In proceedings of "Design and Operation of Gas Carriers",
The Royal Institution of Naval Architects - RINA, September 2004, London, UK

FATIGUE ANALYSIS OF GAS CARRIERS

T. Lindemark, F. Kamsvåg and S. Valsgård, Det Norske Veritas, Oslo

SUMMARY

The paper describes the thinking behind Det Norske Veritas (DNV) ship hull analysis procedures from the 1950'ies and onwards. The focus is on the latest development in computer based design analyses, from hydrodynamic analysis through structural analyses, to strength and acceptance criteria. The current DNV fatigue analysis procedures are illustrated with examples from standard size (138 000 - 140 000 $m^{3)}$ membrane type LNG carriers, spherical type carriers and a CNG carrier.

Hydrodynamic load analysis options and modelling principles are described and examples given for various applications. Structural Finite Element (FE) modelling are made of the whole vessel or parts of the vessel depending on vessel type and which type of analysis are to be carried out. Spherical carriers and open deck type CNG carriers will benefit from analysis using a full global model in order to describe the torsional and warping response of the hull and the restraint offered from the tank covers, whereas closed type vessels like membrane carriers usually have been analysed with a model covering the midship area only.

Automatic load transfer is used when possible to transfer the loads from the hydrodynamic models to the structural FE model. This usually demands a full global structural model to ascertain that all loads are automatically in balance. However, methods are also available enabling automatic load transfer also to midship FE models. The ability to transfer loads automatically to the structural models is a prerequisite for doing full, stochastic (spectral) fatigue analyses.

Fatigue analysis procedures are discussed from the most simplified alternatives to the full stochastic procedures including stochastic screening of fatigue prone areas. Finally, probabilistic simulations are used to illustrate the importance of refining the fatigue analysis method in order to minimise the expected number of fatigue failures during the lifetime of the ship.

1. INTRODUCTION

The existing DNV ✠1A1- rules require a certain amount of direct strength calculations in addition to explicit formula requirements. Vessels with closed cross-sections as Tankers, Bulk Carriers and membrane type LNG carriers can in most cases be analysed using midship models only, whereas open type ships like Container vessels, Open Hatch Cargo Carriers, Spherical tank LNG carriers and open deck CNG carrier designs in principle need a full model in order to properly describe their torsional and warping response. There is a growing realisation / need in the market among operators and underwriters to have visible proof that the ships are built using state-of-the art technology and that fatigue evaluations have been carried out in accordance with the intended trade of the vessel.

In addition to traditional Ultimate Limit State (ULS) design criteria, yield and buckling, *fatigue endurance* (Fatigue Limit State design - FLS) has become an increasingly important design parameter. As compared to a vessel engaged in world-wide trading, operation in harsher environments like the North Atlantic or the North Sea is more demanding in terms of fatigue damage. As a rule of thumb these areas will in average give twice as much fatigue damage to the vessel as compared to a ship engaged in world-wide operation. Other areas, like the US west coast to Alaska trade, may be even worse. Hence, careful local design of fatigue prone areas and details has become exceedingly important. Details that have been known to perform satisfactorily in world-wide operation will not necessarily be up to expectations in more severe environments.

The objective of the present paper is to outline computerised fatigue analysis procedures as applied to modern gas carrier designs.

1.1 RULE DEVELOPMENT

Seen in a long term perspective the development of the ship design rules of the Classification Societies has been one of a slow, experience based evolution with some leaps, or rapid development phases in between. Up to the 1950's the rules of Det Norske Veritas (DNV) were, as those of most classification societies, experienced based in which requirements for structural scantlings were mostly given in tabular form.

The earlier most important leaps were based on the introduction of, and implementation of *new technology*, i.e.:

- Scientifically based rules in the 1950's
- Direct calculations of hydrodynamic loads and stress analysis with Finite Element Methods (FEM) in the late 1960's and early 1970's

Today the full potential of modern computer technology and information systems is pursued in order to streamline information handling. This includes design and analysis procedures, the approval procedures and the day-to-day operations. The Nauticus software is used as the DNV Life-Cycle class production system designed to take care of all pertinent ship information from the concept (pre-contract) stage, through plan approval, the construction and operation phases and to scrapping of the vessel. Nauticus Hull is a computer system for ship hull design and analyses, and is an important building block in this philosophy. A basic concept is that a full geometrical model of the ship is made once. From this geometry model of the ship Finite Element models are generated for use in plan approval and detail analysis of the ship hull. Further, and most important, the geometrical model is used as a graphical information carrier for all relevant information about the ship.

1.2 SHIP HULL ANALYSIS

The ships rules of most Class Societies have been based on putting acceptance levels on isolated stress components. Usually it is distinguished between stresses from hull girder bending (*primary stresses*), stresses from bending of frame and girder systems (*secondary stresses*) and stresses from bending of stiffeners and unstiffened plates (*tertiary stresses*). The two last components are mainly caused by the action of lateral loads, external and internal liquid pressures and loads from cargo and equipment.

This approach has been necessary and very useful from the 1950's and onwards when the rules were first written into an analytical and more scientifically form. Simple analytical formulas have been used to determine requirements to plates, stiffeners and girder systems. These are based on defined lateral loads acting on the various components and defined stress acceptance levels (fractions of yield in tension and simplified buckling criteria) for the various stress components and elements.

When the use of computers picked up momentum the simplified girder/frame requirements were gradually

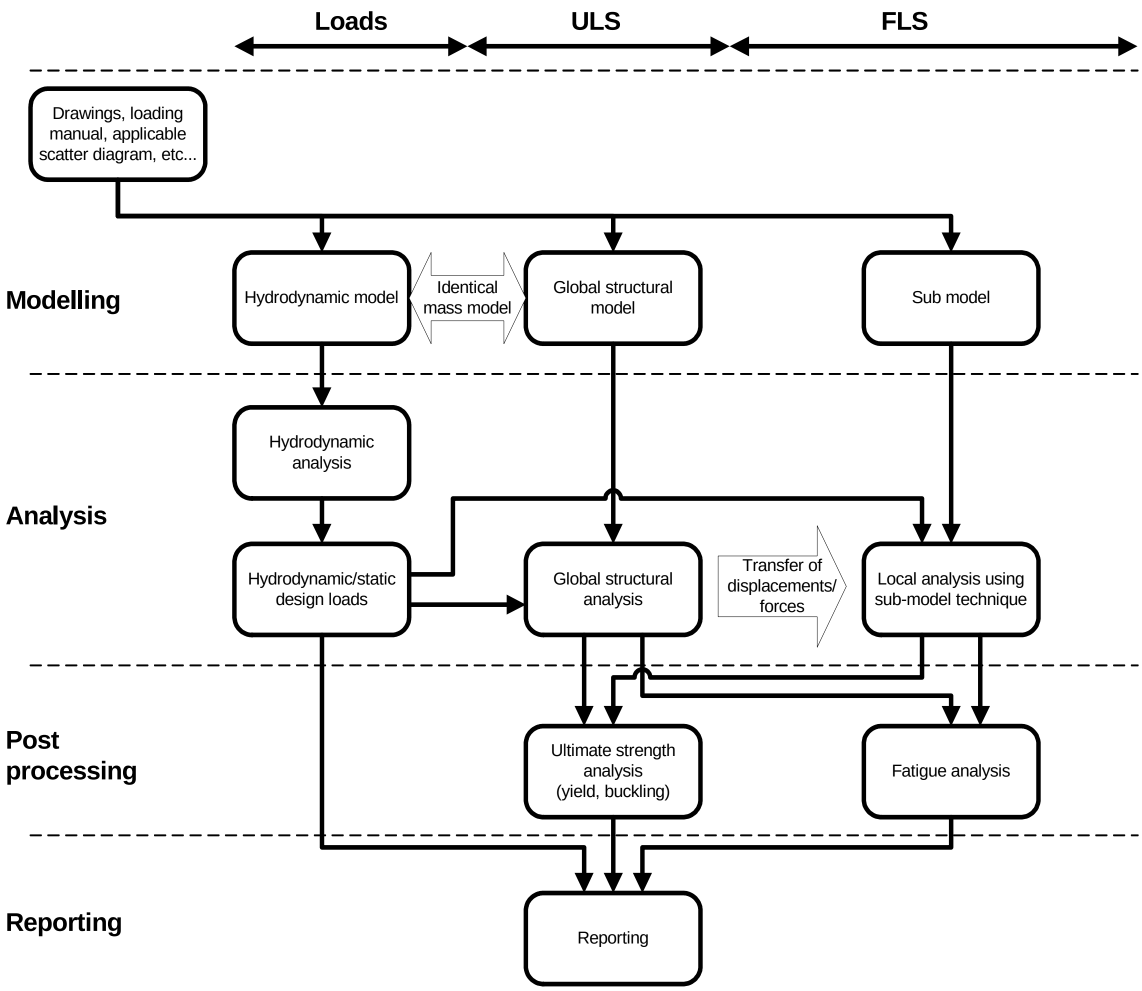


Fig. 1 - Computational Ship Analysis Flow Chart

taken out of the rules and substituted with 2-D and 3-D beam analysis. This practice was in DNV checked out by extensive full scale measurements on the tanker Esso Norway during 1970-71. Extensive computer modelling, direct calculation of hydrodynamic loads and Finite Element Stress response analysis were carried out and comparisons made with the measurement results. The comparisons confirmed that the analysis procedures used were working and based on a sound foundation, [1].

Today extensive calculations are carried out for special ship types and for looking into special problem areas. The DNV rules in theory also accept designs based on direct probabilistic analysis.

Fig. 1 illustrates the main steps in a in a direct Computational Ship Analysis procedure.

Before the early 1990's fatigue was only indirectly considered in the ship hull design by a material factor f_1 that only allowed for a certain fraction of the difference between the yield strengths of High Tensile Steel (HTS) and Mild Steel (MS) to be utilised. This was done in order to keep the general stress level down in a bid to counteract fatigue damage assuming good shipbuilding practice for the structural details. Then, in 1991 a requirement for direct fatigue control of side longitudinals was included in the DNV rules. With this it became possible to quantify the fatigue performance of the actual details.

A comprehensive Classification note 30.7 *for fatigue analysis* of ship hulls were developed throughout the 1990's and thoroughly tested with the major shipyards, [2]. The fatigue analysis approach herein uses *principal stresses* and defines standard calculation procedures which can be used for all kinds of ships. Four sets of S/N fatigue curves are used applicable for standard welded and unwelded material in corrosive and non-corrosive environment. The local notch stress is determined by stress concentration factors applied on the nominal stresses. This allows for a continuous evaluation of the fatigue capacity for incremental changes in the local geometry. Credit may also be given for higher standards of workmanship.

The long term distribution of the stress range response to be applied in the fatigue assessment can be determined at various complexity levels. The simplest method is to estimate the reference stress range at a defined occurrence rate from rule formulas for the wave induced loading and to combine this with simplified expressions for the shape of the stress range distribution. The more accurate method is to carry out a direct long term load and stress frequency response analysis, Fig. 1.

2. GAS CARRIER DEVELOPMENTS

From the very beginning of the gas carrier industry great care has been taken to include all relevant failure modes in the design of the tank systems and fatigue has been one of the most important design parameters.

The seaborne transport of liquefied gases in bulk is older than often realised. Already in 1949 the first dedicated liquefied gas carrier was delivered with DNV class. This was a vessel with fully pressurised cargo tanks for transport of LPG/Ammonia. The vessel, named Herøya, had vertical cylindrical tanks and was built at the Horten Navy Shipyard in Norway. DNV, therefore, became involved very early in the setting of safety standards for these types of vessels, and was the first of the classification societies to publish comprehensive rules for gas carriers. This was in 1962.

A research team on LNG was established in DNV in 1959. A membrane tank system was developed and tested successfully in 1962. The system used double corrugated aluminium sheets as the primary barrier. This system was later taken over and further developed by Technigaz in France.

The Moss spherical tank design was developed by the Kvaerner Group in Norway during 1969-1972. Basic design criteria for type B tank were formulated by DNV in 1972 rules. In order to confirm compliance with the design criteria comprehensive R&D programmes were carried out in DNV, e.g. sloshing loads from liquid movement inside the cargo tanks, crack propagation, fatigue characteristics and buckling strength.

The idea of shipping gas on keel without a costly liquefaction process is equally old as the LNG industry, but have until recently been no success due to the heavy gas containment systems if the tanks (pipes) were to be designed according to conventional pressure vessel codes or the international Gas Code (IGC). This leads to heavy containments systems with virtually no lifting capacity left for cargo unless unreasonably large and costly ships were to be used.

Most CNG concepts apply high pressure (130-275 bars) in a semi-chilled or at ambient condition in order to keep the gas in a gaseous state with basically no liquid hydrate fall-out. CNG tanks are mostly based on the use of cylindrical bottles or pipes with diameters up to 48 inch being designed according to modern limit state pipeline or pressure vessel codes. For such tanks *fatigue becomes the driving design parameter*, ref [3].

2.1 GAS CARRIER OPERATIONS

For long distance seaborne transportation of large volumes of natural gas LNG represents today the most efficient commercial alternative. For large volume, shorter distances pipelines will be the obvious choice.

The cost of a standard size carrier of 135 000 - 140 000 m^3 has during the last decade dropped to about 150-170 MUS$ but is presently on an upward trend. The spot trading market for LNG carriers has been increasing. This has led to speculative ordering of new LNG carriers.

Several of the oil majors are going into LNG-shipping and plans to use increased size carriers up towards 250 000 m^3. Due to draught restrictions the large carriers may be wider than conventional carriers and require new design solutions with respect to hull form, propulsion efficiency and manoeuvring capability.

Most LNG carriers have so far been operating on eastbound trades from the Middle East to Japan/Korea. This is an area with a reasonably benign wave climate. With increased gas import to the US and Europe we see increased cross Atlantic trading which will require carriers capable of operation in more demanding environmental conditions. *Fatigue* considerations and tank *sloshing loads* will be more important design parameters. For Atlantic trading owners today most commonly specify 40 years fatigue life in the North Atlantic.

In the aftermath of the September 11th 2001 it is becoming more difficult to get permission to build new land based LNG receiving terminals, particularly in the US. This will require offshore discharge terminals where safe off-loading can take place far from densely populated areas and busy ports and estuaries. The gas can be either discharged as liquid LNG/LPG into a floating receiving barge or as gas from a *regasification* plant onboard the gas carrier via offshore buoys (STL) or a platform to shore ready for distribution into the onshore gas grid. This will require offloading carriers capable of staying at the discharge buoy for up to a week at the time in order to fully empty the vessel. This requires carriers able to operate without tank filling restrictions at the discharge location in order to avoid the risk of sloshing damages inside the cargo tanks.

The Compressed Natural Gas (CNG) technology offers interesting possibilities for handling of associated gas and for exploitation of marginal gas fields (stranded gas). The system does not require a gas liquefaction plant and LNG storage tanks, nor will LNG storage and regasification at the discharge location be necessary. A fleet of CNG ships will serve as both storage and transport vehicles and can discharge directly into the land based gas grid via an on/offshore discharge terminal, an offshore platform or offshore buoys. CNG carriers will come in as an economically attractive supplement in the transport range between pipelines and LNG transport (500 – 3000 nautical miles), [3].

The coming larger sizes of carriers and the new operational profiles outlined here make relying on past experience for fatigue performance of the vessel hulls and containment systems rather uncertain. *Hence, the use of state-of-the-art design for ultimate strength and fatigue will be essential for safe and trouble free operation.*

3. RULE FATIGUE REQUIREMENTS

In 1991 direct fatigue control requirements to side longitudinals, main frames and tween deck frames was included in the DNV Rules for Ships with length >100 metres. This is formulated as a minimum section modulus requirement as a function of the dynamic bending moment resulting from local pressures and stress concentration factors for various end connection details.

In 1995 DNV issued the Class Note 30.7 "Fatigue Assessment of Ship Structures describing fatigue standard fatigue analyses procedures ranging from a simplified method for fatigue calculation of stiffener end connections to full stochastic analysis based on fatigue damage calculations through statistical post processing of direct calculated loads and stresses. The fatigue life is calculated based on the S-N fatigue approach under the assumption of linear cumulative damage (Miner-Palmgren's rule). The simplified approach was made mandatory for vessels with the additional (voluntary) class notation CSA-1 and the direct long term load and stress frequency domain analysis was required for vessels with the additional class notation CSA-2. In addition to end connection of longitudinals, the CSA-2 notation also include fatigue calculations of plating and highly stressed structural details in the cargo area such as panel knuckles, bracket and flange terminations of the main girder system and hatch corners.

In 1998 the class notation NAUTICUS(Newbuilding) replaced the CSA-1 notation, and simplified fatigue calculations of longitudinals were made mandatory for tank, bulk and container vessels with length >190 metres. Fatigue strength calculation of the hopper knuckle was made mandatory for oil tankers.

In 1999 DNV introduced the additional class notations PLUS-1 and PLUS-2 (voluntary). The class notations require fatigue calculations for an extended number of cyclic loads (30 years in world wide operation for PLUS-1 and 40 years in world wide operation for PLUS-2) and additional details of stiffener end connections (web stiffener, cut outs and collar plate), main frames and deck openings.

4. FATIGUE ANALYSIS OPTIONS

The fatigue analysis options described in the DNV Classification Note No. 30.7 "Fatigue Assessment of Ship Structures" [2] are:

- Simplified fatigue calculations
- Stress component stochastic fatigue analysisi
- Full stochastic analysis

4.1 SIMPLIFIED FATIGUE CALCULATIONS

4.1 (a) NAUTICUS(Newbuilding) Fatigue Analysis

The simplified fatigue calculation method, mandatory for the class notation NAUTICUS(Newbuilding), is commonly used for fatigue calculations of longitudinals, butt joints and hopper knuckles. The method is in general suitable for structural members where the total stress level on a simple basis can be expressed as a sum of individual stress components due to global wave bending moments, external sea pressures and internal tank pressures.

Rule formulas are used to calculate dynamic wave loads including global wave bending moments, external sea pressure and internal tank pressure at a 10^{-4} probability level of exceedance. Only basic ship parameters and drawings are required as input, and the calculations can be carried out at and early stage in the design process.

The stress response may be calculated by simplified formulas developed from beam theory combined with tabulated values of stress concentration factors. Alternatively, if the stress concentration factors are not known, stresses may be calculated based on detailed finite element analyses. Stresses in longitudinals due to relative deflections between transverse supports (at transverse bulkheads) are calculated based on a cargo hold FE analysis.

The combined stress response is calculated through a square root summation using simplified formulas for the average correlation between external sea pressure, internal tank pressure and global wave bending moments.

The long term stress range distribution is defined using simplified formulas to calculate the Weibull stress range shape distribution parameter and the long term average zero crossing frequency.

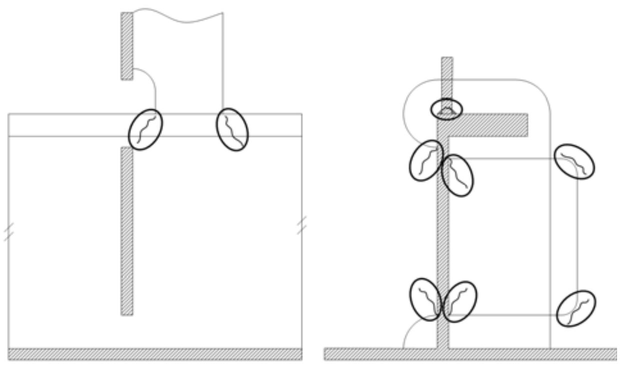


Fig. 2 – Hot Spots Considered in Traditional Fatigue Approach and PLUS-1&2 calculations

4.1 (b) PLUS-1 or 2 Fatigue Analysis

The traditional ship fatigue approach focus on fatigue cracks that are initiated at bracket toes and propagate through stiffener webs. The fatigue strength of these details is controlled by the section modulus of the stiffener and the stress concentration factor related to the bracket geometry. In areas with relatively high girder shear stresses, hot spots located at lugs and web openings may however be more critical. This is reflected in the scope of the class notations PLUS-1 and PLUS-2. Typical hot spots that are investigated are shown in fig. 2.

The calculations are based on simplified calculations of loads combined with finite element analyses to determine the stress response.

4.2 STRESS COMPONENT STOCHASTIC FATIGUE ANALYSIS

Stress component stochastic fatigue analysis are required for fatigue strength of longitudinals and plating for vessels with the additional class notation CSA-2.

In the stress component stochastic fatigue method the formulas for calculation of loads and correlation factors used in the simplified method are substituted by a linear complex summation of stress transfer functions and statistical post processing using wave climate data. The simultaneous occurrence of the different load effects is thus preserved through the calculations. The method is suitable for the same type of connections as the simplified method.

The combined stress response is expressed as:

$$H_\sigma(\omega \mid \theta) = \sum A_{\mathbf{k}} H_{\mathbf{k}}(\omega \mid \theta)$$

where

$H_\sigma(\omega \mid \theta)$ = transfer function for total combined stress

$A_{\mathbf{k}}$ = stress response factor for the considered detail subjected to a unit load for component k

$H_{\mathbf{k}}(\omega \mid \theta)$ = transfer function for the load component k

The load components include hull girder sectional loads, sea pressures and tank pressures. The stress response factors A_k may be determined by simplified formulas or finite element analyses.

The transfer function for the combined stress response, the wave climate (scatter diagram, spectrum, spreading and heading profile) and S-N data serves as basis for the statistical post processing and fatigue damage calculation. The fatigue damage for each loading conditions is calculated as a sum of part fatigue damages for each cell in the scatter diagram using a Rayleigh distribution within each short term condition.

4.3 FULL STOCHASTIC FATIGUE ANALYSIS

Full stochastic fatigue analysis is carried out for specific details of the hull structure. The locations to be considered are decided based on the ship type experience, review of drawings and/or fatigue screening of the global model.

The calculations are based on direct load transfer from the wave load analysis to finite element models of the vessel and employ both global models to determine nominal stresses and deflections and local stress concentration models to determine hot spot stresses. All load effects are taken care of, and hence the method can be used for any type of structure. Full stochastic fatigue calculations are typically used for members with a relatively complex stress response such as hatch corners, discontinuous panel knuckles, tank covers and stiffeners subjected to large relative deformations. The method is also valuable for identifying fatigue prone areas through fatigue screening of the models.

The loads from the hydrodynamic analysis including hydrodynamic sea pressure, internal tank pressures and inertia loads are automatically transferred to the finite element models by the wave load program. Typically 12 headings and 20-25 wave periods for each heading are considered, resulting in 240 to 300 complex load cases.

The finite element analysis is run for all load cases to determine the stress transfer functions at each element of the model, $H_\sigma(\omega|\theta)$, which expresses the relationship between the stress response and the wave frequency (ω) and heading (θ).

As for the stress component stochastic fatigue the transfer functions, wave climate and S-N data are used to calculate the fatigue damage as a sum of part damages for each cell in the scatter diagram. Additional stress concentration effects which are not included in the finite element model, e.g. welding and misalignment, may be included.

4.3 (a) Global Analysis

The global model is a relatively coarse FE model of the entire vessel used to calculated nominal stresses and deflections in the main structure. Plating is represented by shell elements and stiffeners are represented by beam elements. In the midship region the mesh density is typically one element between longitudinals, four elements between web frames and three elements over the height of girder webs. A coarser mesh is used in the fore and aft body to limit the size of the model; typically three to four stiffeners are lumped together and one element is used between web frames. The total size of the models may be up to 70,000 nodes/420,000 degrees of freedom.

The lightship weight is represented by material densities applied to the shell and beam elements. Heavy components such as machinery may be represented by point masses. The model is divided into a sufficient number of sections and the mass of each section is adjusted according to the actual weight distribution. By iteration the convergence of the displacement, longitudinal and vertical centre of gravity, shear force and bending moment distribution is checked for compliance with the loading manual. The deadweight is represented by internal tank pressures automatically transferred by the wave load programme.

A prerequisite for correct load transfer from the hydrodynamic program is there is sufficient compatibility between the hydrodynamic and the global model:

- The total mass and mass distribution is similar
- The total buoyancy and buoyancy distribution is similar

Similar mass properties are ensured using the structural model as mass model in the hydrodynamic analysis.

Having performed the load transfer the final load equilibrium is checked by comparing transfer functions and longitudinal distribution of bending moment and shear forces for different wave headings. Unbalanced forces will disturb the global response, and the final check is critical for the reliability of the results.

4.3 (b) Local Analysis

Local models are used as sub models to the global analysis and the displacements from the global analysis are automatically transferred to the local model as boundary displacements. In addition the local internal and external pressure loads and inertia loads are transferred from the wave load analysis.

From the local stress concentration models local geometric stress transfer functions at hot spots are determined. Element sizes in the order of the plate thickness are used for the details investigated to properly pick up the geometric stress increase.

5. WAVE LOAD ANALYSIS

5.1 CALCULATION PROCEDURE

The dimensioning stresses in a ship structure in a seaway are generally a complex combination of external and internal forces resulting from both global and local loads. When combining individual load components it is important to have information about their magnitude as well as their simultaneous occurrence. The total combined load is generally not the sum of the maximum value of individual load components. The resultant load giving rise to dimensioning stresses in the structure will be determined by combination of the following individual components:

- global hull girder bending
- internal and external pressure loads
- inertial loads from the hull, cargo and equipment

The relative importance of individual load components will vary between different ship types, location on the vessel and type of detail. The procedure for wave load analysis can be described in the following steps:

Fatigue and ultimate strength:
- Hydrodynamic modelling and calculation of motions
- Calculation of roll damping viscous forces
- Calculate transfer functions for 6 d.o.f. motions and global loads in selected sections

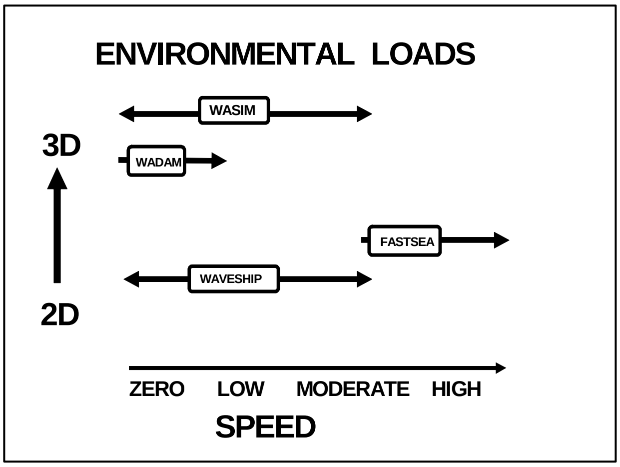
Fatigue:
- Transfer of pressures and accelerations to the structural model

Ultimate strength:
- Prediction of long term values (20 year return period)
- Determination of regular design wave (heading, height and period)
- Perform non-linear analysis for the selected design wave (based on regular waves or MLER [4] waves)
- Calculation of pressure distribution and accelerations for the design wave and transfer to structural model with included non-linear effects

Clearly, the use of a single static design wave to represent the long term design condition is a very substantial simplification. However, until it becomes possible to directly couple the hydrodynamic analysis with the structural model and run the analysis in the time domain, the design wave approach represents a reliable method for applying the loads to the structural model.

5.2 ANALYSIS OPTIONS

DNV pioneered the development of direct load computations some 30 years ago with the development of the Strip Theory and integration with structural analysis using finite element methods. Due to the complexity of the problem, prediction of the loads acting on the vessel in a seaway by direct computations has been one of the most uncertain areas in ship design analysis. Particularly this applies to predictions in extreme weather conditions as basis for ultimate strength analysis and loads due to above water geometry such as bow flare and other curvature at the ship ends. For a Classification Society it is important to have available a range of tools for wave load prediction covering the large number of ship types and technical issues encountered. The main programs in use by DNV are shown in Fig. 3. WASIM [5] represents the latest development in technology for wave load analysis and is developed in response to the need for improvements in the accuracy and reliability of the hydrodynamic load combinations for ships. WASIM is a 3-dimensional time domain program for arbitrary shaped ships (including multi-hulls) or other marine structures in waves. The ship may have an arbitrary forward speed, the waves can come from any direction and the responses can be computed in all six degrees of freedom. The program is based on a three-dimensional Rankine Panel method, where also the free surface is modelled. Radiation conditions are treated by including a zone where the free surface condition is modified such that the waves are absorbed, i.e. a numerical beach.



WAVESHIP:	Linear Strip Theory, frequency domain
WADAM:	3-D linear diffraction theory with zero forward speed, frequency domain
FASTSEA:	21/2 D high speed theory, valid for Fn above 0.4, frequency domain
WASIM:	3-D linear and non-linear, frequency domain with zero or forward speed, time domain with zero or forward speed

Fig.3 – DNV Hydrodynamic Analysis Programs

WASIM can be run in both a fully linear mode and in a non-linear mode. The transfer functions are derived from linear computations by Fourier analysis. The implemented non-linear option still solves the linear radiation and diffraction problem. In its non-linear mode, time series of the specified responses are generated, and additional Froude-Krylov and hydrostatic forces from wave action above still-water level are included. This 3-D approach provides a more accurate and correct basis for computation of loads and subsequent structural dimensioning. The included non-linearities can give significant contributions to both global loads and motions in large waves.

WAVESHIP, WADAM, FASTSEA and WASIM are used as important tools for calculation of dimensioning global loads. These loads serve as input to subsequent Finite Element analysis of global as well as local loads and deflections in connection with the direct calculations as for the CSA-2 Notation. Other important application areas are Rule development and Rule calibration.

Still, linear strip theory may be used to calculate global loads as global hull girder bending. For local pressures, however, 3-D linear diffraction theory has been found to produce more reliable results than the strip theory approach. WASIM is used for transfer of wave induced loads to the FE models, ensuring accurate pressure distributions for both zero and forward speed

5.3 ENVIRONMENTAL CONDITIONS

Probably the single most important aspect relating to the determination of design values for global hull girder loads is the selection of wave climate. The present approach is to employ an actual scatter diagram of wave height/wave period combinations for the sea area considered. This means that the actual contribution of each wave height/period combination is taken into account and the scatter diagram may be used for ultimate strength as well as fatigue analysis.

Standard scatter diagrams for different sea areas have been developed. The standard North Atlantic diagram is used for Ultimate Strength analysis. For the fatigue analysis the wave scatter diagram is selected based on world-wide service, unless a harsher environment is specified. The scatter diagrams are based upon BMT Wave Climate Statistics [6], but incorporate some important modifications to the wave height/period combinations in order to ensure that the wave steepness of the largest waves are not outside the breaking wave limit and that maximum expected wave height over 20 years is in agreement with other sources such as instrumental and hindcast data.

5.4 FORWARD SPEED EFFECTS

Forward speed effects have shown to be important for both ultimate strength and fatigue. Especially the vertical wave bending moment shows an increasing tendency with increasing speed. It is therefore of importance to find an as correct speed as possible to use in the analysis.

The vessel speed is a function of design speed and speed reduction in heavy weather. The speed reduction consists both of voluntary and involuntary reduction. The main contributors to speed reduction are:

- added wave resistance in heavy weather
- bottom slamming
- bow impact
- bow submergence (green water effects)
- unfavourable acceleration levels
- extreme roll motion

The speed reduction will be different for ULS and FLS may vary significantly since the important wave conditions varies significantly between the two design criteria. For the ULS this will normally mean that low speed may be used for determination of design loads. However, for torsion, extreme response may occur in oblique following sea conditions at higher speed (approximately 2/3 of design speed). For FLS, where lower wave periods/heights are of importance a speed in the range 2/3 to 3/4 of design speed should be used.

5.5 NON-LINEAR EFFECTS

The basic assumptions made in linear wave theory computations are that the wave amplitude is negligible and that the hull geometry is wall sided. Clearly, this set of assumptions is incorrect for most ship types under normal operating conditions.

Different types of non-linearity's are important for FLS and ULS calculations. For FLS calculations, where wave heights contributing to the fatigue damage are relatively small, the most important non-linearity is the intermittent wet and dry surfaces in the waterline region. This effect can be included in the transferred loads by changing the pressure loads in this area on the structural model after the load transfer. Scaling of the individual pressures on each element, and for each period and heading, can be performed according to the procedure given in CN 30.7.

For ULS calculations, other effects as bow flare forces, slamming and water on deck may be more important for the design loads. In WASIM, the Froude-Krylov and hydrostatic forces are calculated by integration of the incident wave pressure over the instantaneous wetted surface of the hull and gives thus a nonlinear contribution. This instantaneous wetted surface is defined by the instantaneous position and orientation of the ship in the incident wave. In addition, slamming and water on deck may be solved by using a special in-house version of WASIM. The slamming forces are calculated using a pre-processor conducted for a set of 2-dimensional strips of the hull using the 2DBEM program as developed under the MARIN-CRS research program. These results are subsequently used in the nonlinear WASIM simulation. Thus nonlinear contributions for damping and added mass are incorporated from the above-water-part.

6 FATIGUE SCREENING OF A PRESSURISED NATURAL GAS CARRIER (PNG)

Fig. 4 shows the hydrodynamic model used for seakeeping and wave load analysis of a standard size offshore loading and discharge version of a Pressurised Natural Gas (PNG®) carrier for Knutsen OAS Shipping. The concept was developed together with independent naval architect consultants and with input from DNV [3].

The standard type offshore loading vessel has the following characteristics:

- Net cargo transportation capacity, 22 to 24 million Sm^3 (775 to 850 million scuft)
- Vessel length, about 290 meter
- Vessel beam, 54 meter

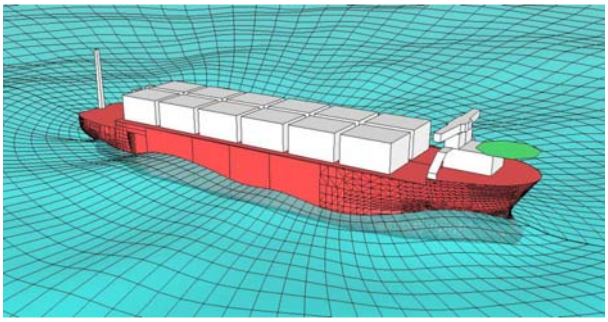


Fig. 4 - Hydrodynamic Model

In order to support the development process DNV carried out initial load, load response and strength analyses of the new concept. Fig. 5 shows the global structural FE-model used in the fatigue screening analysis of the vessel.

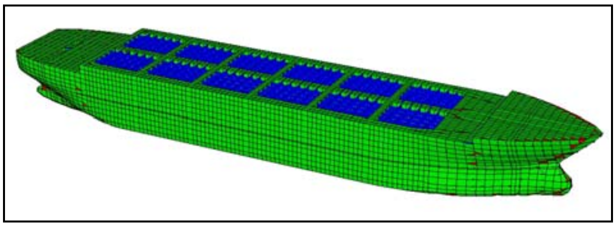


Fig. 5 - Global Structural FE Model

A direct structural response analysis of the hull in accordance with DNV's CSA-2 procedure for direct load application was carried out in order to properly account for torsion and warping effects. Being an open deck ship very much resembling a container vessel this is an essential analysis in order to being able to assess the fatigue endurance of the deck corners.

The scope of the analyses included the following analyses:

- Seakeeping and Wave Load Analysis
- 3-D FEM Analysis of Ship Hull
- Stochastic fatigue analysis

A fatigue screening using full stochastic (spectral) fatigue analysis was performed in order to map the fatigue strength of the design. Being a new type of vessel design this proved to be a very useful analysis in order to pinpoint important areas for more in-depth fatigue analysis.

7 FATIGUE ANALYSIS OF A SPHERICAL TANK LNG CARRIER

A complete fatigue analysis of a spherical tank LNG carrier was carried out on behalf of Mitsui Engineering and Shipbuilding Co. Ltd. in accordance with the DNV CSA-2 and PLUS-2 requirements. The vessel, being built for gas transportation from the Snøhvit field, has the following characteristics:

- Net cargo transportation capacity 140,000 m^3
- Vessel length, about 280 meters
- Vessel beam, about 48 meters

The vessel is designed for 25 years of trading in North Atlantic wave climate.

7.1 WAVE LOAD ANALYSIS

A linear frequency domain analysis were carried out for the full load and ballast condition for wave headings from 0 to 360 degrees with an increment of 30 degrees and 20 wave periods from 5 to 30 seconds. The viscous roll damping coefficients were calculated based on a probability level of 10^{-4} of roll in North Atlantic wave climate. The structural FE model was used as mass model in order to ensure compatibility for load transfer to the finite element analysis. Fig. 6 shows the hydrodynamic model used in the wave load analysis.

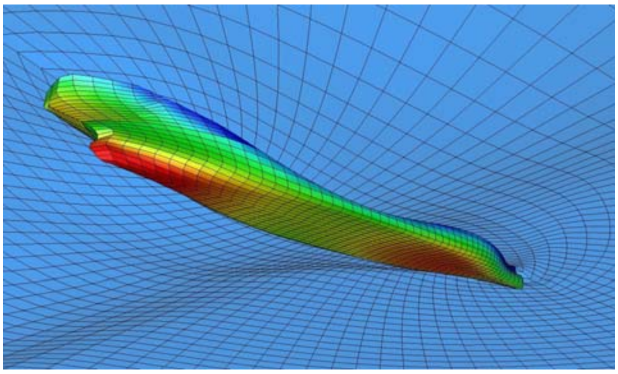


Fig. 6 – Hydrodynamic Analysis of a Spherical Tank LNG Carrier

7.2 COMPONENT STOCHASTIC ANALYSIS

Component stochastic fatigue analyses were carried out for longitudinal stiffeners and plating for three cross sections in the cargo area considering the following stress components:

- Vertical hull girder bending
- Horizontal hull girder bending
- Axial hull girder loading
- Axial and bending stresses due to double hull bending and relative deflections
- Local bending stress due to lateral pressure

Due to the complex structure of spherical tank LNG carriers (large deck openings, non-continuous tank covers and spherical tanks supported on skirts), nominal stress calculations based on beam theory will not give a correct representation of the global stress distribution in the hull girder. Stress response factors due to global bending and axial loading was therefore determined by a cargo hold analysis applying unit bending moments and axial loads. The global stress distribution due to vertical bending is shown in fig. 7.

The cargo hold model was also used to calculate secondary stresses due to double hull bending and relative deflections between frames. The direct calculated external and internal pressure distributions at a 10^{-4} probability level of exceedance were applied to the model. The results showed notch stress magnitudes up to 20-30MPa (at 10^{-4} probability level of exceedance) which gives a significant contribution to the total stress level. An adequate assessment of secondary bending stresses is therefore important for the reliability of the fatigue results.

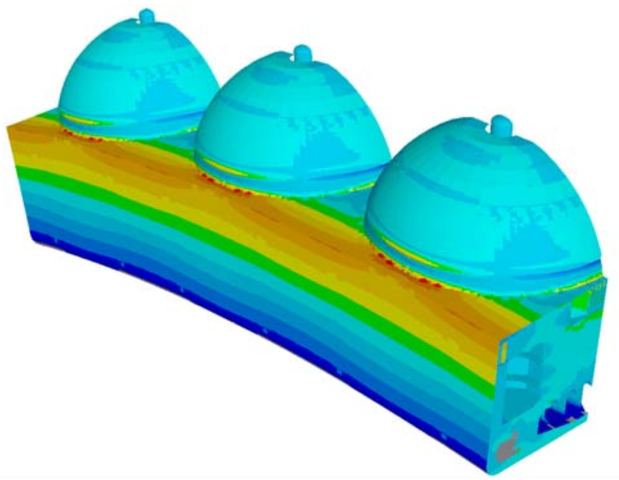


Fig. 7 – Global Stress Distribution from Vertical Bending of a Spherical Tank LNG Carrier

7.3 FULL STOCHASTIC ANALYSIS

7.3 (a) Global Analysis

Fig. 8 shows the global FE model with pressures mapped from the hydrodynamic analysis. In the splash zone of the midship region the pressure distribution is reduced below and extrapolated above the mean waterline.

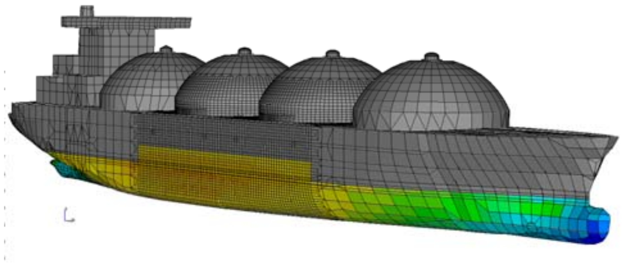


Fig. 8 – Global Model with Pressure Transferred from the Wave Load Analysis

The compatibility between the hydrodynamic model and structural model was proved by comparing the transfer functions for global sectional loads, fig. 9.

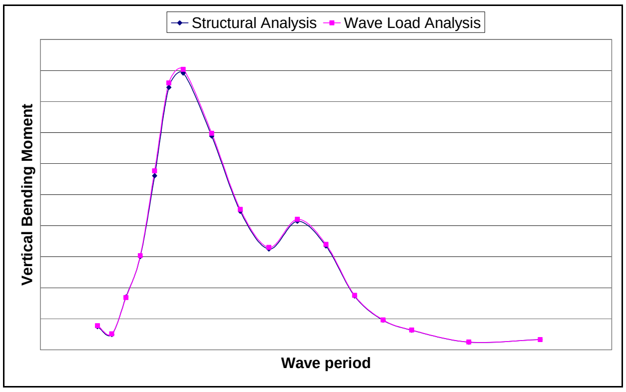


Fig. 9 – Comparison of Transfer Function for Midship Vertical Bending Moment in Structural and Wave Load Analysis (Wave Heading 45º)

A fatigue screening analysis was carried out in order to identify fatigue prone areas. As the fatigue strength was decisive for the minimum deck section modulus particular attention was given to the upper deck. The results were used to determine critical stress concentration factors as basis for detailed design of openings, deck attachments and pipe penetrations. A contour plot of calculated fatigue lives in the midship region is shown in fig. 10.

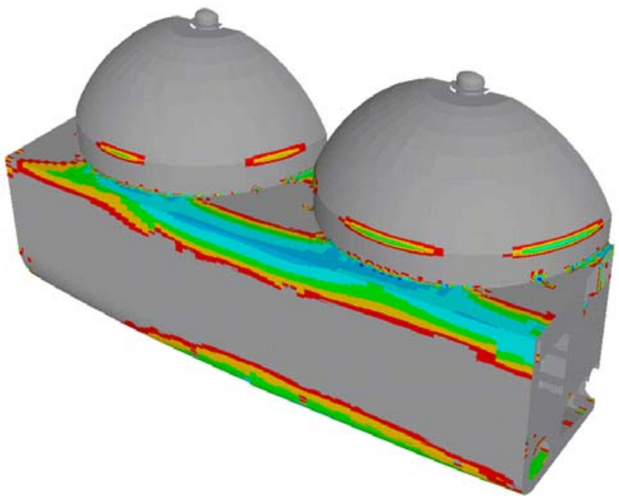


Fig. 10 – Fatigue Screening of Global Model

7.3 (b) Local Analyses

Full stochastic analyses based on local sub models were carried out for the hopper knuckle, tank skirt foundation, inner side connection to foundation deck, tank cover connection to main deck and one side longitudinal, fig. 11. For these details fatigue reinforcements were applied to meet the design fatigue life. The extent of the reinforcements was evaluated using the results of the global screening analysis.

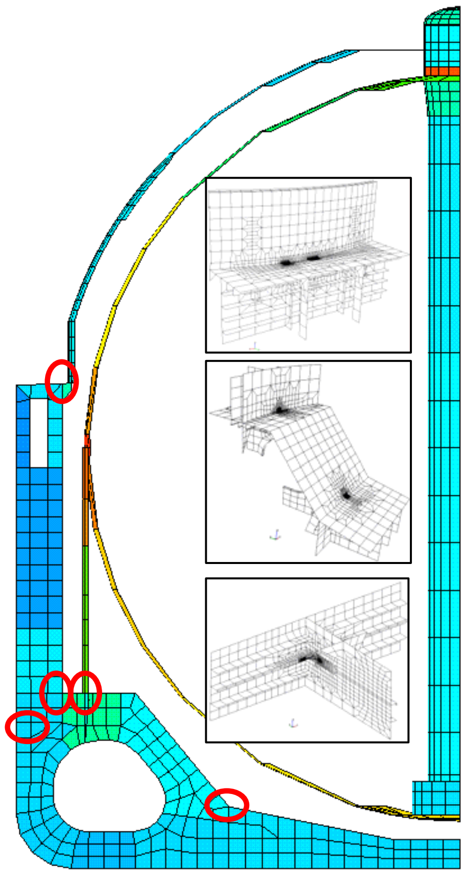


Fig. 11 – Details Subjected to Full Stochastic Analysis

7.4 PLUS-2 ANALYSIS

The fatigue calculations of the stiffener and web frame connections were based on simplified calculations of loads combined with finite element analyses to determine the stress response.

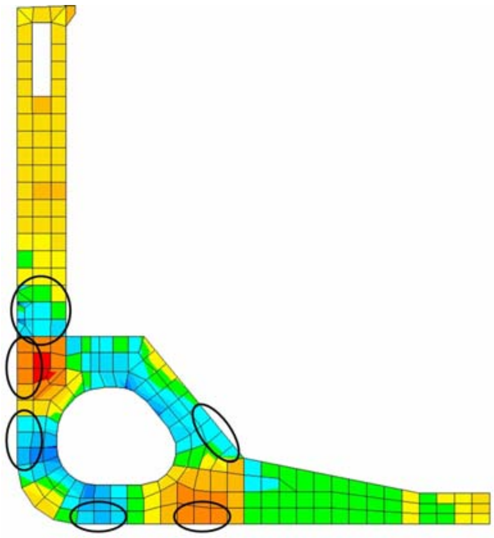


Fig. 12 – Shear Stress Distribution of the Midship Web Frame of a Spherical Tank LNG Carrier Highlighting Fatigue Prone Areas

A 3-D cargo hold analysis was employed to determine nominal stresses and deflections in the main girder system. In areas with high nominal shear stress, see fig. 12, local stress concentration models were used to calculate the geometric stress at hot spots. A typical local model is shown in fig. 13. Fig. 14 shows the hot spot stress at a lug corner which in most cases is the most critical position.

The results of the local analysis are used to calculate the critical nominal shear stress level in order to meet the fatigue design requirements as basis for fatigue assessment of other similar locations.

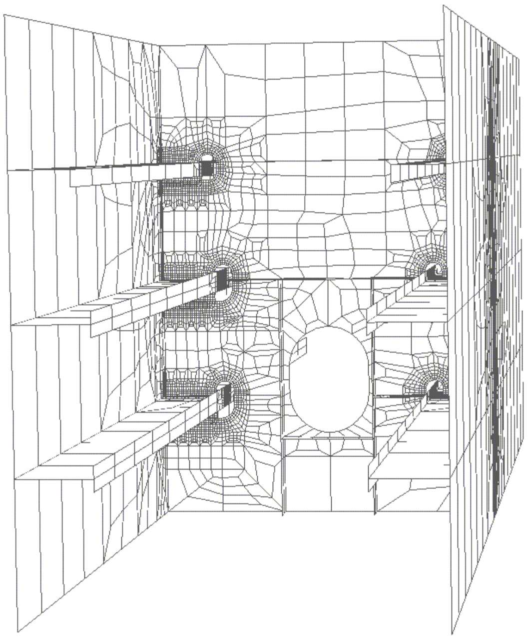


Fig. 13 – Local Model for PLUS-2

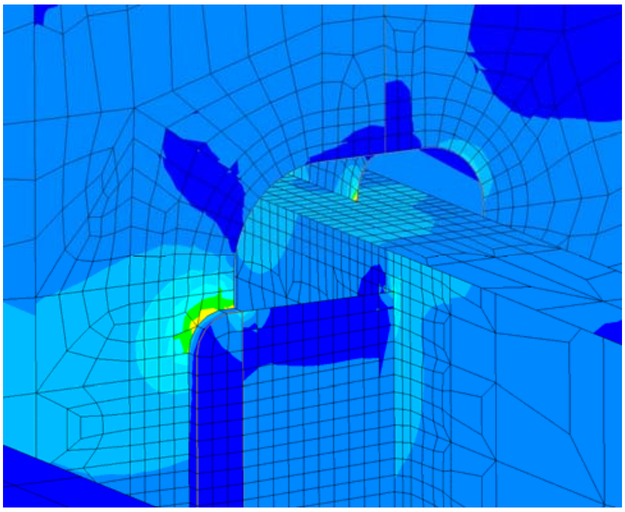


Fig. 14 – Hot Spot Stress at the Lug Corner of a Side Longitudinal

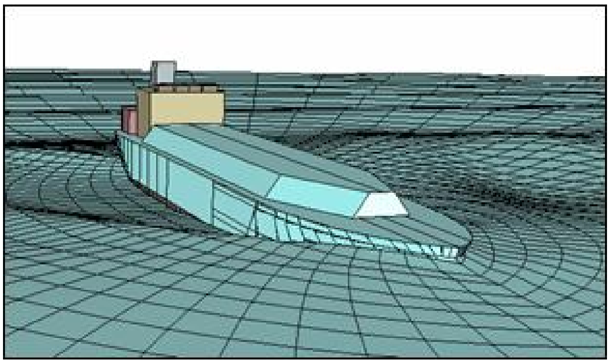


Fig. 15 – Wave Load Analysis of a Membrane Tank LNG Carrier

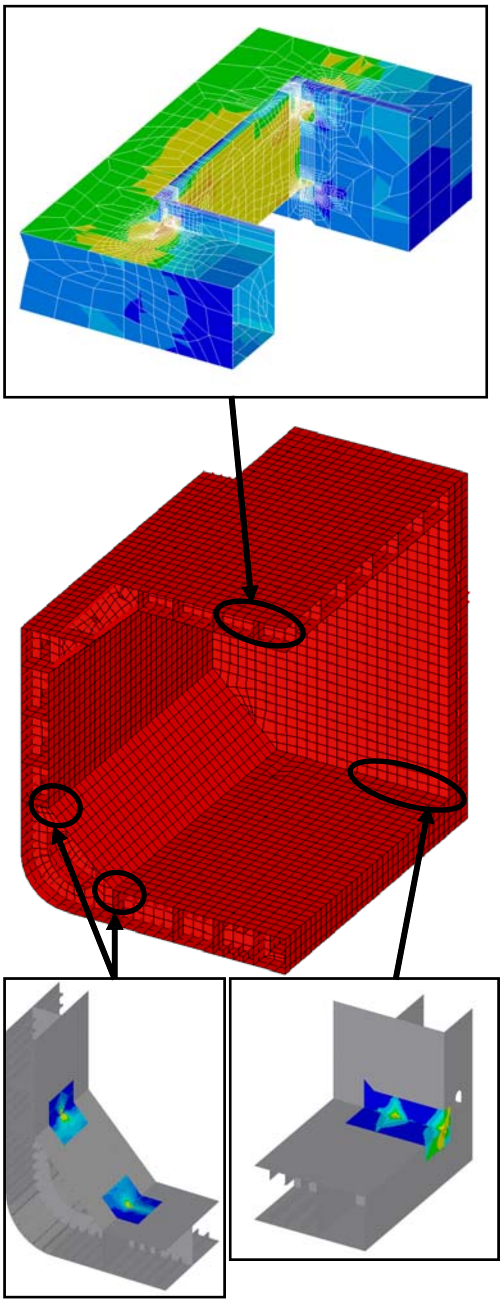


Fig. 16 – Critical Areas for Membrane Tank LNG Carriers

8. FATIGUE ANALYSIS OF MEMBRANE TANK LNG CARRIERS

Fig. 15 shows an illustration of a hydrodynamic model used for load calculation of a LNG membrane carrier.

Fig. 16 shows typical fatigue prone areas for membrane tank LNG carriers:

- Hopper tank lower and upper knuckle
- Connection of inner bottom and transverse bulkhead
- Deck openings (Dome)

In addition to fatigue checks of stiffener end connections component stochastic or full stochastic calculations are used to check the fatigue strength of the critical areas. Analyses have shown that reinforcements and weld dressing are needed in most cases to meet the fatigue requirements for vessels operating in harsh environment.

9 UNCERTAINTIES IN FATIGUE LIFE PREDICTIONS

There are a number of different uncertainties associated with fatigue life predictions. The most important are:

- S-N Curves
- Wave loading
- Stress calculations

9.1 S-N CURVES

The design S-N curves given in CN30.7 are associated with a 97.6% probability of survival (mean minus two standard deviations). As the fatigue life of steel structures is considerably shorter in freely corroding conditions than in air, there is also a large uncertainty associated with the selection of S-N curves.

9.2 WAVE LOADING

Uncertainties in wave loading are associated with the wave climate and load calculation method. Fig. 17 illustrates uncertainties with respect to operational wave environment showing calculated fatigue damages (relative to the largest damage) at various locations of a spherical tank LNG carrier for world wide and North Atlantic operation using direct calculated wave loads. The fatigue damage ratio for the two wave climates varies from 1.6 to 2.4.

Fig. 18 shows calculated fatigue damages based on simplified load calculations and component stochastic calculations. The results of the simplified calculations deviate significantly from component stochastic calculations based on more accurate wave load calculations.

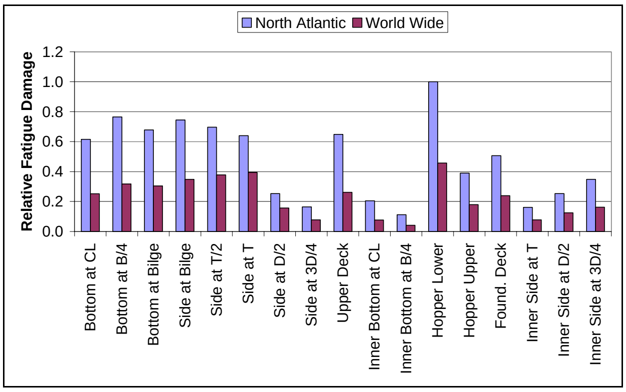


Fig. 17 – Calculated Fatigue Damage in North Atlantic and World Wide Operation for Longitudinals in a Spherical Tank LNG Carrier

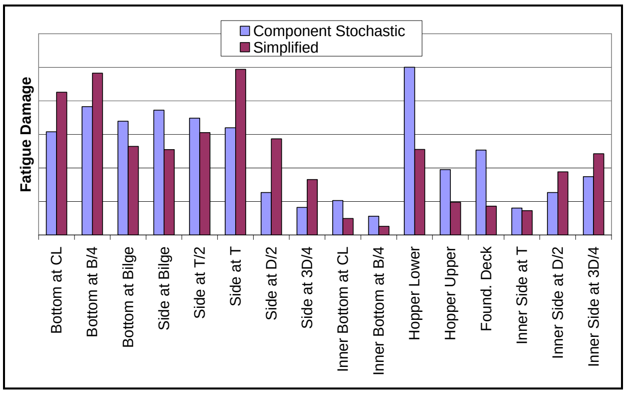


Fig. 18 – Calculated Fatigue Damage based on Component Stochastic and Simplified Calculations for Longitudinals in a Spherical Tank LNG Carrier

9.3 STRESS CALCULATIONS

Small changes in stress results in much greater changes in fatigue life as fatigue damage is proportional to the inverse slope of the S-N curve. Uncertainties with respect to stress calculations and determination of stress concentration factors can be reduced by direct FE calculations, but there will always be uncertainties associated with the finite element method and fabrication.

9.4 SIMULATED CRACK FREQUENCIES

Due to the uncertainties in calculated loads and stresses, the probability of fatigue failure may be much higher than the 2.4% associated with the mean minus two standard deviation fractile of the S-N curves. Fig. 19 shows the simulated crack frequency after 20 years of operation for calculated fatigue lives based on simplified and stochastic (spectral) fatigue analyses, [7]. The probabilistic model has been favourably compared with operational experience from a shuttle tanker operating on the same trade for 18 years in the North Sea, [8].

The predictions show that in order to limit the number of fatigue cracks to 2-3% after 20 years, the design should be based on a calculated fatigue life of 40 years using stochastic analysis. If simplified calculations are used, the design should be based on 80 years.

State-of-the-art fatigue analysis combined with a high calculated fatigue life is an efficient means to reduce the probability of failure and reduces the need for in-service inspection.

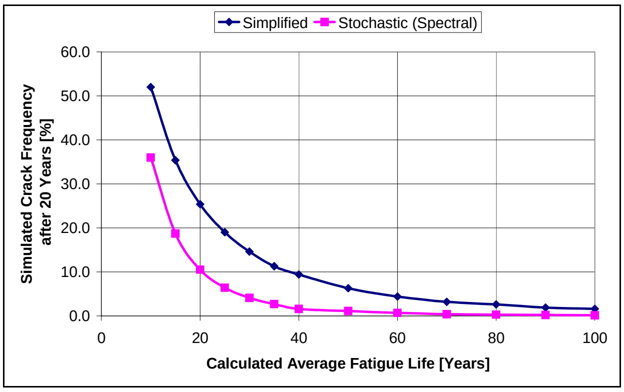


Fig. 19 – Simulated Crack Frequency after 20 Years in Operation for Simplified and Full Stochastic/Spectral Fatigue Calculations

10. CONCLUSIONS

The operation of vessels in harsh environment such as the North Atlantic put strict requirements to the design to achieve sufficient fatigue strength. The fatigue strength of the upper deck should be checked early in the design process as the fatigue requirements may be decisive for the minimum deck section modulus.

The shipbuilding industry has traditionally focused on the fatigue strength of longitudinals which is now very much under control due to the requirements of the classification societies. Details such a laterally loaded plates and stiffener web frame transitions may however in many cases be more critical with respect to fatigue performance. Hence, for successful non-obstructed operation, more extensive fatigue calculations should be performed.

Although simplified calculations may provide a reasonable basis for fatigue design, more sophisticated methods using direct wave load analysis significantly increase the reliability of the results. Simplified stress calculations may serve as basis for fatigue calculations of longitudinal stiffeners and plating, but for details subjected to a more complex stress response full stochastic fatigue calculations based on direct load transfer should be performed.

12. REFERENCES

1. Røren, E. M. Q., Araldsen, P. O. and Holtsmark, G., “Analysis of Tankers with SESAM69, Comparisons between calculations and measurements” (in Norwegian), *Kursdagene, NTH 1971*, The Norwegian Society of Chartered Engineers (NIF)

2. Classification Notes no. 30.7: “Fatigue Assessment of Ship Structures”, *Det Norske Veritas*, February 2003

3. Valsgård, S., Reepmeyer, O., Lothe, P., Strøm, N.K. and Mørk, K.: “The Development of a Compressed Natural Gas Carrier”, *The 9th International Symposium on Practical Design of Ships and other Floating Structures* (PRADS 2004), Lübeck-Travemünde, Germany, September 12-17, 2004

4. Pastoor, L.W.: ”On the Assessment of Nonlinear Ship Motions and Loads”, *PhD Thesis*, Technische Universiteit Delft, 2002.

5. Pastoor, L.W., Helmers, J.B. and Bitner-Gregersen, E.: “Time Simulation of Ocean-Going Structures in Extreme Waves”, *Proceedings of OMAE’03: 22nd International Conference on Offshore Mechanics and Arctic Engineering,* June 8-13, 2003, Cancun, Mexico.

6. British Maritime Technology (primary Contributors Hogben, N., Dachuna, L.F. and Olliver, H.N.) “Global Wave Statistics”, Unwin Brothers Limited, London

7. Kamsvåg, F.: “Probabilistic analysis of uncertainties in fatigue analyses of ships”, *Det Norske Veritas*, June 2000 (Restricted).

8. Hansen, H. R., Nielssen, N. B. and Valsgård, S.: ”Operational Experience with Double Hull Tankers”, RINA Conference on *Design & Operation of Double Hull Tankers*, London, 25-26 February, 2004.

14. AUTHOR’S BIOGRAPHIES

Torbjørn Lindemark, M.Sc., is a Senior Engineer in Det Norske Veritas. He has more than 10 years experience from DNV on R&D, class approval and consultancy on ships including gas carriers and offshore structures.

Frode Kamsvåg, M.Sc., is a Principal Engineer in Det Norske Veritas. He has more than 15 years experience from DNV on R&D and consultancy on various offshore structures and ships, including gas carriers.

Sverre Valsgård, Ph.D., is a Senior Principal Engineer in Det Norske Veritas. He has more than 30 years experience from DNV on R&D and consultancy on ships including gas carriers and offshore structures as well as experience from management positions.