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CSA-2 ANALYSIS OF A 216k LNGc MEMBRANE Carrier

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SUMMARY

The paper describes the CSA-2 analysis of a 216k LNGc with membrane tanks. The paper focuses on direct calculation procedures and principles used in design verification. This includes determination of design loads, global and local strength analysis and fatigue demands. A detailed description of the calculation procedure and the advantages and incentives of applying direct calculations in ship design are presented.

The DNV classification notation CSA-2 is based on direct calculation procedures. The wave loading is calculated by linear and non-linear analysis based on 3D diffraction theory and applied to global and local structural finite element models by direct load transfer. Compared to the traditional approach of using simplified rule formulas, direct wave load calculations by numerical analyses improves the accuracy and reliability of the calculated loads and may be tailored for the intended operational profile and wave environment.

By direct load transfer from the wave load analysis to structural finite element models the simultaneous occurrence of the different load effect is preserved through the calculations. This implies that spectral fatigue calculations can be performed and uncertain assumptions with respect to the correlation of the different load effect are avoided in the fatigue and hull strength evaluations. The stress response in both extreme events and the long term stress distribution was established for the entire vessel by global finite element analyses. As all loads represent actual loading situations the most critical areas are easily identified. Fatigue screening of the hull was used to check for fatigue prone areas. This was used as basis for submodels to include local geometrical stresses in the final fatigue evaluation.

The CSA-2 analysis provides the yard and owner with a full documentation of the hull integrity in extreme events as well as the predicted performance of fatigue prone areas and details.

1. INTRODUCTION

A complete CSA-2 analysis of a membrane tank LNG carrier was carried out on behalf of Hyundai Heavy Industries Co., Ltd. and Samsung Heavy Industries Co., Ltd.

The vessel, being built for gas transportation from Qatar to UK and US, has the following characteristics:

- Net cargo transportation capacity 216,000 m^3
- Vessel length, 303 meters
- Vessel beam, 50 meters

The vessel is designed for 40 years in worldwide operation.

The scope of work included the following tasks carried out according to the DNV Rules for direct strength analyses [1] and fatigue assessment of ship structures [2]:

- Wave load analysis
- Yield and buckling strength evaluation
- Fatigue strength evaluation

All calculations are documented in technical reports and submitted to the owner and class as basis for approval.

2. CSA-2 RULE REQUIREMENTS AND ACCEPTANCE CRITERIA

The additional class notation CSA-2 was implemented in the DNV Rules in 1995. For the CSA-2 notation the wave loads are to be based direct hydrodynamic analysis and the hull strength is to be evaluated based on global and local finite element models.

A set of design loads, typically including extreme bending moments and maximum accelerations and/or pressures, are determined by the wave load analysis and applied to the global structural model using a design wave approach. The loads are calculated for a 20 year return period using wave scatter diagram for the North

Atlantic and serve as basis for yield and buckling strength checks of the hull and main girder system. The acceptance criteria are based on equivalent von Mises stress not to exceed 0.85 or 0.90 of the yield stress depending on the location and a maximum buckling usage factor of 0.9.

Additional to the local yield and buckling strength checks the ultimate capacity of the hull girder is to be checked for both intact and damaged conditions.

Fatigue calculations are to be carried out based on the results from the hydrodynamic analysis for a design life of 20 years in world wide operation or as specified for the vessel in question.

3. WAVE LOAD ANALYSIS

3.1 GENERAL

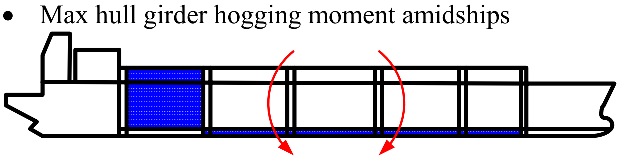
The main purpose of the wave load analysis is to determine the design loads and load transfer functions for the subsequent structural strength and fatigue analysis.

The wave load analysis was based on the linear and non-linear 3D time domain program WASIM [3]. WASIM in its linear mode calculates transfer functions for motions, sea pressure and sectional forces of the vessel. 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. Green water effect is simulated using a simplified model of the Bernoulli's formulation for a breaking dam of water. The simulation of green water flow on deck is coupled with the ship motion in waves. WASIM has no theoretical limitations in vessel speed (as long as the vessel is not planning).

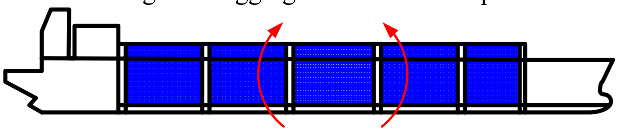
3.2 DESIGN CASES AND SELECTION OF LOADING CONDITIONS

The design loading conditions are selected from the vessel's loading manual. The basis for the selection was to maximise the hull stress response for the following design cases:

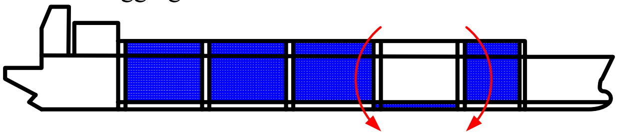
- Max hull girder hogging moment amidships



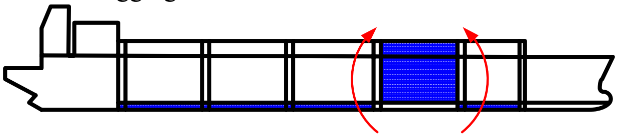
- Max hull girder sagging moment amidships



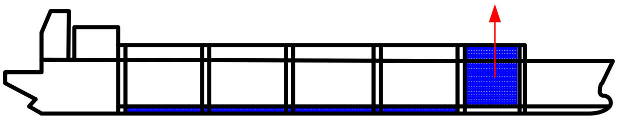
- Max hogging moment in tank no. 2



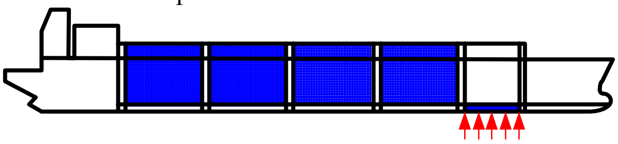
- Max sagging moment in tank no. 2



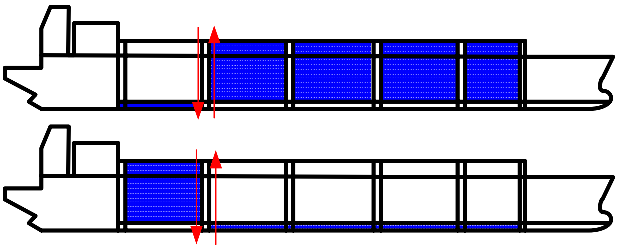
- Max vertical acceleration in tank no.1



- Max bottom pressure in tank no. 1



- Max shear force in tank no. 5



For each design case a corresponding load condition was selected from the loading manual considering stillwater bending moments, shear forces and draught.

The fatigue analysis was based on the normal ballast and full load condition representing the most frequently used load conditions.

3.3 PANEL AND MASS MODEL

The hydrodynamic model consists of a panel model describing the hull shape and a mass model describing the total mass distribution of the vessel for each loading condition.

Figure 1 shows the hydrodynamic panel model. The size of panels was selected to describe the hull shape and resolve the pressure gradients. As the frequencies of the radiated waves are depending on the encounter frequency, the vessel's speed also has to be considered.

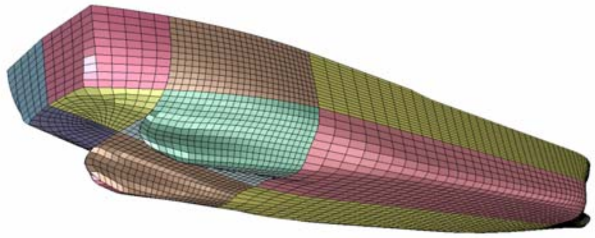


Figure 1: Hydrodynamic Panel Model

The mass model, shown in Figure 2, consists of the structural finite element model representing the lightship weight and bunkering and an additional FE model representing the deadweight. The weight distribution was tuned according to the loading manual by adjusting the material density of the shell elements and nodal mass elements. This approach ensures mass compatibility between the wave load analysis and the structural analysis which is a pre-requisite for correct load transfer as unbalanced loads will cause reaction forces disturbing the global response.

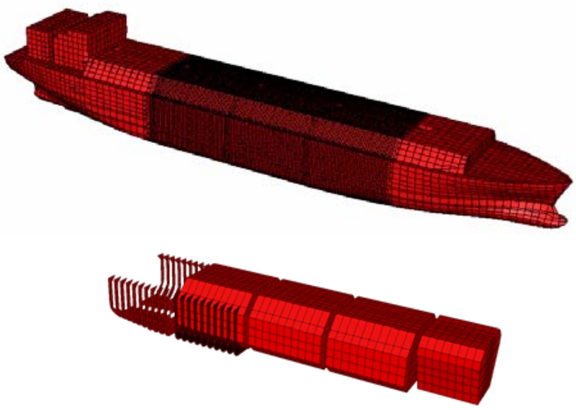


Figure 2: Lightship (Structural Model) and Deadweight Model (Cargo Tank 5 Empty)

The resulting stillwater bending moment and shear forces were compared to the loading manual. The sectional loads calculated by WASIM include the hydrostatic end effect which is not considered in loading manuals. Pressures at the ends of the vessel generate an axial force with a centre of gravity at 2/3 of the draught of the vessel and as the sectional bending moments are calculated based on the position of the neutral axis this force causes a global bending moment as illustrated in Figure 3. This effect was corrected for in the comparison shown in Figure 4.

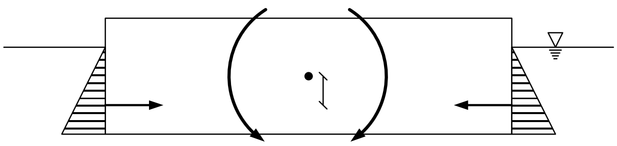


Figure 3: Hydrostatic End Effects

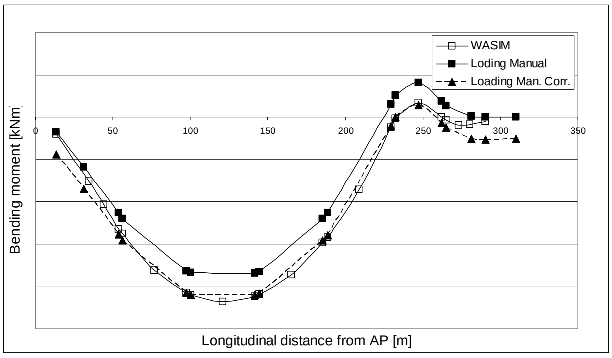


Figure 4: Comparison of Stillwater Loads WASIM vs. Loading Manual

3.4 DETERMINATION OF DESIGN LOADS FOR YIELD AND BUCKLING STRENGTH CHECKS

The design loads were calculated for a 20 year return period based on the IACS North Atlantic scatter diagram. [4]. A vessel speed of 5 knots was used under the assumption that the master will throttle back in severe wave conditions but keep some thrust for manoeuvrability. All wave headings were assumed to have and equal probability of occurrence.
The roll damping viscous forces were estimated using the 2D strip theory program WAVESHIP [3] linearized for a 20year return period. The calculated viscous damping was added to the potential damping.

For each of the design cases given in section 0 long term linear load values (20 year return period) were predicted by a frequency domain analysis and regular design waves were determined for each of the design conditions. Subsequently the conditioned irregular design wave approach or better known as the MLER method (Most Likely Extreme Response) [5] was used to develop irregular design waves. The MLER wave is an irregular sea state of short duration, typically 40s, conditioned to produce the linear 20years response at a given snapshot. For each of the design events the MLER wave is simulated with WASIM in order to calculate the 20years equivalent response with corresponding accelerations, pressures, shear forces and bending moments.

3.5 DETERMINATION OF FATIGUE LOADS

The fatigue loads were calculated based on a forward speed of 2/3 of the design speed and omni-directional headings with an increment of 30 degrees. The viscous damping was calculated as in the design load analysis but linearized at a probability level of 10^{-4}. The reason for this is that the main contribution to the cumulative fatigue damage comes from the smaller waves.

Both two route specific (Qatar-Boston and Qatar-UK) and the DNV scatter diagram for world wide trading [2] were considered. In order to find the most severe of the scatter diagrams with respect to fatigue damage the long term load distributions of the following loads were compared:

- Midship vertical bending moment (VBM)
- Midship horizontal bending moment (HBM)
- Midship pressure distribution
- Midship vertical and transverse acceleration (Av and At)

The ratios of the long term load for each scatter diagram compared to the North Atlantic are shown in Figure 5.

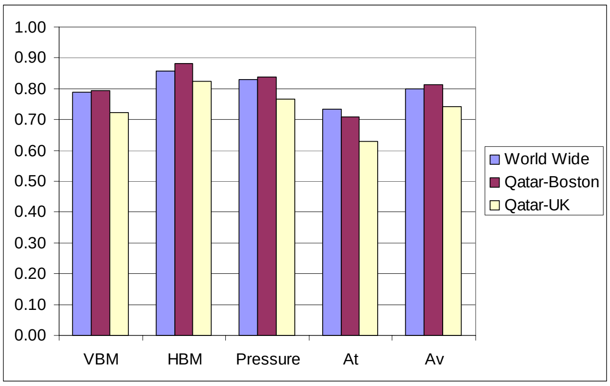


Figure 5: Long Term Load Ratio – Considered Routes vs. North Atlantic

As the Weibull slope parameter and zero crossing periods differ for the different trades, the comparison of the long term load are not fully representative for the fatigue damage accumulation. A more correct comparison was obtained by comparing fatigue damage ratios for each load component relative to a fatigue damage of 1.0 in North Atlantic operation as shown in Figure 6. These were established by multiplying the North Atlantic long term load response by stress factors resulting in a fatigue damage of 1.0. Using the same stress factor on the long term response for the other trades, the fatigue damage relative to a damage of 1.0 in North Atlantic was obtained.

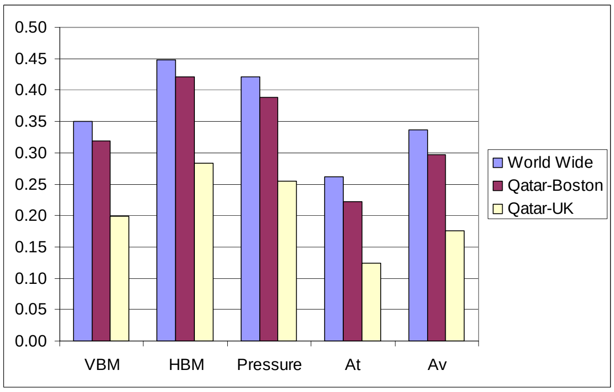


Figure 6: Fatigue Damage Ratios – Considered Routes vs. North Atlantic

The evaluation shows that even though the long term loads are larger for route 1 the world wide scatter diagram is the most severe with respect to fatigue damage.

3.6 RULE LOAD COMPARISON

3.6(a) Design Loads

A comparison of the direct calculated linear and non-linear midship bending moments are shown in Figure 7. The direct calculated design loads are 6% and higher than the IACS values in sagging and 13% in hogging.

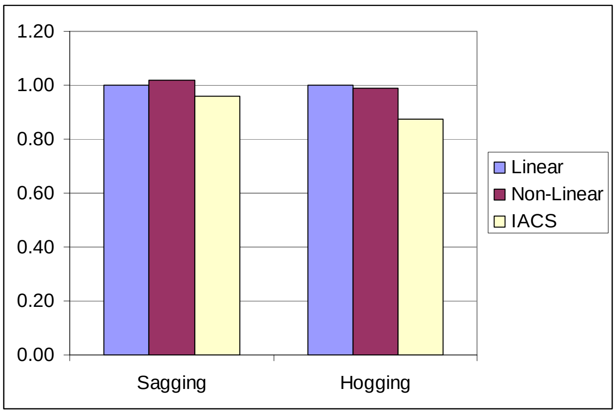


Figure 7: Direct Calculated Midship Wave Vertical Bending Moment vs. IACS Unified Normalized on Linear Hydrodynamic Results

3.6 (b) Fatigue Loads

For comparison of direct calculated fatigue loads with Rule values, long term load were calculated at a probability level of 10^{-4} based on the North Atlantic scatter diagram. The comparison is shown in Figure 8. The Common Scantling Rule loads for oil tankers are also considered, but it should be noted that these will not apply for LNG vessels. In general the DNV and CSR rule values are 10-20% below the direct calculated values except for the CSR vertical acceleration which is somewhat conservative in this case.

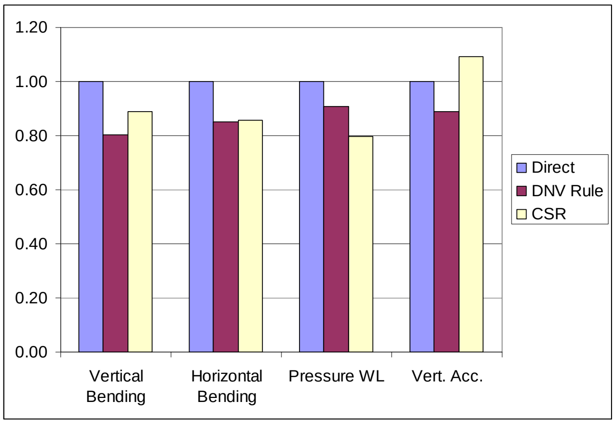


Figure 8: Comparison of Fatigue Loads - DNV Rule and Common Scantling Rules Relative to Direct Calculated Loads (Amidships - North Atlantic)

4. HULL STRENGTH ANALYSIS

4.1 FINITE ELEMENT MODELS

The yield and buckling strength calculations were based on a global structural analysis. A relatively coarse mesh was applied in the fore and aft ship (see Figure 2) due to model size limitations. For the strength assessment of these areas two submodels were made as shown in Figure 9.

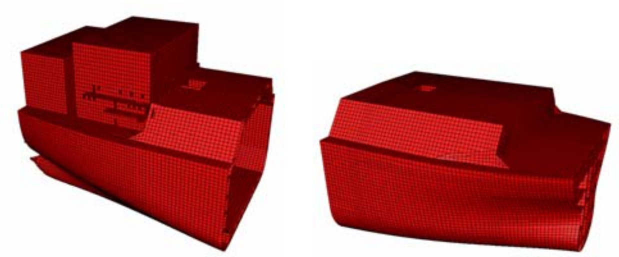


Figure 9: Submodels of Fore and Aft Ship

4.2 DIRECT LOAD TRANSFER

For each of the design cases determined in the wave load analysis the snapshot load transfer option in WASIM was used for load application to the FE models. An example is shown in Figure 10 showing the global model with internal and external pressure loads for one of the design cases.

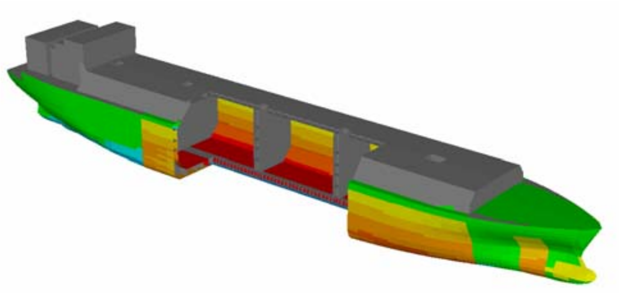


Figure 10: Global Model with Applied Internal and External Pressure Loads

For a final check of load equilibrium and quality of the load transfer the sectional load were calculated and compared to the sectional loads calculated in the wave load analysis as shown in Figure 11 and Figure 12.

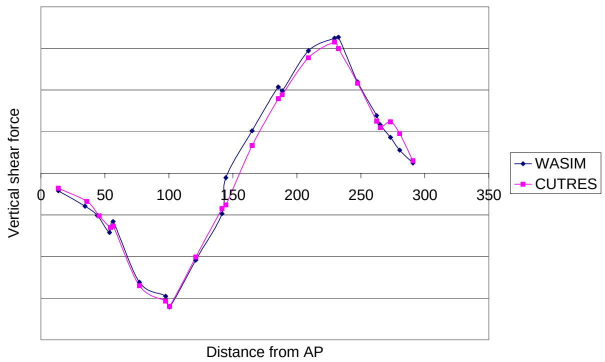


Figure 11: Comparison of Shear Forces in Wave Load analysis (Wasim) and Structural Analysis (Cutres)

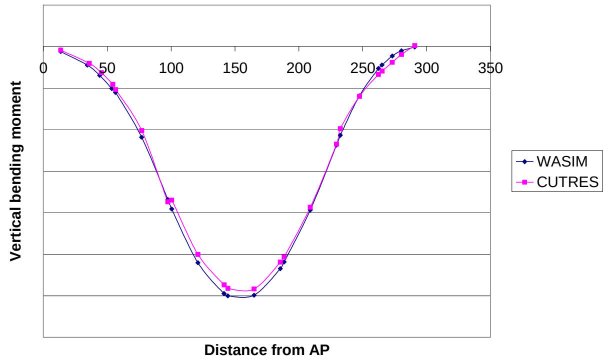


Figure 12: Comparison of Vertical Bending Moments in Wave Load analysis (Wasim) and Structural Analysis (Cutres)

4.3 YIELD AND BUCKLING STRENGTH CHECKS

The stress response of the hull girder was checked for maximum allowable equivalent nominal stress (Von Mises) and a buckling control was carried out using PULS [6].
Some strengthening was required to fulfil the CSA-2 criteria, mainly related to buckling capacity of the double bottom and relatively high stresses in the trunk deck at midship transverse bulkheads, see Figure 13.

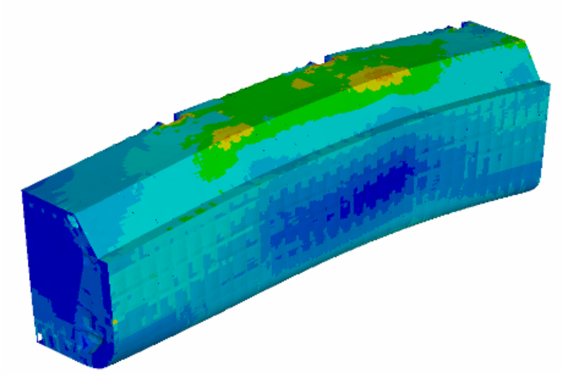


Figure 13: Stress Response in Hold 2-4 in Max Midship Hogging Case

The relatively high stress in this area was found to be caused by bending of the double deck structure shown in Figure 14. This bending occurs due to a combined effect of deformation of the transverse frames due to external pressures and a second order effect of hull girder bending:

- The external bottom pressure results in a rotation of the deck chamfer pushing the deck downwards as illustrated in Figure 15.
- In a global hogging mode, the trunk deck will aspire to the shortest path between the transverse bulkheads which is a straight line resulting in bending moments at the supporting bulkheads.

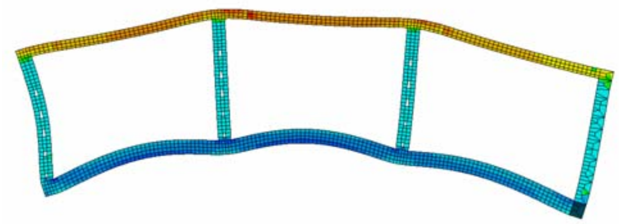


Figure 14: Deflection Pattern of the Girder at 2655 off CL illustrating the Bending of the Double Deck at Transverse BHD

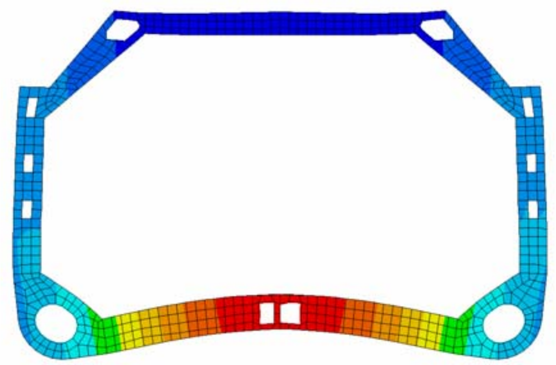


Figure 15: Deflection of Midship Frame in Max Midship Hogging Case

4.4 HULL GIRDER CAPACITY

In the hull girder capacity calculations the ultimate strength of the hull girder was checked for both intact and specified damage scenarios for collision and grounding, see Figure 16.

The ultimate capacity is found by summing up the buckling and yield capacities of structural elements over the whole section:

$$M_U = \sum_k \sigma_{crj} A_j z_j + \sum_k \sigma_{fk} A_k z_k$$

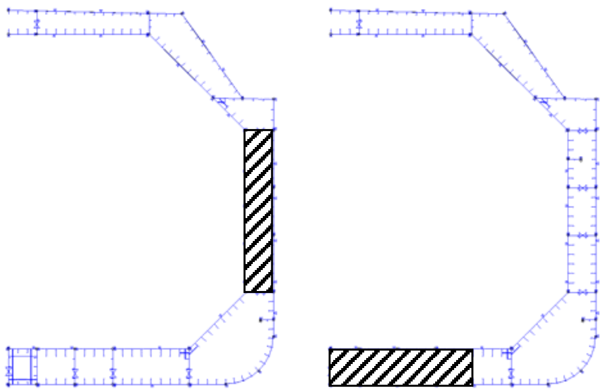
A = area of panel
z = distance from panel to plastic neutral axis
σ_{cr} = critical buckling stress of panel on compression side
σ_f = yield stress of panel on tension side
j = all panels on compression side
k = all panels on tension side

The capacity check was based on the calculated design loads for midship sagging and hogging:

$$\gamma_S M_S + \gamma_W M_W \le \frac{M_U}{\gamma_M}$$

M_S = design stillwater bending moment in sagging or hogging
M_W = design wave bending moment in sagging or hogging
γ_S = stillwater bending moment factor
= 1.0 for intact conditions
= 1.1 for damaged conditions (allowing for moment increase with accidental flooding of holds)
γ_W = wave bending moment factor
= 1.1 for intact conditions (safety factor corresponding to ~100 years North Atlantic)
= 0.67 for damaged conditions (reduction factor corresponding to ~3 months exposure in world wide climate)

The hull girder capacity was found to comply with the requirements for both intact and damaged conditions.



Penetration height 0.6D   Penetration breath 0.55B

Figure 16: Damage extent in Collision and Grounding

5. FATIGUE STRENGTH ANALYSIS

The fatigue strength analysis was carried out based on spectral fatigue calculations. Spectral fatigue calculations are based on complex stress transfer functions established through the direct wave load calculations combined with subsequent stress response analyses. The stress transfer functions express the relation between the wave heading and frequency and the stress response at a specific location and may be determined by either

- Component stochastic analysis
- Full stochastic analysis

Component stochastic calculations may in general be employed for stiffeners and plating and other details with a well defined principal stress direction mainly subjected to axial loading due to hull girder bending and local bending due to lateral pressures. Full stochastic calculations can be applied to any kind of structure.

Spectral fatigue calculations imply that the simultaneous occurrence of the different load effect is preserved through the calculations and the uncertainties are significantly reduced compared to simplified calculations.

The calculation procedure includes the following assumptions for calculation of fatigue damage:

- Wave climate is represented by scatter diagram
- Rayleigh distribution applies for stresses within each short term condition (sea state)
- Cycle count is according to zero crossing period of short term stress response
- Linear cumulative summation of damage contributions from each sea state in the wave scatter diagram

The spectral calculation method assumes linear load effects and responses. Non-linear effects due to large amplitude motions and large waves were neglected under the assumption that the stress ranges at lower load levels (intermediate wave amplitudes) contribute relatively more to the cumulative fatigue damage. In cases where linearization is required, e.g. in order to determine the roll damping or intermittent wet and dry surfaces in the splash zone, the linearization was performed at a load level representative to stress ranges giving the largest contribution to the fatigue damage. In general a reference load or stress range at 10^{-4} probability of exceedance is used.

5.1 COMPONENT STOCHASTIC FATIGUE ANALYSIS

5.1 (a) Calculation Methodology

The component stochastic fatigue calculation procedure is based on linear combination of load transfer functions calculated by the wave load analysis program and stress response factors representing the stress per load ratio. A flow diagram of the calculation procedure is given in Figure 17.

The following load transfer functions were considered:

- Vertical hull girder bending moment
- Horizontal hull girder bending moment
- Hull girder axial force
- Vessel motions in six degrees of freedom
- External (panel) pressures

Load transfer functions for internal cargo and ballast pressures due to accelerations in x-, y- and z-direction were derived from the vessel motions:

$$H_{p\_ax}(\omega|\theta) = \rho \cdot x_s \cdot H_{ax}(\omega|\theta)$$
$$H_{p\_ay}(\omega|\theta) = \rho \cdot y_s \cdot H_{ay}(\omega|\theta)$$
$$H_{p\_az}(\omega|\theta) = \rho \cdot z_s \cdot H_{az}(\omega|\theta)$$

x_s, y_s and z_s is the distance from the centre of free liquid surface to the load point in x-, y- and z-direction defined by the coordinate of the free surface centre minus the coordinate of the load point. The acceleration transfer functions were determined in the tank centre of gravity and include the gravity component due to pitch and roll motions.

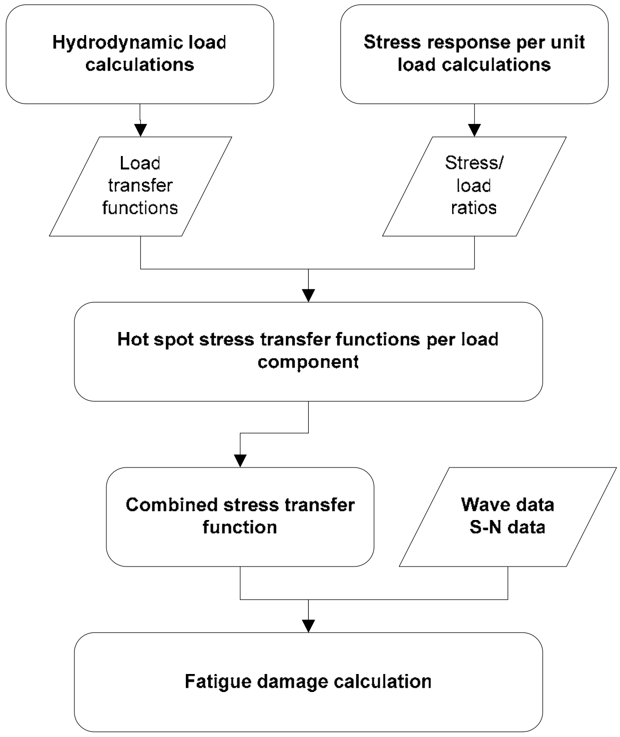


Figure 17: Flow Diagram for Component Stochastic Fatigue Calculations

For each load transfer function the corresponding stress transfer function was determined as

$$H_{\sigma,k}(\omega|\theta) = A_k \cdot H_k(\omega|\theta)$$

where

A_k = Stress/load ratio for load component k

$H_k(\omega|\theta)$ = Load transfer function for load component k

The combined stress response is determined by a linear complex summation of stress transfer functions

$$H_\sigma(\omega|\theta) = \sum_{k=1}^{n} H_{\sigma,k}(\omega|\theta)$$

The following stress component factors may be relevant to determine the combined stress in stiffeners and plating:

A_1 = Axial stress per unit vertical hull girder bending moment

A_2 = Axial stress per unit horizontal hull girder bending moment

A_3 = Axial stress per unit global axial force

A_4 = Bending stress per unit local external pressure

A_5 = Bending stress per unit local internal pressure (to be combined with accelerations in x-, y-and z-direction)

A_6 = Bending stress due to relative deflection of stiffeners between web frames per unit external pressure

A_7 = Bending stress due to relative deflection of stiffeners between web frames per unit internal pressure (to be combined with accelerations in x-, y- and z-direction)

The stress factors A_k may be either positive or negative depending on the position in the structure, type of loading and sign convention of sectional loads used in the wave load programme. As wrong sign will change the phase of the transfer function by 180 degrees it is important to ensure that correct signs are used.

The stress per load ratio was calculated based on the simplified formulas for nominal stress and tabulated stress concentration factors.

Relative deflections at transverse bulkheads were calculated based on a cargo hold analysis applying the direct calculated long term pressure loads at 10^{-4} probability of exceedance.

The combined stress transfer functions were combined with the wave scatter diagram and S-N curves and the fatigue damage was calculated by summation of part damages from each wave period and heading for every sea state in the scatter diagram.

5.1 (b) Details Considered and Findings

Component stochastic fatigue calculations were performed for longitudinal stiffeners and plating in the midship region, tank no. 1 and in the forward deep tank. For the bottom, bilge and lower part of side shell some strengthening of both stiffeners and plating was required in order to meet the fatigue design criterion.

In Figure 18 the calculated fatigue damage for selected longitudinals are compared to results based on simplified calculations according to the DNV rules and common scantling rules for oil tankers. The comparison is based on the calculated fatigue damage for 25 years operation in the North Atlantic and a corrosion protection period of 20 years. The comparison shows that both the DNV Rules and the Common Scantling Rule requirements are non-conservative compared to the spectral fatigue calculation method.

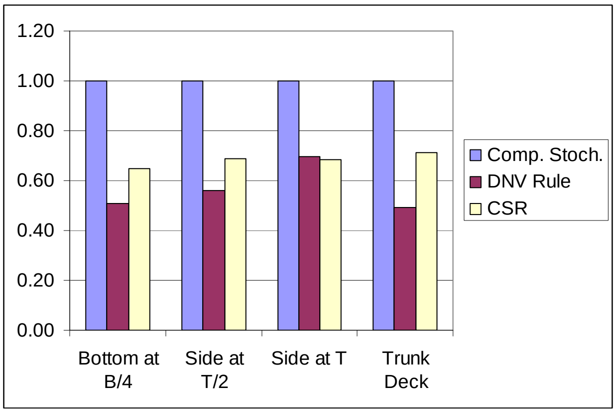


Figure 18: Comparison of Fatigue Damage by DNV Rule and Common Scantling Rules Relative to Component Stochastic Calculations

5.2 FULL STOCHASTIC FATIGUE ANALYSIS

5.2 (a) Calculation Methodology

A flow diagram of the full stochastic fatigue calculation procedure is shown in Figure 19.

Hydrodynamic loads are directly transferred from the wave load analysis program to the finite element models. Hydrodynamic loads include panel pressures, internal tank pressures and inertia forces due to rigid body accelerations. By direct load transfer the stress response transfer functions are implicitly described by the FE analysis results. All wave headings from 0 to 360 degrees with an increment of 30 degrees were included. For each wave heading 19 wave frequencies were included to properly describe the shape of the transfer functions.

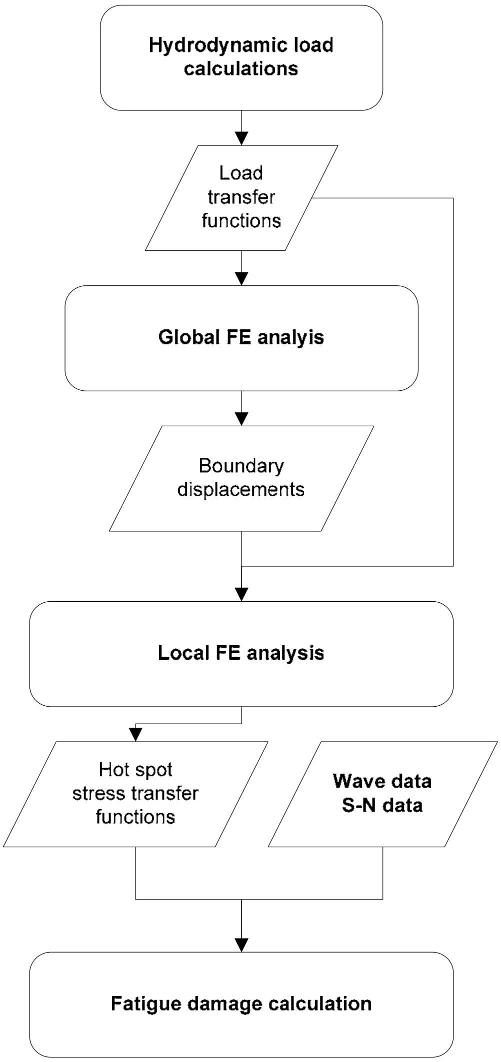


Figure 19: Flow Diagram for Full Stochastic Fatigue Calculations

The analysis was based on a global finite element model of the entire vessel and local models of selected critical details in the hull. The local models were used as sub models to the global analysis and the displacements from the global analysis were transferred to the local model as boundary displacements. From the local stress concentration models geometric stress transfer functions at hot spots are determined using element sizes in the order of the plate thickness to pick up the geometric stress increase.

The hot spot transfer functions are combined with the wave scatter diagram and S-N data and the fatigue damage was calculated by summation of part damages from each wave period and heading for every sea state in the scatter diagram using the program STOFAT [3].

All load effects are preserved through the calculations and hence the method is suitable for fatigue calculations of details with complex stress pattern.

As for the hull strength analysis similar mass properties are ensured using the structural model as mass model in the hydrodynamic analysis. The quality of the load transfer was checked by comparing transfer functions for sectional loads for different wave periods, see Figure 20.

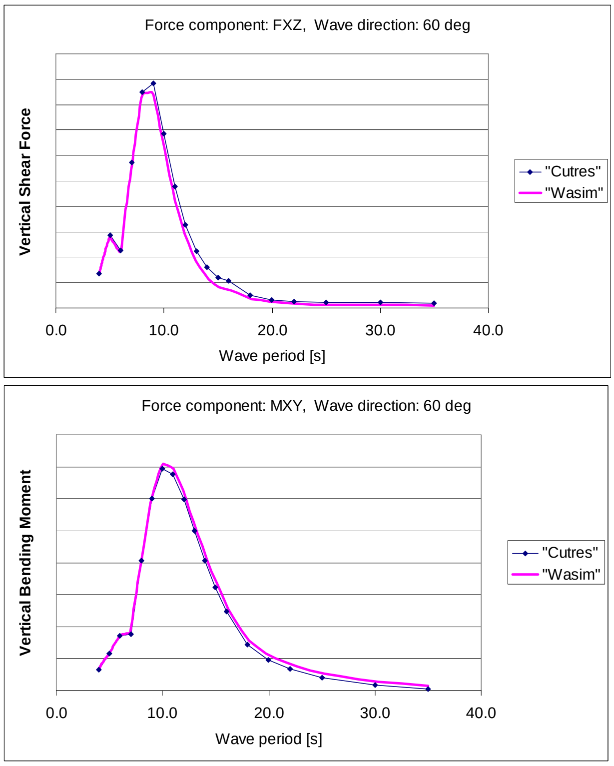


Figure 20: Comparison of Sectional Loads as Computed by Wave Load Analysis (Wasim) and Structural Analysis (Cutres)

5.2 (b) Local Analysis of Critical Details

Detailed fatigue analyses using local stress concentration models were performed for critical joints in the hull including:

- Upper and lower hopper knuckles
- Upper and lower chamfer knuckles
- Double bottom girder connection at transverse bulkhead
- Stringer connection at transverse bulkhead
- Midship dome opening
- Termination of tank no. 1 longitudinal bulkhead

The locations are shown in Figure 21.

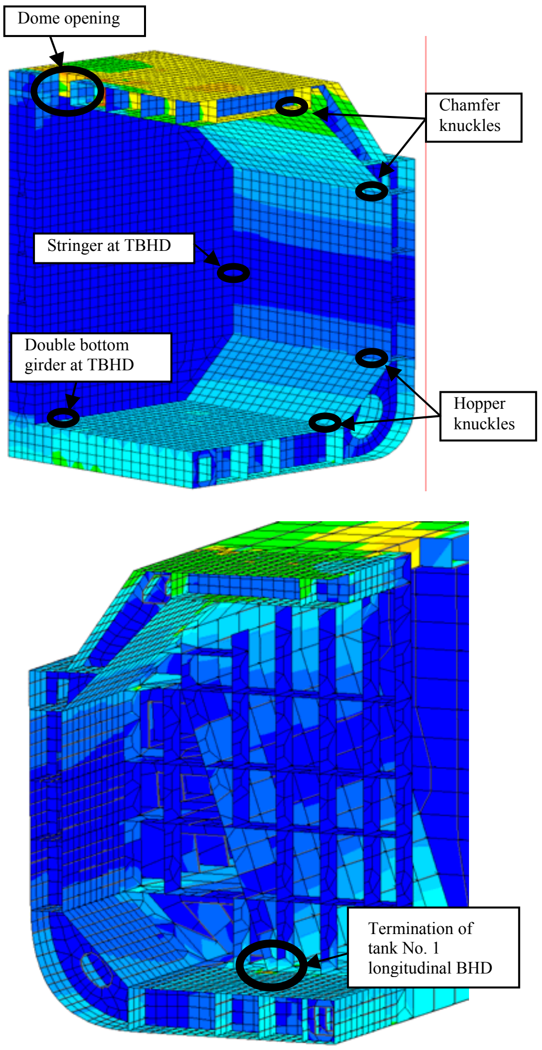


Figure 21: Details Subjected to Detailed Fatigue Analysis by Stress Concentration Models

For each joint a screening analysis was carried out to identify potential crack locations. This is exemplified in Figure 22 showing the fatigue screening results for the dome opening. Finally a detailed analysis was carried out for the potential crack locations including relevant stress concentrations not picked up by the finite element models.

For web stiffened cruciform joint connections shell element models do not fully represent the three dimensional geometry of the joint and 2nd level submodel by solid elements were made to investigate the difference between shell and solid element models. A comparison of the calculated fatigue damages for the double bottom girder connection at transverse bulkhead is shown in Figure 23. In this case the calculated fatigue damage was reduced by a factor of 0.6 based on stress values at a distance equal to half the plate thickness from the hot spot.

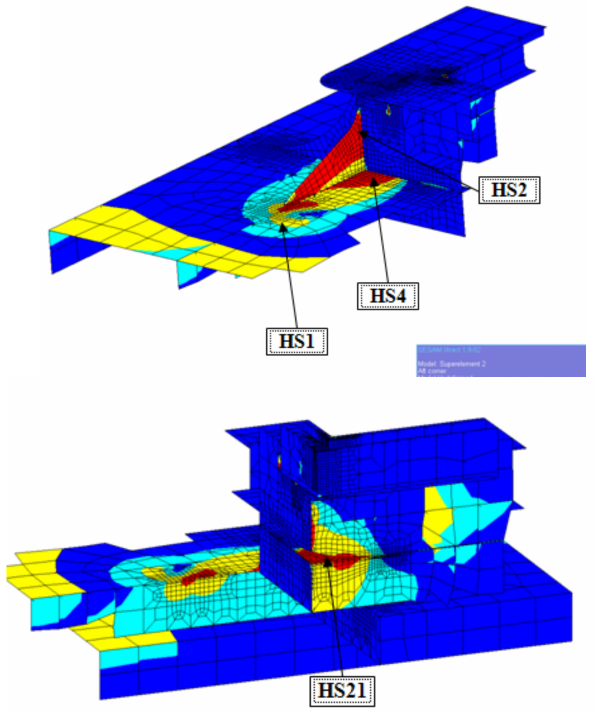


Figure 22: Fatigue Screening of Midship Dome opening

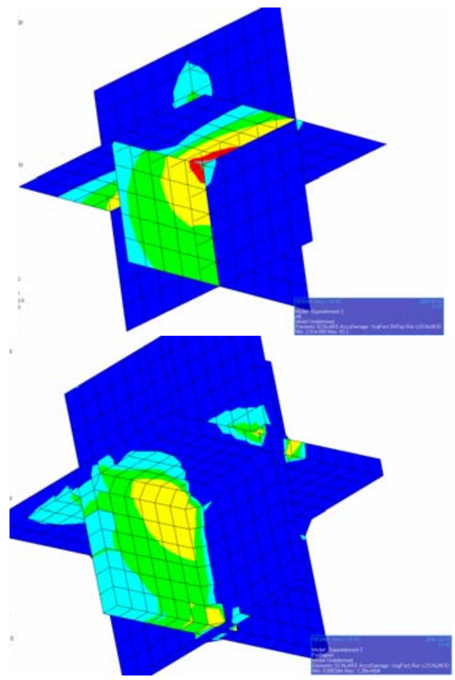


Figure 23: Calculated Fatigue Damage by Shell and Solid Element Model of the Double Bottom Girder Connection to Transverse Bulkhead

In general fatigue strengthening was required in order to fulfil the required fatigue design life of 40 years. The strengthening included weld toe grinding and profiling and local thickness increase by inserts at the most critical locations. For the dome opening double deck bending stresses, see Chapter 4.3, cause a significant stress increase compared to nominal hull girder stresses which should be considered in the fatigue design of gas carriers.

5.2 (c) Global Fatigue Screening

Spectral fatigue screening analyses or the global models was performed to identify fatigue prone areas, determine critical stress concentration factors for the trunk deck and evaluate the fatigue strength of critical joints relative to the results from the local analysis.

The global fatigue screening analysis was based on the nominal stress transfer functions from the global model combined with representative stress concentration factors.

For the trunk deck several runs were performed for different stress concentration factors representing thickness transitions, doubling plates, openings and pipe penetrations and details and outfitting welded to the deck plate. An example plot from the evaluation is shown in Figure 24. The results showed that care should be taken to avoid high stress concentration factors in the area of the transverse bulkhead between tank nos. 3 and 4.

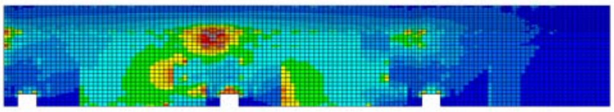


Figure 24: Fatigue Screening Results for the Trunk Deck Tank Nos. 2-4

Through the fatigue screening analysis it was found that relatively high forces were transferred from the trunk deck through the superstructure and into the main deck. This is illustrated by the fatigue damage contour plot shown in Figure 25.

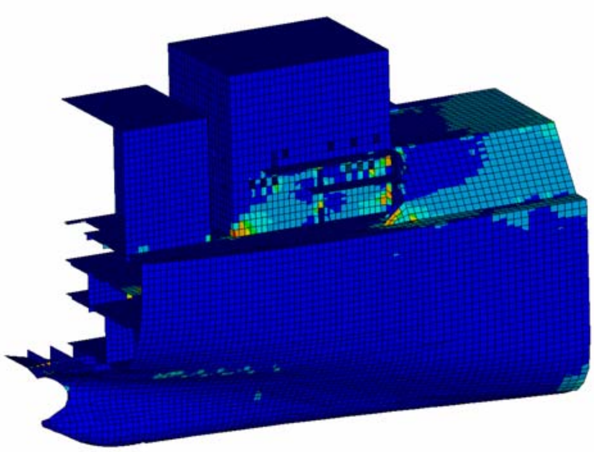


Figure 25: Fatigue Screening Results for Aft Ship

For assessment of the fatigue strength of the joints considered in the detailed analysis at other locations in the cargo area, the fatigue screening results were scaled based on the results from the local analysis. The scaling factor was calculated as the ratio of the calculated fatigue damage for the detailed analysis divided by the fatigue damage from the screening analysis at the same location. This scaling is valid under the assumption that the stress concentration of the detail considered in the local stochastic analysis is representative for similar details at other locations As the stress concentration is depending on the finite element mesh, local geometry, the position and load pattern, some additional uncertainties are introduced. The results, presented in Figure 26, served as basis for deciding the extent of fatigue strengthening.

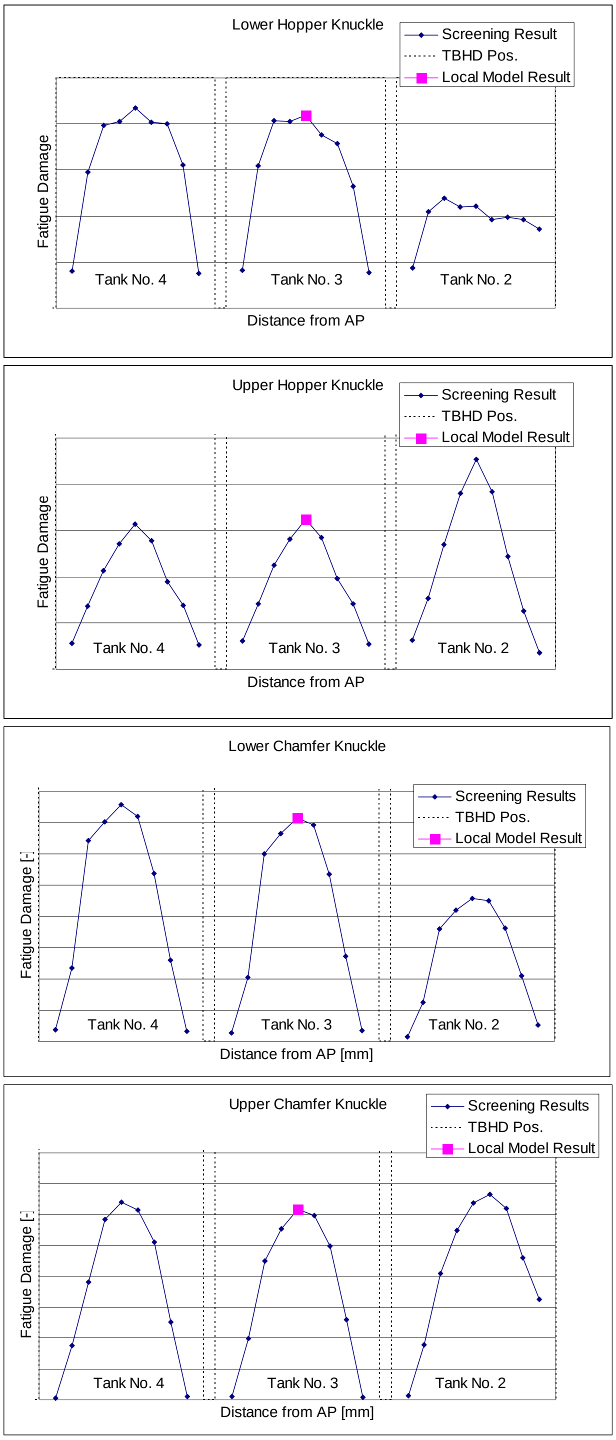


Figure 26: Fatigue Screening Results for Inner Hull Knuckles

6. CONCLUSIONS

The CSA-2 analysis carried out for a 216,000 m^3 LNG carrier is described. Through the CSA-2 analysis the yield, buckling and fatigue strength of the hull are evaluated by direct hydrodynamic load and stress response analyses and fully documented for the owner and class by technical reports.

The design hogging and sagging moments calculated by non-liner wave load analyses exceeded the IACS unified requirements by 6% in sagging and 13% in hogging. Fatigue loads calculated based on the North Atlantic scatter diagram were in the order of 10-20% above the DNV Rule and Common Scantling Rule values.

Through the structural analysis if was found that bending of the double deck at transverse bulkheads cause a significant stress increase compared to nominal hull girder stresses and should be considered both in hull strength and fatigue calculations of gas carriers. It was also found that relatively high forces were transferred from the trunk deck through the superstructure and into the main deck and care should be taken in the design of openings and knuckles in this area to avoid fatigue cracking. With respect to fatigue strength the inner hull knuckles are the most critical area for membrane type gas carriers. This is both due to high stress concentrations and the fact that these locations are covered by the cargo containment system and thus inaccessible for inspection and repair.

In order to fulfil the CSA-2 requirements additional strengthening of the hull was required compared to standard rule requirements, both with respect to yield and buckling strength and fatigue, leading to a more robust vessel.

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