Fatigue methodology of offshore ships
FOREWORD

DNV GL recommended practices contain sound engineering practice and guidance.
CHANGES – CURRENT

General
This document supersedes DNV-RP-C206, October 2012.

Text affected by the main changes in this edition is highlighted in red colour. However, if the changes involve a whole chapter, section or sub-section, normally only the title will be in red colour.

On 12 September 2013, DNV and GL merged to form DNV GL Group. On 25 November 2013 Det Norske Veritas AS became the 100% shareholder of Germanischer Lloyd SE, the parent company of the GL Group, and on 27 November 2013 Det Norske Veritas AS, company registration number 945 748 931, changed its name to DNV GL AS. For further information, see www.dnvgl.com. Any reference in this document to "Det Norske Veritas AS", “Det Norske Veritas”, “DNV”, “GL”, “Germanischer Lloyd SE”, “GL Group” or any other legal entity name or trading name presently owned by the DNV GL Group shall therefore also be considered a reference to “DNV GL AS”.

Main changes July 2015

- General structure
  The revision of this document is part of the DNV GL merger, updating the previous DNV recommended practice into a DNV GL format including updated nomenclature and document reference numbering, e.g.:
  — DNV replaced by DNV GL.
  — DNV-OS-C101 replaced by DNVGL-OS-C101 etc.

  A complete listing with updated reference numbers can be found on DNV GL’s homepage on internet.

  To complete your understanding, observe that the entire DNV GL update process will be implemented sequentially. Hence, for some of the references, still the legacy DNV documents apply and are explicitly indicated as such, e.g.: Rules for Ships has become DNV Rules for Ships.

Editorial corrections
In addition to the above stated main changes, editorial corrections may have been made.
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SECTION 1 DEFINITIONS

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1.1 Latin symbols

Table 1-1 Latin symbols

<table>
<thead>
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<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>cross sectional area</td>
</tr>
<tr>
<td>B</td>
<td>greatest moulded breadth of ship in meters measured at the summer waterline</td>
</tr>
<tr>
<td>(C_B)</td>
<td>block coefficient = (\Delta/1.025LB) according to DNV Rules for Ships Pt.3 Ch.1 Sec.1 B101</td>
</tr>
<tr>
<td>(C_w)</td>
<td>wave coefficient according to DNV Rules for Ships Pt.3, Ch.1.</td>
</tr>
<tr>
<td>D</td>
<td>moulded depth of ship according to DNV Rules Pt.3 Ch.1 Sec.1</td>
</tr>
<tr>
<td>(D_{FF})</td>
<td>design fatigue factor (safety factor on fatigue life)</td>
</tr>
<tr>
<td>(F_{\Delta\sigma}(\Delta\sigma))</td>
<td>Weibull distribution</td>
</tr>
<tr>
<td>(H(\omega))</td>
<td>transfer function</td>
</tr>
<tr>
<td>(H_s)</td>
<td>significant wave height</td>
</tr>
<tr>
<td>I</td>
<td>moment of inertia</td>
</tr>
<tr>
<td>(I_a)</td>
<td>moment of inertia for the transverse frame</td>
</tr>
<tr>
<td>(I_b)</td>
<td>moment of inertia for the longitudinal stringer / girder</td>
</tr>
<tr>
<td>K</td>
<td>stress concentration factor</td>
</tr>
<tr>
<td>(K_g)</td>
<td>geometric stress concentration factor</td>
</tr>
<tr>
<td>(K_n)</td>
<td>stress concentration factor for asymmetrical stiffeners with lateral loading</td>
</tr>
<tr>
<td>(K_{te})</td>
<td>stress concentration factor due to eccentric tolerance (normally plate connections)</td>
</tr>
<tr>
<td>(K_{ta})</td>
<td>stress concentration factor due to angular mismatch (normally plate connections)</td>
</tr>
<tr>
<td>L</td>
<td>length of ship in meters, according to DNV Rules for Ships Pt.3 Ch.1 Sec.1.</td>
</tr>
<tr>
<td>(L_{pp})</td>
<td>length between perpendiculars</td>
</tr>
<tr>
<td>M</td>
<td>Moment</td>
</tr>
<tr>
<td>(M_{wo})</td>
<td>wave induced vertical moment</td>
</tr>
<tr>
<td>(M_H)</td>
<td>wave induced horizontal moment</td>
</tr>
<tr>
<td>(N_S)</td>
<td>number of cross ties</td>
</tr>
<tr>
<td>(Q(\Delta\sigma))</td>
<td>probability level for exceeding the stress range (\Delta\sigma)</td>
</tr>
<tr>
<td>S</td>
<td>sum of plate flange width on each side of the horizontal stringer</td>
</tr>
<tr>
<td>(S_{w}(\omega))</td>
<td>wave spectrum</td>
</tr>
<tr>
<td>(S_{\sigma}(\omega))</td>
<td>stress response spectrum</td>
</tr>
<tr>
<td>T</td>
<td>mean moulded summer draft in meters</td>
</tr>
<tr>
<td>(T_{act})</td>
<td>actual draft</td>
</tr>
<tr>
<td>(T_d)</td>
<td>design life</td>
</tr>
<tr>
<td>(T_z)</td>
<td>zero crossing period</td>
</tr>
<tr>
<td>Z</td>
<td>section modulus</td>
</tr>
</tbody>
</table>
### 1.2 Greek symbols

#### Table 1-2 Greek symbols

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>δ</td>
<td>deformation</td>
</tr>
<tr>
<td>ψ_ij</td>
<td>zero crossing frequency in short-term condition i, j</td>
</tr>
<tr>
<td>ω</td>
<td>wave frequency</td>
</tr>
<tr>
<td>ν₀</td>
<td>long-term average zero frequency</td>
</tr>
<tr>
<td>ρ</td>
<td>correlation coefficient</td>
</tr>
<tr>
<td>σ</td>
<td>stress amplitude</td>
</tr>
<tr>
<td>σ₂</td>
<td>secondary stress amplitude resulting from bending of girder system</td>
</tr>
<tr>
<td>σ₃</td>
<td>tertiary stress amplitude produced by bending of plate elements between longitudinal and transverse frames/ stiffeners</td>
</tr>
<tr>
<td>σ_{nominal}</td>
<td>nominal stress amplitude, e.g. stress derived from beam element or finite element analysis</td>
</tr>
<tr>
<td>η</td>
<td>fatigue usage factor</td>
</tr>
</tbody>
</table>
### Table 1-2  Greek symbols  (Continued)

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\Delta$</td>
<td>moulded displacement (tonnes) in salt water on draft $T$ (salt water density 1.025 tonnes/m$^3$)</td>
</tr>
<tr>
<td>$\Delta \sigma$</td>
<td>stress range</td>
</tr>
<tr>
<td>$\Delta \sigma_s$</td>
<td>stress in the weld normal to the throat</td>
</tr>
<tr>
<td>$\Delta \sigma_g$</td>
<td>global stress range</td>
</tr>
<tr>
<td>$\Delta \sigma_l$</td>
<td>local stress range</td>
</tr>
<tr>
<td>$\Delta \sigma_h$</td>
<td>nominal stress range due to horizontal bending</td>
</tr>
<tr>
<td>$\Delta \sigma_v$</td>
<td>nominal stress range due to vertical bending</td>
</tr>
<tr>
<td>$\Delta \sigma_w$</td>
<td>shear force stress range in the weld to be used together with S-N curve $W$</td>
</tr>
<tr>
<td>$\Delta \tau_s$</td>
<td>shear stress range in the weld throat</td>
</tr>
<tr>
<td>$\Delta \tau_v$</td>
<td>shear stress range in the weld throat parallel to the weld</td>
</tr>
<tr>
<td>$\Gamma(\cdot)$</td>
<td>Gamma function</td>
</tr>
<tr>
<td>$\gamma$</td>
<td>Peak enhancement factor</td>
</tr>
</tbody>
</table>
SECTION 2 OVERVIEW OF HULL FATIGUE ANALYSIS

2.1 Introduction
The following fatigue cracking failure modes are considered in this document (see also Figure 2-1):

Fatigue crack growth from the weld toe into the base material

In welded structures fatigue cracking from weld toes into the base material is a frequent failure mode. The fatigue crack is initiated at small defects or undercuts at the weld toe where the stress is highest due to the weld notch geometry. A large amount of the content in this classification note is made with the purpose of achieving a reliable design with respect to this failure mode.

Fatigue crack growth from the weld root through the fillet weld

Fatigue cracking from root of fillet welds with a crack growth through the weld is a failure mode that can lead to significant consequences. Use of fillet welds should be avoided in connections where the failure consequences are large due to less reliable NDE of this type of connection compared with a full penetration weld. However, in many welded connections use of fillet welds can hardly be avoided and it is also efficient for fabrication. The specified design procedure in this document is considered to provide reliable connections also for fillet welds.

Fatigue crack growth from the weld root into the section under the weld

Fatigue crack growth from the weld root into the section under the weld is observed during service life of structures and it is observed in laboratory fatigue testing. The number of cycles to failure for this failure mode is of a similar magnitude as fatigue cracking from the weld toe in as welded condition. There is no methodology that can be recommended used to avoid this failure mode except from using alternative types of welds locally. This means that if fatigue life improvement of the weld toe is required the connection will become more highly utilized and it is also required to make improvement for the root. This can be performed using a full penetration weld along some distance of the stiffener nose.

Fatigue crack growth from a surface irregularity or notch into the base material

Fatigue cracking in the base material is a failure mode that is of concern in components with high stress cycles. The fatigue cracks often initiate from notches or grooves in the components or from small surface defects/irregularities. The specified design procedure in this document is considered to provide reliable connections also with respect to this failure mode.

a) Fatigue crack growth from the weld toe into the base material
2.2 Fatigue evaluation procedure for hull details

In terms of fatigue design, a design cycle may be considered as consisting of two distinct phases; a Preliminary Design Phase and a Fatigue Design Phase.

A brief description of the different fatigue phases are given below.

Preliminary design phase

The aim of the preliminary design phase is to ensure that the main scantlings are adequately designed with respect to fatigue to allow confidence when ordering steel, therefore avoiding potentially costly modifications later in the construction phase.

The initial scantlings are often based on strength (structural capacity) considerations, with minor attention related to the fatigue capacity requirements. As fatigue capacity is generally more important for unit’s stay on location without possibility of dry docking, e.g. FPSOs, than for conventional tankers, fatigue design should be given sufficient consideration prior to steel order/main scantling approval. Hydrodynamic loads
should be based on the site specific scatter diagram. However, for units intended for unrestricted service (World Wide operation) worldwide scatter diagram should be used for units intended to be dry docked every 5th year. See also DNVGL-OS-C102 Ch.2, Sec.2 The mass of unit's specific equipment/modules can change both global and local stress ranges and hence this mass should be included as early as possible in the analyses.

Simplified fatigue analysis can be performed to calculate initial scantlings for at least four (4) sections outside the midship area. Depending on the wave climate, it may be advisable to perform this analysis using the load component based approach described in 2.5.3.1, in which case a hydrodynamic analysis is also required.

The bulkhead/frame relative deflections and associated nominal stresses can be determined in several ways. It is recommended that nominal stresses are calculated using a beam or shell model, as these models are required during the “Fatigue Design Phase” and also used for general strength analysis.

**Fatigue design phase**

The fatigue design phase of the project is used to document the estimated fatigue capacity for a selection of structural details. A suitable selection of details shall be analysed to ensure that the worst of equal local details meet the fatigue requirements. Screening analysis is therefore important in order to identify the most fatigue prone areas that should be selected for further analysis. At the completion of this phase an overall understanding of the hull fatigue performance should be achieved.

Typical fatigue calculations to be performed in this phase are:

- load component based fatigue analyses for minimum five (5) sections along the unit. Results from dynamic sea pressure calculation at waterline should be basis for selection of cross sections. At least one section at a transverse bulkhead shall be analysed
- fatigue screening, using global or part-ship models, in order to ensure that areas other than those analysed have satisfactory fatigue lives (see [3.2]).

Examples of details that are likely to be selected for analysis are listed in Table 3-1.

Fatigue calculations using stress concentration models may be necessary for fatigue sensitive areas where adequate geometric stress concentration factors, \( K_g \), do not exist. Stiffener lugs connected to transverse frames/bulkheads are a typical case where such calculations are required.

The bulkhead/frame relative deflections and associated nominal stresses should be established from a beam or shell element model of the region under consideration.

In order to obtain a thorough overview of the fatigue performance for the unit’s hull, fatigue calculations need to be completed for many details. Analytical focus on screening is necessary, as described in Section 3. Several approaches are available to the fatigue designer that varies in the level of complexity and required information. At this stage it is anticipated that the hydrodynamic and cargo hold FE analyses (3 holds) have been completed and should be used to establish a more comprehensive overview of the hull fatigue performance.

Three primary fatigue approaches, depending on the detail type, may be used at this stage to identify where further calculations should be conducted. For transverse frame gussets, hopper knuckles, stringer connections, etc. a unit load applied to the cargo hold FE model may be used to identify the hot spot locations (see [3.2]). Combining the results from both the unit load approach and the Preliminary Design Phase the designer will be able to identify the details to be analysed further.

Using the global FE model, the nominal global stresses in the hull girder may be more accurately obtained compared to a section scantlings approach since this will include effects such as shear lag. Similarly the part-ship or cargo hold FE models can be used for this purpose, although the shear lag effects may not be as accurately captured as for a global model. In addition these models provide more accurate relative deflection magnitudes for updating the nominal stresses for the longitudinal stiffeners that intersect a transverse bulkhead. The revised nominal stresses can be combined with appropriate SCFs to obtain the hot spot stress. These stresses may then be used for post-processing with the hydrodynamic loads to determine revised fatigue damages, as described in [2.4.3.1].

If tabulated SCFs are used then the fatigue calculations are limited to the validity of the SCFs as presented
in Classification Notes No. 30.7 Appendix A. If tabulated values are not available, it will be necessary to make local FEM model of the detail in accordance with the requirements to the hot spot method as described in 4.5.

Load component stochastic method is found to be sufficient for longitudinal hull members such as stiffeners, side shell, deck and bottom. Fatigue critical details should be analysed using an integrated model in order to capture all load effects hence reducing uncertainties.

For an internal turret moored FPSO additional fatigue design is necessary. Generally one additional cross section through the turret region should be considered due to the changes in the hull girder section properties and the associated global stress concentration factors (see [8.3.2]). However the approach described above is valid only for the details that are not influenced by the turret loads, as indicated in Figure 8-1. Within the region of influence the hull / turret interface details are to be screened using the approach described in [8.6.2].

If sufficient fatigue life can not be reached using load component stochastic analysis further refinement in analysis methodology using full stochastic analysis can be performed as described in [2.4.2]. Alternatively, redesign of the details shall be considered.

Many of the stress concentration models will be selected based on experience and / or lack of an available geometric stress concentration factor, $K_g$, for the detail. $K_g$ factors may not be available for unit’s specific details such as the foundations for topside stools, riser porches, helidecks, flare towers and caisson, as there is no equivalent detail for trading ships. Development of stress concentration models on the basis of experience may commence at an earlier phase in the project.

### 2.3 Unit configuration and operation

#### 2.3.1 Introduction

In order to select the suitable fatigue analysis method for different structural details and at different design phases, the following issues related to the unit’s mooring configuration (see [2.3.2]) and operation (see [2.3.3]) should be considered.

#### 2.3.2 Mooring configuration

Typical FPSO mooring configurations include the following two types:

* **Spread-moored design**

  The global effect of the mooring may be ignored with respect to hull girder response, but should be considered in the local analysis of chain stopper and hull supporting structure.

  Typically the unit will maintain a constant heading, relative to the compass, with this configuration.

* **Turret-moored design**

  Turrets may either be located internally within the hull envelope or externally and allow the FPSO to orient into the incoming weather. The mooring lines are connected directly to the turret. Internal turrets often significantly affect the hull geometry and scantlings. The global influence of the anchoring loads on the hull girder can normally be ignored. For internal turret configurations the supporting structure in the vicinity of the centre line of the ship will be exposed to extreme loads independent on the wave direction. A typical internal turret is shown in Figure 8-12.

#### 2.3.3 Operation

The unit’s operational profile will determine maximum and minimum draft, and these parameters are important input into the fatigue analysis (see [5.2]). Low cyclic loads due to continuous loading/unloading are also dependent on the unit’s operational characteristics (see [6.13.2]).
2.4 Spectral fatigue analysis methodology

2.4.1 General principles
This RP defines two principal fatigue analysis methodologies:

— the full stochastic method using either a part-ship or global model, and
— the load component method.

These methods are based on a spectral procedure, which includes the following assumptions for calculation of fatigue damage:

— wave climate is represented by scatter diagrams (summation of short term conditions)
— Rayleigh distribution applies for stresses within each short term condition
— cycle count is according to zero crossing period of short term stress response
— Miner summation is according to linear cumulative damage.

The spectral method assumes linear load effects and responses. The hydrodynamic loads and structural responses should be calculated using 3D potential theory and finite element analysis, respectively. Details are provided in Section 4.

Stresses used for calculation of fatigue are based on hot spot stress methods using stochastic methods. The hot spot stress is either calculated using a stress concentration factor model or derived from nominal stresses combined with associated stress concentration factors.

However, the analyses should include relevant non-linear effects that affect stress and have a probability of exceedance level larger than $10^{-4}$. An example of such an effect is the intermittent wetting of the side shell and the resulting effect on the Linearised pressure loads.

Other load effects, such as slowly varying response, impact loads, should be included if they influence the fatigue life. See [5.2] and [6.13].

2.4.2 Full stochastic fatigue analysis methodology
A full stochastic fatigue analysis can be performed using either a global or a part-ship (3 holds) structural model of the unit. This method requires that the wave loads are transferred directly from the hydrodynamic model to the structural model. External wave loads, internal tank loads and inertia loads shall be considered in a consistent manner to maintain equilibrium.

When using a part-ship model special attention is to be given for the boundary conditions. With a part-ship model the dynamic loads outside of the model extent need to be included to obtain a balanced model. The method described in the RP uses section forces and moments applied at the ends of the model, to represent the part of the unit that is not modelled. Shear lag effects can be important for fatigue analysis and need to be addresses when using a part-ship model. Normally this method should be avoided if there are no longitudinal bulkheads present.

Quality assurance is important when executing the full stochastic method. With both the part-ship and global model approach, the structural and hydrodynamic analysis results should have equal shape and magnitude for the bending moment and shear force diagrams. Also, the reaction forces in the structural analysis should be minimal. Guidance on the procedure is given in [4.2] and [4.3].

If the midship model (part-ship) is available early in the design phase, the part-ship model procedure may be used during the “fatigue design phase” as it is an efficient and effective analysis approach.

Figure 2-2 and Figure 2-3 show example flow charts for implementing the full stochastic fatigue analysis using either a global or part-ship model, respectively. References to relevant sections in this RP are given for each step.
Figure 2-2  Example full stochastic analysis procedure flowchart – global model

The advantage of a global direct stochastic analysis is that all linear load effects are automatically included via an integrated hydrodynamic/structural program. The fineness of the panel and finite element mesh should correspond with the type of the analysis. A global structural model is usually constructed using a relatively coarse mesh (e.g. one element between frames), which provides reliable calculation of nominal stresses in shell and deck plating. These stresses do not include tertiary stresses (see App.A.1.3 for definition of tertiary stress). As an example, nominal stress from global load effects may also be derived by using one element between frames.

The advantages to using the part-ship full stochastic analysis are similar to the global approach. Since a 3-hold model is usually sufficient, modelling and analysis time may be significantly reduced. However, some additional manual effort is required to accurately transfer loads and define boundary conditions.
The global FE analysis shall be run for all wave load cases, i.e. 12 - 16 headings and 25 - 30 wave periods per heading, amounting to a total of 300 - 480 load cases for each basic loading condition. Resulting deformations are then transferred from the global FE model to the local stress concentration (SCF) model where they form the boundary displacements for each corresponding load case. In addition, the local internal and external hydrodynamic pressures should be automatically transferred from the wave load program to the SCF model. The fatigue damage contribution from each cell in the wave scatter diagram can then be calculated based on the principal stresses from the local FE model at the hot spot, the wave spectrum, wave spreading function, S-N data, etc.

It is important that the results from the global model are checked before the displacements/forces for each load case are transferred to the boundaries of the SCF model. The boundaries of the SCF model should correspond with element boundaries in the parent model where the displacements/forces are known to be correct (see [4.5]). For example, relatively large element sizes in the global model can give a stiffer frame system that can cause the relative deflection of the frame adjacent to the bulkhead to be incorrect. In such cases, intermediate sub-models with a medium coarse mesh covering a larger area of the global model may be required to capture the correct frame deflections.

The requirements for developing stress concentration factor models are given in [4.5].
2.4.3 Load component stochastic fatigue methodology

In a load component based fatigue analysis, the stress from each load component (load case) is calculated based on the loads from the hydrodynamic program. Phasing between the different load components is included in the load transfer functions and the total stress can then be calculated from a summation of the different stress contributions (see [6.8]). A stochastic fatigue evaluation, where the simultaneous occurrence of the different load effects is preserved, may then be performed for the combined stress transfer function.

There are, in general, no limitations for the use of the load component based method provided that all stress effects contributing to the total stress can be isolated and are included in the specific analysis.

It is important that stress effects are not duplicated in the analyses. Duplication can readily occur in cases where effects are difficult to separate from one another. Application of local loads can also result in undesirable effects, such as fictitious global bending moments. See [4.3] for load application principles and boundary conditions. Practical guidance on the application of unit loads is given in App.A

Use of the load component based method offers the major advantage that non-linear effects, such as the pressure reduction due to intermittent wet surfaces in the waterline region, can be relatively easily taken into account.

The following load transfer functions from the hydrodynamic program are normally included as a basis for the load components:

— hull girder horizontal and vertical bending moments and axial force (shear lag effects may not be fully captured).
— unit motions and accelerations in six degrees of freedom (d.o.f.).
— pressures for all panels of the 3D diffraction model.

The different load effects are modelled as unit static loads. The load/stress ratio for each unit load is calculated and applied as a factor to the appropriate complex load transfer function from the hydrodynamic analysis. The general flowchart shown in Figure 2-4 may be followed for the analysis. More detailed flowcharts for the stress combination and fatigue calculations are given in the relevant subsections below.
Different levels of the load component based method are described in 2.5.3.1 and 2.5.3.2 below.

### 2.4.3.1 Load component based analysis with use of SCF models

Use of SCF models is the most thorough component based fatigue analysis method. Stress concentration models are used for calculation of hot spot stresses, and do not necessitate separation of local axial stress from bending stress. The principles for establishing and using stress concentration models are given in [4.5] and in Figure 2-4.

Figure 2-5 shows a flowchart for combination of stress transfer functions in order to obtain combined stress for use in fatigue calculations.
2.4.3.2 Load component fatigue analysis using available SCFs (K-factors)

This approach will normally be used in the fatigue design phase of the project, but can also be used in the preliminary design phase.

If stress concentration factors for the given detail are available (see Classification Notes No. 30.7 Appendix A), then calculation of fatigue life may be based on the stress concentration factors combined with nominal stresses. The main difference between this approach and that given in [2.4.3.1] is that different K factors may be required, depending on whether the stress is caused by axial load or bending load at the actual location (see App.A).

The flowchart shown in Figure 2-6 presents an overview of the combination of stress transfer functions in order to give a combined stress for use in subsequent fatigue calculations. It should be noted that this approach using SCFs is applicable only for geometries with similar dimensions to those for which the K-Factors are derived. Stress concentration models should be used for geometries outside this range.

2.5 Fatigue damage calculation

The fatigue damage calculations should be performed according to one of the methods described in DNVGL-RP-C203 or DNV Classification Notes No. 30.7. Damage contributions from each of the analysed loading conditions should be summed based on the fraction of time that each load condition is expected to be present.

There is experimental evidence which shows that the Miner sum at fatigue failure varies considerably around the nominal value of 1.0. When using frequency domain analyses, other conservation in the analysis methodology balance potential non-conservatism in the Miner sum. However, this conservatism is not present when rain flow counting of stress cycles is applied. Therefore, when performing fatigue analyses in the time domain, the Miner sum is limited to 0.5. Appropriate safety factors shall also be applied to the damaged areas.
2.6 Definition of stress concentration factor

Stress concentration factors (K-factors) may be determined based on fine mesh finite element analyses as described in 4.5. Alternatively, a suitable tabulated K-factor may be selected, see Classification Notes No. 30.7 Appendix A.

The fatigue life of a weld toe detail is governed by the hot spot stress range. For welded components other than butt welded connections, the hot spot stress is obtained by combining the nominal stress with the geometric K-factor, $K_g$. K-factors are defined as:

$$K_g = \frac{\sigma_{\text{hot spot}}}{\sigma_{\text{nominal}}}$$  \hspace{1cm} (1)

The hot spot S-N curve (curve E) in DNVGL-RP-C203 is given for a butt welded specimen where the hot spot stress is equal to the nominal stress, i.e. $K_g = 1.0$.

The relation between the hot spot stress range to be used together with the S-N curve and the nominal stress range is:

$$\Delta\sigma_{\text{hot spot}} = K_g \Delta\sigma_{\text{nominal}}$$  \hspace{1cm} (2)
All stress raisers (excluding the localized stress concentration due to the weld profile) shall be considered when evaluating the hot spot stress. This is achieved by multiplication of the K-factors arising from different causes. The resulting K-factor to be used for calculation of hot spot stress is:

\[ K = K_g K_{te} K_{ta} K_n \]  

where

- \( K_g \) = stress concentration factor due to the gross geometry of the detail under consideration
- \( K_{te} \) = additional stress concentration factor due to eccentricity tolerance (normally used for plate butt weld connections and cruciform joints only)
- \( K_{ta} \) = additional stress concentration factor due to angular mismatch (normally used for plate connections only)
- \( K_n \) = additional stress concentration factor for asymmetrical stiffeners on laterally loaded panels, applicable when the nominal stress is derived from simple beam analyses

The K-factors for typical details in ships are presented in Classification Notes No. 30.7 Appendix A. Classification Notes No. 30.7 Appendix F gives default values for workmanship tolerances based on what is considered to be normal shipyard practice. If greater tolerances are used, the K-factors should be calculated based on actual tolerances, see DNVGL-RP-C203.

2.7 Definition of S-N curves
Reference is made to DNVGL-RP-C203 for S-N curves for:
- base metal,
- toe-cracking of welded joints, dependent on the predominant direction of principal stress, type on connection detail etc., and
- root weld failure in fillet or partial penetration welds.

2.8 Design fatigue factors
Design fatigue factors (DFF) for the FMS notation are given in DNVGL-OS-C102 Ch.2 Sec.7.2.
SECTION 3 SELECTION OF STRUCTURAL DETAILS FOR FATIGUE ANALYSIS

3.1 Typical area and example of critical details
Since offshore ship shape units are a mix of ship structures and offshore specific details, experience from ship structures with similar details is important input for analysis of the hot spot areas.

The fatigue life is related to the magnitude of the dynamic stress level, the number of load cycles, the corrosiveness of the environment, and the magnitude of stress concentration factors for the structural details. These factors are dependent on the ship design. Hence, some ship types and categories of hull structural elements are known to be more susceptible to fatigue damage than others. The importance of potential fatigue damage is related to its consequence, in terms of safety and economic implications, which may require shut down for repair. The relative importance increases with the increasing number of potential hot spots with similar geometry and loading, e.g. side shell longitudinal stiffener connections.

A large proportion of all ship structure fatigue damage occurs in panel stiffeners on the ship side and bottom and at the tank boundaries of ballast and cargo tanks. The calculated fatigue life depends on the type of stiffeners used and the detail design of the connection to supporting girder webs and bulkheads. Asymmetrical profiles will normally have a reduced fatigue life compared to symmetrical profiles unless the reduced efficiency of the asymmetrical profile is compensated by improving the design for the attachment to transverse girder webs and bulkhead structures.

Typical areas and details to be checked for the FMS notation are given in DNVGL-OS-C102 Ch.2 Sec.7.

3.2 Screening of structural details for fatigue analysis
Fatigue screening is the process of identifying critical locations for detailed analysis. In tanker design, the critical details are fairly well known, given the vast experience base. However, this experienced based screening is not completely applicable to FPSOs, given the limited operational experience, different loading modes, operational requirements and structural details.

Offshore units include many structural details that may be critical from a fatigue design viewpoint. Fatigue screening during the design process is therefore a very important step for achieving fatigue control. Fatigue screening has the following objectives:

— to ensure that all critical structural details meet the fatigue design requirements, and
— to select a limited number of critical details for refined stress analysis.

Screening involves various levels of complexity, depending on the loading, location and previous experience. Consequently, fatigue screening is involved during fatigue design phases discussed in 2.2.

3.2.1 Initial assessment of longitudinal structural members

3.2.1.1 Basic design
In the initial phase of the design process, it is important to establish the main scantlings with reasonable accuracy and with minimal effort. All limit states and requirements that could be governing for the scantlings should be checked.

In order to achieve a sound fatigue life for FPSO hull details, the maximum allowable stress concentration factor, $K_g$, shall be specified. The $K_g$ factor should be determined based on welding details that are normally present in the cargo deck. An economical value for maximum $K_g$ is given in Table 3-2 with S-N curve E should be used in the initial design phase for screening of main scantlings with typical attachments.

Table 3-1 Proposed economical value for $K_g$ used in initial design

<table>
<thead>
<tr>
<th>Main deck details</th>
<th>$K_g$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Penetrations with distance more than one diameter apart from each other. Doubling plates and soft toe support brackets with distance more than 500 mm apart from each other or to penetrations.</td>
<td>1.5</td>
</tr>
</tbody>
</table>

The fatigue life for offshore units shall be based on site specific wave data unless dictated by project-specific
requirements. Simplified fatigue analyses based on site specific wave data may be performed according to the following procedure:

1) A wave-load analysis shall be performed. The global wave bending moments shall be calculated for a return period of 20 years including corresponding Weibull parameters.

2) The global stress in the deck, including the stress concentration factor, can then be calculated and checked against the allowable stress, $\Delta \sigma_{\text{allowable}}$, for the desired fatigue life and relevant S-N curve, as given in [2.8] (Note that the allowable stress is for the $10^{-8}$ probability of exceedance level that corresponds to a return period of 20 years.)

3) Weibull shape parameters shall be based on long term statistics at the site for the FPSO.

The longitudinal nominal stress range in the deck structure is obtained as:

$$\Delta \sigma_V = 2 M_{V20} / W_y$$

$$\Delta \sigma_H = 2 M_{H20} / W_z$$

(4)

where:

$M_{V20}$ = Vertical wave bending moment amplitude

$M_{H20}$ = Horizontal wave bending moment amplitude

$W_y$ = Sectional modulus for hull about a horizontal axis

$W_z$ = Sectional modulus for hull about the vertical axis

$$\Delta \sigma_{20} = \sqrt{\Delta \sigma_V^2 + \Delta \sigma_H^2 + 0.2 \Delta \sigma_V \Delta \sigma_H}$$

(5)

The design requirement is:

$$\Delta \sigma_{20} K_g \leq \Delta \sigma_{\text{allowable}}$$

(6)

The nominal geometry shall be adjusted until equation (6) is satisfied. Bottom and side shell plating should be checked with consideration of relevant loads, see App.A.

Fatigue may be the governing factor for the thickness of side shell plating at the waterline area, and the critical detail is often the fillet weld between the longitudinal and the shell plate. This weld (i.e. the toe) can be checked using $K_g = 1.13$ for plates with thickness < 25 mm and lateral dynamic pressure according to principles described in Appendix A. Both the plate bending from lateral pressure and the global stress from wave bending moment and wave shear forces shall be taken into consideration. It has been found from experience that if the plate thickness is greater than $1/46$ of the stiffener span, the fatigue contribution due to lateral pressure is considered to be minor.

4) The size of the longitudinal stiffeners shall also be checked. The analysis requires an assumption with respect to bracket design versus bracket less design at transverse frames (see Figure 3-1, Figure 3-2).

The required fatigue life for such details can normally be obtained with a bracket design provided that the longitudinal stiffener has the required strength in the ULS condition. If a bracket less design is used, then the size of the longitudinal can be governed by the Fatigue Limit State. This difference arises because a design without brackets may be more susceptible to fatigue cracking in the connection between the longitudinal and the web frame than a design with brackets. However, this depends also on the local geometric design of the connection.

Where brackets are used below the water line, a double bracket design is preferred. A double bracket gives less eccentric load transfer of the shear force into the web frame than a single sided bracket design, and with lower stress concentrations.

5) The interface between the topside and the hull shall be carefully considered in the design and fabrication phase. Actions caused by the topside and interface structures shall be adequately addressed in the ship hull design. In many cases, fulfilment of both the Ultimate Limit State criteria and Fatigue Limit State criteria will require that the hull structure be strengthened beyond that of normal ship designs. It should
be noted that a good interface design is an important aspect for establishing a realistic analysis model to reliably calculate the fatigue lives of these regions.

Figure 3-1  Stiffener and brackets welded to the flange of the longitudinal (bracket design)

Figure 3-2  Longitudinal welded to transverse frame at web only (bracket less design)

3.2.2 Screening of transverse structural members

In order to ensure that all the transverse webs along the unit meet the fatigue requirements, the designer shall perform a screening analysis of the hull. The screening should be based on a coarse FE model and a representative pressure load based on a directly calculated pressure using a 3D diffraction theory.

The transverse web(s) subject to the maximum nominal stresses and/or deflections shall be identified from the FE model. Once the critical frame is identified, potential critical details, shall be analysed using the stochastic fatigue analysis procedure with refined local models.

Longitudinal stiffeners to webs/bulkheads in the side shell, longitudinal bulkheads and bottom plating shall also be subject to fatigue screening Fatigue lives of potential hot spots in each of the critical connections, such as lug/web and cut-outs, shall be assessed. Internal/external pressure parameters can normally be used to screen the exact locations for refined fatigue analysis.

For example, a map of long-term pressure loads at all side shell locations can be developed for the fatigue screening as given in example, ref. Figure 3-3. If the web frame spacing, plate thickness of the lugs, webs and bulkheads or size/shape of the cut-outs vary from location to location, then a geometry factor can be introduced to scale the pressure loads accordingly. Typically, a minimum of 6 or 8 connections (such as those in the bulkheads and the adjacent web located in the waterline area, the turn of the bilge and the bottom) are chosen for refined fatigue analysis. Note that refined modelling of the bracket toes is usually included in the refined fatigue analysis of longitudinal to web / bulkhead connections.
3.2.3 Example screening of offshore specific details

The offshore specific details given in DNVGL-OS-C102 shall be screened to identify those details for refined stress and fatigue analysis. As each of the specific details experiences different load sources, with respect to fatigue design, there is no single screening procedure suitable for all cases. Appropriate $K_g$ values have not been established for most of these details, and hence the following examples are based on first establishing $K_g$ factors for all relevant loads, and then applying the $K_g$ factors in the screening process. It may also be developed comparable approaches to achieve the fatigue screening objective.

**Screening of topside supports**

It is envisaged that the fatigue screening method for longitudinal hull structural members can be used for this application, except that $K_g$ factors for the topside support and deck interface structural design are not provided.

Ideally the topside support should be designed in a way that the longitudinal stresses from wave bending moment etc. will not interact with the stress from the inertia forces from the topside module. This is usually achieved by using large soft brackets at the topside support to main deck intersection.

The locations of the topside supports are generally spread all over the main deck at a large number, often near other sources of SCFs such as doubling plates and penetrations, see Sec.10. In order to identify the worst location it is recommended to make an allowable SCF map of the main deck based on the nominal stress and the appropriate S-N curve (E curve).

When the vertical and horizontal section modulus and the cross sectional area of the hull is known, the stress amplitude per load component can be found as shown in Equation (7). Alternatively the stress
components can be found from the global finite element model after application of a unit load to the model.

\[
\begin{align*}
\sigma_y &= 1/W_y \\
\sigma_H &= 1/W_z \\
\sigma_d &= 1/A
\end{align*}
\] 

(7)

Based on the unit stress per load component a component stochastic analysis can be performed in the statistical post-processor, see [6.8]. Theoretically the whole deck can be analysed but it is usually sufficient to define points for each 15-20 m in the longitudinal direction with 3-4 point across the beam. It is important to have calculated the sectional loads in the hydrodynamic analysis at cross sections close to the deck points considered.

From the component stochastic analysis the stress amplitude at 10^{-4} probability level shall be extracted together with corresponding Weibull parameter and zero-crossing period. Once the three parameters are known some iteration using the fatigue life formulas need to be performed in order to arrive at the correct allowable SCF. The allowable SCFs should be presented as map using the main deck drawing as background. This will enable the designers to easily identify the areas where topside stool design will be critical with respect to fatigue.

The above procedure can be adopted for all fatigue critical areas where local effects are of less importance.

Where local effects could have an effect on the calculated fatigue lives load components such as inertia loads due to unit motion and forced deflections (for hull girder) need to be included.

In order to get a realistic picture of the fatigue sensitivity of each topside stool location, it is necessary to include the stiffness of the topside modules in the cargo hold (see Figure 3-4) or full ship finite element model of the FPSO. Mass balance is not necessary, especially if the model is purely used for screening purposes, as long as the stiffness is correct.

Figure 3-4  Typical cargo hold model with topside modules included
On the FE model the following loads shall be applied:

- unit bending moment, horizontal and vertical as separate load conditions
- unit load at each topside centre of gravity considering vertical, horizontal and longitudinal acceleration (topside mass \( \times \) unit acceleration) as separate load conditions. For three topside modules this will equate to 9 load cases
- deck deformation loads will be accounted for if the correct stiffness has been used for the topside modules
- external sea pressure (unitized) to account for bending of the transverse girder system. This load case can be ignored if the topside stools are placed on top of transverse bulkheads.

Once all load cases are established, a component based fatigue calculation can be performed in accordance with [2.4.3]. The SCFs can be taken from tabulated values or from a detailed SCF model.

It is important to include a coarse representation of the topside stool using shell elements in order to make the extraction of the stress components easy. The connection between the topside modules, usually a beam model, needs to be sufficiently rigid to avoid large unrealistic deflections, which will lead to unreliable results.

After the screening exercise on the cargo hold model is complete, a more refined model shall be made of the topside stool to verify the fatigue lives. Boundary displacement from the cargo hold model can be exported to the refined model and the procedure can be repeated to get the final fatigue lives.

**Screening of riser porches (spread-moored FPSOs)**

The process for fatigue screening of topside supports can also be applied to riser porches. Calculation of \( K_g \) factors for fatigue analysis should include the following two fatigue load sources:

- hull girder bending loads
- first and second order riser loads.

For the details governed by hull girder loads the same procedure as described for topside stool can be adopted.

For certain riser porches and hotspots, the designer may be able to calculate the fatigue damage separately for the hull girder and riser fatigue loads. For designs where both hull girder and riser loads contribute significantly to the fatigue damage at the same location, a combination of the fatigue damages shall be performed. The combination may be performed according to Equation (23). Alternatively the combined spectrum approach as described in [6.9.4] can be used.

**Screening of mooring foundation (spread-mooring system)**

Spread-mooring foundations are typically located in the FPSO's bow and stern areas above the main deck, where the influence of longitudinal hull girder loads is small. The key parameter to be considered with respect to fatigue screening are the first and second order mooring line tensions, acting on the mooring foundations. Calculation of mooring line tension should include particular attention to the estimate of line stiffness, which is a function of the FPSO's mean offset position(s) due to current, wind, wave loads and unit motions.

**Screening of deck penetrations**

Refer to [3.2.1].

**Screening of crane pedestals**

There are two primary fatigue load sources that should be considered in the screening of crane pedestals for refined fatigue analysis. These are:

- longitudinal hull girder bending
- inertia load due to unit's motions.

Maximum hull girder bending usually occurs near midship area, and maximum unit's accelerations can occur near the unit's bow and/or stern area(s), such that it may be necessary to assess multiple pedestal details using the refined fatigue analysis method.
In addition to this it is important to consider the dynamic loading due to crane operations for the crane itself.

**Screening of other specific details**

Refined fatigue analysis for other items should be performed in accordance with this RP.

**Screening of hull / turret interface structure (FPSOs)**

Extreme turret loads (moments and forces from the mooring analysis) are generally available early in the project. These may either be provided as a moment and force at a specified location on the turret or as a load distribution at the bearing level.

Since the FPSO will weathervane, the fatigue locations will vary around the circumference of the moonpool. As a consequence the turret loads need to be investigated for several headings. The hot spots are established in the same manner as for the unit load approach described in DNVGL-RP-C203.

This method requires that the FE model for the hull and interface structure is available, which is typically the case as the same model is used for the extreme load analysis. Requirements for the hull / turret interface modelling is provided in [4.8]. The loads may be transferred to the interface structure using a course representation of the turret, as described in Figure 8-12.

It is intended that this approach is used for the fatigue design phase although the screening may commence upon completion of the extreme analysis of the interface structure.

The fatigue damage may be estimated using the extreme stresses with a long term distribution, as described in DNVGL-RP-C203.

If extreme loads are not available then unit force and moments may be used to determine the relative performance of critical connections.

### 3.2.4 Example screening of brackets

Introduction of soft toe and heels suggest that cracking may occur in the bracket itself or in the fillet weld root connecting the bracket to the flange.

As shown in sketch (see Figure 3-5) the principal stress runs right at the 45º angle and so the details should be checked against the E curve.

Crack initiation from the weld root should also be investigated especially if the throat thickness of the fillet welds is marginal. See [7.3.4] for derivation of stress in fillet welds.

![Figure 3-5 Stress direction in bracket toe](image-url)
SECTION 4  OVERVIEW OF MODELLING TECHNIQUES FOR FATIGUE ANALYSIS

4.1  Structural modelling using finite element methods

4.1.1  Scantlings
It is required to install and maintain adequate corrosion protection (CP) systems for new-build hull structures. Therefore, the fatigue analysis shall be based on gross, or as-built, scantlings.

4.1.2  Overview of structural finite element models
Finite element analysis is required to obtain accurate stress distribution in the hull structure. There are several levels of finite element models used in the various phases of design. The following three different levels of finite element models are referred to in this RP:

*Global structural model:*
A relatively coarse mesh model used to represent the overall stiffness and obtain the global stress distribution of the primary members of the hull. A typical model is shown in Figure 4-1.

*Cargo hold model (3-tank model):*
A model used to analyse the deformation response and nominal stresses for the primary members in the midship area. The model is normally established for the frame and girder strength analysis and may be used for fatigue evaluation purposes. It will normally cover ½ + 1 + ½ cargo hold length in the midship region. Typical models are shown in Figure 4-2.

It may be appropriate to extend the cargo hold model to 3 complete holds (i.e. 1 + 1 + 1) to allow the part-ship model stochastic procedure to be followed.

*Part-ship model:*
Similar to the cargo hold model except it is used for portions of the unit outside of the cargo hold region. In addition the model extent is typically three full cargo hold lengths, i.e. 1 + 1 + 1.

*Stress concentration model (or local model):*
A model used for fully stochastic fatigue analyses and for component based fatigue analyses of details where the geometrical stress concentration is unknown. Typical details to be considered are:
- hopper knuckles (Figure 4-3)
- bracket and flange terminations for main girder systems
- topside stools
- riser supports
- stiffener connections.

The local models are usually referred to as sub-models. Stresses in these models may be derived by transfer of boundary deformations/ boundary forces from the coarser model. Such transfer of data between models requires that the various mesh models are “compatible,” i.e. meshes in the coarse model produce deformations and/or forces applicable as boundary conditions for the finer mesh models, see [4.5.2].

A description of the structural model hierarchy, in terms of analysis, is depicted in Figure 7-1.
Figure 4-1 Global hull finite element model

Figure 4-2 Cargo hold / part-ship finite element models (midship area)
4.2 Structural modelling principles for a global model

4.2.1 Model idealization

The global hull analysis is intended to provide a reliable description of the overall stiffness and global stress distribution in the primary members in the hull. The following effects shall be taken into account:

— vertical hull girder bending including shear lag effects
— vertical shear distribution between ship side and bulkheads
— horizontal hull girder bending including shear lag effects
— transverse bending and shear.

A complete finite element model may also be necessary for the evaluation of the vertical hull girder bending of ships that have a complex arrangement of continuous structures above the main deck, such as long topside modules.

The mesh density of the model shall be sufficient to describe deformations and nominal stresses due to the effects listed above.

The global analysis may be carried out with a relatively coarse mesh. Stiffened panels may be modelled by means of layered (sandwich) elements or anisotropic elements. Alternatively, a combination of plate and beam elements may be used. Modelling shall provide a good representation of the overall membrane panel stiffness in the longitudinal/transverse and shear directions.

An example global finite element model for an oil tanker is shown in Figure 4-1. The model may also be used to calculate nominal global (longitudinal) stresses away from areas with significant stress concentrations. The following features will induce significant stress concentrations:

— termination of girder/bulkheads
— moonpool or other large penetrations.

Small penetrations are normally disregarded in the global model. For consideration of local stresses in web frames, girders or other areas (see [4.5]), fine mesh areas may be modelled directly into the coarse mesh.
model by means of suitable element transitions. However, an integrated fine and coarse mesh approach implies that a large set of simultaneous equations needs to be solved.

The advantage of a sub-model (or an independent local model) is that the analysis is carried out separately on the local model, requiring less computer resources and enabling a controlled step by step analysis procedure to be carried out.

The various mesh models shall be “compatible”, i.e. the coarse mesh models shall produce deformations and/or forces applicable as boundary conditions for the finer mesh models (referred to as sub-models). If super-element techniques are available, the model for local stress analysis may be applied at lower level super-elements in the global model.

Sub-models (e.g. fine mesh models) may be solved separately by use of the boundary deformations/ boundary forces and local internal loads from the coarse model. Load data can be transferred from the coarse model to the local model either manually or, if sub-modelling facilities are available, automatically by the computer program.

The sub-models shall be checked to ensure that the deformations and/or boundary forces are similar to those obtained from the coarse mesh model. Furthermore, the sub-model shall be sufficiently large that its boundaries are positioned at areas where the deformation/ stresses in the coarse mesh model are regarded as accurate. Within the coarse model, deformations at web frames and bulkheads are usually accurate, whereas deformations in the middle of a stiffener span (with fewer elements) are not sufficiently accurate.

The sub-model mesh shall be finer than that of the coarse model, e.g. a small bracket is normally included in a local model, but not in global model.

4.2.2 Extent of model

The full unit extent shall be included in the model.

All main longitudinal and transverse geometry of the hull shall be modelled. Structure not contributing to the global strength of the unit may be disregarded. The mass of disregarded elements shall be included in the model.

Structural components not contributing to the global stiffness, such as superstructure, topsides (small modules), topsides support, etc., are not normally included in the global analysis. However, the mass of these elements should be correctly included in the model.

It should be emphasized that these structures can lead to local/global stress concentrations and it should be checked that omission of these parts does not lead to non-conservative results. Similarly, the omission of minor structures may be acceptable provided that such omission does not either significantly change the deformation of the structure or give non-conservative results, i.e. too low stress, to the structural analysis.

4.2.3 Modelling of girders

Girder webs shall be modelled by means of shell elements in areas where stresses are to be derived. However, flanges may be modelled using beam and truss elements. Web and flange properties shall be according to the actual geometry. The axial stiffness of the girder is important for the global model and hence reduced efficiency of girder flanges should not be taken into account. Web stiffeners in direction of the girder should be included such that axial, shear and bending stiffness of the girder are according to the girder dimensions, see [4.2.4].

The mean girder web thickness at cut-outs may generally be taken as follows:
where

\[ t_{\text{mean}} = \frac{h - h_{\text{co}}}{h r_{\text{co}}} t_{\text{w}} \]

\[ r_{\text{co}} = 1 + \frac{l_{\text{co}}^2}{2,6(h - h_{\text{co}})^2} \]

\( l_{\text{co}} \) = length of cut-out

\( h_{\text{co}} \) = height of cut-out

\( h \) = girder web height

For large values of \( r_{\text{co}} \ (> 2.0) \), geometric modelling of the cut-out is advisable.

### 4.2.4 Modelling of stiffeners

Continuous stiffeners should be included using any of the following options:

- lumping of stiffeners to the nearest mesh line
- inclusion of stiffeners in layered elements (sandwich elements), using 6 and 8 node shell elements for triangular and quadrilateral elements respectively
- inclusion of stiffeners as material properties (anisotropic material properties).

### 4.2.5 Elements and mesh size

The performance of the model is closely linked to the type of elements and the mesh topology that is used. The following guidance on mesh size etc. assumes the use of 4-node shell or membrane elements in combination with 2-node beam or truss elements. The stiffness representation of 3-node membrane or shell elements is relatively poor and their use should be limited as far as practical.

The shape of 4-node elements should be as rectangular as possible, particularly where in-plane shear deformation is important. Skew elements will lead to inaccurate element stiffness properties.

Element formulation of the 4-node elements can require all four nodes to be in the same plane. Unintended fixation of a node can occur if it is “out of plane” compared to the other three nodes. The fixation will be seen as locally high stresses in the actual elements. Double curved surfaces should therefore be modelled with 3-node elements instead of 4-node elements. However, some structural analysis programs adjust the element formulation such that “out of plane” elements does not necessary create significant errors in the structural analysis.
Provided that 4-node element formulations include linear in-plane shear and bending stress functions, the same element size may be used for both 4-node shell elements and 8-node shell elements.

The use of higher level elements such as 8-node or 6-node shell or membrane elements will not normally lead to reduced mesh fineness. 8-node elements are, however, less sensitive to element skewness than 4-node elements, and have no "out of plane" restrictions. In addition, 6-node elements provide significantly better stiffness representation than that of 3-node elements.

Based on the above discussion, use of 6-node and 8-node elements is preferred but can be restricted by computer capacity.

The mesh size should be decided considering proper stiffness representation and load distribution of tank, and sea pressure on shell elements or membrane elements.

The following rules can be used as a normal guideline for the minimum element sizes to be used in a global/stiffness structural model using 4-node and/or 8-node shell elements (finer mesh divisions may be used):

- General: One element between transverse frames/girders
  Quadratic elements are generally preferable
- Girders: One element over the height in areas where stresses are to be obtained
  Beam elements in other areas
- Girder brackets: One element
- Stringers: One element over the width
- Stringer brackets: One element
- Hopper plate: One to two elements over the height depending on plate size
- Bilge: Two elements over curved area
- Stiffener brackets: May be disregarded

All areas not mentioned above should have equal element sizes. One example of suitable element mesh with suitable element sizes is shown in Figure 4-1.

The eccentricity of beam elements should be included. If the program does not support eccentricity of profiles, the modelled bending properties of the beams should include the attached total plate flange.

### 4.2.6 Mass modelling

The mass modelling shall be according to the loading manual, i.e. have the same longitudinal, vertical and transverse mass distribution. The correct mass description is important in order to produce correct motions and sectional forces in the hydrodynamic analysis, and to generate correct global/local stress patterns in the structural analysis.

Identical mass models should be used in hydrodynamic analysis and structural analysis. The structural model should consequently be used as a mass model in the final hydrodynamic analysis to establish pressure loads for the actual load transfer. This ensures that gravity/inertia loads are correctly transferred from the hydrostatic/dynamic analysis to the structural model.

It is generally recommended that:

- mass density is used for structural elements
- pressure is used for external and internal hydrostatic and hydrodynamic loads and
- point masses are used for non-structural members.

The point mass representation shall be sufficiently distributed to provide a correct representation of rotational mass and to avoid unintended results. Point masses should be located at structural intersections to minimize the local response.

The mass from topside structures should be included in the model. See also [4.6].

If supported by the program system, use of non-structural members may be a suitable modelling method for cargo which can be otherwise difficult to model correctly. The mass may then be placed in the centre of gravity of the hold and connected to the hold walls/bottom.

A relatively coarse mass description may be adequate for the global model, whereas a more precise mass
description may be necessary for models where local deflections are of interest. The selected accuracy of the mass description depends on model size, mesh size, local loading and the results to be produced. For some local models, the inertia load from the local model itself will be insignificant, and stresses from more global actions will dominate the response.

Balancing the mass model to give correct mass description is not always a straightforward task. The global structural model usually consists of one or, if the super element technique is available, several super elements. The size of each super element may be relatively large. Correct centres of gravity do not necessarily result in correct mass distribution within each super element. Even small inaccuracies in the mass description can lead to relatively large errors in global forces/moments.

Correct mass balancing may be achieved by dividing the hull into several regions and adjusting masses according to correct mass description in each region.

4.2.7 Modelling of cargo and ballast water

If the global analysis shall be used for detailed calculations of side shell connections or details affected by liquid pressures the cargo/ballast water should be modelled in the global model. The phase difference between internal and external pressure will automatically be taken into account.

When swash bulkheads are present with minimal openings, then the bulkhead may be considered as dividing the hold into two separate compartments for hydrodynamic analysis purposes.

4.2.8 Boundary conditions

The boundary conditions for the global structural model should reflect simple supports that will avoid built-in stresses. A three-two-one fixation, as shown in Figure 4-4, can be applied. Other boundary conditions may be used if desirable. The fixation points should be located away from areas of interest, as the loads transferred from the hydrodynamic load analysis may lead to imbalance in the model. Fixation points are often applied at the centreline close to the aft and the forward ends of the unit.

![Figure 4-4 Example of boundary conditions](image)

4.2.9 Transfer of hydrodynamic loads

The hydrodynamic loads are to be taken from the hydrodynamic load analysis. To ensure that phasing of all loads is included in a proper way for further post processing, direct load transfer from the hydrodynamic load analysis to the structural analysis is the only practical option. The following loads should be transferred to the structural model:

- inertia loads for both structural and non-structural members including topsides
- external hydro pressure loads
- internal pressure loads from liquid cargo, ballast *
— inertia loads from equipment (cranes, topsides, helicopter decks, etc.)
— riser and mooring loads
— viscous damping forces (see below).

* The internal pressure loads may be exchanged with mass of the liquid (with correct centre of gravity) provided that this exchange does not significantly change stresses in areas of interest (the mass shall be connected to the structural model).

Inertia loads will normally be applied as acceleration or gravity components. The roll and pitch induced fluctuating gravity component \((g \sin(\theta) \approx g \theta)\) in sway and surge shall be included.

Pressure loads are normally applied as normal pressure loads to the structural model. If stresses influenced by the pressure in the waterline region are calculated, pressure correction according to the procedure described in 4.4.3 need to be performed for each wave period and heading.

The riser loads may be applied as concentrated loads to one or more locations depending on the mooring/riser configuration. These loads may be omitted for general details (see [2.3.2]).

Viscous damping forces can be important for some units, particularly those units where roll resonance is in an area with substantial wave energy, i.e. roll resonance periods of 6 to 15 seconds. The roll damping may, depending on meteological ocean criteria, be neglected when the roll resonance period is above 20 to 25 seconds.

Viscous damping forces can be transferred either as line loads along the bilge or as distributed surface loads parallel to the hull (roll motion). Omission of the roll viscous damping force leads to unbalanced torsion forces along the hull.

4.3 Structural modelling principles for a part-ship model

4.3.1 Model idealization

The purpose for the part-ship model is to perform a full stochastic fatigue analysis based on direct load transfer as an alternative to the global model approach described in [4.2]. The intention is to reduce modelling size and time by modelling a selected part of the unit rather than the entire unit. Apart from the extent and the load application at the model boundaries of the finite element model, the part-ship approach is similar to the global approach. However, part-ship models may not capture the shear lag effects as well as a global model.

External items such as cranes, derrick, topside support etc. are to be included in the model at their respective positions to ensure the correct transfer of inertia loads.

As for the global model the units used in the finite element model should be consistent with those used in the hydrodynamic analysis.

4.3.2 Extent of model

The procedure described assumes that the part-ship model has a complete transverse bulkhead at the aft and forward end. Consequently the model will normally consist of three complete compartments, i.e. 1+1+1, where the primary fatigue calculations will be completed for the middle compartment as shown in Figure 4-5. The other compartments are to be included to limit the influence of the boundary conditions on the results.
Variations of the 3-hold requirements may occur at certain locations, such as towards the ends of the unit where an additional compartment is not available, e.g. external bow mounted turret. In this instance the model shall extend completely to the end of the unit and include one compartment on the opposite side of the location being considered, as shown in Figure 4-6.

4.3.3 Modelling of girders and stiffeners
Refer to [4.2.3] and [4.2.4].

4.3.4 Elements and mesh size
Refer to [4.2.5].

4.3.5 Mass modelling
Generally the mass modelling requirements are similar to those described in [4.2.6].
Identical mass models should be used in hydrodynamic analysis and structural analysis for the extent of the part-ship model. This ensures that gravity/inertia loads are correctly transferred from the hydrostatic/dynamic analysis to the structural model.
The section forces in several cross-sections should be checked and verified with the results from the hydrodynamic analysis. If the mass balancing is performed correctly, the deviation between the structural analysis and hydrodynamic analysis should be negligible. See [9.4] for more details of the verification procedure.

4.3.6 Modelling of cargo and ballast water
Refer to 4.2.7.

4.3.7 Boundary conditions
To avoid singularities, from imbalance in the applied loads, boundary conditions are required at both ends of the part-ship FE model. To avoid large reaction forces the spring stiffness should be small, i.e. 1/1 000 of actual spring stiffness as described in [4.4.5]. Springs, see Figure 4-7, should be applied in all three translation degrees of freedom and the ratio between them should be such that no unwanted moments are introduced.

![Spring locations for a part-ship model](Image)

**Figure 4-7  Spring locations for a part-ship model**

4.3.8 Application of hydrodynamic loads
Refer to [4.2.9].

Since a substantial part of the unit is not included in the part-ship model concept, sectional loads shall be applied at the ends of the model to represent the effect of the missing hull girder parts. It is therefore important that the hydrodynamic analysis includes one section cut at the aft end of the part-ship model and one at the forward end.

The section forces and moments at these sections shall be transferred to the part-ship model in order to maintain load balance. As a part of the quality control to ensure load balance, the stress in the structural model should be integrated and the moments and shear forces should be compared with the results from the hydrodynamic analysis, as described in [9.4].

Due to the influence of intermittent wet and dry surfaces at the waterline (Figure 4-8) the external pressure loads need to be corrected, as described in [4.4.3].
4.4 Structural modelling principles for the load component fatigue analysis

4.4.1 Model idealization
The midship analysis is used to analyse deformation response and nominal stresses of the primary hull structural members in the midship area. The effect of shear lag is not captured.

The midship analysis is a requirement for 1A class approval.

4.4.2 Extent of model
The finite element model shall normally include the tank/hold under consideration, plus one half of the adjoining tank/hold at each end of the considered tank/hold, i.e. the model extent comprises ½ + 1 + ½ holds or tanks. A model covering the half breadth of the ship may be used provided there is symmetry with both the structure and loading. If there is a symmetry plane at the half-length of the considered tank/hold, then the extent of the model may be taken as one half tank/hold on each side of the transverse bulkhead. This model corresponds to the FE model required for the cargo hold region transverse strength analysis necessary for certain unit types.

Figure 4-2 shows typical models of cargo hold midships (3 tank models) for an FPSO.

For regions outside of the cargo hold region the model should normally include three complete compartments, i.e. 1 + 1 + 1, as described for the part-ship model (see [4.3]).

4.4.3 Load components
Lateral loads from sea pressure, cargo, etc. shall be applied to the model. The applied loads shall be normalized loads representing local loads at different areas in the model. Loads, such as sea loading, shall be separated into several load cases such that effects of local pressure at the different areas of the unit can be combined with the correct phase information.

Hull girder forces and moments shall be applied to the ends of the model and shall be analysed as separate load conditions. Complex summation of transfer functions shall be used to combine the hull girder response with the response from the lateral load distribution.

An example of load cases to be used for 3-tank analysis is shown in Table 4-1.

Effect of intermittent wet surfaces
Due to intermittent wet and dry surfaces, the range of the pressure may be reduced above $T_{act} - z_{wl}$, see Figure 4-8. The dynamic external pressure amplitude (half pressure range), $p_{e}$, related to the draft of the load condition considered may be taken as:

$$P_e = r_p p_d (kN/m^2)$$

where:

- $p_d = \text{dynamic pressure amplitude below the waterline}$
- $r_p = \text{reduction of pressure amplitude in the surface zone}$
- $= 1.0$ for $z < T_{act} - z_{wl}$
- $= \frac{T_{act} + z_{wl} - z}{2z_{wl}}$ for $T_{act} - z_{wl} < z < T_{act} + z_{wl}$
- $= 0.0$ for $T_{act} + z_{wl} < z$

$z_{wl} = \text{relative wave motion (calculated at probability level 10^{-4}) in m, measured from actual water line.}$

The external sea pressure is assumed not to contribute to fatigue damage in the area of side shell above $z = T_{act} + z_{wl}$
4.4.4 Finite element mesh

The fineness of the mesh used for the cargo hold/tank analysis shall be decided based on the method of load application and type of elements used.

The element mesh for the cargo hold/tank model shall represent the deformation response and shall be fine enough to enable analysis of nominal stress variations in the main framing/girder system. The following points may be used as guidance:

— A minimum of 3 elements (4-node shell/membrane elements) over the web height are necessary in areas where stresses are to be derived. With 8-node elements, 2 elements over the web/girder height are normally sufficient. Figure 4-2 illustrates these two alternatives for possible mesh subdivisions in a double skin FPSO.
— For the tanker model shown in Figure 4-2 a), the general element length is equal to half the web frame spacing. This implies that the effective flange/shear lag effect of the plate flanges (transverse web frames) is not properly represented in this model, and that the mesh is not suitable for representation of stress concentrations at knuckles and bracket terminations. If a better representation of flexibility of the frames is desired, the number of elements may be increased to 4, either side of the frame. The modelling of the frames should also be seen in connection with extent of the local model, see [4.2.2].
— The mean girder web thickness in way of cut-outs may generally be taken as in [4.2.3].

4.4.5 Boundary conditions

Each component load case requires that different boundary conditions are applied to the model. For example, to compute the stress response to lateral pressures the model shall be vertically and horizontally supported by distributed springs located at the intersections of the transverse bulkheads with:

— ship sides and the longitudinal bulkheads,
— longitudinal girders, and
— deck, inner bottom and outer bottom.

The spring constants shall be calculated for the longitudinal bulkheads and the ship sides. Calculations shall be based on actual bending and shear stiffness for a model length of three cargo holds. Symmetry conditions shall be applied at the model ends. Note that for a model length of 1+1+1 tank lengths, only half of the spring stiffness should be applied to the end sections.

Transverse stresses can be affected by the spring boundary conditions described above. An example of suitable boundary conditions for analysis of the hopper knuckle is given in App.A, Table A-1.

The boundary conditions to be used for the hull girder loads should be according to the definitions in Appendix A.

4.4.6 Balance of loads

Vertical load balance for the lateral load case can be achieved by introduction of “fictitious” balancing loads. These balancing loads should be introduced into the model in such a way that they do not effect the stress flow at hot spots under consideration. Reference App.A.

4.5 Structural modelling principles for stress concentration factor models

4.5.1 Introduction

Local finite element analyses may be used for calculation of local geometric stresses at hot spots or for determination of associated K-factors. These analyses involve the development of fine element mesh models for details, such as bracket connections, stiffener to web frame connections or local design of frames/girders. It is important to note the definition of the K-factors and their relation to the S-N curve (see also [2.6]).

The finite element analysis is not normally used for direct calculation of the notch stress at a detail. The analysis is usually applied to calculate the geometric stress distribution in the region of the hot spot, such that these stresses can be used either directly in the fatigue assessment of given details or as a basis for derivation of stress concentration factors. The aim of the finite element analysis is to calculate the stress at the weld toe (hot spot) due to the presence of the attachment, denoted hot spot stress, $\sigma_g$. The stress concentration factor due to this geometry effect is defined as:

$$K_g = \frac{\sigma_g}{\sigma_{nominal}}$$

Thus the main objective of the finite element analysis is to provide a reasonably accurate model of the stresses at a region outside the weld affected zone. The model should have a fine mesh for sufficiently accurate calculation of $K_g$. 

Reference is also made to DNVGL-RP-C203. In addition to guidance on FE modelling derivation of hot spot stress DNVGL-RP-C203 presents hot spot target values for a few geometries that may be used for assessment of FE-program and FE-elements.

4.5.2 Sub-modelling technique

Sub-models using boundary deformations/forces from a coarse model that may be used subject to the following rules. The rules aim to ensure that the sub-model provides correct results. These rules can, however, vary for different program systems.

The following items shall be considered in a sub-model:

1) The sub-model shall be compatible with the global (parent) model.
   This means that the boundaries of the sub-model should coincide with those elements in the parent model from which the sub-model boundary conditions are extracted. The boundaries should preferably coincide with mesh lines as this ensures the best transfer of displacements / forces to the sub-model.

2) Curved areas shall be given special attention.
   Identical geometry definitions do not necessarily lead to matching meshes. Displacements to be used at the boundaries of the sub-model will have to be extrapolated from the parent model. However, only radial displacements can be correctly extrapolated in this case, and hence the displacements on sub-model can consequently be wrong.

3) The boundaries of the sub-model shall coincide with areas of the parent model where the displacements/forces are correct.
   For example, the boundaries of the sub-model should not be midway between two frames if the mesh size of the parent model is such that the displacements in this area cannot be accurately determined.

4) Linear or quadratic interpolation (depending on the deformation shape) between the nodes in the global model should be considered.
   Linear interpolation is usually suitable if coinciding meshes (see above) are used.

5) The sub-model shall be sufficiently large that boundary effects, due to inaccurately specified boundary deformations, do not influence the stress response in areas of interest. A relatively large mesh in the “parent” model is normally not capable of describing the deformations correctly.

6) If a large part of the model is substituted by a sub model (e.g. cargo hold model), then mass properties shall be consistent between this sub-model and the “parent” model. Inconsistent mass properties will influence the inertia forces leading to imbalance and erroneous stresses in the model.

7) Transfer of beam element displacements and rotations from the parent model to the sub-model should be especially considered.

8) Transitions between shell elements and solid elements should be carefully considered. Mid-thickness nodes do not exist in the shell element and hence special “transition elements” may be required.

4.5.3 Extent of model

The model shall be sufficiently large to ensure that the calculated results are not significantly affected by assumptions made for boundary conditions and application of loads. If the local stress model is to be subject to forced deformations from a coarse model, then both models shall be compatible as described in 4.5.1. Forced deformations may not be applied between incompatible models, in which case forces and simplified boundary conditions shall be modelled.

4.5.4 Example modelling of special details

**Hopper knuckle**

Figure 4-3 shows an example model of a hopper tank knuckle in a tanker. The stress concentration model uses shell elements in order to directly determine the geometric stress, \( \sigma_g \), at the knuckle line and hence the geometric stress concentration factor, \( K_g \).

**Modelling of lug plate to web**

Normally the mid-plane of a lug plate is eccentric to the mid-plane web frame, see Figure 4-9. This lug plate eccentricity should also be included in a finite element analysis model. Shell elements shall be used to model
the mid surfaces of the lug plates. The weld around the lug plate connection shall be simulated with shell elements that are placed normal to the first shell elements, as indicated in Figure 4-9. A thickness of 2.0 \( t_w \) is used for these elements, where \( t_w \) is the lug plate thickness.

Test specimens have indicated point B (see Figure 4-9) as a critical area subject to a significant bending stress in the lug plate.

![Figure 4-9 Modelling of lug plate to web frames by shell elements](image)

Modelling of flange terminations

The toe of brackets should be modelled by mesh size in accordance with FE modelling principles described in DNVGL-RP-C203. The element size should vary dependent on the plate thickness at the location of the hot spot. The element at bracket toe should be modelled with one element over the toe height to avoid unrealistic large stress gradients (see Figure 4-10). The flanges shall be modelled with shell elements.

![Figure 4-10 Element mesh at bracket toe](image)

Modelling of fillet welds for direct use of stress in fatigue calculation

More than one element is needed for modelling of a fillet weld if the analysis should provide stresses directly in the weld throat for root cracking fatigue assessment. Even with 20 node solid elements, at least 4 elements over the throat thickness are required for derivation of a nominal or engineering shear stress in the weld. However, one element is sufficient for proper modelling of stiffness such that stresses can be read
4.6 Modelling principles for mooring/riser foundations (spread-moored FPSO)

The general arrangement of the mooring system determines where the mooring line loads and riser loads are applied to the hull. Various structural elements may need to be considered, such as fairleads, chain stoppers, winches, riser porches, bend stiffeners, etc. Local structural models of these regions are required in order to determine the hot-spot stresses (see [4.5]).

For each riser or mooring line, a unit load shall be applied to the local structural model. The load shall be applied in a direction defined by the separate mooring system analysis. It is usually sufficient to consider the direction arising for the environmental state which is expected to contribute most to fatigue damage. The stress computed at the hot spot for the unit load defines an influence coefficient $u_i$ for the stress at the hot-spot due to mooring line or riser load $i$.

The same approach may be applied for both tensions and bending moments.

This influence coefficient is frequency independent and therefore differs from the transfer functions discussed elsewhere in the RP.

The standard deviation of the stress at the hot spot can be obtained by multiplying the influence coefficient with the standard deviation of the applied tension or moment. This can be done for each environmental sea state included in the fatigue analysis.

Mooring line tensions and/or riser loads can be strongly correlated, such that if several tensions or moments affect one hot spot, then the resulting stresses should be combined conservatively; e.g. the standard deviation of the combined stress is given by:

$$\sigma_c = \sqrt{\sigma_a^2 + 2\cdot \sigma_a \cdot \sigma_b + \sigma_b^2}$$  \hspace{1cm} (9)

The standard deviations of the two contributing stresses are $\sigma_a$ and $\sigma_b$, and full correlation is assumed.

Similarly, wave-frequency-stresses arising from the mooring lines or risers should normally be combined conservatively with wave-frequency stress arising otherwise in the ship hull.

Wave-frequency stresses and low-frequency stresses arise from different excitation mechanisms. The procedure for determining the combined fatigue damage effect of the two frequency ranges shall be applied as described in 6.9.

4.7 Modelling principles for topsides supports

Modelling of topsides and supporting structure for fatigue assessment shall, as a minimum, consider the following effects:

— influence of global bending moments and forces at the connection of the topsides to the deck
— impact of topsides inertia loads on stresses in longitudinal bulkheads, transverse bulkheads and web frames
— impact of deck deformation loads on deck/support interface structure, including:
  — relative horizontal displacement between the topside modules and the deck,
  — relative vertical curvature of the hull on supports
  — relative displacement due to torsional deformation
  — relative displacement due to the deformation of the cargo holds due to internal and external pressure.

In order to properly consider deck deformation loads on the topside support and deck interface structural design, the hull structural model should include a space frame representation of the topsides supporting module. The hull structural model can be a global or part-ship model developed for the full-stochastic
fatigue analysis, as discussed in [2.4.2]. Refined stress analysis for the most fatigue critical support(s) should be performed using the sub-modelling approach described in [4.5].

The topside supports and module supporting frame designs are sometimes unavailable at an early design stage. It is therefore suggested that a "point load" modelling approach be applied to size the module supports. Point loads acting in longitudinal (x), transverse (y) and vertical (z) directions at the top of the topside support can be considered to represent the topside weights such that the inertia loads due to unit’s motions can be defined. These point loads may also be derived based on the calculated reaction forces from the topside modules or based on an assumed upper bound of loads representing all different modules and weight distributions, as shown in Table 4-1.

4.8 Modelling principles for hull / turret interface structure

4.8.1 Introduction
The modelling principles for the hull / turret interface structure depend on turret type (see [8.2]), turret location (see [8.3]) and the load application. In general, fatigue calculations the FPSO hull may be separated into two categories:

— structure directly influenced by turret loads
— structure not directly influenced by the turret loads

as represented in Figure 8-1, in [8.1.3].

4.8.2 Uninfluenced structure
Where the hull structure is not influenced by turret loads it may be treated in the same manner as the remaining hull structure, using either the global or part-ship FE models (see [4.3] or [4.4]). In this situation it is only necessary to represent the stiffness of the turret supporting structure in order to maintain the correct stiffness.

4.8.3 Influenced structure
To perform fatigue calculations for the region influenced by the turret loads additional modelling requirements are necessary. These are addressed below.

Some or all of the following loads should normally be included in a fatigue analysis of the turret interface structure:

— hull girder loads
— riser/mooring loads
— turret inertia loads.

For turret sections the interaction forces between the turret module and the hull will be important for the local stress flow in the hull, around the turret opening. The interaction forces will depend on the design and relative stiffness of the turret module and the hull parts supporting the turret.

Depending on the turret size and location within the hull, the local stresses casd by the turret interaction loads may need to be superimposed with the global stresses due to the hull loads. This may achieved using the methods described in [6.11].

Stress from wave-frequency and low-frequency loads should be combined according to the methods as described in [6.9.4].

4.8.4 Extent of finite element models
The element model of the hull girder turret section shall normally extend a full compartment length at both ends of the primary hull part to be analysed, i.e. 1+1+1 compartment lengths, to reduce the effects of improper end constraints. This allows for a proper inclusion of longitudinal hull girder stresses and also for stresses from turret loads. The model size should be sufficient to avoid significant influence from the boundary conditions.
Mesh size and element types for the stiffness models should be in accordance with the modelling requirements specified for part-ship model, as given in [4.3]. The model requirements for stress concentration finite elements are provided in [4.5].

**External turret designs**

The external turret is normally located at the bow or stern and, in this case, the hull girder loads are of minor importance for the interface structure. Typically the stresses resulting from the turret dissipate rather rapidly from the connection point, as represented by the “stress influence line” in Figure 4-11. The main issue is, consequently, to transfer the riser/mooring loads to the hull. Separate stress concentration models will normally have to be used for the interface structure.

![Figure 4-11 Example of model extent for an external turret](image)

With the full stochastic fatigue analysis method, using either a global or partial model, the full bow or stern structure shall be modelled. The structural model should extend, as a minimum, to the first full transverse bulkhead aft of the turret arrangement where the boundary conditions have negligible influence on the region being considered, as shown in Figure 4-11.

Simplified calculations, with suitable stress concentration factors, may be satisfactory for certain details depending on geometry and stress level, such as longitudinal stiffener bracket connections in the upper deck of the rigid arm. In this circumstance the structural model need only extend to where the boundary conditions do not influence the detail to be investigated.

It is normally not required that a fine element model of the turret is included for assessing the hull fatigue design.

Examples of coarse and fine mesh (sub-model) external turret FE models are shown in Figure 4-12.
Figure 4-12 Example of a) external stiffness and b) refined turret models (port side shown)

Internal turret designs

To establish the interaction between the hull and the turret structure it is necessary to model both structures, as shown in Figure 4-13. The turret structure may be represented by a fairly coarse finite element model including all main stiffness elements, i.e. shell, decks, bulkheads, frames and girders. The hull finite element model should follow the principles described for the part-ship model (see [4.3]). An example of a hull structural model in the turret area is shown in Figure 4-14, where the model extends to the adjacent transverse bulkhead fore and aft of the turret region.

The interface structure between the part-ship and turret model will require a semi-fine mesh in order to capture all of the local load paths during the analysis. Generally the mesh size will approximately be in the order of 20 times the local plate thickness. All brackets, plating and stiffening in the interface region, as shown in Figure 4-12 and Figure 4-14, shall be included. The connection between the turret and the hull support structure should be made using a spring system representing the bearing or support of the turret. The spring stiffness of the bearing is normally provided by the bearing manufacturer. It will normally be
acceptable to use the same spring stiffness for the whole circumference. However, for some designs it may be necessary to include varying spring constants to account for the effect of the mean riser/mooring force. Similarly, if the turret is supported by elastomers, to isolate the turret from the hull deflection, springs should be used. The magnitude of the spring constants should be provided by the elastomer supplier.

Once the fatigue locations have been determined through the screening process (see [3.2]) separate stress concentration models should be used to estimate the fatigue damage for the interface details. The fatigue analysis will typically be performed using the sub-modelling technique described in [4.5.2].

Figure 4-13  Example of model extent for an internal turret

Figure 4-14  Example of hull girder model around the turret
SECTION 5  DEFINITION OF METEOLICAL OCEAN CRITERIA AND UNIT LOADING CONDITIONS

5.1 Site specific meteological ocean criteria
The basic description of the wave conditions usually takes the form of a 2-dimensional scatter diagram (Hs, Tp diagram), showing the relative frequency of various combinations of significant wave height and peak wave period (or zero-up-crossing period). Each of these combinations corresponds to a wave spectrum, usually expressed by some standard form, such as the JONSWAP spectrum with a cosine type directionality function.

In some instances the significant wave height in a scatter diagram may be presented as a range. In order to conduct the fatigue analysis it is necessary that the mean wave height for each range is specified, which is then used for scaling of the responses. If a mean significant wave height for each range is not provided then the range midpoint may be used.

5.2 Site specific loading conditions

5.2.1 Unit operation profile and number of load cases for analysis
The number of design load cases to be analysed for fatigue analysis is dependent on the design philosophy. Due to large difference in full load and ballast draft FPSOs shall normally be analysed for a minimum of three loading conditions. Additional loading conditions shall to be evaluated if:

— the difference in draft between two loading conditions exceeds 8 m
— the dynamic pressure profile would result in a non-conservative evaluation for a side longitudinal, or cargo tank configuration, i.e. checker board tank arrangement, is such that full tanks are adjacent to empty tanks.

5.2.2 Cyclic loads due to continuous loading/unloading.
Low frequency fatigue loads due to loading/offloading affect internal tank boundary structures such as transverse and longitudinal bulkhead, stringer connections to bulkheads, hopper tank, bottom longitudinal and inner bottom. Low frequency fatigue due to global hull girder bending from loading/offloading may normally be ignored if the contribution to fatigue damage is small, i.e. less than 5%, however local pressure fluctuations due to loading/offloading shall still be evaluated. See also [6.13.2].
SECTION 6  HYDRODYNAMIC LOADS AND UNIT MOTION ANALYSIS

6.1  Modelling of unit directionality

6.1.1  General
The influence of unit heading is important for hull fatigue analysis. Typically, the relative heading of wave loads for a trading tanker is considered to have equal probability for all unit headings, unless specific trade route data is available. This is termed ‘omni-directional’.

The calculation of unit response shall take into account the actual orientation of the unit relative to the wave and swell directions. The operational profile and effects of the mooring system (fixed or weathervane), wind, waves, swell, and current shall be considered when establishing the relative orientation. See [8.7].

Specific factors for the spread-moored and turret-moored arrangements are provided below.

6.1.2  Analysis of spread-moored unit
As described in 8.7.2, for a spread-moored unit, the relative wave heading is based on the mooring arrangement and the site-specific meteorological ocean data. The meteorological ocean data is normally represented in a directional scatter diagram.

In general the information related to the directionality of the environment and the intended orientation of the unit shall be provided in the project specification.

6.1.3  Analysis of turret-moored unit
With a turret-moored unit the unit may be free to weathervane. If the environmental loads are not aligned, i.e. non-collinear, then there will be occasions where the wave loading will be oblique to the unit centreline. It is important to capture this effect as it will have significant influence on the fatigue damage estimates for certain detail types, such as side longitudinals at the waterline.

The unit heading may be established through the mooring analysis by considering the appropriate environmental variables: wind, current, swell and seas (see Figure 6-1).

As described in 8.7.3 the following mean environmental loads, for a turret-moored unit, are usually considered for determining the relative heading:
— current loads
— wind loads
— wave drift loads.

![Diagram showing environmental variables acting on a turret moored ship](image)

Figure 6-1  Environmental variables acting on a turret moored ship

If the environmental loads are not aligned they may set up a resultant moment, tending to rotate the unit about the turret axis. If all environmental actions are collinear, i.e. they act in the same direction, and then
the unit orientation will tend to ‘head’ into the conditions. This is an ideal situation but, in practice, the environmental loads can be non-collinear, resulting in environmental loads oblique to the unit centreline. For active turret-moored units control forces may also affect the heading, such as:

— thruster forces
— forces due to any locking device or friction on the turret, and the corresponding torsional reaction moment from the attached mooring lines and risers.

All relevant mean forces should be taken into account when the heading angle of the unit is computed. Knowledge of the unit heading relative to the waves is necessary for the evaluation of the stresses due to 1st-order wave loads, which normally dominate the fatigue loading. In this case the heading angle shall be determined for each set of short-term, stationary, environmental conditions that are considered in the fatigue analysis.

As a result two approaches for taking the unit heading into account are considered:

— rule-of-thumb, and
— direct calculation.

Typically the simplified unit heading approach will be used as its application is similar to the spread-moored unit approach. The use of the simplified approach shall be validated for site-specific conditions.

The issues related to each method are provided below.

6.1.3.1  Headings for weather vaning units
If wind, wave and current conditions at a site are always collinear, then a turret-moored unit will orient into the waves, also called weather vaning. For sites where the environment tends to be collinear, the directionality may be simplified and presented in a format similar to a spread-moored system. Under these circumstances an experience-based allowance for directionality has often been applied.

Conservative assumptions with respect to wave headings shall be used in lieu of site specific data. A unit with external turret in the front end will have larger probability for head sea. If the turret is located near the hull quarter length, the unit will have less head sea. The relative proportions for each heading may differ for each environment and location of the turret.

The hull shall be analysed using the heading profile given in DNVGL-OS-C102 Ch.2 Sec.2 unless other reliable site specific data is available.

6.2  Hydrodynamic analysis modelling

6.2.1  Wave load and sea-keeping theory
The hydrodynamic load model shall give a good representation of the wetted surface of the ship, with respect to both geometry description and hydrodynamic requirements.

The sea keeping and hydrodynamic load analysis shall be carried out using 3D potential theory with a recognized computer program. The program shall calculate response amplitude operators (RAOs, transfer functions) for motions and loads in long crested regular waves. Requirements for the range of wave headings and wave periods to be applied in a hydrodynamic analysis are stated in [6.3.1] and [6.3.2] respectively. For calculation of viscous roll damping see [6.6].

6.2.2  Hydrodynamic panel model for a spread-moored unit (FPSO)
The element size of the panels for the 3D hydrodynamic analysis shall be sufficiently small to avoid numerical inaccuracies. In general, suitable accuracy is normally achieved using a mesh of at least 40 to 60 stations along the length of the ship, each of at least 15 to 20 nodes, giving a total of 600 to 1200 elements per half ship side. The mesh should provide a good representation of areas with large transitions in shape, and so the bow and aft areas are normally modelled with a higher element density than the parallel midbody area. A higher density of elements should also be applied around the bilge and close to the still water level (for all load conditions). As a rule of thumb, the mesh should have at least 5 elements per the shortest wave length in the ship’s lateral direction, assuming that the shortest wave length is less than or equal to approximately 10% of the ship’s length. An example:
For a ship of 250 meters in length with the shortest wave period equal to 4 seconds, the equation
\[ \lambda = \frac{g}{2 \cdot \pi} \cdot T^2 \]
gives the shortest wave length of 25 meters. Using 5 elements per shortest wavelength yields an element size of 5 meters and a total of 50 stations along the unit's length.

Special considerations shall be made if the unit is to operate in shallow waters. Shallow water will increase the magnitude of the first order response as the wave will "squeezed" under the hull. At first the panel size in the hydrodynamic model should be halved with respect to the above requirements. If a significant increase in the short term response is registered the panel size should be halved again. Dependent on the actual water depth this procedure should converge quite rapidly with the minimum panel size as ¼ of the original. Intermediate steps can be applied as appropriate as the increased number of panels will lead to additional analysis time. The roll motion should be the primary response to check but other responses should also be checked.

When a bilge keel is installed to reduce the roll motion of the ship it is important that the effect is captured in the hydrodynamic analysis. Both eddy making and viscous damping effects shall be considered. These effects are quite complex and a proper software tool shall be used to include the bilge keel effect. In order to properly model the bilge keel several parameters need to be defined such as the length, width and geometric location. It is also necessary to define the angle of the bilge keel relative to the horizontal plane. The effects from mooring lines and risers should be included as discussed in 6.2.4.

6.2.3 Hydrodynamic panel model for a turret-moored unit (FPSO)
There are specific issues that need to be considered when developing a panel model for a turret-moored unit:

**External turrets**
As indicated for the mooring and riser responses (see [8.7.1]) the external turret may be ignored for the purpose of the hull hydrodynamic analysis.

**Internal turrets**
Similar to external turrets the panel models for internal turrets are generally not necessary for the global hydrodynamic analysis since the vertical dynamic buoyancy forces can normally be ignored.

Experience show that there are minimal differences to the hull responses if the moonpool is included in the panel model compared to if the moonpool were ignored.

Since this RP is concerned with the turret interface structure, and not the turret itself, the turret may be modelled as structural elements in order to provide correct distribution of mooring/riser forces into the interface structure, i.e. the buoyancy forces on turret can be ignored.

For more general modelling information see 6.2.2.

6.2.4 Mass model
The mass model shall ensure a proper description of local and global moments of inertia around the longitudinal, transverse and vertical global ship axes. The determination of sectional torsion loads can be particularly sensitive to the accuracy and refinement of the mass model. For the hydrodynamic analysis, the mass distribution between 2 successive model stations should include, at least, 3 mass points in the longitudinal direction and 2 mass points in the transverse direction. In order to give a proper description of both local and global moments of inertia, these mass points should have the correct longitudinal position relative to the station coordinates, correct transverse position relative to the ship centreline, and correct vertical position relative to the baseline. The hydrodynamic model and the mass model should be in proper balance, giving still water shear force distribution with zero value at FP and AP.

Moorings and risers are not expected to influence first order motions significantly. However, in order to preserve static equilibrium, their static contribution shall be included in the analysis. The static contribution may be included either by vertical pre-tension according to static weight of these items, or as point masses. Care should be taken when distributing the point masses to avoid any unintended mass asymmetry which will have influence on the roll motions.
Where a turret is installed the mass can be included by adjusting the specific density of the steel assuming the turret is idealized as a coarse FE model or beam elements. Where beam elements are used typically there should be at least 5 beam elements over the height of the turret. This is applicable for both external and internal turrets.

Especially spread-moored FPSOs tend to have an asymmetric mass distribution due to a combination of side risers, ballast tanks and asymmetric topside modules. This asymmetry can influence the roll response even for bow on or stern on waves. This effect is usually not considered for the mass matrix used as input to the hydro analysis, so in most cases a full discrete mass model is necessary to capture the effects of any asymmetry in the mass distribution.

If the fatigue for the hull / turret interface is conducted using a combined analysis method with RAOs from mooring/riser analysis, as described in 8.6.2, then the mass is included in the mooring/riser analysis, and the mooring / riser masses should not be included in the hull hydrodynamic model.

6.2.5 Linking of hydrodynamic and structural models

There should be adequate correlation between hydrodynamic and structural models, i.e. both models should have:

— equal buoyancy
— coinciding mesh geometry (as far as possible)
— equal mass, balance and centre of gravity for both the hydrodynamic and structural models.

Any slight imbalance between the mass model and hydrodynamic model should be corrected by modification of the mass model. In many cases, the hydrodynamic analysis will be performed prior to completion of the structural model. The topside weights, which are often estimated during the design phase, will be part of the final structural model applied for load transfer. Topside weights based on preliminary loading manuals shall be included in the mass model applied in the hydrodynamic analysis. Depending on the current detailing of the design, the need for a contingency mass should be considered.

The final structural mass model should be used as the mass model in the final hydrodynamic analysis to establish pressure loads for the load transfer. See also [4.6].

It is recommended that the structural mass model in the final hydrodynamic analysis for load transfer be applied to the structural model. See also 4.6.

A slight modification of the mass model is usually needed in order to balance the hydrodynamic and mass model. If the unbalanced load is less than 5% of the wave excitation force, then the load balance may be achieved by adjusting the point masses close to FP and AP. Alternative procedures for balancing the models, such as inertia relief, may be used provided that the procedure is approved by the classification society and owner.

6.3 Calculation of wave loads and unit motions

6.3.1 Definition of wave headings

Stochastic fatigue life calculations shall normally be based on the following principles:

— for a spread moored unit (FPSO): Wave headings 0 to 360°, step 22.5° to 30°, (see also discussion of wave headings in [8.7.2])
— for a turret moored unit (FPSO): Wave headings 0° to 360°, step 22.5° to 30° (see also discussion of wave headings in 6.1)

In order to calculate the effect of short-crestedness the hydrodynamic analysis should include wave directions from –90° to +90° relative to the actual analysed heading.

Although it may be determined that a turret-moored unit will only experience loads from discrete headings (see 6.1.2) the fatigue analysis requires the wave spreading to be taken into account. In addition, the hydrodynamic analysis may be used for other purposes, such as the transit condition, which requires all headings to be analysed. Therefore, it is recommended that for a turret-moored unit the hydrodynamic analysis is completed for all headings.
6.3.2 Definition of wave periods

The hydrodynamic load analysis shall consider a sufficient range of regular wave periods (frequencies) so as to provide an accurate representation of wave energies and structural response. The shortest wavelength (lowest wave period) should be at least five times longer than either the panel diagonal of the smallest hydrodynamic element or 10% of the ship length. In deepwater conditions, for example, the corresponding period is given by equation (10).

\[ T_{\text{min}} = \left( \frac{2 \cdot \pi \cdot 0.1L}{g} \right)^{0.5} \]  

(10)

In addition, the following general requirements apply with respect to wave periods:

— the range of wave periods shall be selected in order to ensure a proper representation of all relevant response transfer functions (motions, sectional loads, pressures, drift forces) for the wave period range of the applicable scatter diagram

— a proper wave period density should be selected to ensure a good representation of all relevant response transfer functions (motions, sectional loads, pressures, drift forces), including peak values. Typically, for stochastic fatigue life calculations, 25 to 30 wave periods are used for a smooth description of transfer functions.

6.4 Derivation of motion induced accelerations and inertia loads

Acceleration and inertia induced dynamic loads can be significant for topsides, flare towers, swivel stacks, etc. Loading conditions for fatigue analysis for these detail types shall be selected to represent typical loading situations that will be relevant during most of the operational lifetime of the unit.

Motions in 6 degrees of freedom should be determined based on the environmental criteria given in Sec.5.

Accelerations in longitudinal and transverse direction should include the g-component induced by ship pitch motion and ship roll motion.

The stress ranges at lower load levels (intermediate wave amplitudes) give a relatively large contribution to the cumulative fatigue damage such that non-linear effects due to large amplitude motions and large waves can be neglected in the fatigue analysis.

In cases where linearization is required, it is recommended that the linearization is performed at a load level representative for the stress ranges that contribute the most fatigue damage, i.e. stress ranges at probability of exceedance level between 10^{-2} to 10^{-4}.

6.5 General response issues

Load effects that are symmetrical with respect to the ship’s centreline (e.g. vertical bending moment for tanker shaped structures) need only be evaluated for one half (either port or starboard) of the ship.

The vertical bending moment should be calculated about the neutral axis. The neutral axis may vary substantially along the unit length.

6.6 Calculation of roll damping

The roll damping computed by 3D linear potential theory includes moments acting on the hull as a result of the creation of waves when the unit rolls. At roll resonance, however, the 3D potential theory will under-predict the total roll damping. The roll motion will, consequently, be grossly over-predicted. In order to adequately predict total roll damping at roll resonance, the effect from damping mechanisms not related to wave-making, such as vortex-induced damping (eddy-making) near sharp bilges, drag of the hull (skin friction), and bilge keels (normal forces and flow separation), should be included. Such non-linear roll damping models have typically been developed based on empirical methods, using numerical fitting to model test data. Example non-linear roll damping methods for ship hulls includes those published by Tanaka, Kato and Himeno.
Results from experiments indicate that non-linear roll damping on a ship hull is a function of roll angle, wave frequency and forward speed. As the roll angle for fatigue analysis is generally unknown and depends on the scatter diagram considered for fatigue analysis, an iteration process is required to derive the non-linear roll damping. The following 4-step iteration procedure may be used for guidance:

**Table 6-1**

<table>
<thead>
<tr>
<th>STEP</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Input a roll angle, ( \theta_{\text{input}} ), to compute non-linear roll damping</td>
</tr>
<tr>
<td>2</td>
<td>Perform unit motion analysis including damping from STEP 1</td>
</tr>
<tr>
<td>3</td>
<td>Calculate long-term roll motion, ( \theta_{\text{update}} ), with probability level ( 10^{-4} ), using design wave scatter diagram.</td>
</tr>
<tr>
<td>4</td>
<td>If ( \theta_{\text{update}} ) from STEP 3 is equal to ( \theta_{\text{input}} ) in STEP 1, stop the iteration. Otherwise, set ( \theta_{\text{input}} = \theta_{\text{update}} ), and go back to STEP 1.</td>
</tr>
</tbody>
</table>

### 6.7 Definition of transfer function

Linear modelling of the ship response is usually sufficient for fatigue assessment purposes, provided that non-linearity due to wet and dry surfaces at the waterline is accounted for separately (see also [4.4.3]). Under this assumption of linearity, the response is then described by a superposition of the responses from all regular wave components that describe the irregular sea. The resulting stress is obtained by summation over all contributing dynamic loads/load effects.

Since most fatigue damage is related to moderate wave heights, the linear frequency domain results are normally applied without any corrections for large wave effects.

The transfer function (frequency response function) \( H(\omega, \theta) \), represents the response to a sinusoidal wave with unit amplitude for different frequencies \( \omega \) and wave heading directions \( \theta \). Transfer function values shall be determined for a sufficient number of frequencies and wave headings. Consistent representation of transfer function phase and amplitude is necessary in order to achieve a correct modelling of the combined local stress response. The mass model should reflect the steel weight distribution and the distribution of cargo in the vertical, longitudinal and transverse directions.

The vertical bending moment may be estimated using a hydrodynamic model (see [6.1]) and by running a wave load program that determines the response for a set of wave frequencies and heading directions. The vertical bending moment transfer function is computed as the vertical bending moment \( M_{v}(\omega, \theta) \) per unit wave amplitude \( (H/2) \).

\[
H_{v}(\omega, \theta) = \frac{M_{v}(\omega, \theta)}{H/2} \quad (11)
\]

The horizontal bending moment transfer function, \( H_{h}(\omega, \theta) \), shall be determined in a similar manner to the determination of the vertical bending moment transfer function and shall have consistent phase relations.

The external pressures shall be determined in a similar manner to determination of the vertical bending moment, and shall have consistent phase relations. In the waterline region, a reduction of the pressure range is applicable due to intermittent wet or dry surfaces, see [4.4.3].

The internal tank pressures may be obtained by combining the accelerations described in App.A (substituting computed acceleration for the given acceleration estimates) with combined transfer functions for motions and accelerations relative to the unit axis system.

Consistent representation of transfer function phase and amplitude is necessary in order to achieve a correct modelling of the combined local stress response.
6.8 Combination of transfer functions for the load component method

The combined stress response can be determined by a linear complex summation of stress transfer functions. The combined local stress transfer functions may be found by combining the complex response transfer function for unit loading conditions as:

\[
H_o(\omega|\theta) = A_1 H_{1o}(\omega|\theta) + A_2 H_{2o}(\omega|\theta) + A_3 H_{3o}(\omega|\theta) + A_4 H_{4o}(\omega|\theta) + A_5 H_{5o}(\omega|\theta) + A_6 H_{6o}(\omega|\theta)
\]

where:

- \(A_1\) = stress per unit vertical bending moment.
- \(A_2\) = stress per unit horizontal bending moment.
- \(A_3\) = stress per unit relative lateral external pressure load.
- \(A_4\) = stress per unit relative lateral internal pressure load.
- \(A_5\) = stress per unit axial load.
- \(A_6\) = stress per unit acceleration load.
- \(H_{1o}(\omega|\theta)\) = transfer function for combined local stress.
- \(H_{2o}(\omega|\theta)\) = transfer function for vertical bending moment at a representative section.
- \(H_{3o}(\omega|\theta)\) = transfer function for horizontal bending moment.
- \(H_{4o}(\omega|\theta)\) = transfer function for external pressure in centre of the considered panel.
- \(H_{5o}(\omega|\theta)\) = transfer function for liquid cargo pressure in centre of the considered panel.
- \(H_{6o}(\omega|\theta)\) = transfer function for axial load.
- \(H_{acc}(\omega|\theta)\) = transfer function for acceleration loads (includes also topside loads).

\(A_k\) is the local stress response due to a unit sectional load for load component \(k\). The \(A_k\) factors may be determined either by FEM analyses or by a simplified method for replacement of the described loads by unit loads as described in Appendix A. Note that it is important to ensure compatibility between the reference (coordinate) systems used in both the load model and the stress analysis model.

The factors \(A_k\) may be determined by adding unit sectional loads at the considered sections, to determine the effect of each individual load component when the hull pressure distribution is determined from a wave loading program.

Negative dynamic pressures do not occur at the waterline (intermittent wet and dry surfaces) and hence the stress range is proportional to the pressure amplitude. The effective stress range for longitudinal details in the waterline region may be estimated using the reduction factor, \(f_p\), as described in 4.4.3. Alternatively, the stress range distribution may be determined from the pressure ranges by integration of pressures in each wave height (or sea state) in the long-term environmental distribution.

Quality assurance using this approach is very important so as to ensure that the correct phase relationships between the different loads are maintained, see [9.4.3].

6.9 Short-term response analysis

6.9.1 Wave-frequency response

The short term distribution of load responses for fatigue analyses may be estimated using the wave climate, represented by the distribution of \(H_s\) and \(T_z\) in a sea scatter diagram for the actual area.

The environmental wave spectrum for the different sea states can be defined using the Pierson-Moschowitz wave spectrum:

\[
S_p(\omega|H_s,T_z) = \frac{H_s^2}{4\pi} \left( \frac{2\pi}{T_z} \right)^3 \omega^4 \exp \left( -\frac{1}{\pi} \left( \frac{2\pi}{T_z} \right)^2 \omega^4 \right), \quad \omega \geq 0
\]
The ship response spectrum based on the linear model is directly given by the wave spectrum, where the relationship between unit wave height and stresses, the transfer function $H_{\sigma}(\omega|\theta)$, is established as:

$$S_{\sigma}(\omega|H_z,T_z,\theta) = |H_{\sigma}(\omega|\theta)|^2 S_{\sigma}(\omega|H_z,T_z)$$

The spectral moments of order $n$ of the response process for a given heading are calculated as:

$$m_n = \int_0^\infty \omega^n S_{\sigma}(\omega|H_z,T_z,\theta) d\omega$$

The spectral moments may include wave spreading as:

$$m_n = \int_{\theta-90^\circ}^{\theta+90^\circ} \sum_{\theta-90^\circ}^{\theta+90^\circ} f(\theta) \omega^n S_{\sigma}(\omega|H_z,T_z,\theta) d\omega$$

using a spreading function $f(\theta) = k \cos^n(\theta)$, where $k$ is selected such that,

$$\sum_{\theta-90^\circ}^{\theta+90^\circ} f(\theta) = 1$$

and $n$ is normally equal to 2.

The wave spreading should include wave headings from $-90^\circ$ to $+90^\circ$.

The stress range response for ship structures can be assumed to follow the Rayleigh distribution within each short-term condition. The stress range distribution for a given sea state $i$ and heading direction $j$ is then derived as:

$$F_{\Delta\sigma ij}(\sigma) = 1 - \exp \left(-\frac{\sigma^2}{8m_{0ij}}\right)$$

where $m_{0ij}$ is the spectral moment of order zero.

A summation of the fatigue damage within each sea-state and heading direction can be applied, see DNVGL-RP-C203.

### 6.9.2 Swell response

Response to swell may be calculated similarly to the response to wind-waves in 6.3.1, using the Jonswap spectrum with peak enhancement factor equal to 5 and $\cos^2\theta$ spreading unless otherwise stated. The response to wind-waves is independent of response to swell, and the combined effect can be obtained by adding the variances of these responses. The up-crossing period of the combined response through the mean level can be computed using the sums of the respective spectral moments.

### 6.9.3 Low-frequency response

Low-frequency response contributions are discussed in 8.7.1. The contributions are normally determined from the separate analysis of mooring or riser systems, and are not integrated in the global hull analysis. The hot-spot stresses due to low-frequency mooring or riser forces are obtained via influence coefficients as described in 4.6. Standard deviations of stresses are directly established from the analysis.

### 6.9.4 Combination of wave-frequency and low-frequency response

The frequency difference between the wave-frequency and low-frequency response components increases the complexity related to the computation of fatigue damage. The procedure in DNVGL-RP-C203 has been developed for a stress process with a narrow band of frequencies. If the wave-frequency and low-frequency processes are both of appreciable magnitude, then combination of the two components leads to a wide-banded stress process. A short piece of the stress time history may be visualized as a sine wave with a long...
period, superimposed by sine wave with a short period; one tenth of the long period, as indicated in Figure 6-2.

Figure 6-2 Typical mooring line tension time history, with dominant wave-frequency component

Separate calculation of the fatigue damage from the two frequency bands is known as "simple summation." However, this calculation procedure does not account for the augmentation of the low-frequency amplitudes by the wave-frequency amplitudes. Simple summation is therefore non-conservative and shall not be used.

An alternative procedure, which is both conservative and convenient to use, is known as the "combined spectrum" method. The spectra of the wave-frequency and low-frequency components are added together. The standard deviation of the combined stress process is given by:

$$\sigma_C = \sqrt{\sigma_W^2 + \sigma_L^2}$$

where $\sigma_W$ and $\sigma_L$ are the standard deviations of the wave-frequency and low-frequency stresses respectively. The up-crossing rate of the combined stress process through the mean level is given by:

$$v_C = \frac{1}{\sigma_C} \sqrt{2 \sigma_L^2 v_L^2 + \sigma_W^2 v_W^2}$$

where $v_W$ and $v_L$ are the up-crossing rates of the wave-frequency and low-frequency processes respectively. If both the low-frequency and wave-frequency events cause equivalent damage, then the wave-frequency will dominate the combined process frequency, and the relatively large number of stress cycles will lead to conservatism in the fatigue damage summation. The level of conservatism will be excessive (>200%) when frequency ratio is larger than 10. However, if either of the low-frequency or wave-frequency process contributes negligible damage, the conservatism is insignificant.

The conservatism of the above combined spectrum method can be greatly diminished by multiplying the fatigue damage with a correction factor for dual narrow-banded processes. However this factor should not be used when the frequency ratio is smaller than 4. The expression for this correction factor is complex, but only the standard deviations and up-crossing frequencies of the two component processes are needed as input. The correction factor is given by:

$$\frac{v_p}{v_c} \left[ \frac{n + 1}{\lambda_L \sqrt{2 \pi}} - \frac{\lambda_W}{\sqrt{2 \pi}} + \frac{m \Gamma \left( \frac{m + 1}{2} \right)}{\Gamma \left( \frac{m}{2} + 1 \right)} - \frac{v_W}{v_C} (\lambda_W)^{\frac{m}{2}} \right]$$

where $n$, $m$, $\lambda_L$, $\lambda_W$, and $v_C$ and $v_W$ have the usual meanings.

---

**Illustration of line tension time history**

![Line tension time history](image)
where $\Gamma(t)$ is the gamma function, $m$ is the slope parameter of the S-N curve.

The $\lambda$'s are normalized variances given by:

$$
\lambda_L = \frac{\sigma_L^2}{\sigma_C^2}, \quad \lambda_W = \frac{\sigma_W^2}{\sigma_C^2} \tag{21}
$$

and the up-crossing rate of the envelope of the stress process is given by:

$$
V_P = \sqrt{\lambda_L^2 V_L^2 + \lambda_W \lambda_W V_W^2 \delta_W^2} \tag{22}
$$

with $\delta_W = 0.1$.

If the process standard deviations are in different regions of the S-N curve a single slope S-N curve should be considered by ignoring the change of slope. Note that Equation (23) can also be used to combine fatigue damage, with similar results, when the frequency ratio is larger than 4.

The following equation can be used for calculation of total fatigue damage from low frequency and high frequency fatigue at a hot spot:

$$
D = D_1 (1 - \frac{V_2}{V_1}) + V_2 \left( \frac{D_1}{V_1} \right)^{1/m} + \left( \frac{D_2}{V_2} \right)^{1/m} \tag{23}
$$

where

$D_1$ = calculated fatigue damage for the high cycle fatigue (from wave action)

$D_2$ = calculated fatigue damage for the low cycle fatigue (from loading and unloading)

$V_1$ = mean zero up crossing frequency for the high cycle fatigue

$V_2$ = mean zero up crossing frequency for the low cycle fatigue

$m$ = negative inverse slope of the S-N curve.

Alternatively, additional accuracy in the fatigue damage can be obtained by making a time simulation of the combined stress process, and applying the rain flow counting method to establish the short term distribution of stress ranges in each environmental state. These numerical short term distributions can be weighted with the probability of occurrence of each environmental state, and then combined to obtain the long term distribution of stress ranges.

For time domain simulations, it is not practical to run all sea-state bins (100 – 150) and multiple headings, typically 800 – 1 200 time domain simulations. Simplifications are usually made to reduce this number significantly. The simplifications implies that a very few basic sea states are selected, e.g. for riser response, selected sea states - typically Near, Far and a few Cross headings (relative riser plane) - are simulated. The reduced number (could be in the order of only 30 – 40 summed over all headings) of sea-states have to be carefully selected and it will be the responsibility of the riser designer to document that the selected reduced number of sea-states are sufficient. It should be attempted to select sea-state bins producing similar amount of damage.

One procedure which has been used for SCR’s (steel catenary riser) hung off from Spar platforms may provide some guidance. This procedure showed that the fatigue damage in the TDP (touch down point) region of a SCR was approximately proportional to $H_s$ squared.
Selection of $H_s$

$$H_s = \left( \frac{\sum_{i=1}^{N} n_i \cdot H_{s_i}^m}{\sum_{i=1}^{N} n_i} \right)^{1/m}$$

- $N$ = number of sea-states included in the sea-state bin.
- $H_{s_i}$ = significant wave height for the $i$th sea-state.
- $n_i$ = number of occurrences of the $i$th sea-state.
- $m$ = exponent in power law relationship between damage and $H_s$, i.e. damage proportional to $(H_s)^m$.

Selection of $T_p$

An equivalent peak period may be calculated using the same method, but the exponent ($m$) was set to one.

Care should taken when selecting peak periods to make sure that the roll resonance period is well covered by the selected sea states.

Note that this is just an indication of how a reduced number of sea-states can be selected and that this will vary with actual riser system (SCR’s, flexible risers etc.), location (TDP, hang off etc.) and floater hang off (turret, spread mooring). The ultimate goal off with this approach is to achieve reliable results with the partial scatter diagram and therefore to avoid any correction factors between a full and a partial scatter diagram.

An alternative procedure is an interpolation process described as follows:

1. Select a few environmental states that span the entire range and a few states spread out around the inside of this range.
2. Calculate the damage rate per unit time in each of these states.
3. Develop an interpolation function to estimate the damage rate in any environmental state.
4. Sum up the fatigue damage over the entire set of environmental states, using the interpolation function for the damage rate, together with the time duration of each state.
5. Calculate the damage rate in a few, new intermediate states. Ideally these should be states that contribute significantly to the total damage and are distant from the states already calculated.
6. Check the performance of the interpolation function and refine as necessary, by repeating these steps until adequate accuracy is obtained.

6.10 Calculation of long-term stress range response

This section of the RP may be applied for verification purposes, and can also be used for investigation of accumulated damage from a combined response. This section is not required for a stochastic fatigue analysis.

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

$$F_{\Delta \sigma}(\sigma) = \sum_{i=1}^{N} \sum_{j=1}^{M} r_{ij} F_{\Delta \sigma ij}(\sigma) p_{ij}$$

where:

- $p_{ij}$ = is the probability of occurrence of a given sea state $i$ combined with heading $j$
- $r_{ij} = \frac{\nu_{ij}}{\nu_0}$ = is the ratio between the response crossing rates in a given sea state and the average crossing rate.
For derivation of the accumulated fatigue damage, the estimated long-term stress range distribution can be applied. A Weibull distribution, with shape parameter $h$ and scale parameters $q$, provides a good representation of the long-term stress range distribution. The fitting of the Weibull distribution to the sum of Rayleigh distributions should preferably be based on a least square technique for a number of stress ranges $\sigma$. The Weibull distribution is described as:

$$F(\sigma) = 1 - \exp \left[ -\left( \frac{\sigma}{q} \right)^h \right]$$

As guidance for definition of the Weibull distribution parameters, the stress levels corresponding to a cumulative probability of 95% and 99% will divide the fatigue damage in three approximately equal parts, damages, indicating the most important range of the response distribution.

### 6.11 Combining long-term responses

Given two long-term distribution functions $F_1(\sigma)$ and $F_2(\sigma)$ for the stress components $\sigma_1$ and $\sigma_2$ the distribution of the combined stress is obtained by adding stress components at the same probability level. Thus, for a probability level $P_i$, the distribution functions are inverted to give the combined stress $\sigma_c$.

$$\sigma_{c_i} = F_1^{-1}(P_i), \quad \sigma_{c_2} = F_2^{-1}(P_i)$$

$$\sigma_{c} = \sigma_{c_1} + \sigma_{c_2}$$

The distribution of the combined stress can be established by carrying out this computation for a range of probabilities, $P_i$. A Weibull distribution can be fitted to this empirical distribution. This combination is exact when the two stress processes are linearly dependent, i.e., both stress processes have the same periods. If the periods of the two processes are different then it is prudent to apply the shorter period in the fatigue analysis.

### 6.12 Application of hydrodynamic loads on structural FE models

The loads from the hydrodynamic load analysis shall be directly applied on the global FE model, as described in 2.5.2. The calculated design loads shall be applied as a combination of pressure forces on the hull and inertia forces. If loads are applied correctly, the global FE model will be close to equilibrium, and the reaction forces should be close to zero. The effect of simultaneously acting dynamic ship loads should be accounted for in the analysis.

Loads due to viscous damping shall be included and transferred to the structural model.

The sectional loads shall be checked in order to ensure that the sectional loads in the structural FE model are similar to the sectional loads in the hydrodynamic load analysis (see [9.4]).

The model balance should be checked according to [6.2.5].

### 6.13 Other load effects

#### 6.13.1 Introduction

Other load effects that may be included are loads due to loading/unloading, slamming, sloshing, green seas and springing effects. However, it should be emphasized that most of these load effects represent analysis areas still under development with respect to both analytical theory and practical application. Slamming, sloshing and green seas include impact effects for which values and application to fatigue analysis are complicated.
6.13.2  Cyclic loads due to loading/unloading

Fatigue damage resulting from global bending during loading/unloading need not normally be calculated provided that the unit is designed according to 1A and with less than 1000 loading/unloading cycles during the lifetime. The number of cycles expected during the service life of the unit shall be verified by the operator.

It is not normally necessary to calculate fatigue damage from low frequency fatigue in transverse or longitudinal bulkheads due to filling and emptying operations provided that all the following conditions are met:

— the pressure is cycling from one side only (stress range = stress amplitude),
— number of loading/unloading cycles during the lifetime is less than 1000 (few cycles), and
— the $K_0$ factor is less than 2.0 for normal strength (NS) material and 1.44 for high strength (HS) material.

If these conditions are not satisfied, then the fatigue damage from the cyclic load source should be calculated as for high cycle fatigue, e.g. by using one stress block in the equation for fatigue damage in DNVGL-RP-C203. Then, the fatigue damage from cyclic loading/unloading can simply be added to damage from other sources.

A low frequency fatigue analysis should be performed for design of bulkheads between tanks that experience a full load reversal according to the loading steps in Table 6-2. (The difference between load scenarios in step 2 and step 4 gives a significant stress range).

**Table 6-2 Loading sequence that may give contribution to fatigue damage**

<table>
<thead>
<tr>
<th>Loading steps</th>
<th>Tank</th>
<th>Neighbour tank</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Empty</td>
<td>Empty</td>
</tr>
<tr>
<td>2</td>
<td>Full</td>
<td>Empty</td>
</tr>
<tr>
<td>3</td>
<td>Full</td>
<td>Full</td>
</tr>
<tr>
<td>4</td>
<td>Empty</td>
<td>Full</td>
</tr>
</tbody>
</table>

The welded connections between stiffeners in transverse frames to longitudinals in double bottoms have been found to be susceptible to low frequency fatigue cracking when designed according to standard ship practices.

The calculated hot spot stress should be combined with the hot spot stress S-N curve E. However, due to large stress cycles implying local yielding at the hot spot the calculated hot spot stress from a linear elastic analysis should be increased by a plasticity correction factor before the S-N curve is entered for calculation of fatigue damage.

The plasticity correction reads:

\[
k_p = 1.0 + 0.4 \left( \frac{\Delta \sigma}{2 \sigma_y} - 1 \right)
\]

for $\Delta \sigma > 2 \sigma_y$

\[
k_p = 1.0
\]

for $\Delta \sigma \leq 2 \sigma_y$

where $\sigma_y = \text{yield strength of the base material (nominal or characteristic value)}$. 

\[(26)\]
SECTION 7 HOT SPOT ANALYSIS

7.1 General
To conduct fatigue analysis using the hot spot stress method, the designer may select either of the following procedures:

a) all structural models described in Section 4 are combined into one model, or
b) fine mesh models are solved by use of sub-model analyses where deformations/forces from the global analysis are transferred to the fine mesh model. (This load transfer can be performed either manually or, if sub-modelling facilities are available, automatically by the computer program).

It should be noted that different terminology may be used to describe the same type of finite element model. The analysis process may be described as starting from a coarse global element model, see Figure 7-1, with subsequent finite element models that have refined mesh density. Sub-models may be denoted as intermediate or, if used directly for hot spot stress calculation, it may simply be denoted as a sub-model. Such a model may also be denoted as an SCF model, local model or a fine mesh model.

![Figure 7-1 Schematic description of finite element model terminology](image)

7.2 Calculation of nominal stress
If 4 or 8-node shell elements are used for modelling of plate behaviour in a local model, then the membrane stress may be taken at the “middle surface”.

Similarly if one 20-node element is used over a plate thickness, the nominal stress may be derived from the midside node in the middle of the thickness.

Note that the definition of nominal stress states that the nominal stress for a particular calculation should be evaluated using sound engineering judgement.

7.3 Hot spot analysis of local structure

7.3.1 General
Quality assurance (QA) of local structural analysis shall be performed in order to ensure that boundary loads or displacements are properly defined and that the model is in equilibrium. Complex structural details, such as side shell connections and riser porches, where multiple fatigue hot spots exist, shall be screened to ensure that all critical hot spots are identified for fatigue assessment.

7.3.2 Local screening of hot spots for fatigue assessment
As a minimum requirement, local fatigue screening should be performed for all critical hot spots using fine-mesh finite element models with mesh size in the order of $t \times t$. Fatigue screening calculations should use nodal and/or Gaussian stresses in conjunction with an SCF of 1 with the appropriate S-N curve, as given in DNVGL-RP-C203.
The hot spot stresses at critical locations should be calculated as specified in 7.3.2 to 7.3.4. These hot spot stresses can be used to update the fatigue life, see [8.3].

7.3.3 Derivation of hot spot stress

**Stress normal to the weld**

For modelling with shell elements without any weld in the detailed geometry the following procedure shall be used:

— the hot spot stress is taken as the stress at the read out point 0.5 \( t \) away from the intersection line and combined with SN-curve E.

For modelling with three-dimensional elements with the weld included in the detailed geometry the following procedure shall be used:

— the hot spot stress is taken as the stress at the read out point 0.5 \( t \) away from the weld toe and combined with SN-curve E.

It is recommended to link the derived hot spot stress to the E-curve for welded joints in DNVGL-RP-C203.

**Stress parallel to the weld**

When plate/shell elements are used and the weld is not modelled, the stress shall be taken at the plate and tubular intersection lines, for shell elements.

If the weld is modelled using 3D finite elements, the stress shall be taken at the weld toe for 3D elements.

See DNVGL-RP-C203 for selection of S-N curve.

![Figure 7-2 Read out point for stress in relation to direction of main stress and weld](image)

7.3.4 Derivation of stress for fillet welds

The design criterion for fillet welds make use of different stress components in the fillet weld, as shown in DNVGL-RP-C203 and illustrated in Figure 7-3.

The relevant stress range for potential cracks in the weld throat of load-carrying fillet-welded joints and partial penetration welded joints may be found as:

\[
\Delta\sigma_w = \sqrt{\Delta\sigma_p^2 + \Delta\tau_p^2 + 0.2 \Delta\tau_p^2}
\]  

(27)

The stress components used in this equation are shown in Figure 7-3. Equation (30) is a general equation for fatigue design of fillet welds subjected to a complex loading.

At some locations of the welds there are stress in the plate normal to the fillet weld, \( \sigma_{wp} \), see Figure 7-4, and a shear stress in the plate parallel with the weld \( \tau_{wp} \).
Equilibrium of plate in section parallel with the weld gives:

\[ \tau_{//} = \frac{2a \cdot \tau_{//p} \cdot t_p}{2a} \]  \hspace{1cm} (28)

where \( \tau_{//} \) is mean nominal shear stress in the weld as shown in Figure 7-3.

\[ a = \text{throat thickness of weld} \]
\[ t_p = \text{plate thickness} \]

The shear stress in the weld is then obtained from equation (28) as

\[ \tau_{//} = \frac{\tau_{//p} \cdot t_p}{2a} \]  \hspace{1cm} (29)

Equilibrium of plate in section normal to the weld (see Figure 7-4) gives:

\[ (\tau_{\perp} + \sigma_{np}) \frac{1}{2} \sqrt{2} \ 2a = \sigma_{np} \cdot t_p \]  \hspace{1cm} (30)

And assuming a reaction force on the weld throat in the direction of \( \sigma_{np} \) then gives:

\[ \tau_{\perp} = \sigma_{\perp} = \frac{\sigma_{np} \cdot t_p}{2\sqrt{2} \ a} \]  \hspace{1cm} (31)

Then from Equation (30) a combined stress is obtained with use of equation (32), and (34). This resulting stress range shall be used together with the W3 curve.

![Figure 7-3 Explanation of stresses on the throat section of a fillet weld](figure_url)
Figure 7-4 Stress in section normal to the weld

The concept of "Engineering shear stress" can be used when assessing fatigue in fillet welds. In the following it is shown that this methodology gives the same answer as the procedure outlined above. Reference is made to Figure 7-5. When the connection is subjected to an axial force $P$ with tangential force $T = 0$. The nominal stress in the fillet weld is understood to be the combined stress:

$$
\sigma_n = \sqrt{\sigma_\perp^2 + \tau_\perp^2}
$$

(32)

where

- $\sigma_\perp$ = mean normal stress acting on the throat section as shown in Figure 7-5.
- $\tau_\perp$ = mean shear stress acting on the throat section as shown in Figure 7-5.

In some literature this stress is also denoted as "Engineering shear stress" when it is derived for the fillet weld in Figure 7-5 as:

$$
\sigma_n = \frac{P}{2aL}
$$

(33)

i. e. the engineering shear stress is simply obtained as force divided by weld throat area.

The parallel shear stress is similarly derived as:

$$
\tau_p = \frac{T}{2aL}
$$

(34)

where the notation is illustrated in Figure 7-5. Thus, following Equations (27), (29) and (31) the following shear stress for fatigue in the fillet weld is obtained

$$
\sigma_\perp = \frac{1}{2aL} \sqrt{P^2 + 0.2T^2}
$$

(35)
The nominal stress defined in DNVGL-RP-C203 shall be used for fatigue analysis of a fillet weld. It is recommended that this stress is derived from the nominal stress in the plate connected by the fillet weld. If a 3-dimensional analysis of a welded connection has been performed, such as for the example shown in Figure 7-6, then the nominal stress, \( \sigma_d \), should be derived from the 3 dimensional FE analysis. The nominal stress, \( \sigma_d \), can then be used to calculate the stress in the weld. (The calculation can be based on considerations of equilibrium as shown in example 1 below.)

If a 3D FE analysis is used to determine the stress in a fillet weld, then the fillet weld should be modelled with element sides through the section of the throat thickness such that the calculated stress can be derived directly at the element nodes. At least 4 elements should be used over the throat thickness, ref. Section 4.5.3. The mean stress over the throat thickness should be calculated for design purposes based on the element stresses for each stress component.

**Example 1: Doubler plate**

Note: The following stresses are derived based on the definitions given in DNVGL-RP-C203.

Equilibrium in the transverse direction:

\[
\sigma_\perp = \tau_\perp
\]  
(36)

Equilibrium in the longitudinal direction:

\[
\sigma_d t_d = \frac{1}{2} \sqrt{2(\sigma_\perp + \tau_\perp)} a
\]  
(37)

From equations (36) and (37), the stress to be used with the W3 curve is:

\[
\sigma_w = \sigma_y \frac{t_d}{a}
\]  
(38)
For a bi-axial stress field, the hot spot with the largest principal stress range should be analysed. The resulting stress components at this hot spot should be combined as given in DNVGL-RP-C203.

**Example 2: Fillet welds at scallops**

Due to the weld shape and a possible bending moment over the bracket thickness, the stress in the weld at the scallop (see Figure 7-7) shall be assessed by the following procedure.

The shear stress in the weld is estimated by considering a region around the edge of the welded specimen, see Figure 7-8. A length of weld, \( d \), is included on both sides of the specimen. The length \( d \) may be taken as equal to the thickness, \( t \). The weld area within the section under consideration is calculated as:

\[
A_w = 2da + at + 2a^2
\]  
(39)

The corresponding nominal specimen section area is calculated as:

\[
A_s = dt
\]  
(40)

From the requirement for equilibrium:

\[
\sigma_n A_s = \sigma_w A_w
\]  
(41)

giving:

\[
\sigma_w = \frac{A_s}{A_w} \sigma_n
\]  
(42)
The nominal stress in the bracket is now calculated based on the stress at the Gaussian points 3 and 4 as shown in Figure 7-9. The nominal stress is calculated as:

\[
\sigma_n = \frac{1}{4}\left(\left(\sigma_3 + \sigma_4\right)_{\text{at element surface}} + \left(\sigma_3 + \sigma_4\right)_{\text{at middle of element}}\right)
\]  

(43)

The stress range in the weld expressed by \(\sigma_w\) should be used together with the W3-curve.

![Figure 7-7 Potential fatigue locations for a side longitudinal connection to a transverse frame](image)

![Figure 7-8 Section through weld at bracket scallop considered](image)

![Figure 7-9 Element at scallop showing numbering of Gaussian points](image)

7.3.5 Derivation of stress for base material

The stress range in the base material is the maximum calculated stress range, normally taken at some edge to include the effect of stress concentration factors such as cut outs (see example in Figure 7-10). For analysis with 8-node shell elements, the stress for use with the S-N curve can be derived directly from the element nodal stresses. If 4-node elements are used for the analysis, the stress is usually calculated inside the element. Fictitious bar elements may be used in order to derive the stress along the surface provided that the bar elements have negligible area but stiffness equal to the plate, i.e. the same Young’s modulus. The axial stress from the bar element between A and B in Figure 7-10 can then be used directly together with the S-N curve.
7.3.6 Derivation of hot spot stress concept for simple connections

It should be noted that the definition of the stress field through the plate thickness implies that the described hot spot stress methodology is not directly recommended for simple cruciform joints, simple T-joints in plated structures or simple butt joints that are welded from one side only. Analysing such connections with for example shell elements would result in a hot spot stress equal the nominal stress. This is illustrated by the shell model shown in Figure 7-11.

Figure 7-11 Illustration of difference to attract stresses normal to and in plane of a shell element model

For stresses in the direction normal to the shell (direction I) there will be no stress flow into the transverse shell plating as it is represented only by one plane in the shell model. However, it attracts stresses for in-plane (direction II) shown in Figure 7-11.

As the nominal stress S-N curve for direction I is lower than that of the hot spot stress S-N curve, it would be non-conservative to use the hot spot concept for this connection for direction I while hot spot concept would be acceptable for direction II at position a) in Figure 7-11. For direction I at position c) in Figure 7-11 the calculated stress from finite element analysis should be combined with S-N curve F in order to calculate the fatigue damage.
SECTION 8  HULL/TURRET INTERFACE FATIGUE DESIGN

8.1  Introduction

8.1.1  General
In order to control the unit’s heading and wave-induced motions, particularly roll, it is sometimes appropriate to allow the unit to rotate into the weather. This is termed ‘weather vaning’ and typically a turret-mooring arrangement is used to achieve this functionality. Under certain circumstances turret-mooring arrangements may also be necessary for units that need to disconnect from the mooring system in extreme weather conditions, e.g. tropical storms, or to allow flexibility for exploiting marginal fields.

Based on the current industry practice the turret design is generally the responsibility of the turret designer and hull falls under the hull designer (shipyard) scope. However, the interface structure between the turret and the hull may be the responsibility of either party. As a result the management of the interface structure design is a crucial issue. This RP requires that the structural interface design be defined in the project plans in terms of technical and management activities at the project start.

Compared to a spread-moored unit, the hull fatigue design effort required for a turret-moored system increases due to two main considerations:

— unit heading due to weather vaning, and
— interaction of the turret and the hull structure.

8.1.2  Turret functionality
The turret serves three main functions:

— maintain the unit at the desired location
— allow the unit to weather vane, and
— support the risers and flow lines.

Typically the turret is moored to the seabed by means of mooring lines and is connected to the unit by bearings and supporting structure. The bearing support structure within the hull is designed to resist the vertical and horizontal forces from the turret. The turret offset relative to the seabed will depend on the water depth and the characteristics of the unit and mooring system.

Most turrets allow the unit to rotate 360° to keep the unit’s orientation aligned with the incoming waves, wind and current. The rotation of the unit around the turret may either be ‘passive’ or ‘active’. If the unit is allowed to freely rotate about the turret it is termed a ‘passive’ system. An ‘active’ system uses assistance from thrusters to control the unit heading.

The location of the turret in the unit, bearing design, loading condition, environmental conditions and the hull and topside design will govern the weathervaning characteristics. The chosen arrangement depends on a number of factors including: operating environment and shuttle tanker operations. However completely passive systems may utilize thrusters to minimize fishtailing effects and control heading during offloading to a shuttle tanker.

The turret is equipped with risers and umbilicals for oil or gas flow to a swivel stack on the turret allowing the oil/gas to be routed to the stationary piping system on the unit.

8.1.3  Turret loads to be considered
Several load types are to be considered for proper hull / turret interface fatigue design consideration:

— mean loads due to riser and mooring pretension and mean environmental actions
— wave-frequency loads arising from first order wave loads (both wind-waves and swell)
— low-frequency loads arising from second order wave loads and wind gust loads, with periods of the order of a few minutes.
8.1.4 Turret induced hull structure loads
Generally the loads from the mooring and riser system have negligible influence on the hull girder forces and moments. However, the hull / turret interface structure is required to support the reaction forces and moments from the turret itself. Different design philosophies for supporting the turret loads exist. Irrespective, the additional loads to the hull, via the turret, shall be addressed in the interface design. An example of the area of immediate influence for an internal turret is schematically represented in Figure 8-1. It is important to note that the extent may vary, depending on the connection and supporting structure arrangement.

![Figure 8-1 Example region of turret load influence](image)

8.2 Overview of turret designs
Turret design arrangements may be divided into two types: external and internal. Internal refers to the turret being contained within the watertight envelope of the hull whereas an external turret is located outside of the hull envelope. Each one of these is further described in the following sections.

8.2.1 External turrets
External turrets are attached to the hull at the bow or stern. Such turrets are often used for conversions in either benign waters or tropic storm locations where disconnection is necessary. The interface with the hull is generally localized. A representation of an external turret configuration is presented in Figure 8-2. Often the number of risers for an external turret is less than 20.

![Figure 8-2 Example external turret arrangement](image)

8.2.2 Internal turrets
Internal turrets are generally categorized into permanent or disconnectable types. Internal permanent turrets are designed to be an integrated part of the hull. These types are generally used in harsh
environments and / or newbuildings. An internal turret can generally support a larger number of risers compared to an external turret.

The hull connection design for an internal permanent turret may differ significantly depending on the designer’s connection philosophy and the location of the turret within the hull envelope. Some designs use upper supports to resist both the vertical and horizontal forces from the mooring and riser system. In additional a lower bearing supporting structure may also be provided. In this situation the upper support structure bearings resists the vertical and part of the horizontal mooring and riser forces and the lower supporting structure bearings resist the remaining horizontal forces. The moment generated by the resultant force from the mooring and riser system will be, in such cases, small as it is taken both by the lower and upper bearings. Two examples for a turret design configurations with both upper and lower support are shown in Figure 8-3 and Figure 8-4.

As an alternative some designers dispense with the upper horizontal bearing, allowing the lower bearing to resist the full horizontal loads. As a consequence the upper support structure only takes the vertical loads.

![Figure 8-3 Example internal permanent turret with upper and lower support](image)

The design of the supporting structure for the bearings may also vary between turret configurations. Some designs utilize a number of radial brackets above the main deck to transfer the vertical forces directly to the moonpool cylinder, transverse web frames and vertical shear panels. Other turret designs use a torsion box located at the upper support to achieve a uniform load distribution from the turret through the bearings to the interface structure, as shown in principle in Figure 8-4.

![Figure 8-4 Example internal permanent turret with torsion box](image)

Internal disconnectable turrets are connected to the hull such that the complete turret, with the associated mooring lines and risers, may be released in a simple manner. The turret will then submerge to a water depth at which a mass/buoyancy equilibrium is achieved; allowing the unit to sail away. The system is commonly used where quick release is needed in case of an emergency, such as possible collision with icebergs or extreme weather conditions. The concept is also used for storage units and/or shuttle tankers.
The turret can be re-connected to the unit by hoisting the turret up into a recess located in the bottom of the hull structure. An example of the turret design principle is shown in Figure 8-5.

Similar to the arrangement described in Figure 8-3 the horizontal load components are resisted by the lower bearing and the small vertical forces are taken by the upper bearing.

Figure 8-5  Example internal disconnectable turret in the installed position

8.3 Impact of turret location on hull fatigue design

The location of the turret will influence the weather vaning characteristics and, therefore, the structural modelling requirements for the unit, see [6.1.3.1].

8.3.1 Modelling requirements for turrets located near the bow or stern

In principle, turrets located near the bow or stern provide a passive system that is sensitive to external forces from the wind, current and waves. The system will ensure that the unit will be in moment equilibrium about a vertical axis through the turret, without the use of thrusters. In sea-states where wave, wind and current act in the same direction the unit will mainly orient itself into the head sea direction. Since the turret is frequently located towards one end and on the centreline of the unit pitch and heave motion induced loads typically dominate the design.

In a collinear environment the roll and transverse accelerations are normally small. However these motion induced loads may be a design concern in a non-collinear environment or during transit. The inertia forces from roll and pitch can be significant when the turret is arranged with a large swivel stack above the main deck.

Units with turrets located close to the bow or stern may be subjected to fish tailing as these arrangements are sensitive to rapid changes in the environmental conditions, e.g. current, wind, etc. This effect can be counteracted by thrusters located at the opposite end to the turret. This applies to both internal disconnectable and internal permanent turret types.

The mass of the turret, swivel stack and mooring lines will influence the wave load analysis and should, at least, be included as an additional mass, as described in [6.2.4]. The interface structure will experience the reaction forces from the mooring and riser system. The effects from global hull girder bending are normally of less importance, but should be included in an integrated hull/turret analysis.

8.3.2 Modelling requirements for turrets located near amidships

Turrets located closer to amidships can also allow the unit to act as a passive system if the bearing system has low friction properties. In small sea states the moment induced by external loads may not be sufficient to overcome the internal frictional resistance between the turret and the hull or the rotational stiffness of the mooring / riser system. In this situation the unit may not fully orient into the weather and it may be necessary to manually rotate the turret or use thrusters to obtain the desired direction.

In active systems the thrusters are necessary to maintain directionality. Thrusters are sometimes installed in passive systems to improve heading control in all sea states.
Some internal permanent turrets are located closer to amidships. The required moonpool opening can be large and represents a significant stress concentration factor for longitudinal bending stresses as well as large dynamic stresses to the turret itself. Such stress concentrations in the hull due to global geometry changes will be included in the FE models (part-ship or global). In such locations the wave bending moment may be at its maximum value and it is, therefore, necessary to determine the longitudinal stress distribution by means of a finite element model. The effect on the turret supporting structure from moonpool ovalisation (see Figure 8-6) due to hull bending shall be determined. The stiffness and the stiffness distribution of the turret, bearings and hull supporting structure will govern the stress distribution at the interface structure due to mooring and riser loads. Both static and dynamic loads will affect the ovalisation of the moonpool and thus affect the clearance in the lower bearing, which again will affect the dynamic load distribution on the lower bearing.

![Figure 8-6 Plan view of deformed turret moonpool due to a vertical bending moment](image)

### 8.4 Managing the hull/turret interface analysis

Turrets are connected to the hull by different means depending on the turret design. Some typical turret design configurations are presented in [8.2]. Common to all designs is that the supporting structure shall have sufficient capacity to take the vertical and horizontal reaction forces imposed by the mooring and riser forces. Also, the hull deformations, due to global hull girder bending response and local structural response, will impact on the design of the turret. The importance of hull deformation for the turret design will differ between the turret designs, but still needs to be considered. Likewise, the presence of the turret may affect both the global and local hull deformation.

With the interaction of both structures it is, therefore, necessary that compatibility of loads and displacements is achieved throughout the fatigue design process. Often there are two main parties involved in this process: the hull designer and the turret designer. An example of the information required to complete the hull/turret interface fatigue design is presented in Figure 8-7. Typically, for new-buildings, the hull/turret interface responsible is the hull designer. The figure demonstrates the importance for establishing close working relationships between the respective parties and familiarity with the counterparts requirements so as to avoid any misunderstanding.

The Prime contractor has the overall responsibility for the unit design. To ensure that the hull / turret interface is properly considered it is necessary to prepare and follow up a thorough analysis management plan. To ensure correct execution, a project-specific procedure describing the analysis approach, flow of information between relevant parties, holding points and milestones shall be developed. This plan should be discussed and agreed between all involved parties.

The specific requirements for attaining compatibility may differ for the various turret types and design approaches, as discussed further in 8.2 and 8.6, respectively.
Figure 8-7  Example hull / turret information flow

As shown in Figure 8-7 the turret designer shall be given hull scantlings and hull dynamic response early in the design cycle to allow the first pass of the mooring, riser and turret analysis to be completed. Next the hull designer shall receive, from the turret designer, scantlings and arrangement of the turret, bearing system and mooring / riser properties and design loads.

8.5 Fatigue analysis methodology for the hull/turret interface

The fatigue capacity for the hull/ turret interface structure may be determined using two methods:

— simplified fatigue using long term distribution of the mooring and riser loads, or
— combined analysis using the mooring and riser RAOs.

An example Gantt chart of the general flow and sequence of information for the hull / turret interface analysis is shown in Figure 8-7. Specific examples flow charts, related to the analysis method, are provided in [8.6]. As shown in Figure 8-7, both the hull and turret designer independently complete the unit’s hydrodynamic analysis.

Generally the simplified method shall be used during the fatigue design phase with the combined approach used in the fatigue verification phase.

If the turret structure is modelled separately to the unit hull structure then several iterations may be necessary to ensure that convergence of deflections between the interface and the turret is achieved.

8.6 Hull/turret interface calculation methods

8.6.1 Introduction

Acceptable methods for fatigue calculations of the turret/hull interface structure are:

— simplified fatigue using long term distribution of the mooring and riser loads
— combined analysis using the mooring and riser RAOs.
These approaches are presented in Figure 8-8 and Figure 8-9. For the purpose of illustration the design flow charts assume that the interface structure is the responsibility of the hull designer.

The turret designer shall incorporate the unit’s hydrodynamic analysis results in the mooring and riser analysis to establish the associated mooring and riser loads and responses at the turret.

8.6.2 Simplified fatigue using long term distribution of mooring/riser dynamic range

Normally this approach will be used during the Fatigue Design Phase (Section 2). An example flow chart for this method is presented in Figure 8-8. The use of this method is to be limited to the fatigue analysis of the hull / turret interface structure only.

The approach requires the mooring designer to develop a long term distribution of the mooring and riser loads at the turret. The long-term distribution of mooring or riser loads are derived from the short-term response in a similar way as applied to the hull structural response (see [6.10]). The extreme forces and moments are then applied to the structural FE model of the hull/turret interface at different headings, as described in [3.2.3] and Figure 3-5. Additional attention may be required concerning the response period due to combined wave-frequency and low-frequency response, and the heading angle response of a turret-moored system.

The hot spot stresses for the selected details are then used to produce a long term distribution of dynamic stress using the Weibull parameter approach, as described in [6.10]. The Weibull parameter should be estimated for all important load effects, where the dominating load effect is used to describe the long-term distribution of stresses in the hull / turret interface detail. The stress obtained from the FE analysis is to be considered as the stress amplitude, i.e. the equivalent loading may occur on the opposite side on the hull/ turret interface.

It is generally the responsibility of the turret designer to produce the extreme responses, the associated Weibull parameters and the response correlation to the hull designer. The hull designer will then establish the long term stress distribution and determine the fatigue capacity using the approach provided in DNVGL-RP-C203.

Depending on the location of the turret there may be an interaction between the hull induced loads and the turret loads (refer [8.3]) for the hull/turret interface structure. The hull load induced fatigue damages may be estimated separately using the methods described in [2.5]. If the hull load induced fatigue damages are less than 10% of the estimate turret induced fatigue damage then the interaction may be ignored.

In the event that the damage proportions are similar in magnitude then it is necessary to consider a combination of responses to estimate the fatigue damage. Although conservative it is acceptable to sum the stresses at a particular hot spot location for both the hull and turret induced loads and create a long term stress distribution considering the larger Weibull parameter.

If acceptable fatigue damage estimates cannot be achieved then the hull and turret responses may be combined using the methods described in [6.11].

8.6.3 Combined analysis using mooring and riser response amplitude operators

The objective for this approach is to use the full stochastic fatigue procedure. This is achieved by obtaining the response amplitude operators (RAOs) for both the moorings and the risers and applying these to the interface structural analysis, coupled with the hull RAOs (external and internal pressure, global bending, accelerations, etc.).

It is assumed that a coupling between the mooring / riser RAOs and the hull responses can be established so that the correct phase relationship from all responses is maintained. This will allow the fatigue capacity to be established using the full stochastic fatigue method.

Both the wave frequency and the low frequency responses are considered separately and combined using the methods described in [6.9.4].

Reference is also made to [2.4.2] where the full stochastic fatigue analysis approach using part-ship model is described.
Wave-frequency response

To establish the wave-frequency responses the following steps are necessary:

The hydrodynamic loads, excluding the mooring and riser loads, are transferred from the hydrodynamic analysis to the structural model (global or partial), as described in [6.2.5]. The hydrodynamic analysis is performed without stiffness from the riser/mooring system. Generally this is the responsibility of the hull contractor.

The linearised riser/mooring loads from the mooring analysis are transferred to the structural model. The linearization of the mooring is discussed in [8.7.1]. Generally this is the responsibility of the turret designer.

Loads/stresses from both load types are combined either before or after the structural analysis, as described in [6.8]

Low-frequency response

Refer to [6.9.3].

Combining responses

The wave frequency and low frequency responses shall be combined as described in [6.9.4].

Fatigue calculation

The full stochastic fatigue analysis is completed with the calculated stresses using the methods described in [6.9].

Figure 8-8 Combined analysis based on RAOs for each riser and mooring line

8.7 Mooring and riser systems modelling

8.7.1 Modelling for response analysis of mooring and riser system

The detailed responses of mooring lines and risers are usually analysed separately from the hull. The following brief introduction to mooring line and riser dynamics provides background for modelling of mooring and riser induced stresses in the structural details around the hull connection points.

Risers and mooring lines are long, slender elements. Axial stiffness and forces due to tension, weight, buoyancy, inertia and drag forces are significant for both types of element. Unlike mooring lines, risers usually have significant bending stiffness. Risers are typically larger in diameter than mooring lines, such that riser excitation forces due to the incoming waves shall be taken into account. Excitation of the mooring lines is dominated by the motions imposed at the upper end (i.e. by the attached unit).
The forces and moments imposed on the unit by risers and mooring lines may be described by the following categories:

— mean loads due to pretension and mean environmental actions
— wave-frequency loads arising from first order wave loads (both wind-waves and swell)
— low-frequency loads arising from second order wave loads and wind gust loads, with periods of the order of a few minutes.

The riser and mooring systems should be subject to separate analyses for each direction and each short-term environmental condition considered in the hull fatigue analysis. The following information is required for the analyses:

— tensions for each line (computed for the point of first interaction with the hull)
— riser bending moments
— mean angular orientation of the mooring lines and riser relative to the hull.

In each case, the predicted tensions and moments should be provided in the following form, indicating the mean position and the angular orientation of the hull in each case:

— mean value
— standard deviation of wave-frequency component
— standard deviation of low-frequency component
— up-crossing period of the wave-frequency component through the mean level
— up-crossing period of the low-frequency component through the mean level
— mean angular orientation relative to axes fixed in the hull.

**Wave-frequency mooring and riser response**

The wave frequency mooring and riser tensions are small in comparison with the first order restoring loads on the hull and may be ignored in the determination of the hull motion response.

It is possible to partially integrate the wave-frequency response of the mooring system into a hull structural analysis, provided that the restoring characteristics of the individual lines are linearised. Such linearization is, to some extent, dependent on the mean position of the system and the direction of the environmental actions, i.e. selection of a linearization point is subject to compromise.

**Low-frequency mooring and riser response**

The low-frequency motions of a unit are essentially a form of resonant response in the horizontal plane. The mooring lines (and risers) provide the restoring forces necessary to create a resonant system. The low-frequency exciting forces are relatively small compared to the first-order forces, and can be neglected unless the resonant system is sensitive to these exciting frequencies.

**Mean mooring and riser response**

The mean mooring and riser loads control the mean position of the unit in a short-term environmental state. The mean tension will vary between the mooring lines, dependent on the direction of the environmental actions. Although these tensions can be large, they are usually negligible compared to the weight and buoyancy forces on the hull, i.e. they do not normally impose any significant change in draft, trim or heel and are static in nature. Consequently, as specified in 8.1.3, the moorings and risers do not need to be included for the hull motion response analysis in the wave-frequency domain.

8.7.2 Spread-moored FPSO mooring loading issues

An example of a spread-moored FPSO arrangement is shown in Figure 8-10.

The relative direction of environmental forces acting upon an FPSO is dependent on:

— the arrangement of the spread mooring
— the probability of directionality of wind, waves and current, and the combined probability that these act within the same sector.
For fatigue analysis of a spread moored FPSO hull, the unit may be assumed to be floating at its moored position, which implies that the unit’s change in position and heading due to environmental forces is ignored. However, local fatigue design of mooring line anchor regions in the FPSO should be based on fatigue analysis where mooring line forces include the effect of relative displacement of the FPSO due to current and drift.

### Figure 8-9 Spread mooring design

#### 8.7.3 Turret-moored FPSO mooring loading issues

The relative direction of the environment to a turret-moored FPSO is dependent on two parameters:

- the probability of directionality of wind, waves and current and the combined probability that these act within the same sector
- the operation of the FPSO including fixation of turret and use of thrusters.

The position of the FPSO relative to waves and swell should be based on an assessment of the environmental parameters and the planned operation of the FPSO, as discussed further in [6.1].
This RP specifies two methods for considering the FPSO heading for a turret-mooring design:

- rule-of-thumb approach
- direct calculations.

Details of both methods is provided in [6.1].

An example of an internal turret-moored design is shown in Figure 8-12.

Hull / turret interaction

Low-frequency tensions in adjacent lines tend to be in phase, whereas tensions in lines in opposite directions tend to be out of phase. These phase differences complicate the combination of load effects from a large number of lines and/or risers. Combined load effects are necessary in order to capture the total effect of the mooring system on the turret-hull interaction.

Two approaches for modelling the combined mooring system loads may be considered:

a) total mooring system forces are obtained from the mooring system analysis, and the resulting forces may be represented at the centre of the mooring / riser connection point, as shown in Figure 8-11. Mooring system response analyses usually provide only the horizontal forces. These forces should be conservatively combined with the vertical components of the line tensions

b) by making a suitable assumption concerning dependencies between tensions in individual lines, short-term, low-frequency tension statistics for individual mooring lines can be combined to provide overall low-frequency turret loads. The low-frequency tension is a function of the horizontal platform displacement relative to the anchor ends of the lines, such that adjacent lines are strongly correlated, opposed lines are negatively correlated, and orthogonal lines tend to be independent. This form of dependency can be modelled using the cosine function. If the angles between the horizontal projections of the \( n \) mooring lines and the longitudinal axis of the FPSO are denoted by \( \alpha_i \), \( i = 1, 2, \ldots, n \), then the correlation coefficient between the low-frequency tensions in lines \( i \) and \( j \) is approximately given by the cosine of the angle between the lines; i.e. \( \rho_{ij} = \cos (\alpha_i - \alpha_j) \). Further, if the angles between the lines and the vertical, at the point of contact with the turret, are denoted by \( \gamma_i \), \( i = 1, 2, \ldots, n \) (\( 0 < \gamma < 90^\circ \)), then the sinus of these angles can be used to obtain the horizontal force components of the tensions. Thus, a tension standard deviation \( \sigma_{T_i} \) in one line induces a longitudinal force in the FPSO with standard deviation \( \sigma_{X_i} = \sigma_{T_i} \cdot \sin \gamma_i \cdot \cos \alpha_i \). The standard deviation of the combined longitudinal force on the turret, due to low-frequency tension in the lines can then be computed as

\[
\sigma_x = \left( \sum \sum \sigma_{T_i} \cdot \sin \gamma_i \cdot \cos \alpha_i \right)
\]

This expression assumes that the time-variation of the mooring line angles can be neglected.

Note that this correlation model is not applicable to the wave-frequency tension, as the tension component is also dependent on the vertical motion of the turret.
8.7.4 Risers system

The type of connection selected for use between the risers and the unit depends on the mooring system.

For spread mooring systems, the risers are typically fixed to a hang-off platform at the main deck level, see Figure 8-12 (b). The riser platform is exposed to the resulting force from the drag and inertia forces on the risers. The resulting total forces from the risers will be dimensioning for the global riser platform, and the forces and moments from each riser will govern the local supporting structure. Typically, the riser loads will have negligible effect on the global response of the unit.

For turret mooring systems, the risers are connected to the lower part of the turret. The local supporting structure for each riser should be considered in a detailed finite element analysis, as shown in Figure 8-12.
8.7.5 Coupled mooring analysis

Programs are available for coupled analysis of a floating unit together with the mooring lines and risers, as described by Ormberg et al (1998). Normal diffraction analysis methods are used to provide the hydrodynamic forces on the rigid hull, while finite element models are used to describe the risers and moorings. The equations of motion are solved for the coupled system in the time domain.

At present, coupled analysis is primarily used to improve the accuracy of the ship motions, and provide a better basis for structural analysis of the risers. The use of coupled analysis for conducting fatigue calculations of the hull is normally not required.

Figure 8-12 (a) Local model of turret riser/fairlead supporting structure, and (b) typical spread-mooring riser hang-off platform
SECTION 9 DOCUMENTATION AND VERIFICATION OF ANALYSIS

9.1 Documentation of analysis

The analysis shall be verified in order to ensure accuracy of the results. Verification shall be documented and enclosed with the analysis report.

The documentation shall be adequate to enable third parties (e.g. owner and the class society) to follow each step of the calculations. For this purpose, the following should, as a minimum, be documented or referenced:

- basic input (drawings, loading manual, weather conditions, etc.)
- assumptions and simplifications made in modelling/analysis
- models
- loads and load transfer
- analysis
- results (including quality control)
- discussion, and
- conclusion.

Checklists for quality assurance shall also be developed before the analysis work commences. It is suggested that project-specific checklists be defined before the start of the project and be included in the project quality plan. These checklists will depend on the shipyard’s engineering practices and associated software.

9.2 Documentation of hydrodynamic properties

Due to many involved parties it is important that the hydrodynamic properties used in the analysis are properly documented. Typical properties to be documented are listed below and should be based on the selected probability level (10^-4 is recommended) for long-term analysis:

- viscous damping level
- mass properties (radii of gyration)
- motion reference point
- scatter diagram/sea states, wave spectrum, wave spreading
- sectional loads with corresponding Weibull shape parameter and zero-crossing period
- accelerations with corresponding Weibull shape parameter and zero-crossing period
- sea pressure loads with corresponding Weibull shape parameter and zero-crossing period.

It is recommended that the documentation of the hydrodynamic parameters is initiated in the FEED phase in order to have comparable numbers throughout the project.

9.3 Verification of structural models

For proper documentation of the model, guidance is given below.

Assumptions and simplifications are required for most structural models and should be listed such that their influence on the results can be evaluated. Deviations in the model compared with the actual geometry according to drawings shall be documented.

The set of drawings on which the model is based should be referenced (drawing numbers and revisions). The modelled geometry shall be documented preferably as an extract directly from the generated model. The following input shall be reflected:

- plate thickness
- beam section properties
- material parameters (especially when several materials are used)
- boundary conditions
— out of plane elements (4-node elements, see [4.2.5])
— mass distribution/balance.

9.4 Verification of loads

9.4.1 Introduction

Inaccuracy in the load transfer from the hydrodynamic analysis to the structural model is among the main error sources for this type of analysis. The load transfer can be checked on basis of the structural response (see [9.4.3]) or on basis on the load transfer itself.

It is possible to ensure the correct transfer in loads by integrating the stress in the structural model and the resulting moments and shear forces should be compared with the results from the hydrodynamic analysis.

![Comparison between section loads from hydrodynamic calculation and the applied loads on the finite element model](image)

**Figure 9-1 Example of QA for sectional loads between hydrodynamic and FE calculations**

As demonstrated in Figure 9-1, 10 sections are usually sufficient in order to establish a proper description of the bending moment and shear force distribution along the hull. The first and last sections shall correspond with the ends of the finite element model.

9.4.2 Verification of long term distribution of mooring and riser loads

Due to uncertainty with establishing the long term distribution for the mooring and riser loads from the time domain simulation, independent analysis and supporting documentation is required.

9.4.3 Quality assurance using the load component analysis approach

As a part of the quality assurance using this approach the following issues need to be clearly defined and understood by all involved parties:

— hydrodynamic model roll tuned for a probability level (10⁻⁴ or less)
— integration direction for all sections defined and communicated between the structural and hydrodynamic analyst (see Figure 9-2)
— position, number and neutral axis of sections agreed with structural analyst
— panels for external pressure defined at correct positions
— element normal on panel model.
9.4.4 Verification of scaling factors for load component fatigue analysis

Based on experience, there are several possibilities for error when using the load component based fatigue approach. It is therefore important to ensure that thorough checks are completed and documented for, at least, the scale factors and load sign convention.

In order to establish the correct scaling factors to be used for converting the unit load condition results into the correct magnitude for post-processing, the following factors shall be considered:

— units of the stress analysis
— units of the hydrodynamic analysis results
— units necessary for entering the S-N curve calculations
— integration direction and resulting direction of moment from the hydrodynamic analysis.

When defining the correct sign for the scaling factors, it is imperative that the integration direction and resulting sign of the force and moment are discussed and agreed between the hydrodynamics specialist and the fatigue analyst. If incorrect sign conventions are used, then the phase relationship between different load components will not be maintained.

9.5 Verification of response

The response should be verified at several levels to ensure that the analysis is correct. The following aspects should be verified as applicable for each load considered:

— global displacement patterns/magnitude
— local displacement patterns/magnitude
— global sectional forces
— stress levels and distribution
— sub model boundary displacements/forces
— reaction forces and moments.

Global displacement patterns/magnitude

In order to identify any serious errors in the modelling or load transfer, the global action of the unit should be verified against expected behaviour/magnitude.
Local displacement patterns

Discontinuities in the model, such as missing connections of nodes, incorrect boundary conditions, errors in Young’s modulus etc., should be investigated on basis of the local displacement patterns/magnitude.

Global sectional forces

Global bending moments and shear force distributions for still water loads and hydrodynamic loads should be according to the loading manual and hydrodynamic load analysis respectively. Small differences will occur and can be tolerated. Larger differences (>5% in wave bending moment) can be tolerated provided that the source is known and compensated for in the results. Different shapes of section force diagrams between hydrodynamic load analysis and structural analysis indicate erroneous load transfer or mass distribution and hence should not normally be allowed.

Stress levels and distribution

The stress pattern should be according to global sectional forces and sectional properties of the unit, taking into account shear lag effects. More local stress patterns should be checked against probable physical distribution according to location of detail. Peak stress areas in particular should be checked for discontinuities, bad element shapes or unintended fixations (4-node shell elements where one node is out of plane with the other three nodes).

Where possible, the stress results should be checked against simple beam theory checks based on a dominant load condition, e.g. deck stress due to wave bending moment (head sea) or longitudinal stiffener stresses due to lateral pressure (beam sea).

Sub-model boundary displacements/forces

The displacement pattern and stress distribution of a sub-model should be carefully evaluated in order to verify that the forced displacements/forces are correctly transferred to the boundaries of the sub-model. Peak stresses at the boundaries of the model indicate problems with the transferred forces/displacements.

Reacting forces and moments

Reacting forces and moments should be close to zero for a direct structural analysis. Large forces and moments are normally caused by errors in the load transfer. The magnitude of the forces and moments should be compared to the global excitation forces on the unit for each load case.

9.6 Verification for the hull/turret interface

With two parties involved with the design of the hull / turret interface structure it is important to ensure that same information is used and the same results are produced. Primarily there are two main issues to be checked:

- hydrodynamic results, and
- deflections.

Since both the turret designer and the hull designer shall perform a hydrodynamic analysis for the hull this information should be used for verification. As a minimum the following items are to be checked:

- accelerations at midships and turret location
- sectional forces and moments at midships and turret location
- wave drift forces
- roll damping
- mass / buoyancy, and
- centre of gravity.

Upon completion of the FE analysis for the hull model deflections at the bearing level shall be compared. Any significant deviations are to be corrected before the analysis is finalized.
SECTION 10  FABRICATION AND FATIGUE

10.1 Introduction
This section describes the link between fatigue results and good fabrication practices to ensure that fabrication of fatigue critical details is satisfactory. The actual fatigue life performance of fabricated details in service is dependent on fabrication quality down to the shape of the weld surface. The SN curves are mainly based on fatigue testing of welds made in well-controlled environments and favourable welding positions. During the actual fabrication at a yard, the fatigue critical welds are often performed just as non-critical welds. However, fatigue critical welds need increased focus to ensure that the fabricated weld adheres to the assumptions made for calculated fatigue life of the connection.

10.2 Inspection of fatigue critical details
Information about critical details should be included in separate drawing or in the drawings made for planning of the NDE of welds, whatever is found appropriate. The drawings will give information from the fatigue designer to the yard workers and QA department dealing with fabrication. Details with fatigue life less than two times the design fatigue life should be marked in the drawing. The yard should plan for inspections of these details before they are sand blasted and painted.

10.3 Bracket toe welding
By experience, the most fatigue damages have shown to be starting at bracket toes or similar geometries. Increased quality during fabrication and welding of this local detail will reduce the risk of fatigue cracks significantly.

The weld around a bracket toe shall be continuous welding, without start stop position in a distance 2 times the thickness of the thickness of the bracket. The weld geometry shall be without undercuts and a geometry that merges well into the base material, see Figure 10-1.

![Preferred weld shape at bracket toes and grinding direction](Image)

Figure 10-1  Preferred weld shape at bracket toes and grinding direction

The yards should remove all undercuts and surface defects up to 2 mm depth by grinding instead of by welding. "Cosmetic" welding for restoring a surface may give very high hardness, which can results in micro cracks in weld toe. Grinding direction should be parallel to the direction of the dynamic stress. Hence, the grinding should be with a pencil grinder rather than a disc grinder.

10.4 Temporary and permanent attachments and scallops

10.4.1 Attachments with welding
For all areas that are fatigue critical (calculated fatigue life less than two times design life), no attachments are allowed. Instructions should be made in documents or drawings where no attachments are allowed.
10.4.2 Temporary attachments
Temporary attachment is however allowed where the C curve gives critical fatigue life given that the attachment is completely removed without undercut and the remaining weld ground flush with grinding direction parallel to the direction of the dynamic stress. Information about grinding direction is also needed on the drawings. NDE by use of magnetic particle inspection (MPI) or liquid penetrant testing should be used to confirm that there are no cracks in the surface. However, to avoid risk of fatigue cracks from the numerous temporary attachment which can be difficult to manage to follow up, it is recommended to restrict the temporary attachments. This can be done by expanding the “no attachment” area to include where the fatigue life is regarded as critical based on the S-N curve C. Temporary attachment can only be allowed based on a case by case acceptance.

10.4.3 Remaining attachments
Some yards have a practice to partly remove lifting brackets by cutting the bracket about 20 mm from the plate and leave it as it is. If this practice is allowed, the attachment should be considered permanent and analysed accordingly.

10.4.4 Scallops and drain holes
Scallops used for block erection joints have a high SCF ($k_g$ up to 3) if the geometry is not well controlled. In order to avoid risk of fatigue from scallops, a small scallop in the longitudinal material should be used and the scallop in should be closed by welding.

10.4.5 Permanent attachments to the hull and deck
Fatigue calculations is normally focused on connections of side longitudinals, attachment to the underwater hull such as openings, anodes and bilge keel, support structures in the upper part of side shell such as mooring wing/stopper support and riser connections. On main deck, fatigue calculations are performed for doubling plates, openings, topside supports etc. For FPSO’s where the numbers of supports and attachments are large, it is difficult to avoid that doubling plates are placed close to an opening or another support. The fatigue analysis should ideally be performed for a combination of SCF’s due to attachments and openings so that the attachments can be located where they are originally planned. The challenge is to keep control of all the supports and openings on the main deck as the fatigue calculations are performed prior to the detail decisions of the locations of all the holes and doubling plates.

The combined SCF can be determined by local FEM model or by multiplication of SCF from pre-defined SCF’s for holes as given in Peterson’s Stress Concentration Factors or in attached Figure 10-2.
Figure 10-2  Variation SCF due to hole in plate

It should be remembered to include standard ship details in the fatigue assessment. Deck supports that are often forgotten are:

— bollards for key side mooring
— chocks
— winches for key side mooring
— ship side railing stanchions
— attachment of accommodation ladder
— attachments to deck strake
— deck houses and misc.

Standard bollards and chocks are often a closed box structure which also can be located in the midship area. The fatigue analysis should take into consideration that these details often have one sided welding and none access for inspection inside.

In practice, it is difficult to manage where all cut outs, supports and doubling plates are located. The fatigue analysis should determine how close an opening could be to a topside support and how close a doubling plate could be to an opening. This information could be included in a separate drawing for main deck for attachments, supports, openings, doubling plates etc. As an alternative to calculate every attachment, a minimum distance should be determined, say, 0.5 meter free distance between any openings, supports and attachments. This information is easy to implement in engineering and follow up at the yard. Deviations to this rule have to be approved by the hull fatigue designers case by case.

10.5 Non-destructive examination to reduce risk of fatigue

10.5.1 Introduction

Non-destructive examination (NDE) is normally used to detect systematic errors with the welding, delaminating of plates and to some extent, detecting initial cracks at fatigues critical areas. NDE is a common term for:

— visual inspection
— ultrasonic testing of internal defects of welds
— X-ray testing of internal defects of welds
— magnetic particle inspection for surface cracks
— liquid penetrant inspection for surface cracks.

10.5.2 Use of non-destructive examination to reduce risk of fatigue

For various parts of the structure, the amount of NDE is decided in accordance with Rules of the classification society which have limited guidance as to where the NDE shall be performed. Welds in main deck and bottom plate within $0.4 \times \text{Lpp}$ of midship are required to have approximately 20% NDE of the welds. Welds of longitudinal stiffeners and brackets have normally requirement to approximately 5% of the welds. Since these latter welds are fillets, only magnetic particle inspection (MPI) is applied.

The fabrication specification should specify the extent of MPI used on bracket toes and similar details. It is also required to have mandatory testing of all bracket toes and heels for details in fatigue critical areas (fatigue life less than 2 times the design life time). Fatigue critical areas will normally include bracket toes and heels for side longitudinals in the midship area in the splash zone.

10.6 Information from fabrication to in-service inspection plans

In theory, a newbuilding is free from any defects at delivery. In practice, any substandard details that are not improved or weld repairs in fatigue critical areas should be recorded in separate as-built drawing for fatigue details. If the details after re-analysis have fatigue lives less than the required fatigue life (safety factor multiplied by design service life), these details shall be subject to more frequent inspections compared to the standard 5 year cycle that is basis for classed units.
SECTION 11 REFERENCES

<table>
<thead>
<tr>
<th>Reference</th>
</tr>
</thead>
<tbody>
<tr>
<td>Dahle T. <em>Spectrum Fatigue Life of Welded Specimens in Relation to the Linear Damage Rule</em>. Proceedings of a Conference held 26-27 August 1993 at the University of Denmark.</td>
</tr>
<tr>
<td>DNV Recommended Practice RP C203. <em>Fatigue Strength Analysis</em>. August 2005. (this is a revised version of CN 30.2 <em>Fatigue Strength Analysis for Mobile Offshore Units</em>, August 1984)</td>
</tr>
<tr>
<td>Gran S. <em>Full Scale Measurements of Wave Excited Vibrations of a 255000 DWT Tanker in Ballast Condition (Progress Report No.5)</em> Det Norske Veritas, Report No. 74-41-5.</td>
</tr>
</tbody>
</table>
### Table 11-1

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<thead>
<tr>
<th>Reference</th>
</tr>
</thead>
</table>
APPENDIX A LOAD APPLICATION FOR STRESS COMPONENT ANALYSIS METHODS

A.1 Calculation of load/stress ratios

A.1.1 Introduction

In order to perform a stress component based stochastic fatigue analysis, the load/stress ratio between the hydrodynamic load effects and the stress calculated in the structural analysis shall be established.

This ratio shall, for the final results, be based on the structural FE model. Initial calculations (and verification analysis) may be based on simplified formulas.

Figure A-1 Definition of stress components

The following stress components should normally be considered for the fatigue control (the different components may be omitted if documented to be insignificant):
where:

\[ \sigma = (\sigma_g) + (\sigma_2 + \sigma_2A + \sigma_3) \]
\[ = (\sigma_v + \sigma_h + \sigma_i + \sigma_a) + (\sigma_e + \sigma_i + \sigma_{acc}) \]

\( \sigma_g \) = total nominal stress from global actions
\( \sigma_2 \) = total nominal stress from secondary bending (double bottom bending)
\( \sigma_2A \) = total nominal stress from local bending of stiffeners (including relative deflections)
\( \sigma_3 \) = total nominal stress from local tertiary plate bending
\( \sigma_v \) = total nominal stress from vertical hull girder bending
\( \sigma_h \) = total nominal stress from horizontal hull girder bending
\( \sigma_t \) = total nominal stress from torsion hull girder bending (if applicable)
\( \sigma_a \) = total nominal stress from axial hull girder forces
\( \sigma_e \) = total nominal local stress due to external pressure
\( \sigma_i \) = total nominal local stress due to internal pressure
\( \sigma_{acc} \) = total nominal local stress due to acceleration

The nominal stress levels shall be combined with their appropriate stress concentration factors prior to any summation. Be aware that the stress concentration factors may depend on the local action (axial or bending).

Torsion and acceleration of hull steel may normally be neglected in a component based fatigue analysis.

A more thorough description of each component is given in the relevant sections below.

The application of unit loads is described in [A.2].

A.1.2 Global stress

Global stress \( (\sigma_g) \) is caused by global action of the hull girder. The global stress components for vertical and horizontal bending and axial loads may be achieved by using the methods as described in [A.2].

A.1.3 Secondary bending stress

Local secondary bending stresses \( (\sigma_2) \) are the results of bending due to lateral pressure of stiffened single skin or double hull cross-stiffened panels between transverse bulkheads, see Figure A-1. This may be relevant for bottom or deck structures, sides or longitudinal bulkheads.

Stresses from stiffener bending \( (\sigma_2A) \) are caused by pressure induced bending of the stiffeners and associated plate flanges between transverse supports (e.g. frames, bulkheads), ref. Figure A-1 and Figure A-2.
FEM analysis is the preferred way of determining secondary stresses.

Using only the results from a FE analysis with direct load transfer, it is difficult to separate stresses caused by relative deflection (tertiary stresses) from stresses caused by stiffener bending. Both of these effects give rise to local bending of the stiffener caused by internal/external pressure. However, it is possible to separate the different stresses by means of a component based method where line loads are applied at transverse bulkheads/frames such that no stiffener bending is induced. (This separation may be desirable as stresses from stiffener bending may be derived accurately from parametric equations.)

The procedure below may be used to calculate local bending stress from a model with adequate mesh density. It should be noted that frame deflections can be estimated relatively accurately using four 4-node elements or two to three 8-node elements between bulkhead and frame. Correct bending moments in the stiffeners may need more longitudinal elements. Six 4-node elements or three to four 8-node elements may be necessary. If the mesh density is too coarse to extract bending moments directly then the deflections should be applied to a finer meshed model.

The procedure below can be used to calculate secondary bending stress.

1) Apply unit line loads on the transverse frames/bulkheads to three adjacent tanks *.
2) Apply counteracting line loads in deck (above longitudinal bulkheads) such that equilibrium in loads is achieved.
3) Read the axial force in the longitudinals and convert to stress in stiffener.
4) Equivalent procedures may be used for side shells.

*The line loads should be applied at the transverse bulkheads and frames with a magnitude equivalent to the pressure distribution at 10^-4 probability level, see [4.5.4].

Pressures calculated at the mid-position of each cargo tank should generally be used.

Dynamic secondary bending stresses should be calculated for dynamic sea pressure, \( p_e \) and for internal dynamic pressure \( p_i \). The pressures to be used should normally be determined at the mid-position for each cargo hold or tank.

The local bending stress of stiffeners with effective plate flange between transverse supports (e.g. frames, bulkheads) may be approximated in lieu of FEM analysis by:

\[
\sigma_{zd} = K \frac{M}{Z_x} + K \frac{m}{I} \frac{E I}{l^2} \cdot r \delta \tag{A1}
\]

**Figure A-2 Stiffener stress definitions**
where:

\( K \) = stress concentration factor
(Note that the stress concentration factor in front of each term in eq. (A1) may be different, e.g. \( K_n \) shall to be included in the first term, but not in the second term.)

\( M \) = moment at stiffener support adjusted to the hot spot position at the stiffener (e.g. at bracket toe)

\[
M = \frac{ps^2}{12} r_p
\]

\( p \) = lateral dynamic pressure
= \( p_e \) for dynamic sea pressure
= \( p_i \) for internal dynamic pressure

\( s \) = stiffener spacing

\( l \) = effective span of longitudinal/stiffener as shown in Figure A-3

\( Z_s \) = section modulus of longitudinal/stiffener with associated effective plate flange. For definition of effective flanges, see DNVGL-OS-C101

\( I \) = moment of inertia of longitudinal/stiffener with associated effective plate flange

\( m_\delta \) = moment factor due to relative deflection between transverse supports

For designs where all the frames obtain the same deflection relative to the transverse bulkhead, e.g. where no stringers or girders supporting the frames adjacent to the bulkhead exist, \( m_\delta \) may be taken as 4.4 at the bulkhead.

At termination of stiff partial stringers or girders, \( m_\delta \) may be taken as 4.4.

When the different deflections of each frame are known from a frame and girder analysis, \( m_\delta \) should be calculated due to the actual deflections at the frames by using a beam model or a stress concentration model of the longitudinal. A beam model of a longitudinal covering \( \frac{l}{2} + \frac{l}{2} \) cargo hold length is shown in Figure A-5. A representative value of \( m_\delta \) for side and bottom can normally be calculated using one load condition according to:

\[
m_\delta = \frac{M_\delta l^2}{\delta EI}
\]

where:

\( M_\delta \) is the calculated bending moment at the bulkhead due to the prescribed deflection at the frames, and \( \delta_1, \delta_2, ... \delta_n \)

\( \delta_1 \) is the relative support deflection of the longitudinal at the nearest frame relative to the transverse bulkhead. The frame where the deflection for each longitudinal in each load condition, \( \delta_1 \) is to be taken, should be used.

\( \delta \) = deformation of the nearest frame relative to the transverse bulkhead (positive inwards - net external pressure)

\( r_\delta, r_p \) = moment interpolation factors for interpolation to hot spot position along the stiffener length, Figure A-4

\[
r_\delta = 1 - 2\left( \frac{x}{l} \right) \quad ; \quad 0 \leq x \leq l
\]

\[
r_p = \delta \left( \frac{x}{l} \right)^2 - \delta \left( \frac{x}{l} \right) + 1.0 \quad ; \quad 0 \leq x \leq l
\]

where:

\( x \) = distance to hot spot, Figure A-4.
Figure A-3  Definition of effective span lengths
In order to obtain a reliable fatigue assessment, it is important not to underestimate the relative deformation bending stresses in longitudinals between supports. The appropriate value for relative deformation, $\delta$, shall be determined in each particular case, e.g. by beam or finite element analyses.

For preliminary calculations, the following coarse approximation of relative deflection between transverse bulkheads and adjacent web frames in ship side of double hull tankers may be used:

$$
\delta = \left(1 - \left(1 - \frac{2z}{D}\right)^2\right)\delta_m
$$

(A2)

where:

$$
\delta_m = \frac{0.3S\ell^2D}{E\sqrt{i_{hh}}\sqrt{1 + N_S}}
\quad \text{for designs with side stringers}
$$
Secondary longitudinal bending stresses are generally small compared to other stress components and can be omitted if simplified analysis shows that they are insignificant. The separation of secondary bending stress from other effects is not normally necessary, see [A.1.4].

A.1.4 Stress from plate bending
Longitudinal local tertiary plate bending stresses are well known from the literature and need not be calculated from the structural FE model.
Longitudinal local tertiary plate bending stress amplitude at the weld at the plate/transverse frame/bulkhead intersection is midway between the longitudinals may be given by:

\[ \sigma_3 = 0.343 p (s / t)^2 K \]  \hspace{1cm} (A3)

where:
- \( p \) = lateral pressure
- \( p_e \) for dynamic sea pressure
- \( p_i \) for internal dynamic pressure
- \( s \) = stiffener spacing
- \( t \) = plate thickness

Similarly, the transverse stress amplitude at stiffener mid-length is

\[ \sigma_T = 0.50 p \ (s / t)^2 K \]  \hspace{1cm} (A4)

### A.1.5 Stress caused by external/internal pressure

If a stress concentration model is used, then the resulting hot spot stress can be directly applied for a fatigue analysis and it is not necessary to divide the pressure-induced stress in the specific effects above. The same also applies if other fine mesh models are used to assess the nominal stress. The mesh for both types of model shall be such that effective breadth of the plating is included in the stress results.

Stresses from stress concentration models are used directly in the fatigue calculations. Whether the stress in a stiffener is caused by stiffener bending, relative deflection or secondary bending is irrelevant and will have no effect on the calculated fatigue life. The load/stress ratio can therefore be calculated for external/internal pressure regardless of which structural effect causes the actual stress.

However, it should be noted that the use of the same pressure transfer function for all effects caused by internal/external pressures may not be correct. This is explained in [A.2.4].

### A.1.6 Hopper knuckle/transverse stresses

Transverse stresses are generated mainly by internal/external pressures distribution around the actual longitudinal section of the ship. It is therefore important to describe the pressure loads correctly. The external pressure distribution in the surface region should normally be modified to account for intermittent wet and dry surfaces, see [4.5.4]. For details where the transverse pressure distribution is the dominant stress contributor and where modified pressure distribution in the surface region is important, stresses can be found using the procedure given below.

1) Apply unit external pressure to a minimum of \( \frac{1}{2} \) tank + \( \frac{1}{2} \) tank. *
2) Apply unit internal pressure loads to filled tanks for longitudinal, transverse and vertical acceleration.
3) Apply unit acceleration for steel and point masses (if relevant).
4) Perform structural analysis with the unit load cases.
5) Establish load/stress ratio for each load.
6) Combine stresses according to [6.2].

*The pressure to be according to \( 10^{-4} \) pressure distribution, see [4.5.3].

With the exception of the hopper tank knuckle, stress effects due to exclusion of pressure variation along the hull can usually be neglected. The pressure variation along the hull will lead to smaller total external pressures and, consequently, to smaller stresses such that omission of the pressure variation leads to conservative results. This may also apply to other transverse stresses. See also [A.2.5].
Boundary conditions as shown in Table A-1 have been shown to give good agreement with results where the global model is used to represent the stiffness outside the “load model” for the hopper knuckle.

**Table A-1  Boundary conditions for transverse stresses**

<table>
<thead>
<tr>
<th>Direction</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Longitudinal translation</td>
<td>Fixed at one end</td>
</tr>
<tr>
<td>Transverse translation</td>
<td>Horizontal springs at the intersection of deck, inner bottom and outer bottom to the transverse bulkhead</td>
</tr>
<tr>
<td>Vertical translation</td>
<td>Vertical springs at the intersection of side, inner side and longitudinal bulkhead to the transverse bulkhead</td>
</tr>
<tr>
<td>Rotation about longitudinal axis</td>
<td>Free</td>
</tr>
<tr>
<td>Rotation about transverse axis</td>
<td>Symmetry conditions: fixed at the ends of the model</td>
</tr>
<tr>
<td>Rotation about vertical axis</td>
<td>Symmetry conditions: fixed at the ends of the model</td>
</tr>
</tbody>
</table>
A.2 Application of unit loads

A.2.1 Introduction

The following loads are normally applied to the model:

— uniform vertical and horizontal bending moments
— uniform axial load
— external sea pressure
— internal cargo pressure
— inertia loads due to acceleration in x-, y- and z-direction.

All loads are applied as unit loads.

The bending moments and the axial load represent all the global load effects, whereas the internal/external pressure and the inertia loads represent the local load effects and so should not induce any global effects.

The cargo tank model operates with two sets of boundary conditions: one set of boundary conditions for the global loads, and one set for the local loads. The boundary conditions applied to the model are discussed in [4.4.5].

A.2.2 Vertical and horizontal bending moments

Unit vertical and horizontal bending moments are defined uniformly over the length of the cargo tank model.

The cargo tank can be modelled as a cantilever beam; fixed for all degrees of freedom at one end, and free at the other end. The bending moments are applied at the free end as a force pair acting in the opposite directions and applied at two points positioned vertically above each other.

The free end where the moments are applied should be kept plane such that the displacements of the plane are as a rigid body.

The magnitude of the force pair is given by:

\[ F = \frac{M}{h} \]

where:

\[ F \] = magnitude of force at the two points
\[ M \] = magnitude of bending moment = 1
\[ h \] = distance between the two forces.

Shear forces are not applied to the structural model and hence shear lag effects in the hull girder are not included in the results. The shear lag effect will vary according to unit configuration and the ratio between bending moment and shear force. The shear lag effect is normally small and may be omitted for midship sections with one or more longitudinal bulkheads. For cross sections without longitudinal bulkheads, the shear lag effect will be larger and should be included. The effect on longitudinal stress will normally be less than 10% in the midship area.

Figure A-7 illustrates the application of vertical bending moment to the cargo tank model.
A.2.3 Torsion loads
Torsion (warping) loads may usually be neglected for a FPSO. In cases where torsion (warping) is of importance, a global model should be used. Direct load transfer will normally be the best option.

A.2.4 Axial loads
A unit uniform axial load is defined over the length of the cargo tank model.

The cargo tank model should be modelled as a cantilever beam; one end kept plane and rigid and one end totally fixed. The axial load should be applied such that no global moments are introduced, e.g. the moments about the neutral axis $Z_{CG}$ and $Y_{CG}$ are zero.

Alternatively, specified displacements representing a unit axial load may be given.

Figure A-8 illustrates application of axial load.

A.2.5 External pressure
The external sea pressure distribution is defined as normal pressure according to the long-term values calculated at $10^{-4}$ probability level. The distribution in the surface region is modified due to the effect of intermittent wet and dry surfaces.

Due to intermittent wet and dry surfaces, the pressure range above $T_{act} - z_{wl}$ is reduced, see Figure A-9. The dynamic external pressure amplitude (half pressure range), $p_e$ related to the draft of the load condition considered, may be taken as:

$$p_e = r_p p_d \quad (\text{kN/m}^2)$$

\hspace{0.7cm} (AS)
where

\[
p_{d} = \text{dynamic pressure amplitude below the waterline taken from hydrodynamic analysis.}
\]

\[
r_{p} = \text{reduction of pressure amplitude in the surface zone}
\]

\[
= 1.0 \quad \text{for } z_{w} < T_{\text{act}} - z_{wl}
\]

\[
= \frac{T_{\text{act}} + z_{wl} - z_{w}}{2z_{wl}} \quad \text{for } T_{\text{act}} - z_{wl} < z_{w} < T_{\text{act}} + z_{wl}
\]

\[
= 0.0 \quad \text{for } T_{\text{act}} + z_{wl} < z_{w}
\]

\[
z_{wl} = \text{distance in m measured from actual water line. (In the area of side shell above } z = T_{\text{act}} + z_{wl} \text{ it is assumed that the external sea pressure will not contribute to fatigue damage.)}
\]

\[
p_{dT} = \frac{3}{4} \frac{p_{d} g}{\rho}
\]

\[
p_{dT} = p_{d} \text{ at } z = T_{\text{act}}
\]

\[
T_{\text{act}} = \text{the draft in m of the considered load condition}
\]

\[
\rho = \text{density of sea water} = 1.025 \text{ (t / m}^2\text{)}
\]

**Figure A-9 Reduced pressure ranges in the surface region**

The pressure distribution is assumed constant in the longitudinal direction.
The assumption of constant pressure distribution in the longitudinal direction is conservative. The dynamic sea pressure depends on the wave amplitude, and the assumed pressure distribution means that the wave elevation is constant and equal to the wave amplitude over the whole length of the midship model, as illustrated in Figure A-11.

An example where the external pressure is split into three load cases (external pressure at bottom, external pressure at bilge/lower side and external pressure at side) is shown below.

The actual external pressure shall be scaled according to the location where pressures are extracted from the hydrodynamic analysis.

For the example in Figure A-12, fatigue shall be calculated for a stiffener located at position P1. The following three different effects caused by the external pressure shall be captured:

— local bending of the stiffener
— relative deflection
— secondary bending.

Pressures at position P2 may normally be used for all effects above. However, the long term pressure distribution (Weibull slope parameter) can vary quite significantly over the height of the unit side. To account for this variation, the pressure at position P1 may be used to calculate the local bending stress. The pressure at this position shall then be scaled according to the following expression:

\[ p_{P1\_one} = \frac{1}{p_{P1\_unit}} \]

where:
- \( p_{P1\_one} \) = unit pressure at point P1 (\( p_{P1\_one} = 1 \))
- \( p_{P1\_unit} \) = pressure at point P1 according to the unit load case

The pressure positions (points) most representative for relative deflection and secondary bending should be used to calculate the pressures for these effects. For load case LC1, this will be a position around position P2. Pressure transfer functions for this position should consequently be used to calculate stresses from these effects. The pressures have to be scaled according to the same procedure as for point P1.

![Figure A-12 Definition of load cases for external pressure](image)

**A.2.6 Internal cargo pressure**

The dynamic pressure from liquid cargo or ballast water should be calculated based on the combined accelerations related to a fixed coordinate system. The gravity components due to the motions of the FPSO should be included.

The dynamic internal pressure amplitude, \( p_i \) in kN/m², may be taken as the maximum pressure due to acceleration of the internal mass:

\[
 p_i = \max \left\{ p_1 = \rho a_i h_i \right. \left. \right\} \quad \text{(kN/m}^2) \quad \text{(A6)}
\]

where:
- \( p_1 \) = pressure due to vertical acceleration (largest pressure in lower tank region)
- \( p_2 \) = pressure due to transverse acceleration
- \( p_3 \) = pressure due to longitudinal acceleration
- \( \rho \) = density of sea water, 1.025 (t/m³)
- \( x_s \) = longitudinal distance from center of free surface of liquid in tank to pressure point considered (m)
The effect of ullage (void space in top of tank) will add to the pressure in one half cycle and subtract from the pressure in the other half cycle, and is therefore omitted in the above description of the half pressure range. Similarly, the effect of the tank top geometry may be omitted. For partly subdivided tanks where the fluid is prevented to flow through swash bulkheads during one half motion cycle, the pressures may be reduced accordingly.

For similar tank filling conditions on both sides of a bulkhead, e.g. for a bulkhead between two cargo tanks, the following apply:

a) The effect of vertical acceleration is cancelled and may be set to zero.
b) The pressures due to motion are added for bulkheads normal to the direction (plane) of the motions.

The net pressure range may be taken as:

\[ p_i = 2 \, p_2 \] for longitudinal bulkheads between cargo tanks and,
\[ p_i = 2 \, p_3 \] for transverse bulkheads between cargo tanks.

(Note that \( \Delta p = 2 \, p_1 \) when liquid on both sides).

As a simplification, sloshing pressures may normally be neglected in fatigue computations.

In the case of partly filled tanks the pressure range on both sides of a bulkhead may be taken as the sum of the pressure amplitudes in the two tanks. Otherwise the range may be taken as equal to the pressure amplitude.

\[ y_s \] transverse distance from center of free surface of liquid in tank to the pressure point considered (m), see Figure A-5
\[ h_s \] vertical distance from point considered to surface inside the tank (m), see Figure A-11
\[ a_v, a_t \text{ and } a_l \] accelerations in vertical-, transverse-, or longitudinal direction (m/s²)
\[ h \] = \( h_0 + 0.05 \)
\[ h_0 = 2.26 - 0.54 \log_{10}(L) \]

\[ \pi \] for longitudinal bulkheads between cargo tanks and,
\[ \pi \] for transverse bulkheads between cargo tanks.
A.2.7 Inertia loads

The inertia loads due to acceleration in x-, y- and z-direction are defined as loads due to unit accelerations in the respective directions, see fig Figure A-15 and Figure A-16.

Note that the effect on stresses due to inertia loads is normally small, but may be of significance for areas close to heavy equipment.
Figure A-15  Illustration of acceleration components

Figure A-16  Illustration of acceleration components and center of mass for double hull tankers with connected top wing and hopper/bottom ballast tanks
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