatar - UK
5. Murmansk - Boston
6. Murmansk - Rotterdam
7. St. Petersburg - Canada
8. Sakhalin - Mexico port close to California USA
9. Sakhalin - Shanghai, China

The new scatter diagrams are made according to Table 5.

IX. TRADING ROUTE EVALUATION

In order to find the worst case of the trading routes the long term load distributions at a 10^{-4} probability of exceedance are compared. The following loads are evaluated (DNV, 2006):

(1) Midship vertical bending moment
(2) Midship horizontal bending moment
(3) Shear force and axial elongation
(4) External pressure distribution in hold no. 3
(5) Vertical and transverse acceleration in no. 3 cargo and ballast tank

Environmental reductions factors (fe factor) are calculated as the ratio of the long term load for each scatter diagram compared to North Atlantic operation. However, as the Weibull slope parameter and zero crossing periods are different for the different trades, the comparison of the long term load is not fully representative for the fatigue damage accumulation.

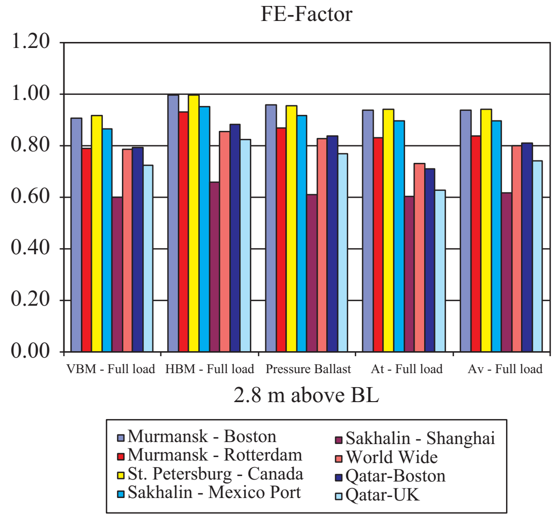

Fig. 6. Long term loads relative to North Atlantic.

A more correct comparison is obtained by comparing fatigue damage ratios for each load component compared to a fatigue damage of 1.0 in North Atlantic operation. These are established by multiplying the North Atlantic long term load response by stress factors resulting in a fatigue damage of 1.0. Using the same stress factor on the long term response for the other trades, the fatigue damage relative to a damage of 1.0 in North Atlantic is obtained. The calculations are based on the DNV S-N curve I for welded joints in air (DNV, 2015).

X. CONCLUSION

A comparison of the FE factors and fatigue damage are presented in Figs. 6 and 7.

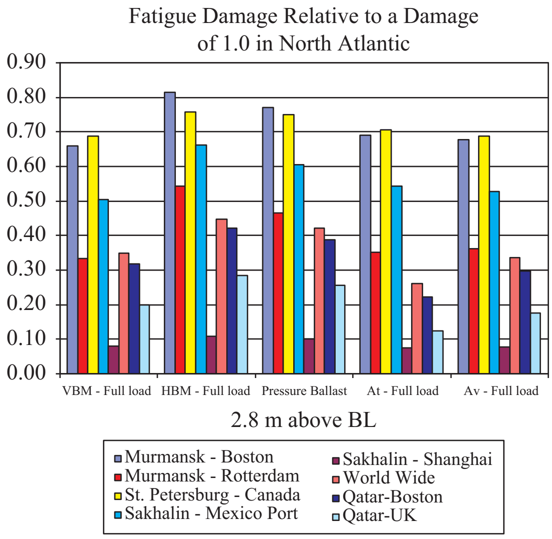

Fig. 7. Fatigue damage relative to North Atlantic.

It can be drawn that in general Murmansk - Boston and St. Petersburg - Canada are the most severe routes after North Atlantic trade for fatigue strength.

In addition, it may be concluded that dynamic pressure is maximum at waterline, and approximately ½ the value at bottom line. Fatigue damage is often more critical just below the draught and around 3-4 m below.

Further, from the analyses undertaken the world-wide trade of 25 years' fatigue life may be equivalent North Atlantic/North Sea trade of around 12.5 years' fatigue life and corrosive environment may also reduce to around 7 years' fatigue life in North Atlantic/North Sea.

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