FATIGUE ASSESSMENT OF A DAMAGED LPG CARRIER

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ABSTRACT

The objective of this work is to assess the fatigue strength of a damaged LPG carrier deck structure. The hot spot stress approach, based on three dimensional finite element models, is used to calculate the maximum principal stresses. The stress concentration factors are estimated using a linearization of the calculated principal stresses around the hot spots. Fatigue damage is calculated based on Palmgren-Miner approach using different mathematical formulations defined by Classification Society Rules and direct approaches. The vertical wave induced bending moment is calculated based on the strip theory. The fatigue analysis is performed for three different damage scenarios.

1. INTRODUCTION

Fatigue damage assessment is an important issue for ship structures and special attention to this failure mode should be paid. Two basic approaches have been developed for ship structural applications: rule based and direct calculation approaches.

The Rules based approach is based on previous experience and has the advantage of its simplicity, but the specific features of an individual ship are not fully considered. Fricke et al. (2002) made a comparative assessment of fatigue damage of the hatch cover-bearing pad using different approaches proposed from different classification societies showing a large scatter, between 2 and 21 years in predicting fatigue lives.

In the direct calculation, the structural characteristics of any individual ships are accounted for, leading to an improved fatigue strength assessment.

Current practice to fatigue analyses of weld components includes different approaches. The main steps in fatigue analysis based on direct calculations involve the description of the wave induced loading, the stress distribution in the structure, the model of fatigue damage or fatigue crack growth and the probabilistic evaluation of the different steps to arrive at a safety index or time dependent reliability.

Hydrodynamic loads can be computed by either the strip theory or the panel method. In the strip theory method, the ship hull is treated as a series of strips that represent the cross sections along the longitudinal direction. Wave loads are first treated separately for each strip without considering the influence of connecting sections.

It has been reported by many authors that the results predicted by the strip theory are in good agreement with experiments in the vertical plane (Jensen et al., 1994). In the present analysis the strip theory is used to calculate the vertical wave induced bending moments.

The analysis of stresses is a complex task due to the complexity of a typical ship structure. Fatigue analysis must be based on local stress response to the global and local loads. The global analysis may be carried out by a finite element model with a relatively coarse mesh.

Sophisticated finite element models can predict local stress response but typical hierarchical multilevel modelling is used where a lower level global model is used for calculating the load and boundary condition for the high level local model. The accuracy of calculated local stress is governed not only by the mesh density of the local finite element model but also the accuracy of the loads applied to the local model (Garbatov et al., 2010).

The aim of the local finite element analysis is normally not to calculate directly the stress at a detail, but to calculate the stress distribution in the region of the hot spot. The second step is to use these stresses as a basis for the derivation of the geometric stress concentration factors.

In this respect the literature has a large spectrum of the applied finite element modelling techniques such as for example (Niemi et al., 2004, Fricke and Kahl, 2005, Chakarov et al., 2008).

For ship structures the fatigue analysis is usually developed accordingly to procedures with predefined long term stress range distribution and assuming a design life corresponding to the service life typically for 25 years, which means that fatigue cracking during service life is accepted and the fail-safe principle is applied.

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The approaches for the fatigue strength assessment utilized the S-N curves for classified joints being a common basis, normal, hot spot and notch stress approach being generally accepted and Palmgren-Miner linear damage summation being applied. Differences remain on the definition of wave loads, the S-N curves and the calculation of hotspot (see Figure 1) and notch spot stress concentration factors (Fricke et al., 2008).

The simplified procedures of the Classification Societies are sufficient for fatigue screening of the details of a ship structure for pointing out the potentially fatigue critical ones but that they cannot be realistically applied to the design of new types of structures (catamarans, fast ships etc). For realistic fatigue life assessment these procedures need to be further calibrated for the different types of ships and the database of ship structural details should be completed and integrated with the results of both tests and theoretical investigations or can be used direct calculation approach (Guedes Soares et al., 2003).

The objective of this work is to analyse the fatigue damage of knee deck structural details as a consequence of detachment between the cargo tank and the principal hull of an LPG ship (three cases). The fatigue analysis is performed using the Rules defined and direct approach based on the S-N formulation. The fatigue damage calculations are performed for two hot spots. Three damage scenario of cargo tank attached to the hull ship have been analysed for two loading conditions. The fatigue damage, analysed by the spectral approach, has been used the Pierson - Moskowitz or JONSWAP spectrum, for 13 different heading angles and 10 speeds and the North Atlantic or WWT scatter diagrams.

2. WAVE INDUCED LOADING

The ocean surface can be represented as a superposition of a large number of regular waves having different heights, lengths, direction and random phase differences. Such consideration allows the ocean surface to be described mathematically and also allows the use of statistical methods to predict the loads in a ship’s life.

Assuming stationarity over a short period of time 1~3 hours, the sea elevation can be described as a stationary, relatively narrow-banded, Gaussian random process, where the distribution of wave energy over different frequencies is expressed by a wave spectrum. In usual practice it is assumed that the sea states are described by a single peaked spectrum, which is well modelled by ISSC parametric of the Pierson and Moskowitz (1964) form.

A more precise formulation would use the Pierson Moskowitz spectrum only for developed sea states and to adopt the JONSWAP spectrum for developing seas and also a double peaked spectrum for mixed seas, Guedes Soares (1984).

![Figure 1: Hot spot stress](image)

![Figure 2: Pierson - Moskowitz spectrum for different significant wave height](image)

General practice in establishing the design wave loading has been to adopt the North Atlantic wave climate as the reference situation based on the argument that it has the most extreme sea states of all ocean going areas (Guedes Soares and Moan, 1991, Cramer et al., 1993).

From the worldwide mission profile of the ship the relative time period within each Mardsen zone is estimated and the frequency of occurrence of different sea conditions is found as the weighted average of the available wave statistics in the different zones.

If the ship response to wave excitation is linear the total response in a seaway is described by a superposition of the responses to all regular wave components that constitute the irregular sea, which can be performed in a frequency domain analysis. Given the linearity, the response is described by a stationary
and ergodic but not necessarily the narrow-banded Gaussian process.

**Figure 3: The North Atlantic wave scatter diagram**

Linear strip theory is the established method of predicting wave-induced load effects despite the emerging availability of computer codes based on the discretization of the hull in panels and on the application of three dimensional diffraction theory (Guedes Soares et al., 1997).

There are several non-linear theories, which result in an appropriate modelling of the non-linear dependency of the wave-induced vertical bending moment and wave amplitude.

The theory of Jensen and Pedersen (1978) is based on a perturbation approach, which keeps terms beyond the linear ones. Other approximation theories model the non-linearity in the hydrostatic restoring force and examples are the method of Fonseca and Guedes Soares (1998), Guedes Soares and Schellin (1998).

Another line of work uses the panel method associated with time simulation, which however is very time consuming.

However, since for fatigue it is the stress range that is important, wave induced loads can still be calculated by linear theories in that the stress range shows a very small degree of non-linearity.

The transfer function modelling is the response due to a sinusoidal wave with unit amplitude of different frequencies is usually obtained from calculations based on the theory of ship motion in potential flow with linearized free surface conditions. The transfer function is however valid for a specified ship velocity, wave heading angle and loading condition. The loading conditions are typically represented by two discrete cases of full load and ballast load while a more detailed discretization of the parameters speed and heading angles is required.

In the evaluation of the dynamic stress levels of the structural joint of consideration, the total dynamic stress components need to be considered. The stress component included in the present analysis is as a result of wave-induced vertical hull girder bending stresses and the local stress.

An adequate approximation for the long-term distribution of wave induced stress range can be described by the Weibull distribution (Guedes Soares and Moan, 1991).

**Figure 4: Stress transfer function, \( v = 18 \) knots**

Figure 4 shows the stress transfer function for a speed of 18 knots, for 13 different heading angles. This calculation has been made here for 10 speeds (from 0 to 18 knots with a step of 2 knots).

3. STRUCTURAL ANALYSIS

The analysis of stresses is a complex task due to the complexity of ship structures. The principal dimensions of the LPG carrier analysed here are presented in Table 1. The ship is designed to transport gas refinery in prismatic type cargo tanks designed for a pressure of about 0.25 Bari and with a minimum temperature of -48 °C . The midship section, already modelled for the finite element analysis is shown on Figure 5.
### Vessel hull characteristics

<table>
<thead>
<tr>
<th>Type of ship</th>
<th>LPG Carrier</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hull type</td>
<td>Monohull</td>
</tr>
<tr>
<td>Length overall, $L_{OA}$</td>
<td>236.8 m</td>
</tr>
<tr>
<td>Length between perpendiculars, $L_{PP}$</td>
<td>230.4 m</td>
</tr>
<tr>
<td>Length of water line at T, $L_{WL}$</td>
<td>227.2 m</td>
</tr>
<tr>
<td>Breadth moulded, B</td>
<td>38.2 m</td>
</tr>
<tr>
<td>Depth of main deck, H</td>
<td>23.2 m</td>
</tr>
<tr>
<td>Design draught (full loaded), $T_{T}$</td>
<td>13.8 m</td>
</tr>
<tr>
<td>Ballast draft (state of ballast draft), $T_{b}$</td>
<td>6.8 m</td>
</tr>
<tr>
<td>Block coefficient, $C_{b}$</td>
<td>0.77</td>
</tr>
<tr>
<td>Maximum service speed</td>
<td>18 kn</td>
</tr>
<tr>
<td>Web frame spacing, $l$</td>
<td>3.2 m</td>
</tr>
<tr>
<td>Moment of inertia about the vertical axis, $I_{v}$</td>
<td>796.4 m$^4$</td>
</tr>
<tr>
<td>Moment of inertia about the horizontal axis, $I_{h}$</td>
<td>320.1 m$^4$</td>
</tr>
<tr>
<td>Height of neutral axis above baseline, $n_{w}$</td>
<td>13.8 m</td>
</tr>
<tr>
<td>Vertical wave hogging bending moment, $M_{w,h}$</td>
<td>118.5 MNm</td>
</tr>
<tr>
<td>Vertical wave sagging bending moment, $M_{w,s}$</td>
<td>-130.784 MNm</td>
</tr>
<tr>
<td>Horizontal wave bending moment (full load)</td>
<td>748.3 MNm</td>
</tr>
<tr>
<td>Horizontal wave bending moment (ballast loaded)</td>
<td>550.1 MNm</td>
</tr>
</tbody>
</table>

The finite elements used for generating the model are solid elements (see Figure 5 and Figure 6). Each solid element is defined by 20 nodes having three degrees of freedom per node resulting in nodal translation in $u$, $v$ and $w$ directions.

The global finite element model considers a reliable description of the overall stiffness and global stress distribution in the hull. It takes into account vertical hull girder bending including shear lag effects, vertical shear distribution between ship side and bulkheads, horizontal hull girder bending including shear lag effects, torsion of the hull girder and transverse bending and shear. The global analysis may be carried out with a relatively coarse mesh.

The aim of the local finite element analysis is normally not to calculate directly the stress at a detail, but to calculate the stress distribution in the region of the hot spot and to use these stresses as a basis for the...
derivation of the geometric stress concentration factors. In this aspect in the open literature has a large spectrum of the applied finite element modelling techniques.

The stress concentration factors analysed here are defined based on the hotspot stress assessment of the welded connection between the bracket on longitudinal web frame and the longitudinal stiffener from main deck at Fr.116 (Figure 7) for each Case 1 - 3.

The local finite element model extends from Fr. 115+200 to Fr.121 longitudinal direction and longitudinal web frame CL to longitudinal web frame 3200mm/CL in a transverse direction as well as from 216000 mm/BL to 25800 mm/BL.

The mesh size is approximately equal to the plate thickness (11 mm). The structure is modelled by solid type quadrilateral elements Figure 7 shows the local finite element model at the connection between the bracket from transverse web frame and longitudinal stiffener from main deck at FR 117 and the particular interested area are shown in Figure 6 to Figure 8 respectively.

The boundary conditions of the complete model are presented in Figure 9. Symmetry boundary conditions are applied in the centreline plane (CL). A uniaxial constant stress it applied in the aft part of the model, this uniaxial constant stress it is given by the cross-section of the model. In the forward part of the model all the displacements are fixed.

4. FATIGUE DAMAGE ANALYSIS

4.1. Stress Concentration Factor Assessment

Hot-spot stresses of the welded bracket of longitudinal web frame and the longitudinal stiffener are analyzed based on the finite element method (see Figure 7). The analysed hot spots are located on both sides of the weld, where the weld is located on the plate surface and on the bracket (Model I and Model II). A linear elastic material of a modulus of elasticity of 206 GPa and the Poison coefficient \( \nu = 0.3 \) are used for the calculation. The structures are subjected to a uniformly distributed unit tensile stresses.
Based on $\sigma_{hs} = 1.5\sigma_{o,St} - 0.5\sigma_{1,St}$, a linear stress extrapolation is performed for the principal stresses, as indicated in Figure 11 for the HS1 and for the HS2 respectively using the same procedure. The hot spot stresses and stress concentration factors are estimated at the point of the maximum principal stress at the weld toe line. The results obtained for the stress concentration factors are HS1 = 1.6 and HS2 = 1.9.

4.2. Rule based fatigue damage approach

Fatigue stresses for the Rules based fatigue damage approach is derived from the beam theory, with the wave-induced section loads as input. Section load used for the present analysis comprises only the vertical wave induced bending moment as defined by the Rules (RB).

Alternatively, the hydrodynamic loads may be transferred to finite element models, and the fatigue stress to be calculated from finite element analysis. A direct calculation involving finite element analysis implies much more effort than a simple beam theory.

The fatigue damage calculation is based on the Palmgren-Miner (Miner, 1945) rules summation. For ocean structures the probability density function of long-term stress range is assumed to be distributed by the two-parameter Weibull distribution and the fatigue damage be calculated as (Nolte and Hansford, 1976):

$$D = \frac{a T_d}{T} \sum_{n=1}^{N} p_n q_n \frac{1}{\ln n_o} \left(1 + \frac{m}{h_n}\right)$$  \hspace{1cm} (1)

where $T_d$ is the design life of ship in seconds, $T_d = 25 \text{ years} \cdot 0.85 = 6.7 \cdot 10^5 \text{s}$, where the coeff. 0.85 accounts for the time that the ship is operating in sea, which is as a result of the fact that the ship will be about 15% of its service life in port for cargo operations or in the shipyard for maintenance and repair, $\nu_o$ is the long-term average response zero-crossing frequency calculated as 0.107 Hz by:

$$\nu_o = \frac{1}{4 \log_{10} (L)}$$  \hspace{1cm} (2)

where $L$ is the length of the ship, $\bar{a}$ is the material constant describing the S-N curve II, $\bar{a} = 10^{12.76} = 5.75 \cdot 10^{12}$ and $m = 3$ (DnV, 2003). $N_{load}$ is the total number of load conditions considered (full load and normal ballast conditions), $p_n$ is a coefficient defining a part of the lifetime spent in each of loading conditions, $h_n$ is the Weibull stress range distribution shape parameter for each load condition, $q_n$ is the Weibull stress range distribution scale parameter for each load condition, which is defined as:

$$q_n = \frac{\Delta \sigma_o}{(\ln n_o)^\frac{1}{h_n}}$$  \hspace{1cm} (3)

where $n_o$ is the number of cycles over the time period for which the stress range level $\Delta \sigma_o$ is defined at a probability level of $10^{-4}$ and it is considered as $n_o = 10^4$ cycles. The Weibull distribution shape parameter is $h = 0.94$ m.

The global stress range, without accounting for the local response of structure, may be calculated as:

$$\Delta \sigma_o = f_m \Delta \sigma$$  \hspace{1cm} (4)

where $f_m$ is a reduction factor accounting for the stress effect of the mean stress. The stress range, $\Delta \sigma$, may be calculated as:

$$\Delta \sigma = SCF \Delta \sigma_n$$  \hspace{1cm} (5)

where $SCF$ is the stress concentration factor representing the ratio of the notch stress to the nominal stress at the notch. According to DnV (2003) the $SCF$ of the welded joint can be calculated as:

$$SCF = K_g \cdot K_w$$  \hspace{1cm} (6)

where $K_g$ is the hot-spot stress concentration factor due to the gross geometry of the detail considered and $K_w = 1.15$ is the notch stress concentration for the weld (DnV, 2003).
The total fatigue damage during the service life is calculated by summing the fatigue damage for each load condition and multiplied by a factor $\chi$ accounting for the corrosive environment.

Fatigue damage assessment of the intact ship, Cases 1 for the two hot spots considered here in dry and wet sea water condition is shown on Figure 12. HS 2 demonstrates bigger fatigue damage.

### 4.3. Direct calculation approach

In the current study, two direct fatigue damage calculation approaches are compared: (i) the wave induced load is calculated by the strip theory and the short term distribution of stress ranges are defined by the use of the special formulation and the fatigue damage calculations (DCA_ST) are performed based on Eqn 8 (ii) the wave induced load is calculated by the strip theory and the long term distribution of stress ranges are defined by the use of the special formulation and the fatigue damage calculations (DCA_LT) are performed based on Eqn 1.

When the long term stress range distribution is defined through a short term Rayleigh distribution within each short term period for the different loading conditions the stress range distribution for a given sea state $i$ and heading direction angle $j$ is given as:

$$F_{\Delta\sigma}(\Delta\sigma) = 1 - \exp\left(-\frac{\Delta\sigma^2}{8m_{ij}}\right)$$  \hspace{1cm} (7)

where $m_{ij}$ is the spectral moment of order zero and for a one slope S-N curve the fatigue damage is calculated as (DnV, 2003):

$$D = \frac{\sqrt{\pi T}}{a} \left[1 + \frac{m_a}{2}\right] \sum_{\text{all headings}} \sum_{\text{all sea states}} \rho_{ij} \left(\frac{\Delta\sigma}{\sqrt{2m_{ij}}}\right)^n$$  \hspace{1cm} (8)

The stress ranges at the middle section of the ship as a result of the vertical wave induced bending moments are computed for the three damage cases analysed here, including the two load cases full and ballast conditions, sea states with significant wave height $H_s$ from 1 to 16 m for the North Atlantic scatter diagram and from 1 to 15 m for the World Wide Trade scatter diagram.

The wave headings are modelled for 13 different heading angles (from 0 to 180 degrees with a step of 15 degrees), 10 speeds, from 0 to 18 knots with a step of 2 knots and two wave spectra Pierson-Moskowitz (PM) and JONSWAP are also used in the fatigue damage calculations.
Fatigue damage calculated for the two hot spot points HS1 and HS2 are presented in Figure 13 to Figure 16. It can be observed that the fatigue damage accumulated by HS2 is bigger than the one for HS1 and the damages calculated when the scatter diagram for the North Atlantic is used is more severe than in the case of the WWT scatter diagram.

In order to establish the long-term distribution of stress ranges, the cumulative distribution may be estimated by a weighted sum over all sea states and heading directions. The long-term stress range distribution is then calculated as:

\[
F_{\Delta \sigma} (\Delta \sigma) = \sum_{i=1}^{all\,\text{seastates}} \sum_{j=1}^{all\,\text{headings}} r_{ij} F_{\Delta \sigma_i} (\Delta \sigma) p_{ij} \tag{9}
\]

where:

- \( p_{ij} \) is the probability of occurrence of a given sea state \( i \) combined with a heading \( j \),
- \( r_{ij} = \frac{v_i}{v_o} \) is the ratio between the response crossing rates in a given sea state and the average crossing rate,
- \( v_o = \sum p_i v_{ij} \) is the average crossing rate,
- \( v_{ij} = \frac{1}{2\pi} \sqrt{\frac{m_{2ij}}{m_{0ij}}} \) is the response zero-crossing rate.

The parameters \( h \) and \( q \) representing the shape and scale factor respectively of the fitted long-term Weibull distribution is applied in the calculation for the accumulated fatigue damage by Eqn (1).

Figure 17 to Figure 24 show the fatigue damage calculation for different inputs related to the hot spots, spectra, scatter diagrams and fatigue damage approaches used for calculations (RB, DCA_ST, DCA_LT).

The severe fatigue damage is obtained for all the possible variations of the governing factors in the Case 3 when the cargo tank is attached to the hull ship only by the support 3 (Figure 10).

Figure 25 represents the fatigue damage calculated by the Rules based approach (RB) and by the direct approach using the long term distribution of the stress ranges (Eqn 1) (DCA_LT).
CONCLUSIONS

In this study, two direct approaches of fatigue assessment have been compared through a case study of a damaged LPG carrier ship with 90000 m$^3$. A real wave environment and operation conditions were employed as input of direct calculations, and the fatigue damages obtained from the Rules based approach are used to verify the direct calculation results. The following conclusions can be made:

Fatigue damage calculated for the North Atlantic is severer than the one calculated for the World Wide Trade scatter diagram.

From all the possible variations in governing factors, the biggest fatigue damage is achieved from the one obtained in the Case 3, coupled with the North Atlantic scatter diagram and the Pierson - Moskowitz spectrum.

The Rules based approach demonstrates the more conservative fatigue damage prediction.
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