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FATIGUE ASSESSMENT OF LONGITUDINAL STIFFENER END CONNECTIONS FOR AGEING BULK CARRIERS

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Key words: fatigue calculation, S-N curves, bulk carriers, finite element analysis, close-up inspections.

ABSTRACT

The world bulk carrier fleet is ageing. In this situation it is required to ensure that the transportation of cargo is carried out by quality vessels. This is possible with old bulk carriers as long as their condition can be properly evaluated. Ship structures are subjected to variable cyclic loading during voyage. Areas which are subjected to cyclic stresses may fail due to fatigue damage. Fatigue cracking usually appears on places with high stress concentration such as welds, notches and sharp geometric transitions. A fatigue crack starts at a localized spot and will with cyclic stress gradually increase over the cross section of the component. This study aims to document the calculated fatigue life of longitudinal members amidships. These fatigue calculations are theoretical calculations that should be used as a guidance for close-up inspections when ships are surveyed periodically to verify that they are maintained in an acceptable condition in accordance with international conventions, the Rules of Classification societies, etc.

I. INTRODUCTION

Large and efficient bulk carriers, designed and built mainly in the 1980's and 1990's, are now reaching the end of their service life. In the last decade a large number of bulk carriers were lost. From 1990 to mid-1997 a total number of 99 bulk carriers were lost, with the death of 654 people, where several structural defects were revealed that strongly affected on the ships' safety.

Many of bulk carriers in operation were old and had suffered structural damage. A study by IACS (International Association of Classification Societies) found that after flooding in the foremost hold, the bulkhead between this hold and the adjacent hold can collapse from the pressure of cargo and water, leading to progressive flooding and sinking.

Deterioration of vessels hull/structure through corrosion, fatigue and damage was identified as a principal factor in the loss of many ships carrying cargo in bulk . Failing to identify such deterioration might lead to sudden and unexpected accident The crews may be unaware of the vulnerability of these bulk carrier vessel types. The consequential loss of a ship carrying heavy cargo could be expected to be very rapid, and a major failure could take place.

Fatigue cracks and fatigue damages were known to ship designers for several decades. Initially the obvious remedy was to improve detail design. With the introduction of higher tensile steels (HTS-steel) in hull structure, at first in deck and bottom to increase hull girder strength, and later on in local structure, the fatigue problem became more imminent. During recent years a growing number of fatigue crack incidents in local tank structures made from HTS steels have demonstrated that a more direct control of fatigue was needed.

Damage to side shell, externally through contact with docksides or tugs and, internally from impact by cargo dislodging equipment during discharge, can result in initiating fractures and/or fatigue of the structure. In single side-skin bulk carriers, bulkheads, trunks and ballast tank boundaries, can present "hard spots" that concentrate forces where the change in construction occurs (e.g., longitudinal to transverse framing). This may lead to undetected fractures.

Further, ageing is a contributing factor in the loss of bulk carriers. Statistically, bulk carriers 20 years or older exhibit a greater chance of total loss than their younger counterpart. This forced IMO to think about the safety of bulk carriers and a new chapter has been implemented in Safety of Life at Sea (SOLAS) (IACS, 2012).

Bulk carriers are prone to many modes of cyclic forces that combine with other dynamic forces acting on vessel's structure. Cyclic wave pressure acts on the side frames of the vessel in a constant cycle of loading and unloading forces. Areas which are subjected to cyclic stresses may fail due to fatigue damage. Fatigue cracking usually appears on places with high stress concentration such as welds, notches and sharp geometric transitions. A fatigue crack starts at a localized spot and will with cyclic stress gradually increase over the cross section of the component

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(Hansen and Winterstein, 1996).

Ships are prone to fatigue damage due to high cyclic loads mainly caused by waves and changing dynamic loading conditions. Hence, fatigue is an important criterion during design. Fatigue damages reduce the load-carrying capacity of the structure, and may cause leakages, resulting in pollutions, cargo mixing or gas accumulating in enclosed spaces, in severe cases, such structural damage may conceivably lead to catastrophic failure or total loss of ships. While initial crack characteristics could be analyzed by the fatigue analysis using the S-N curve approach or detected by the survey (Ozguc, 2016; 2017).

Fatigue cracking damage has been a primary source of costly repair work of aging ships. Cracking damage has been found in welded joints and local areas of stress concentrations, e.g., at the weld intersections of longitudinals, frames and girders. Initial defects may also be formed in the structure by fabrication procedure and may conceivably remain undetected over time. The structural models for predicting fatigue cracking damage have been developed as a function of vessel age.

Vessel longitudinals are important structural elements in the side shell structure of ships. The wave loads introduce significant dynamic stresses in the side shell below the mean water level. This has led to a number of fatigue cracks in the welded connections between side longitudinal stiffeners and transverse frames and bulkheads of ships (Li et al., 2013).

During the last 10-15 years the industry has put significant focus on fatigue analysis methodologies for ship-shaped structures. The reason for this is a large cost consequence associated with fatigue cracks in these structures. During these years, experience has been gained from classification of ship-shaped structures, and recommendations from a number of detailed fatigue analyses of ships have been developed through joint industry projects. A brief overview of fatigue analysis methodology used were presented together with some of the recent advances in analysis methodology (Fricke et al., 2012).

Lotsberg (2006) presented a summary of the finite element analyses performed for assessment of hot spot stress with link to one hot spot S-N curve in the FPSO Fatigue Capacity Joint Industry Project (JIP). Recommendations were indicated on how to perform fatigue assessment of plated structures based on finite element analysis combined with one hot spot S-N curve.

More than 40% of the registered fatigue cracks in ship structures were observed to occur in the side shell, more specifically in the connections of longitudinals to transverse web frames. The fatigue damage was caused partly by vertical and horizontal wave-induced hull bending and partly by outside water pressure on the side shell (Lotsberg and Landet, 2005).

Classification Societies developed different tools to ensure a high quality standard of ageing vessels. The computer programs and procedures available today are sufficient to avoid most fatigue problems related to ship shaped structures. An example of a more rigorous procedure by means of DNV Nauticus Hull and Sesam program packages (DNV, 1999) was presented, where a simplified method was used in accordance with Class Rules for ships.

In this study, the objective of the calculations is to document the calculated fatigue life of longitudinal members amidships. These fatigue calculations are theoretical calculations that should be used as a guidance for close-up inspections. Since cracks can conceivably lead to catastrophic failure of the structure, it is required to properly consider implementation of close-up survey strategy.

II. FATIGUE CUMULATIVE DAMAGE

The fatigue life may be calculated based on the S-N fatigue approach under the assumption of linear cumulative damage (Palmgrens-Miner rule).

When the long-term stress range distribution is expressed by a stress histogram, consisting of a convenient number of constant amplitude stress range blocks $\Delta\sigma_i$ each with a number of stress repetitions n_i the fatigue criterion reads;

$$D = \sum_{i=1}^{k} \frac{n_i}{N_i} = \frac{1}{\bar{a}} \sum_{i=1}^{k} n_i \cdot \left(\Delta\sigma_i\right)^m \leq \eta \tag{1}$$

where,

D = accumulated fatigue damage
$\bar{a}$, m = S-N fatigue parameters
k = number of stress blocks
n_i = number of stress cycles in stress block i
N_i = number of cycles to failure at constant stress range $\Delta\sigma_i$
η = usage factor. Accepted usage factor is defined as $\eta = 1.0$

Applying a histogram to express the stress distribution, the number of stress blocks, k, is to be large enough to ensure reasonable numerical accuracy, and should not be less than 20. Due consideration should be given to selection of integration method as the position of the integration points may have a significant influence on the calculated fatigue life dependent on integration method.

When the long-term stress range distribution is defined applying Weibull distributions for the different load conditions, and a one-slope S-N curves is used, the fatigue damage is given by,

$$D = \frac{v_0 T_d}{\bar{a}} \sum_{n=1}^{N_{load}} p_n q_n \Gamma\left(1 + \frac{m}{h_n}\right) \leq \eta \tag{2}$$

where,

N_{load} = total number load conditions considered
p_n = fraction of design life in load condition n, $\Sigma p_n \leq 1$, but normally not less than 0.85
T_d = design life of ship in seconds (20 years = 6.3 × 108 sec.)
h_n = Weibull stress range shape distribution parameter for load condition n

$q_n =$ Weibull stress range scale distribution parameter for load condition n

$v_o =$ long-term average response zero-crossing frequency

$\Gamma\left(1+\frac{m}{h_n}\right) =$ gamma function.

The Weibull scale parameter is defined from the stress range level, $\Delta\sigma_o$, as,

$$q_n = \frac{\Delta\sigma_0}{\left(\ln n_0\right)^{1/h_n}} \tag{3}$$

where n_o is the number of cycles over the time period for which the stress range level $\Delta\sigma_o$ is defined. ($\Delta\sigma_o$ includes mean stress effect) the zero-crossing-frequency may be taken as,

$$v_o = \frac{1}{4 \cdot \log_{10}(L)} \tag{4}$$

where L is the ship rule length in meters.

Alternatively, in combination with calculation of stress range $\Delta\sigma_o$ by direct analyses, the average zero-crossing-frequency.

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, and a one-slope S-N curve is used, the fatigue criterion reads as,

$$D = \frac{v_o T_d}{\bar{a}} \Gamma\left(1+\frac{m}{2}\right) \sum_{n=1}^{N_{load}} p_n \cdot \sum_{i=1,\, j=1}^{\substack{\text{all seastates} \\ \text{all headings}}} r_{ijn} \left(2\sqrt{2m_{0ijn}}\right)^m \leq \eta \tag{5}$$

where,

$r_{ij} =$ the relative number of stress cycles in short-term condition i, j

$v_o =$ long-term average response zero-crossing-frequency

$m_{oij} =$ zero spectral moment of stress response process

The Gamma function, $\Gamma\left(1+\frac{m}{2}\right)$ is equal to 1.33 for m = 3.0.

III. S-N CURVES

The fatigue design is based on use of S-N curves which are obtained from fatigue tests. The design S-N curves which follow are based on the mean-minus-two-standard-deviation curves for relevant experimental data. The S-N curves are thus associated with a 97.6% probability of survival.

The S-N curves are applicable for normal and high strength steels used in construction of hull structures.

The basic design S-N curve is given as,

Table 1. S-N Curve with air or with cathodic protection.

S-N Curve	Material	$N \leq 10^7$		$N > 10^7$	
		$\log\bar{a}$	m	$\log\bar{a}$	m
I	Welded joint	12.65	3.0	16.42	5.0
III	Base Material	12.89	3.0	16.81	5.0

Table 2. S-N Curve with corrosive environment.

S-N Curve	Material	$\log\bar{a}$	m
II	Welded joint	12.38	3.0
IV	Base material	12.62	3.0

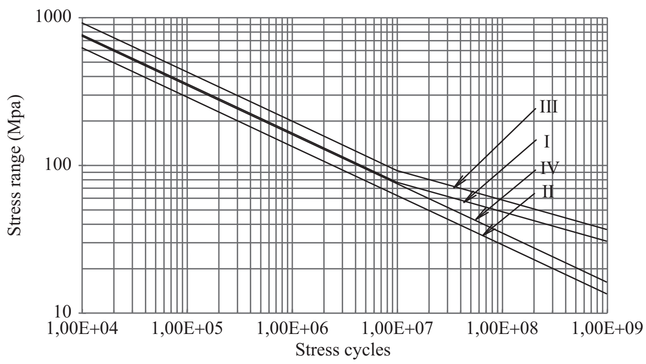

Fig. 1. Design S-N Curves.

$$\log N = \log\bar{a} - m\log\Delta\sigma \tag{6}$$

N = predicted number of cycles to failure for stress range $\Delta\sigma$

$\Delta\sigma =$ stress range

m = negative inverse slope of S-N curve

$\log\bar{a} =$ intercept of logN-axis by S-N curve

$$\log\bar{a} = \log a - 2s$$

where,

$a =$ is constant relating to mean S-N curve

$s =$ standard deviation of log N

$s =$ 0.20

In combination with the fatigue damage criteria given in Table 1 and Table 2. (Fig. 1).

The fatigue strength of welded joints is to some extent dependent on plate thickness and on the stress gradient over the thickness. Thus for thickness larger than 22 mm, the S-N curve in air reads as,

$$\log N = \log\bar{a} - \frac{m}{4}\log\left(\frac{t}{22}\right) - m\log\Delta\sigma \tag{7}$$

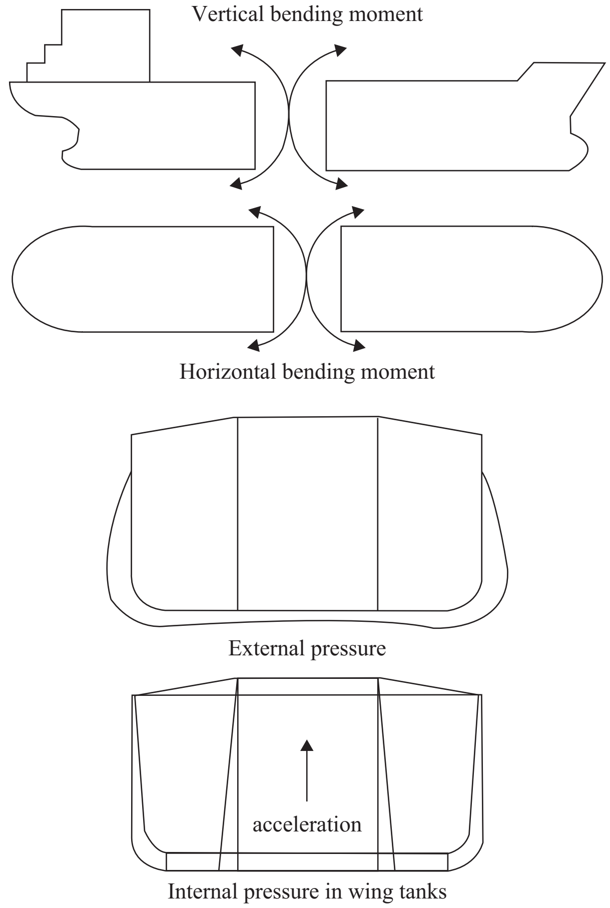

Fig. 2. Typical loads used in Fatigue for loaded and ballast condition.

where t is thickness (mm) through which the potential fatigue crack will grow.

IV. FATIGUE CALCULATIONS

Fatigue calculations are performed in accordance with DNV Classification Note 30.7: "Fatigue Assessment of Ship Structures". Cross section properties and relative deflections are calculated by FE-analysis. A brief explanation of the calculation procedure is given as follows.

The loads consist of:

(1) global vertical bending moment,
(2) global horizontal bending moment,
(3) internal pressure based on accelerations in vertical, transverse and longitudinal direction,
(4) external pressure from waves.

These loads are schematically presented in Fig. 2.

The calculations are performed by use of parametric formulas based on the DNV Ship Rules. These are found from DNV Classification Note 30.7 (DNV, 2012). Two or three loading conditions may be included such as typically fully loaded and ballast condition.

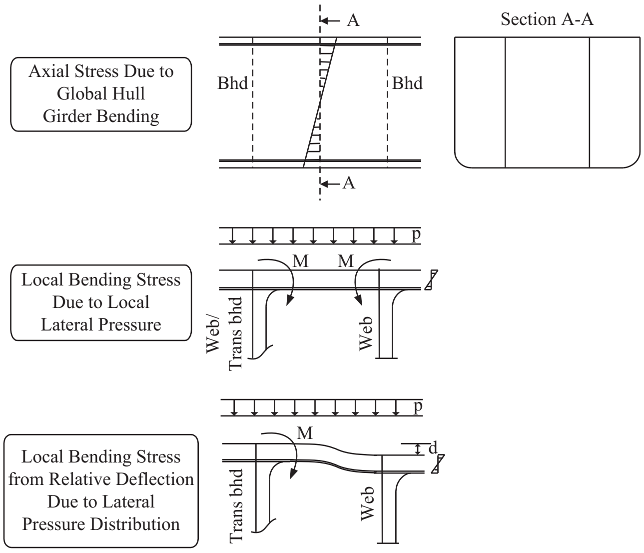

Fig. 3. Stress contributions in double side/bottom from different loads.

The loads give the following stress contributions:

(1) Axial stress due to global bending moments.
(2) Local bending stress of stiffeners due to local lateral pressure.
(3) Relative deflection (deflection between transverse bulkhead and adjacent frame) due to lateral pressure distribution.
(4) Axial stress from bottom/side bending of longitudinal girders/stringers due to lateral pressure distribution (Not shown below or used in the fatigue calculations).

These stress contributions are schematically presented in Fig. 3.

Effective flanges of stiffeners are accounted for. The stress components are combined using correlation coefficients in order to take phase relations between the different loads into account. The coefficients are dependent on which loads that are combined and the location of the stiffener (DNV, 2014).

Stress concentration factors (SCF) are calculated by parametric formulas or taken from tables using reference of CN 30.7 (DNV, 2012), which include typical transition details. The total SCF, K, is typically determined in the following way:

$$K = K_g \cdot K_w \cdot K_n \cdot K_e \tag{8}$$

where,

K_g = Geometric SCF due to geometry of the detail.
K_w = SCF due to the presence of a weld ≈ 1.0 as default.
K_n = SCF due to skew bending of an L-profile (by parametric formula).
K_e = SCF due to an eccentricity, e.g., an overlap.

The K_g depends on the type of loading, hence, K_g from axial tension/compression may be different compared to K_g from lateral pressure or relative deflection. K_n is only included for

Table 3. Assumptions and simplifications.

Assumption or simplification	Comments
World-wide trade has been assumed.	The fatigue life will improve if the vessel has a trading pattern in ocean areas with less severe wave conditions. Vessels trading in the North Atlantic only will have lower fatigue life than calculated in this analysis.
Cargo tanks are assumed coated and ballast tanks are assumed coated.	Uncoated cargo tanks will give reduced fatigue life compared to coated tanks.
The time in different loading conditions is according to CN 30.7 and DNV Rules for ships Pt.3 Ch.1.	For the ship this implies; Full load: 45%, Ballast: 40%, Harbor: 15% (No fatigue damage is calculated in harbor condition)
Effective coating for a period of 15 years has been assumed both in the cargo tanks and in the ballast tanks.	SN-curve in air (non-corrosive) is used for the first 15 years. SN-curve in corrosive environment is used for the remaining life.
Relative deflection is neglected for a typical web-frame.	The influence from relative deflections between web-frames are assumed to be negligible.
Relative deflection is calculated using a 3-D FE-model of one cargo hold. Deflections are calculated for and all longitudinals in way of the transverse bulkhead.	Relative deflection is important for the stress magnitude in the stiffener connection to the transverse bulkhead.
The fatigue evaluation is based on the "net scantlings" dimensions. The calculations have been performed using the dimensions based on direct calculations (non-corrosion control dimensions on longitudinals), using thickness reductions as stated in the DNV Rules.	This will reduce the fatigue life compared to "as built" (original) scantlings slightly.
Only wave induced loads are included as contributing factors to fatigue. The non-linear splash zone (wet/dry zone) is accounted for.	Other dynamic loads, as loading/unloading, vibration, whipping may contribute to the fatigue damage at certain locations. The damage from these loads is not included.
Density of cargo is set to 1,602 tons/m^3 Density of water is set to 1,025 tons/m^3	These are the loads that the ship is intended for. Possible lower densities are not taken into account for. If the vessel during its lifetime has carried cargo with lower density than 1,062 e.g., 1,025 it would have a significant impact on calculated fatigue lives for longitudinals directly influenced by internal tank pressure. Fort his ship this means the inner side longitudinals.
The fatigue life is calculated based on the mean-minus-two-standard deviation SN-curves.	Details are associated with a 97.6% probability of survival based on test data (cracks are expected on 2-3% of similar details within the calculated fatigue life).

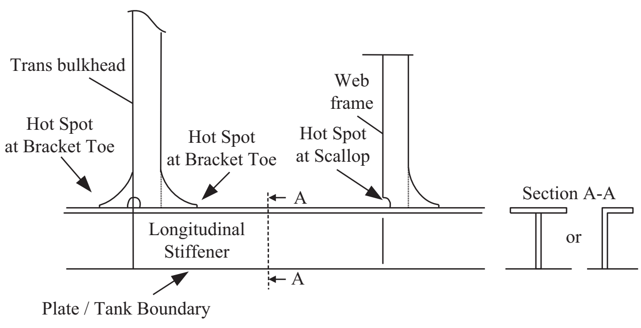

Fig. 4. Typical hot spots for longitudinals.

lateral pressure (but could also be relevant for axial tension in case of sniped stiffener flange).

The Nauticus fatigue program (DNV, 1999) calculates the fatigue life expectancies of hot spots (high stress concentration areas) on top of the stiffener flange. Typical locations are bracket toes and scallops as shown in Fig. 4.

Detailed description of the formulas and the calculation procedure is presented in CN 30.7 (DNV, 2012).

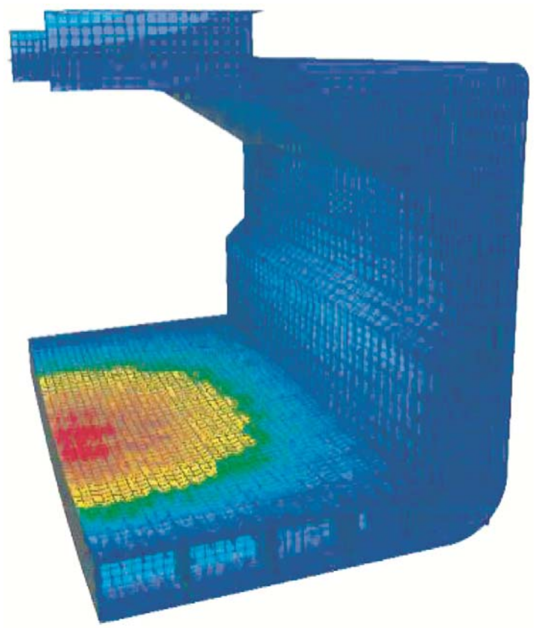

Fig. 5. Relative deflection along one hold, external pressure, fully loaded.

V. BASIC ASSUMPTIONS AND SIMPLIFICATIONS

The fatigue damage is a result of accumulated damage throughout the entire lifetime of the ship. This introduces uncertainties in the calculations as:

(1) Vessel trade (wave environment).
(2) Coating history, corrosion.
(3) Loading condition history (static and dynamic stresses).
(4) Type of cargo (sweet/sour oil, ballast water, chemical fluids).

Assumptions in the stress calculations themselves also have to be made in order to be able to perform the fatigue analysis within a reasonable scope/cost. The main assumptions used in the analysis are listed below together with an explanation of the implication of the assumption.

The fatigue life in this study is described as the crack initiation time (crack starting to grow perpendicular to the main stress direction). This is less than the time it takes for a crack in the weld toe (the SN-curves used are valid for the weld toes) to grow through the thickness of the material. The cracks may grow further before it is detected by visual inspection, typically around 100-200 mm. (Table 3).

VI. RELATIVE DEFLECTION CALCULATION

Additional stress caused by relative deflection may be an important contributor to the total stress for longitudinals connected to transverse bulkheads. Relative deflections are assumed to have a significant influence on the stress level and thus also on the fatigue life. In order to calculate the stresses at a satisfactory level of accuracy, a 3-D FE-model of one cargo tank in the midship area is used to calculate the relative deflections. Rule loads for internal and external dynamic pressures are applied to the model. The relative deflection values are introduced to the transverse bulkhead frames only. The model, with deflections according to external hydrodynamic pressure distribution for fully loaded condition, is presented in Fig. 5. It should be noted that the deflection magnitude is increased so as to visualize the effect.

Table 4. Bulk Carrier Vessel Main Dimensions.

Item	Value	
Length over all, L_{oa}	230.00	m
Length between perpendiculars, L_{bp}	218.00	m
Breadth moulded, B	32.2	m
Depth moulded, D	18.2	m
Draught design, T	12.2	m

Table 5. Analyzed Loading Conditions.

Loading Condition	Draught [m]	GM [m]
LC 1 (loaded)	12.66	1.52
LC 2 (ballast)	7.26	5.56

VII. CASE STUDY

In order to determine a vessel's fatigue life expectancy, the vessels trading pattern throughout its lifetime will decide the dimensioning environmental parameters to be used in the fatigue analysis. This will differ from vessel to vessel and all modifications in design must be encountered for to establish the residual lifetime.

The method described and used in this fatigue analyses is calculating the life time expectancy as from when the vessel was delivered from the yard (irrespective of possible design modifications throughout the years) and assumes 20 years worldwide operation (irrespective of the actual trading pattern).

By this approach the different vessels may be compared as new-buildings with respect to anticipated fatigue life. This is performed through fatigue analysis of the longitudinal material amidships.

The calculations have been carried out for a tanker with main characteristics as presented in Table 4.

The loading conditions used are presented in Table 5.

VIII. CALCULATED FATIGUE LIVES

The results of the fatigue analysis are provided. The calculated fatigue life expectancies are presented with color boxes symbolizing different fatigue life intervals.

The calculated fatigue life expectancies for a typical frame in the midship area are presented in Fig. 6. The actual calculations are performed for frame #130.

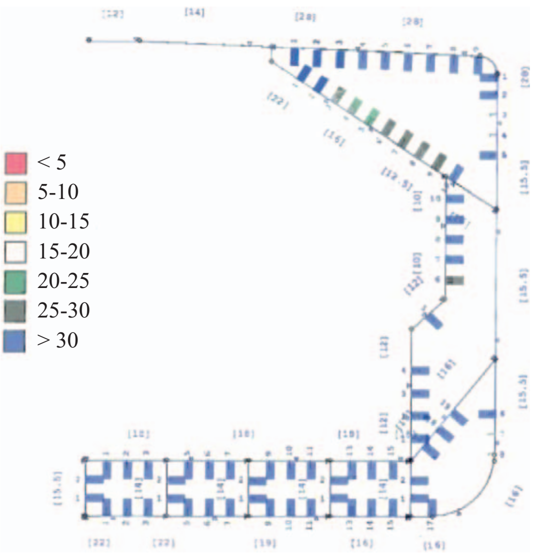

Fig. 6. Calculated fatigue life (years) expectancies of longitudinal stiffener end connections at a typical frame in the midship area.

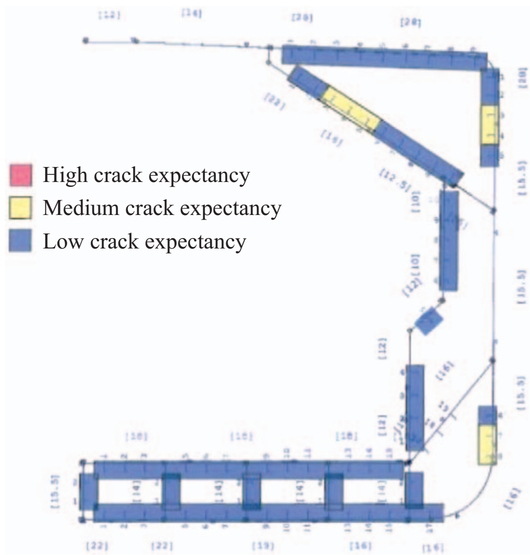

Fig. 8. Inspection guidance for longitudinal stiffener end connections at ordinary frames in the midship area.

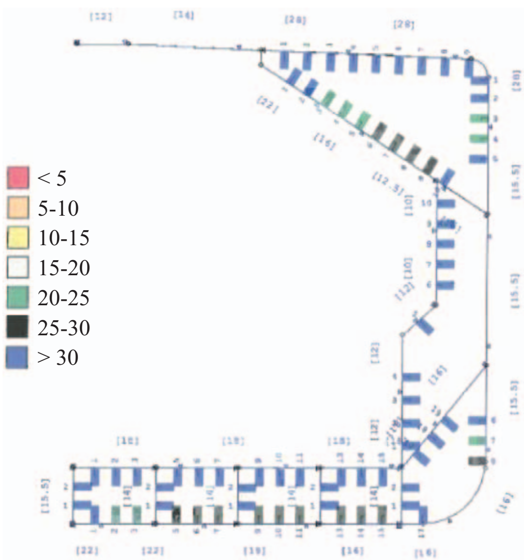

Fig. 7. Calculated fatigue life (years) expectancies of longitudinal stiffener end connections at a transverse bulkhead.

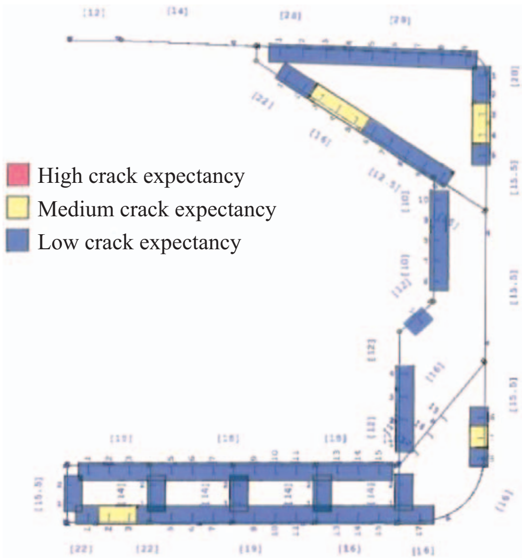

Fig. 9. Inspection guidance for longitudinal stiffener end connections at bulkheads in the midship area.

The calculated fatigue life expectancies for a typical bulkhead in the midship area are demonstrated in Fig. 7. The actual calculations are performed for the bulkhead at frame #135.

IX. AREAS OF POSSIBLE CONCERN

The calculated fatigue life expectancies as presented here are based on the assumptions given in Table 1. The correlation between these assumptions and the actual conditions for the vessel will influence the actual fatigue life of the details. The fatigue lives should therefore not be used as exact number, but more as indications of which end connections that are most vulnerable to fatigue cracks.

This part therefore gives advice regarding which areas that should be given special attention during the close-up inspections. The advice is based on the calculated fatigue life expectancies reported here and comparison between calculated fatigue life expectancies and experience from inspections on similar ships. Based on this finding, stiffener end connections are sorted in three categories as follows:

(1) High crack expectancy
Given for end connections with calculated fatigue life below 10 years
(2) Medium crack expectancy
Given for end connections with calculated fatigue life between 10 to 25 years

(3) Low crack expectancy
Given for end connections with calculated fatigue life above 25 years

The categorization as shown in Figs. 8 and 9 should be used as basis for selection of end connections where the close-up inspections should be performed, such that this is included in the inspection planning. However, if cracks are found for other end connections, the inspection plan should be changed to allow for more extensive inspection of similar connections.

The different categories for a typical frame are presented in Fig. 8 as below.

The different categories for a typical bulkhead are presented in Fig. 9 as follows.

X. FATIGUE RE-DESIGN ANALYSIS

If the calculated fatigue lives are less than acceptable, the following examples provide guidance for improving the fatigue capacity.

Example 1: Deck and Bottom Area

If the calculated fatigue life for a significant number of details is too low, then increase of the gross scantlings (plate thickness) may be considered. Alternatively, local geometry may be improved in order to reduce stress concentration factors. Although typically it is not accepted during the design phase, fatigue life can be improved by grinding or hammer peening.

Example 2: Side Shell

If the calculated fatigue life for a significant number of connections in the side shell is too low, then the bracket sizes may be softened and scantling increased. However, it is important to note that increasing the bracket size for areas subject to relative deflection stresses will actually degrade the fatigue performance.

As an alternative the size of the longitudinal stiffener may also be increased.

If the calculated fatigue life for a significant number of lug connections is too low, then the design should be modified to improve the fatigue capacity.

Example 3: Base Material

If the calculated fatigue damage in the base metal is larger than acceptable, the following options may be considered:

(1) Reduction of maximum stress by means of increased plate thickness or reduced stress concentration factor,
(2) Grinding the edge and application of a durable corrosion protection to improve the S-N curve.

XI. CONCLUSION

This paper describes a fatigue analysis of the bulk carrier as a case study, which is based on DNV Classification Note 30.7: "*Fatigue Assessment of Ship Structures*" (DNV, 2012).

The analysis comprises a check of the longitudinal material amidships, and is based on the "as built" drawings applying "net" scantlings.

The fatigue analysis is in accordance with BP/AMOCO`S requirements. Based on the fatigue calculations, the following guidance is provided for the planning of the close-up inspection.

The study identified longitudinal members with potential fatigue problems. By focusing the inspection on these areas, the inspection can be carried out more efficiently, and the probability of detecting damages will increase. It will also make easier to plan the maintenance of the vessel.

It should be noted that the industrial experiences indicate that if the results show "high crack expectancy" on the upper part of longitudinal bulkhead and/or inner side then the "high crack expectancy" area should be marked with the comment and changed to "medium crack expectancy".

The calculated fatigue life expectancies are presented with color boxes symbolizing different fatigue life intervals. Those simplified fatigue calculations have been carried out to check the situation. Table 6 shows the impact on different groups of stiffeners.

Table 6. The summary of findings for different groups of stiffeners.

Position of longitudinal	Comments
Outer bottom, void tank	Low fatigue risk
Outer bottom, water ballast tank	Low fatigue risk
Outer side, water ballast tank	Medium fatigue risk
Outer side, top side tank	Low fatigue risk
Deck	Low fatigue risk
Inner bottom, void tank	Low fatigue risk
Inner bottom, water ballast tank	Low fatigue risk
Bottom of top side tank	Low or medium fatigue risk
On double bottom longitudinal girders	Low fatigue risk

Further it may be drawn 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 approximately 3-4 m below due to splash effect. It should be noted if the ship has big differences in draught between full load and ballast conditions, the whole ship side is equally important to survey.

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