



RECOMMENDED PRACTICE
DNV-RP-C102

STRUCTURAL DESIGN OF
OFFSHORE SHIPS

FEBRUARY 2002



DET NORSKE VERITAS

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CHAPTER 1

WORLD WIDE OPERATION

CONTENTS

| | | |
|----------|--|-----------|
| 1 | INTRODUCTION | 4 |
| 2 | SYMBOLS, DEFINITIONS AND REFERENCES | 5 |
| 3 | OVERALL DESIGN PHILOSOPHY | 6 |
| 3.1 | GENERAL | 6 |
| 3.2 | WORLD-WIDE OR BENIGN WATERS | 6 |
| 4 | SELECTION OF MATERIALS | 9 |
| 4.1 | MATERIAL AND INSPECTION CATEGORIES AND WELDS | 9 |
| 5 | COMPLIANCE WITH BASIC σ1A1 REQUIREMENTS | 10 |
| 5.1 | TYPICAL DESIGN LOOP | 10 |
| 5.2 | LONGITUDINAL MATERIAL | 10 |
| 5.3 | TRANSVERSE BULKHEADS | 11 |
| 5.4 | RULE STILL WATER BENDING MOMENTS AND SHEAR FORCES | 11 |
| 5.4.1 | <i>Shear force correction factors</i> | 11 |
| 5.5 | HULL GIRDER SYSTEM | 12 |
| 6 | ULS – HULL GIRDER | 13 |
| 6.1 | DESIGN PRINCIPLES | 13 |
| 7 | FLS - HULL STRUCTURAL DETAILS | 15 |
| 7.1 | GENERAL | 15 |
| 7.2 | PRINCIPLES AND METHODOLOGY | 15 |
| 7.2.1 | <i>Simplified fatigue analysis</i> | 15 |
| 7.2.2 | <i>Spectral fatigue methods</i> | 15 |
| 7.3 | DESIGN PROCESS | 15 |
| 7.3.1 | <i>Design Brief</i> | 15 |
| 8 | ALS | 17 |
| 8.1 | GENERAL PRINCIPLES | 17 |

Appendix A Materials, inspection and welds

Appendix B Design loads

Appendix C Hydrodynamic analysis

Appendix D Ultimate structural capacity

Appendix E Fatigue

Appendix F Structural analyses

Appendix G Mooring and riser systems

Appendix H Crane pedestals

1 INTRODUCTION

Ch.1 of the DNV-RP-C102 *Structural Design of Offshore Ships* addresses the structural design of new-build units intended to operate world wide. Ch.1 is thus describing in detail the application of technical requirements given in Ch.1 of the DNV-OS-C102. The Recommended Practices are in principle applicable to all types of mono-hull ship shaped units of conventional shape.

The main body of Ch.1 gives a macro description of the activities in a typical design process. Detailed technical procedures for the various activities are given in the appendices. The appendices are written primarily to describe the design requirements for world-wide operation, and are thus supporting the design process given in Ch.1 of the Recommended Practice.

2 SYMBOLS, DEFINITIONS AND REFERENCES

Definitions

✱1A1 : Structural requirements as defined in Rules for Classification of Ships Pt.3. Ch. 1. These are often referred to as “Main Class Requirements” as they represent common minimum requirements to all world wide ocean-going ships.

Symbols

W_{\min} : minimum midship section modulus
 M_{WR} : rule wave bending moment at a probability of exceedance of 10^{-8} (20 years return period)
 M_S : design still water bending moment
 f_l : material factor dependent on yield strength.
 M_{WB} : linear wave bending moment from direct calculations
 γ_{fi} : partial load coefficient
 γ_{nc} : non linear correction factor
 γ_m : material factor
 σ_y : characteristic yield strength of the material.

References

DNV-RP-C102 Structural Design of Offshore Ships
DNV-OS-C101 Design of Steel Structures
DNV-RP-C203 Fatigue Strength Analysis of Offshore Steel Structures
DNV-OS-C401 Fabrication and Testing of Offshore Structures
Classification Strength Analysis of Hull Structures in Tankers
Notes 31.3
Classification Fatigue Assessment of Ship Structures
Notes 30.7

3 OVERALL DESIGN PHILOSOPHY

3.1 General

The DNV-OS-C102 is partly based on the Load and Resistance Factor Design method (LRFD), as described in DNV-OS-C101 *Design of Steel Structures*, and partly on the design principles given in Rules for Classification of Ships Pt.3 Ch.1. The latter apply to local requirements to plates and profiles. Local requirements are typically acceptance criteria for plate thickness or section modulus of profiles when subjected to tank pressure, external sea pressure or local design loads. The acceptance criteria will depend on the expected global stresses in the element. Such global stresses are implicitly given in the Rules for Classification of Ships Pt.3 Ch.1 in terms of allowable stress for the element when subjected to local loads.

The ship shall in principle be designed to withstand environmental loads and temperatures at a specified location implying that direct calculation of wave loads, ship motions and accelerations shall be carried out. If the ship is to operate in several locations, the most unfavourable condition shall be used in the design. If the ship is to operate world wide, or if no specific location is designated, the North Atlantic scatter diagram shall be used. For world-wide operation the ship shall also comply with the ✕1A1 requirements.

3.2 World-wide or benign waters

The ✕1A1 requirements to longitudinal strength is expressed as requirement to minimum midship section modulus according to equation (3.1).

$$W_{\min} = \frac{M_{WR} + M_S}{175f_1} \quad (3.1)$$

- W_{\min} = minimum midship section modulus
- M_{WR} = rule wave bending moment at an annual probability of exceedance of $10^{-1.3}$ (20 years return period)
- M_S = design still water bending moment
- f_1 = material factor dependent on yield strength. (1.0 for mild steel)

The corresponding requirements according to the LRFD format in the ULS b) combination is given in equation (3.2).

$$W_{\min} = \frac{\gamma_{f1} * \gamma_{nc} * M_{WB} + M_S}{\frac{1}{\gamma_m} * \sigma_y} \quad (3.2)$$

- W_{\min} = minimum midship section modulus
- M_{WB} = linear wave bending moment from direct calculations at an annual probability of exceedance of 10^{-2} (100 years return period)
- M_S = design still water bending moment

| | | |
|---------------|---|--------------------------------|
| γ_{fi} | = | partial load coefficient |
| γ_{nc} | = | non linear correction factor |
| γ_m | = | material factor |
| σ_y | = | yield strength of the material |

When the requirements to the longitudinal strength according to equation (3.2) based on a specific scatter diagram is equal to the $\forall 1A1$ requirements as in equation (3.1), this is referred to as the boundary between world wide operation and benign waters.

Thus benign waters is defined according to equation (3.3).

$$M_{WB} * \gamma_{fi} * \gamma_{nc} \leq 1.17 * M_{WR} + 0.17M_S \quad (3.3)$$

Ch.1 of DNV-OS-C102 is applicable to world wide operation, and Ch.2 is specifically addressing the requirements to operation in benign waters. Based on to equation (3.3), a number of ship lengths have been analysed for 3 different Breadth/Depth, Length/Breadth and Depth/Draught ratios have been analysed in order to find the environmental condition that will give the left hand side of the equation equal to the right hand side. The upper band of the results from these analyses is shown in Figure 3-1.

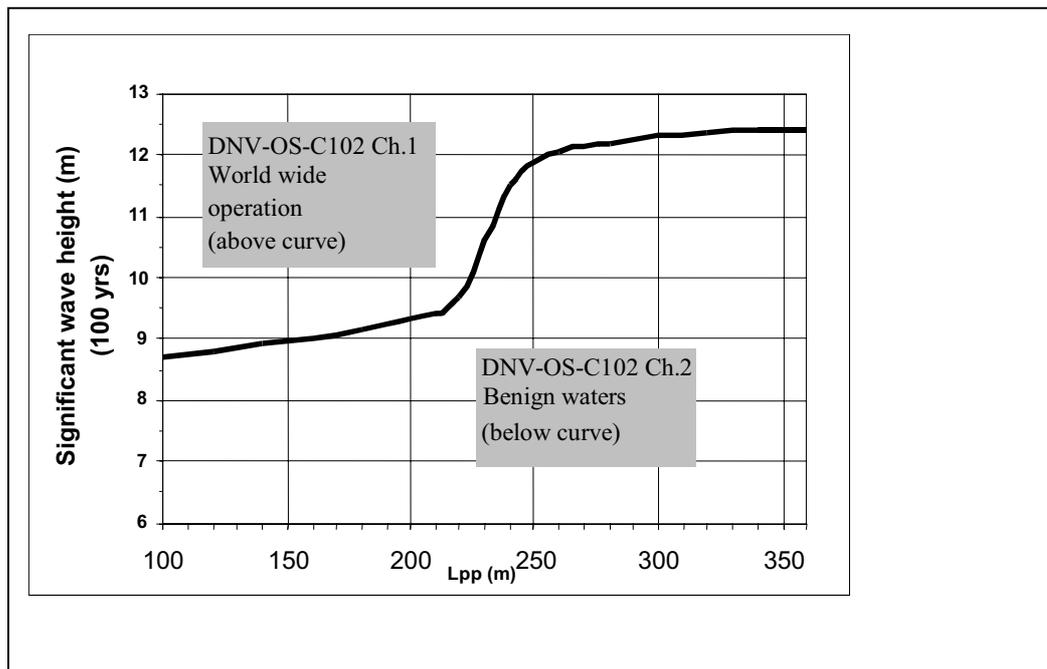


Figure 3-1 World wide and benign waters

Figure 3-1 indicates when the $\forall 1A1$ requirements to longitudinal strength is sufficient (under the curve), and when the LRFD format gives stricter requirements. The figure is only indicative, and the applicable part of DNV-RP-C102 is determined according to the equation (3.3). based on wave bending moment from direct calculations. However, when the 100 years significant wave height at a specific location is less than 8.5m, the $\forall 1A1$ requirements are dimensioning for the hull girder strength, and direct calculations of wave loads and responses are not required. Such direct calculations may, however, be

desirable in order to establish more favourable values of the accelerations for topside design. This is further addressed in Ch.2.

4 SELECTION OF MATERIALS

4.1 Material and inspection categories and welds

Materials shall be divided into material categories. The requirements to the material quality are dependent on material category according to the principles given in DNV-OS-C101. The main structural elements in offshore ships and the associated material category are given in DNV-OS-C102. Further examples of material category, inspection category and welds are given in Appendix A in terms of sketches of typical structural details.

The required extent of inspection is given in DNV-OS-C401 *Fabrication and Testing of Offshore Structures*. For convenience, Table C1 of DNV-OS-C401Ch.2 Sec.3 is given below.

| Inspection category | Type of connection | Test method | | | |
|---------------------|--|-------------|------------------------|----------------------------|--------------------------|
| | | Visual | Magnetic ¹⁾ | Radio-graphy ²⁾ | Ultrasonic ³⁾ |
| I | Butt weld | 100 % | 100 % | 100 % | - |
| | Cross- and T-joints, full penetration welds | 100 % | 100 % | - | 100 % |
| | Cross- and T-joints, partly penetration and fillet welds | 100 % | 100 % | - | - |
| II | Butt weld | 100 % | 20 % ⁴⁾ | 10 % | - |
| | Cross- and T-joints, full penetration welds | 100 % | 20 % | - | 20 % |
| | Cross- and T-joints, partly penetration and fillet welds | 100 % | 20 % | - | - |
| III | Butt weld | 100 % | Spot ⁵⁾ | Spot ⁵⁾ | - |
| | Cross- and T-joints, full penetration welds | 100 % | Spot ⁵⁾ | - | Spot ⁵⁾ |
| | Cross- and T-joints, partly penetration and fillet welds | 100 % | Spot ⁵⁾ | - | - |

1) Liquid penetrant testing to be adopted for non ferro-magnetic materials.
2) May be partly or wholly replaced by ultrasonic testing upon agreement.
3) Ultrasonic examination shall be carried out for plate thicknesses of 10 mm and above.
4) Spot check will be accepted as sufficient for main hull girder members assembled by means of automatic welding process
5) Approximately 2-5%.

5 COMPLIANCE WITH BASIC +1A1 REQUIREMENTS

5.1 Typical Design Loop

The ship shall comply with the +1A1 requirements. In order to document compliance with the Rules for Classification of Ships Pt.3 Ch.1, a typical working process is shown in Figure 5-1.

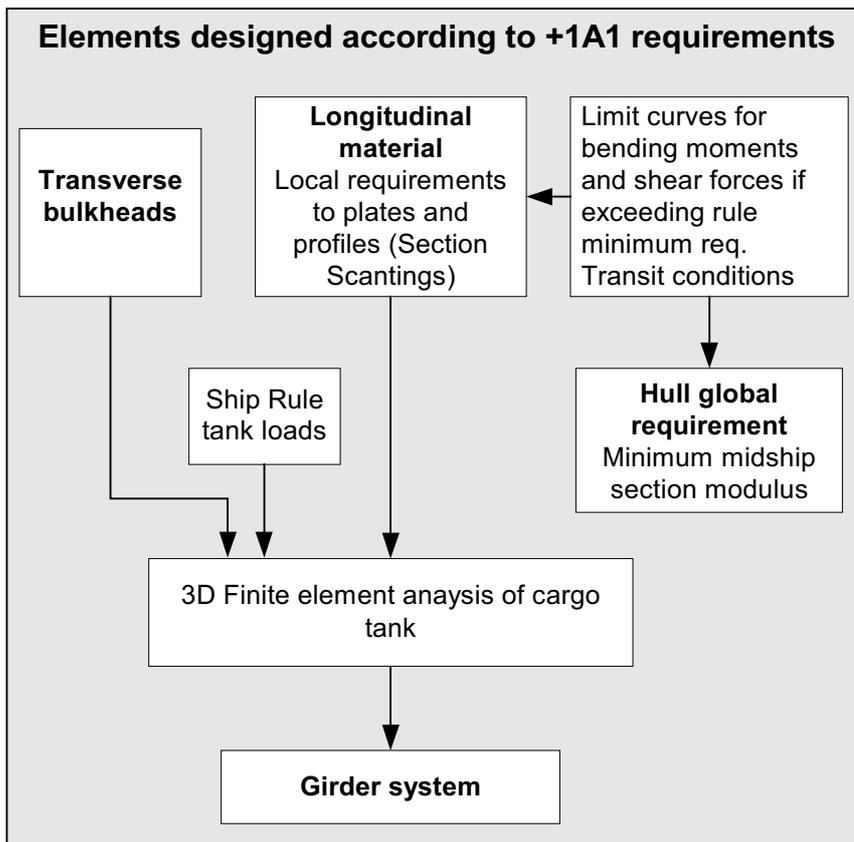


Figure 5-1 Elements designed according to +1A1 requirements

5.2 Longitudinal material

The NAUTICUS Section Scantling program may be used to determine the rule requirements to section scantlings for longitudinal sections and for transverse bulkheads. The scantlings shall in principle be calculated at every section where there are changes in dimensions or geometry. This will typically be at least 4 sections within the cargo area as shown in Figure 5-2.

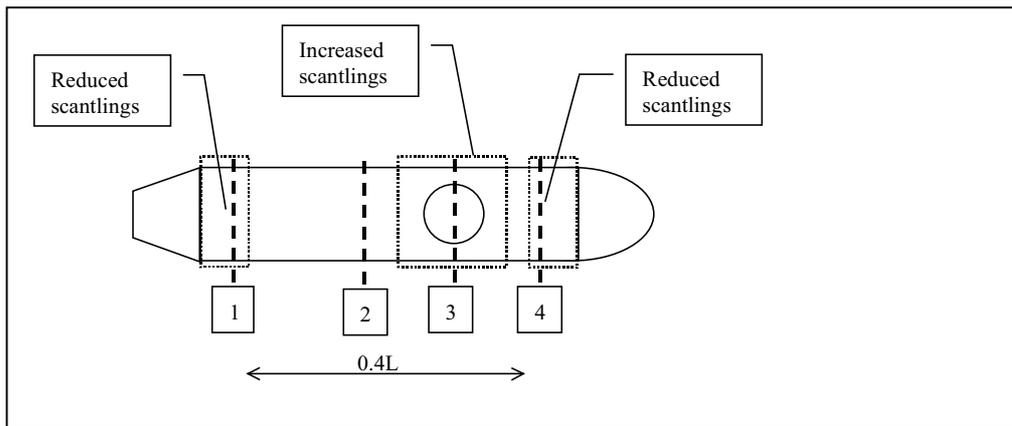


Figure 5-2 Typical sections to be considered for \forall 1A1 requirements

The sections shown are:

- aft end of the cargo area outside 0.4 L where scantlings normally is reduced
- in the vicinity of amidships representative for the midship cargo area
- through the turret area where scantlings are increases and geometry changed
- fwd end of the cargo area outside 0.4 L where scantlings normally is reduced.

5.3 Transverse bulkheads

The NAUTICUS Section Scantling program may also be used to determine the scantlings of the transverse bulkheads. The program does not consider the girder system itself, but the vertical webs and/or horizontal stringers act as supports for the stiffeners and plates. The girder system is designed by means of direct calculations.

5.4 Rule still water bending moments and shear forces

The requirements to midship section modulus is based on given distribution of hull girder minimum bending moments and shear forces for both still water hogging and sagging conditions as well as wave loads. The still water moment and shear force distribution is referred to as the limit curves or envelope curves. The design still water bending moment and shear forces at any position along the length of the unit is in accordance to the Rules for Classification of Ships Pt.3 Ch.1. If the maximum or minimum still water bending moments from the actual load conditions deviate significantly from the rule values, different limit curves may be specified according to guidance given in Appendix B.

5.4.1 Shear force correction factors

The shear force distribution along the hull girder for the separate load conditions is based on the assumption that the hull is infinitely stiff. However, the net vertical loads acting on the bottom structure between transverse bulkheads may partly be transferred via the bottom longitudinal girders to the transverse bulkheads. This implies that the shear forces need to be corrected in order to account for this 3-dimensional effect. This is further explained in Appendix B.

5.5 Hull girder system

The hull girder system includes transverse web frames, longitudinal girders (like CL-girder, side girders and side shell stringers), vertical webs and horizontal stringers on the transverse bulkheads. The girder system is designed by means of a 3D finite element analysis in the same manner as described in Classification Notes 31.3 *Strength Analysis of Hull Structures in Tankers*.

6 ULS – HULL GIRDER

6.1 Design principles

The ULS hull girder capacity checks shall be based on direct calculations of wave loads. The bucking and yield capacity of the hull girder is typically checked at sections with maximum wave bending moment, maximum wave shear, in way moonpool/turret openings and where geometry, scantlings or loads are different. Figure 6-1 depicts the different design tasks in the ULS capacity check of the hull girder and the interface to the design tasks based on 1A1 requirements.

The ULS design process is described in detail in Appendix D.

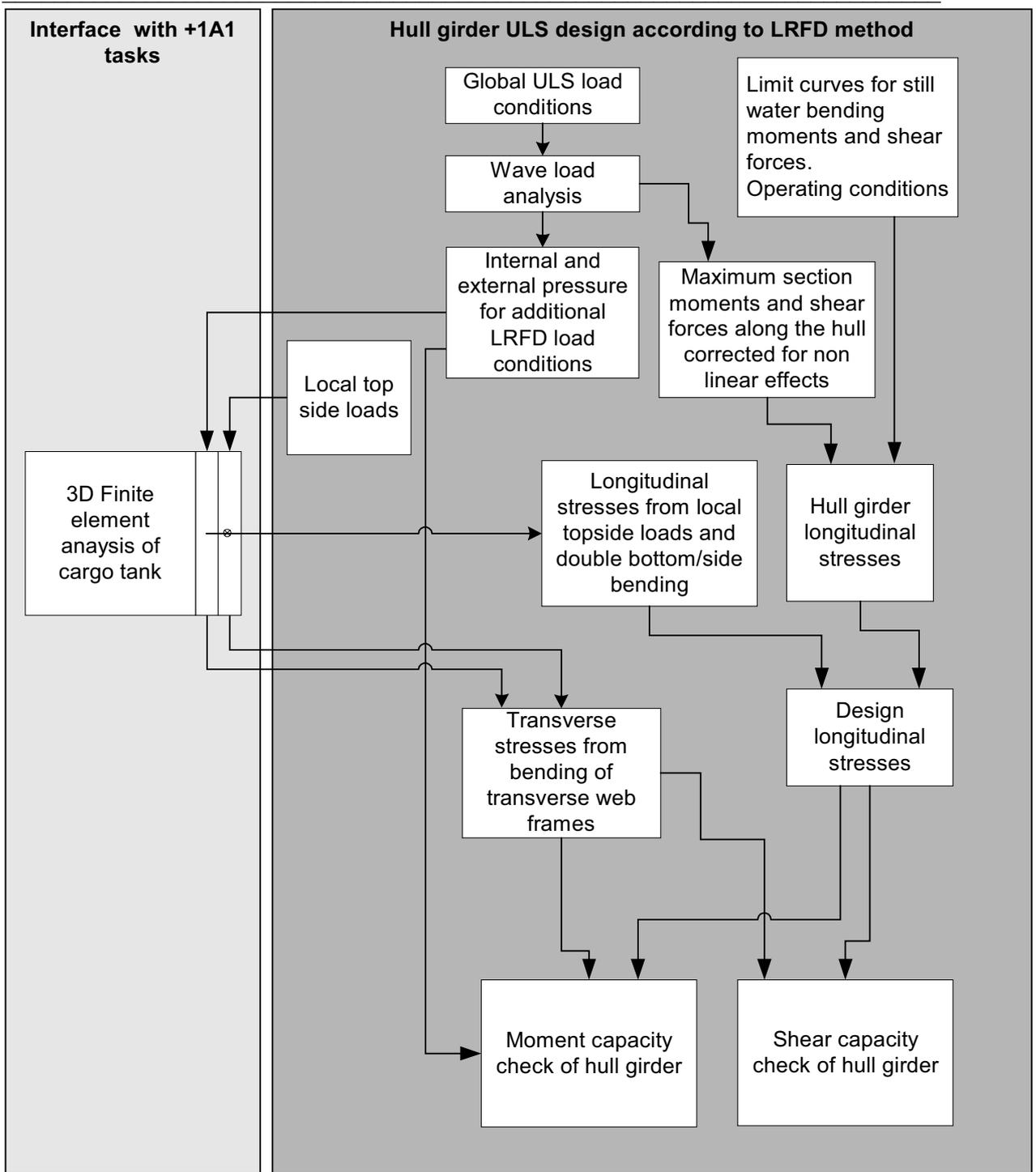


Figure 6-1

Design process for ULS hull girder capacity checks

7 FLS - HULL STRUCTURAL DETAILS

7.1 General

The fatigue life calculations of different structural details are specifically described in Appendix E. A short description of principles and typical fatigue sensitive details are given below.

7.2 Principles and methodology

The fatigue capacity is documented according to the principles given in DNV Classification Notes 30.7 *Fatigue Assessment of Ship Structures*. The fatigue capacity is calculated assuming that the linear accumulated damage (Palmgrens – Miner rule). Classification Notes 30.7 describe several methods for fatigue life calculations. The different methods are used at different stages in the design loop. Applicable method can also be selected dependent on the results from a screening process to identify fatigue critical details.

7.2.1 Simplified fatigue analysis

The simplified fatigue analysis is based on the assumption that the long term distribution of stresses can be described by the maximum dynamic stress amplitude and a Weibull shape parameter. The method is described in Classification Notes 30.7 section 2. and 3. The amplitude value and the Weibull parameter should be determined by means of direct calculations based on a given scatter diagram. These methods are in principle described in Classification Notes 30.7 section 4. and 5. respectively, but values for longitudinal bending moment, external sea pressure, Weibull parameter and ship accelerations should be based direct analyses as the Classification Notes 30.7 assume North Atlantic 20 years conditions.

7.2.2 Spectral fatigue methods

The most accurate methods are either a full stochastic analysis or a stress component based stochastic analysis. These methods are described in Classification Notes 30.7. The procedure requires a given scatter diagram, a known hot spot stress or a nominal stress dependent on the S-N curves used. The full stochastic analysis is a comprehensive task involving typically 300-400 load conditions. All phase information between different loads are kept. The full stochastic analysis are less suitable if some significant loads are non linear. The stress component based analysis are often sufficient for some details (e.g. side longitudinals).

Both methods require a finite element model to determine the stresses. A stress component based stochastic analysis require at least a local finite element model for the local response. This can be part of the typical 3-tank model used for strength assessment.

7.3 Design process

7.3.1 Design Brief

The design brief is a document comprising the design basis and premises such as:
— applicable rules

-
- design fatigue life
 - fatigue safety factors for different structural elements
 - environmental data including wave scatter diagram
 - required structural analysis including method description
 - required documentation .

The design brief incorporate the requirements given in the owner specification implying that the latter should in detail describe the required analyses and associated methods.

8 ALS

8.1 General principles

DNV-OS-C102 does not specifically define any accidental events. From a stability point of view, collision with other vessels is assumed to be covered by requirements of international bodies like MARPOL and IMO and the Classification Society. For normal design, the extent of the damage due to collision with a supply vessel is not dimensioning for the scantlings due to structural redundancy provided the damage stability requirements are complied with.

If the vessel is to be considered for accidental conditions, prescriptive requirements are given in DNV-OS- A101. These requirements are intended to take account of accidental events, which have been identified through previous risk studies and through experience. The selection of relevant design accidental loads is dependent on a safety philosophy considered to give a satisfactory level of safety. The generic loads defined represent the level of safety considered acceptable by DNV, and are generally based on accidental loads affecting safety functions which have an individual frequency of occurrence in the order of 10^{-4} per year.

The most relevant design accidental loads considered in DNV-OS-A101 are:

- impact loads (including dropped object loads and collision loads)
- unintended flooding
- loads caused by extreme weather
- explosion loads
- fire loads

CHAPTER 2

Benign waters

CONTENT

| | | |
|----------|---|-----------|
| 1 | INTRODUCTION | 20 |
| 2 | SYMBOLS AND DEFINITIONS..... | 21 |
| 3 | OVERALL DESIGN PHILOSOPHY..... | 22 |
| 3.1 | GENERAL..... | 22 |
| 4 | SELECTION OF MATERIALS | 23 |
| 4.1 | MATERIAL AND INSPECTION CATEGORIES AND WELDS..... | 23 |
| 5 | STRUCTURAL CAPACITY..... | 25 |
| 5.1 | GENERAL DESIGN PRINCIPLES..... | 25 |
| 5.2 | ALTERNATIVE 1 - HULL STRUCTURE BASED ON ✕1A1 REQUIREMENTS | 25 |
| 5.2.1 | <i>Hull</i> | 25 |
| 5.2.2 | <i>Topside supporting structure</i> | 25 |
| 5.3 | ALTERNATIVE 2 - HULL STRUCTURE BASED ON LRFD PRINCIPLES..... | 27 |
| 6 | FLS - HULL STRUCTURAL DETAILS | 28 |
| 6.1 | GENERAL..... | 28 |
| 7 | ALS..... | 29 |
| 7.1 | GENERAL PRINCIPLES | 29 |

1 INTRODUCTION

Ch.2 of DNV-RP-C102 addresses the structural design of *new-build* units intended to operate in benign waters. The definition of benign waters is given in Ch.1.

Ch.2 is thus describing in detail the application of technical requirements given in Ch.2 of DNV-OS-C102. The Recommended Practices are in principle applicable to all types of mono-hull ship shaped units of conventional shape.

Ch.2 gives a macro description of the activities in a typical design process. Relevant parts of the appendices are referred to in this chapter.

2 SYMBOLS AND DEFINITIONS

Definitions

✱1A1

Structural requirements as defined Rules for Classification of Ships Pt.3 Ch.1. These are often referred to as “Main Class Requirements” as they represent common minimum requirements to all world-wide ocean-going ships.

3 OVERALL DESIGN PHILOSOPHY

3.1 General

Ch.3 of the Recommended Practices applies to units restricted to operate in benign waters. This implies that provided the σ_{1A1} requirements are complied with, the hull girder global requirements are sufficient. Applying the LRFD principles for the ULS global capacity to such a design will result in reduced section scantlings.

The design of the hull can be done completely according the Rules for Classification of Ships Pt.3 Ch1 or based on the LRFD principles. In case of the latter, the minimum midship section modulus according to σ_{1A1} requirements can be reduced by maximum 25%.

The support structure for the topside facilities can either be based on accelerations from direct calculations or on the values given by σ_{1A1} requirements.

The alternative design processes are further described in section 5.

4 SELECTION OF MATERIALS

4.1 Material and inspection categories and welds.

DNV-OS-C102 gives two optional principles for selection of materials. The materials can either be selected according to the Rules for Classification of Ships Pt.3 Ch.1 Sec.2 or according to Ch.1 Sec.2 in DNV-OS-C102. The reason for this option is that the requirements to materials used in benign waters are the same for newbuildings as for conversions. It was the intention of the standard to recommend the principles as given in Ch.1 for newbuilding, and to accept the principles in the Rules for Classification of Ships Pt.3 Ch.1 Sec.2 for existing hulls, i.e. conversions. It is thus advised that selection of materials, inspection and welds be done as described in Ch.1 of this Recommended Practice for newbuildings in benign waters.

Further examples of material category, inspection category and welds are given in Appendix A in terms of sketches of typical structural details.

5 STRUCTURAL CAPACITY

5.1 General Design principles

Independent on the alternative design principles as described below, the scantlings according to the ∇ 1A1 requirements must be established. This may be done as described in Ch.1 Sec.5.

5.2 Alternative 1 - Hull structure based on ∇ 1A1 requirements

5.2.1 Hull

The structural capacity (strength) of the hull can be determined according to the Rules for Classification of Ships Pt.3 Ch.1. Both transit and operating load conditions should be considered. No wave load analysis is required for the design of hull structural capacity as the wave bending moments and global accelerations are given in the Rules in terms of empirical formulae. It is not allowed to use accelerations from direct calculations in this alternative regarding hull structural capacity.

5.2.2 Topside supporting structure

The supporting structure for the topside facilities shall be designed according to the LRFD principles. This will be identical to the procedure given in Ch.1. A typical design procedure for the "Alternative 1" approach is given in Figure 5-1.

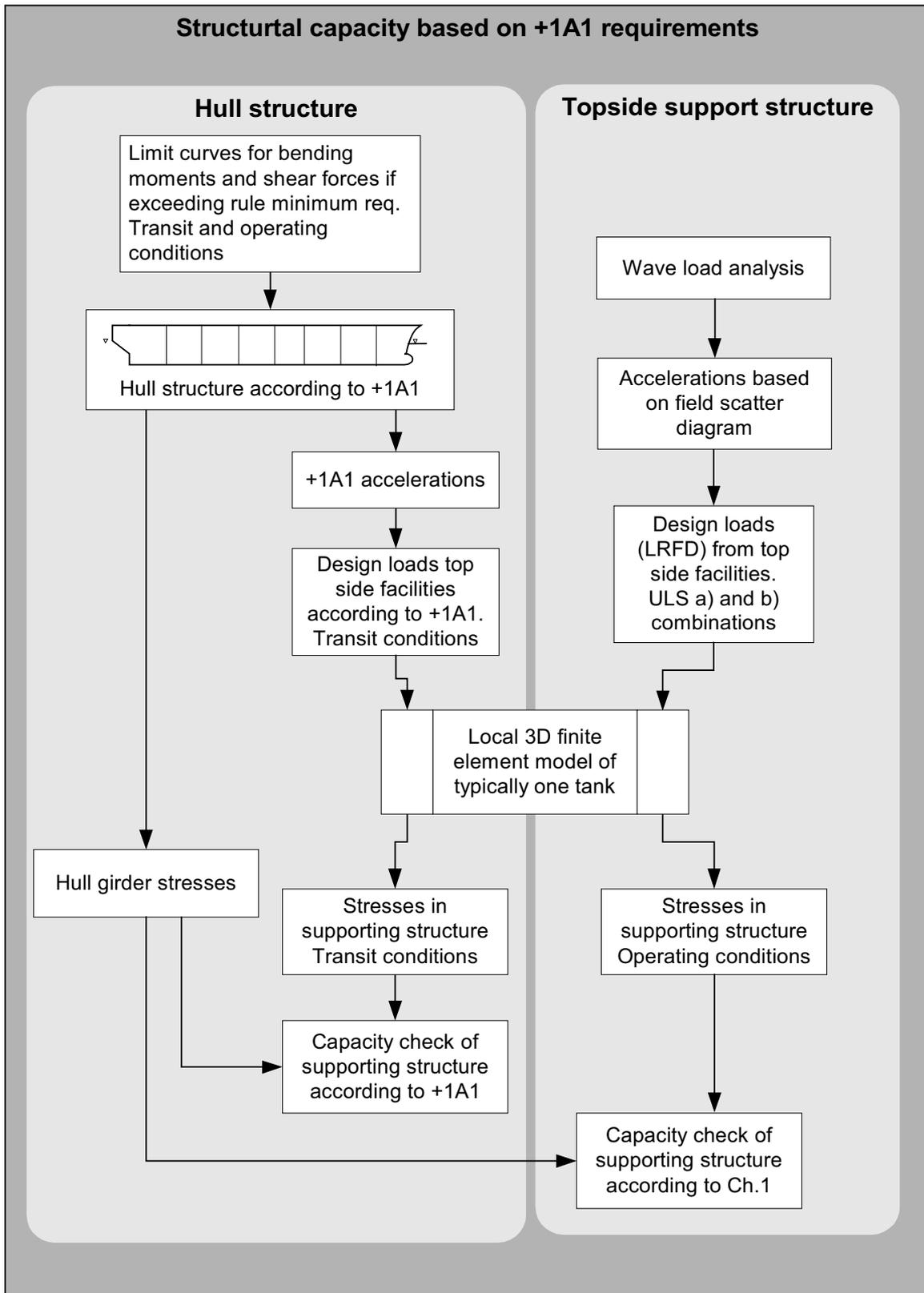


Figure 5-1 Design procedure for “Alternative 1” – Hull design according to +1A1

5.3 Alternative 2 - Hull structure based on LRFD principles

This alternative is identical the design procedure given in Ch.1 with detailed description given in Appendix D.

The minimum section modulus according to the Rules for Classification of Ships may be reduced by maximum 25% if the hull is designed according to the LRFD principles.

6 FLS - HULL STRUCTURAL DETAILS

6.1 General

Fatigue capacity of the structural details are in principle carried out in the same manner as given in Ch.1. The calculations of fatigue life can normally be done by means of the Simplified Fatigue Methods as given in Classification Notes 30.7 for details on deck and bottom area as such are governed by global stresses.

Side longitudinals and other details dominated by local responses should be considered by means of spectral methods.

7 ALS

7.1 General principles

The Accidental Limit State design procedure is the same as given in Ch.1.

CHAPTER 3

CONVERSIONS

CONTENTS

| | | |
|----------|---|-----------|
| 1 | INTRODUCTION..... | 32 |
| 2 | OVERALL DESIGN PHILOSOPHY | 33 |
| 2.1 | GENERAL..... | 33 |
| 2.1.1 | <i>Harsh environment.....</i> | <i>33</i> |
| 2.1.2 | <i>Benign waters.....</i> | <i>33</i> |
| 3 | FLS - HULL STRUCTURAL DETAILS | 34 |
| 3.1 | INTRODUCTION..... | 34 |
| 3.2 | CONDITION OF THE HULL WITH RESPECT TO GLOBAL SECTION PROPERTIES | 34 |
| 3.3 | CONDITION OF LOCAL ELEMENTS | 34 |
| 3.4 | USE OF SURVEY REPORTS TO CALCULATE LOAD HISTORY | 34 |

1 INTRODUCTION

Ch.3 of the Recommended Practice addresses the design process related to conversions of tankers or conventional merchant ships into an offshore unit.

Ch.3 gives a macro description of the activities in a typical design process for conversion projects and refers to relevant parts of the appendices.

Both world wide operation and benign waters are covered.

2 OVERALL DESIGN PHILOSOPHY

2.1 General

2.1.1 Harsh environment

In general the same requirement to safety level applies to conversions as to new-build ships. This means that conversions intended to operate in harsh environments must comply with the requirements given in DNV-OS-C102. This is described in Ch. for world wide operation, and in Ch.2 for operation restricted to benign waters. In order to document the material quality of NVA mild used in primary elements like deck, bottom, side and longitudinal bulkheads, samples should be taken and tested. Possible reinforcement of the hull girder may be done by replacing or adding steel to the main deck/upper side and possibly also to the bottom area.

2.1.2 Benign waters

The structural capacity of conversions for use in benign waters is carried out in the same manner as the “Alternative 1” design method as described in Ch.2 section 5.2.

3 FLS - HULL STRUCTURAL DETAILS

3.1 Introduction

The fatigue capacity of the hull is a matter of fatigue capacity of each structural detail. To determine the fatigue capacity for a detail, the following should be considered:

- condition of the hull with respect to global section properties
- condition of local elements
- survey reports
- expected loads for the converted ship
- scatter diagram for the field of operation after conversion
- inspection program for the converted ship.

3.2 Condition of the hull with respect to global section properties

The condition of the ship with respect to corrosion of plates and profiles and welds must be known in order to evaluate the need for replacement of steel, and to calculate the actual global section properties. Typically, a Nauticus Section Scantlings, or similar programs, is executed for this purpose.

3.3 Condition of local elements

The condition of local elements must be carefully considered. In benign waters, this is especially important for elements exposed to fatigue damage due to local loads such as side longitudinals. The required weld size from a fatigue capacity point of view, can be calculated based on DNV-RP-C203.

3.4 Use of survey reports to calculate load history

The reports from previous inspections can be used to identify critical details. The inspection reports may also be used to calibrate the load history according to reported cracks. The calibrated loads can then be used as a basis for calculation of expected fatigue life of other details exposed to the same load type. Care should be taken if the reported cracks occurred within the first 5 years. Such cracks could be due to poor workmanship and is not suitable for calibration of fatigue loads provided no similar pattern are observed for similar details.

The procedure given below assumes that the cracks observed are suitable for loads calibration.

Referring to Classification Notes 30.7 it can be seen that when the long term stress range distribution is described in terms of a Weibull shape parameter and by means of a one-slope S-N curve, the fatigue life is in inverse ratio to the stress range to the power of 3.

The fatigue life of structural details exposed to the same loads will thus be:

$$F_u = L \left(\frac{SCF_c}{SCF_b} \right)^3$$

F_u = “used” life in years of considered detail

L = number of years from ship delivery until the crack occurred

SCF_c = stress concentration factor of considered detail

SCF_b = stress concentration factor of detail with crack.

The remaining fatigue life can be calculated based on the scatter diagram for the new location.

Appendix A

Materials, inspection and welds

Contents

1. INTRODUCTION.....2

2. DEFINITIONS AND SYMBOLS3

3. TYPICAL STRUCTURAL DETAILS.....4

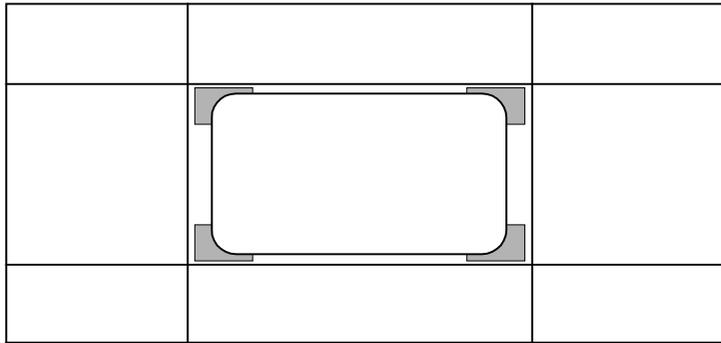
1. INTRODUCTION

The material category, steel grade, inspection category and welds are determined according to the DNV-OS-C102 Offshore Ships. This appendix elaborates on the requirements by means of sketches of typical structural details. The structural details selected should be considered as design principles and are thus applicable to similar designs, e.g. selection of material and inspection category given for a support stool for topside structure can also be applied to interface between drillfloor substructure, or flare structure, and the main deck.

2. DEFINITIONS AND SYMBOLS

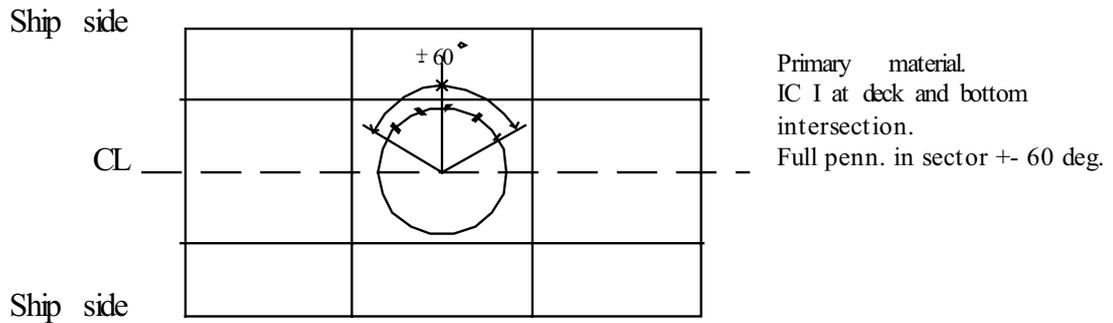
| | |
|----|---|
| IC | Inspection category |
| | Hashed regions in figures relate to full penetration weld |
| | |
| | |

3. TYPICAL STRUCTURAL DETAILS



Special materials in corners plates.
IC I for butt welds

Figure A- 1 Deck and bottom plating, moonpool corners.



Primary material.
IC I at deck and bottom
intersection.
Full penn. in sector ± 60 deg.

Figure A- 2 Deck and bottom plating, moonpool.

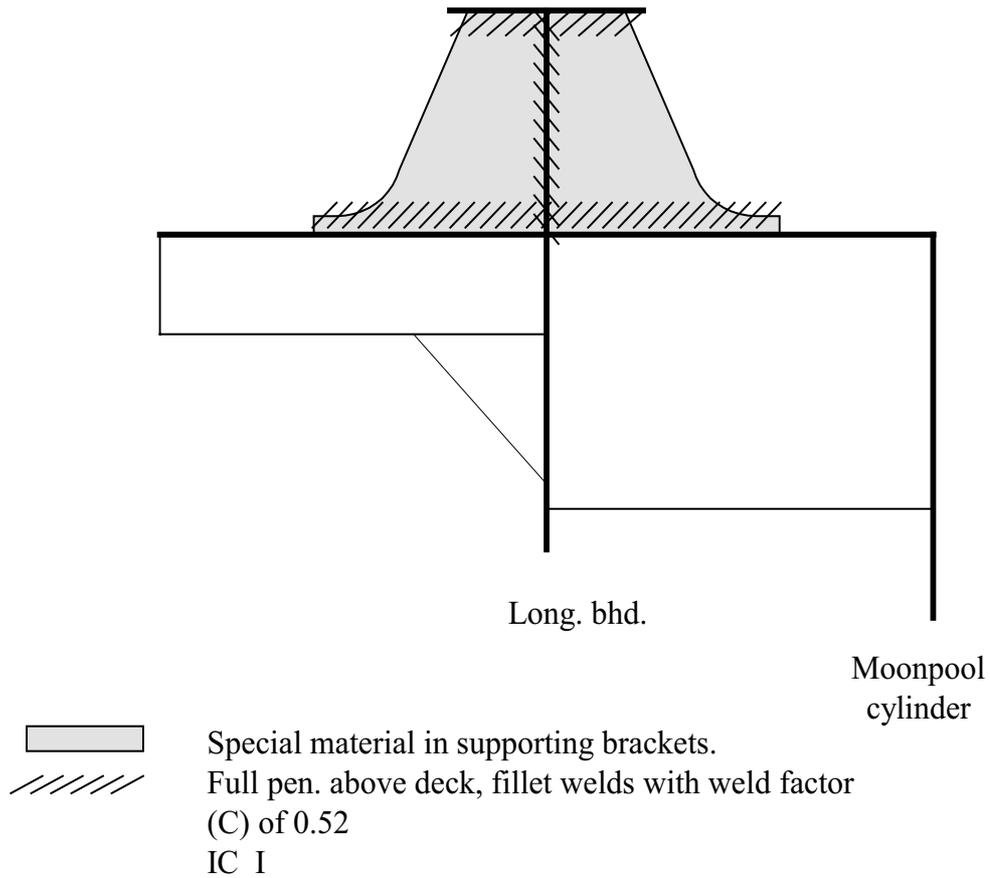


Figure A- 4 Support for topside equipment

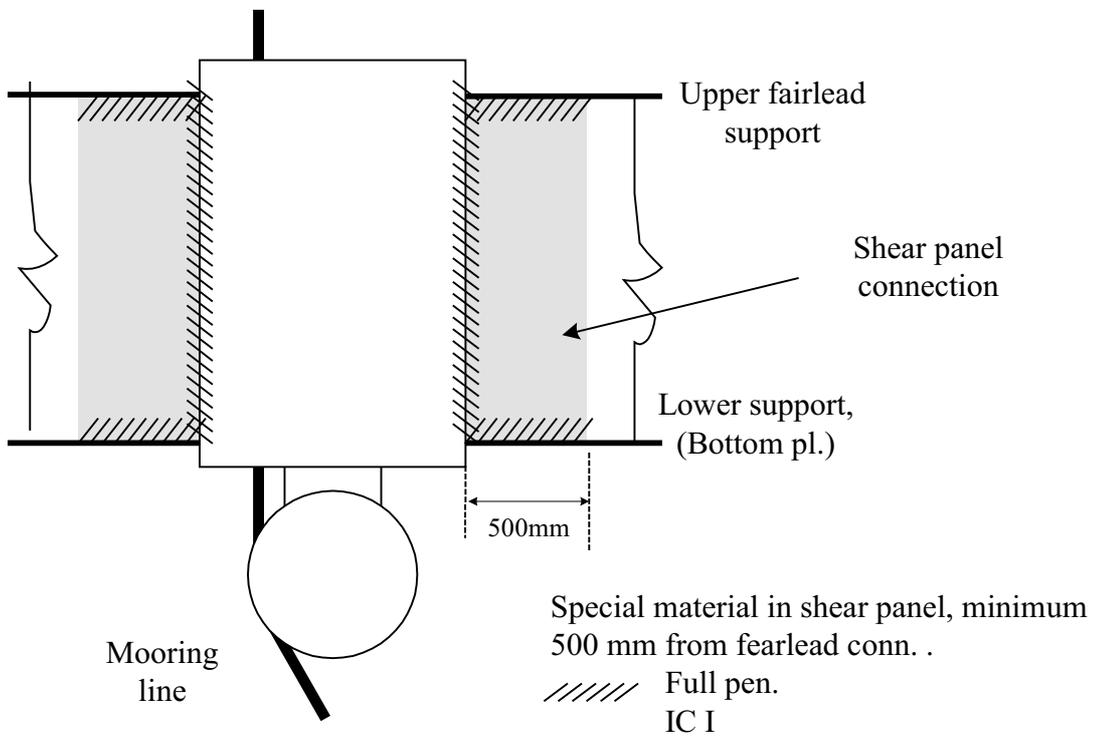


Figure A- 3 Fairlead connections to turret

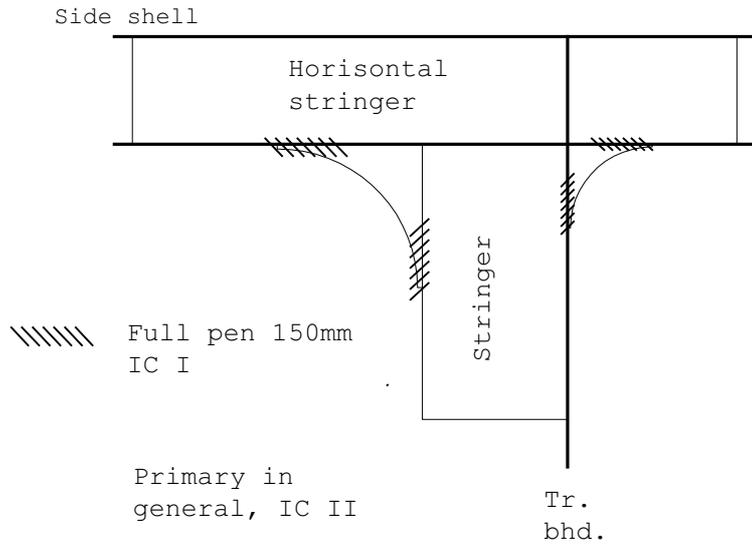


Figure A- 5 Horizontal stringer connection

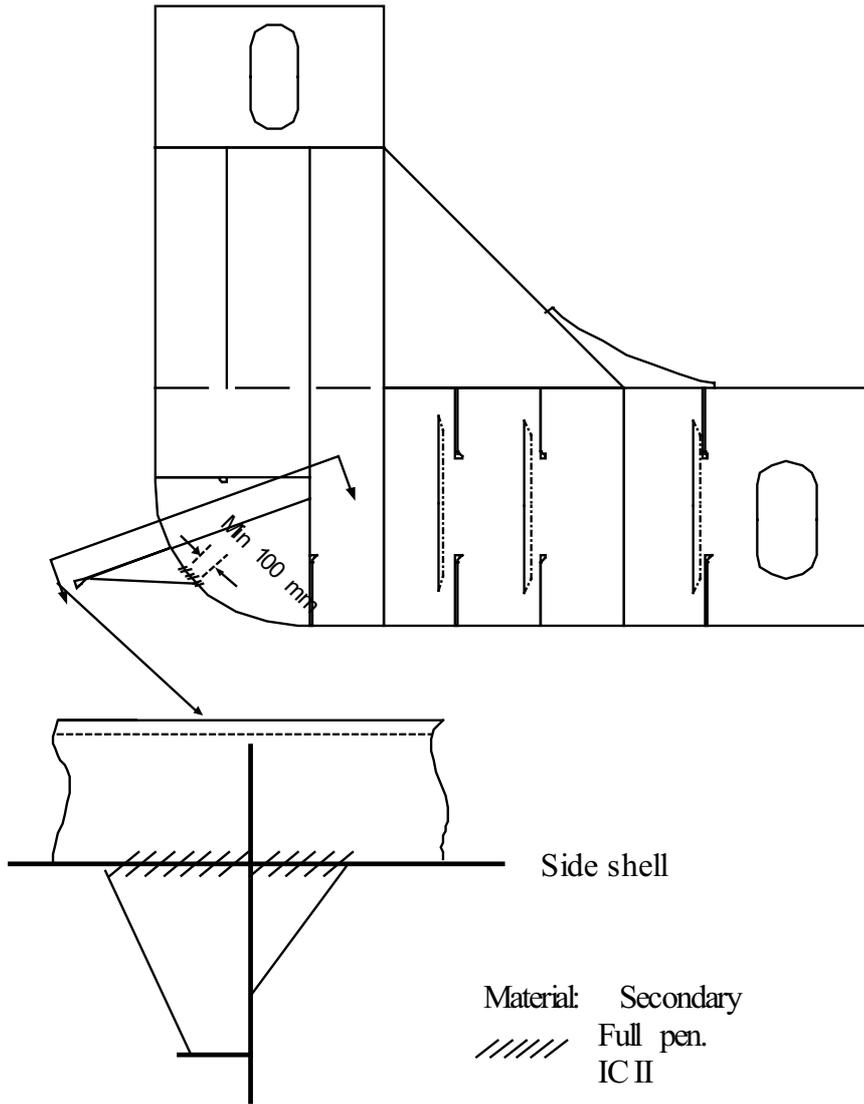


Figure A- 6 Large bilge keel (FPSO)

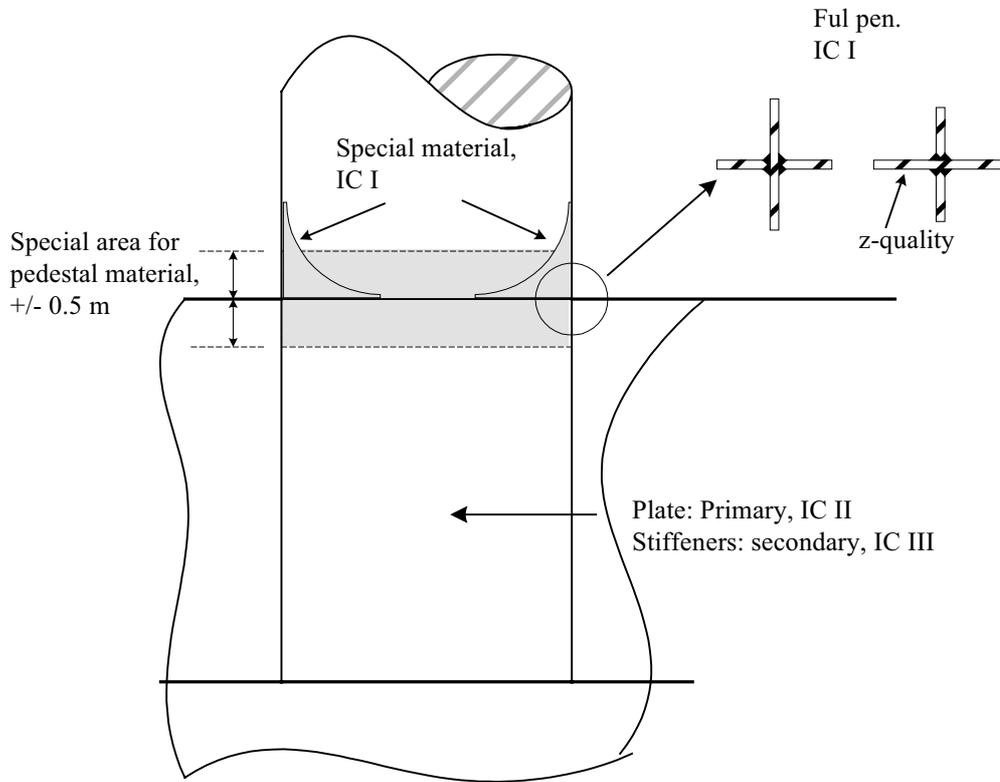
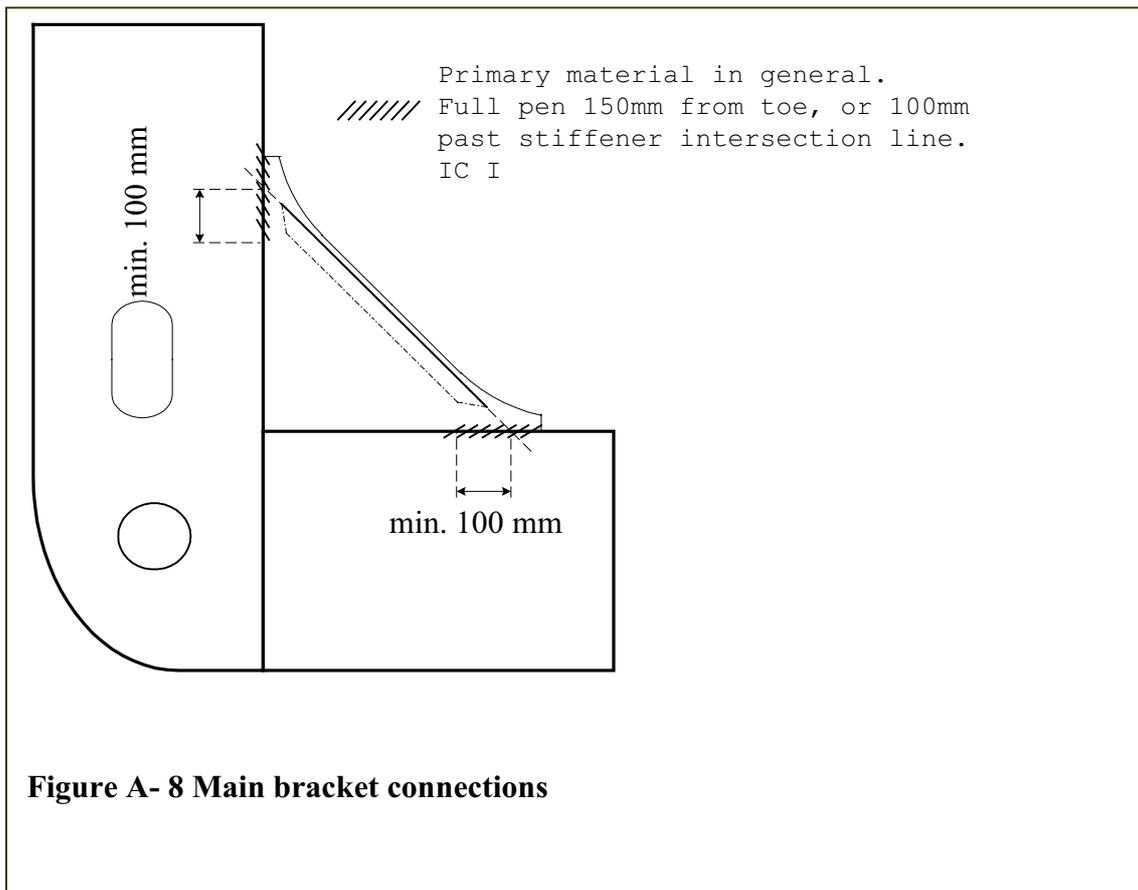


Figure A- 7 Crane pedestal connection to hull



It should be noted that the buckling stiffener should not be too close to the free edge. A distance of $10*t - 15*t$ may be used, where t is the thickness of the bracket. The toe of the stiffener should be minimum 100 mm from the tank top/inner side.

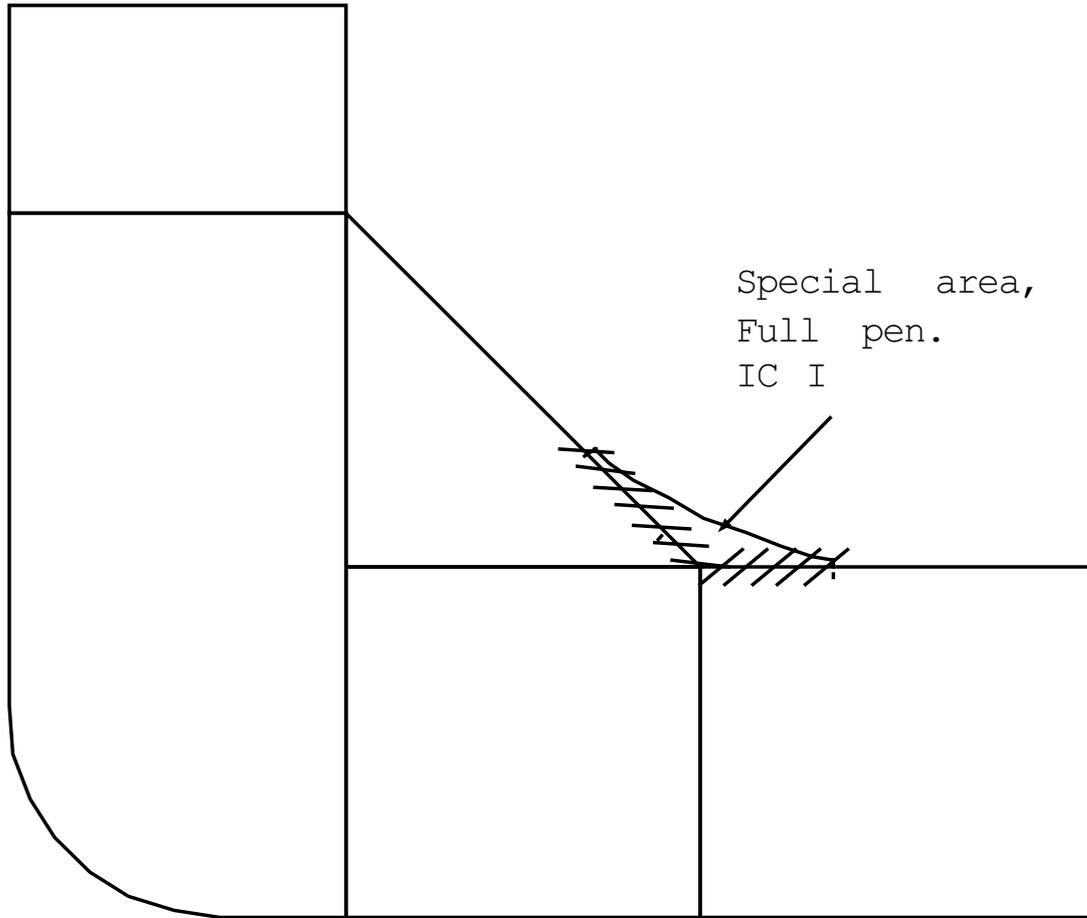


Figure A- 9 Softening bracket at hopper tank corner

APPENDIX B

DESIGN LOADS

CONTENTS

| | |
|---|-----------|
| 1. INTRODUCTION | 3 |
| 2. DEFINITIONS AND SYMBOLS | 4 |
| 3. STILLWATER LOADS | 6 |
| 3.1. PERMANENT LOADS | 6 |
| 3.2. VARIABLE LOADS | 7 |
| 3.3. STILLWATER BENDING MOMENT AND SHEAR FORCE | 7 |
| 3.3.1. <i>Still water bending moment</i> | 8 |
| 3.3.2. <i>Still water limit curves based on actual loading conditions</i> | 9 |
| 3.3.3. <i>Still water shear force</i> | 10 |
| 3.3.4. <i>Shear force transverse distribution</i> | 11 |
| 3.3.5. <i>Shear force correction</i> | 12 |
| 3.3.6. <i>Shear force correction factor</i> | 16 |
| 4. ENVIRONMENTAL LOADS | 19 |
| 4.1. GENERAL | 19 |
| 4.2. WAVE INDUCED FORCES | 19 |
| 4.3. WIND | 20 |
| 4.4. SLOSHING LOADS IN TANKS | 20 |
| 5. COMBINATION OF STILL WATER AND ENVIRONMENTAL LOADS | 21 |
| 5.1. GENERAL | 21 |
| 5.2. GLOBAL BENDING MOMENTS, SHEAR FORCE AND AXIAL FORCE | 21 |
| 5.2.1. <i>Vertical bending moment</i> | 21 |
| 5.2.2. <i>Horizontal bending moment</i> | 21 |
| 5.2.3. <i>Global Shear force</i> | 22 |
| 5.2.4. <i>Axial force</i> | 22 |
| 5.3. EXTERNAL SEA PRESSURE AND INTERNAL TANK PRESSURE | 23 |
| 5.3.1. <i>External sea pressure</i> | 23 |
| 5.3.2. <i>ULS Condition</i> | 23 |
| 5.3.3. <i>FLS Condition</i> | 26 |
| 5.4. INTERNAL TANK PRESSURE | 27 |
| 5.4.1. <i>ULS Condition</i> | 28 |
| 5.4.2. <i>FLS Condition</i> | 30 |
| 5.5. INDUCED FORCES ON TOPSIDE AND DECK EQUIPMENT | 32 |

1. INTRODUCTION

This appendix describes in more detail the loads to be considered in the design of offshore ships. According to the principles in DNV-OS-C102 Offshore Ships, characteristic loads shall be used as reference loads in the partial coefficient method (LRFD format). However, the ship shall also comply with ✕1A1 requirements. This implies that the acceptance criteria are associated with the wave bending moment at an annual probability of exceedance of $10^{-1.3}$ (20 years return period) and with local dynamic sea pressure on plates and stiffeners at a probability of exceedance of 10^{-4} (daily return period).

In the design of offshore ships for world wide operation one set of load conditions to prove compliance with the ✕1A1 requirements and one set of load conditions to prove compliance the LRFD format must be established. Ships intended for restricted operation in benign waters may be designed only according to the ✕1A1 requirements, and just one set of load conditions is required. However, if such ships are also designed according to the LRFD format either for reduced section modulus or topside design, two sets of load conditions are needed.

The tank loads specifically related to the ✕1A1 requirements are not dealt with in this Appendix as they are given directly in Rules for Classification of Ships Pt.3 Ch.1. Reference is also made to Classification Notes 31.3 Strength analysis of hull structure in tankers, for design according to the ✕1A1 requirements. Some still water loads used in connection with the ✕1A1 requirements are also the basis for the ULS capacity checks and are described in this appendix.

The loads may be divided into the following categories:

Static loads:

- Permanent loads
- Variable loads

Environmental loads

2. DEFINITIONS AND SYMBOLS

| | | |
|-------------|---|--|
| C_{wu} | : | Wave coefficient |
| F_{WB} | : | Linear axial force calculated by hydrodynamic analysis |
| I | : | Moment of inertia |
| K | : | Shear force correction factor |
| K_L | : | Shear force correction factor for longitudinal bulkhead |
| K_S | : | Shear force correction factor for ship side |
| L_t | : | Length of centre cargo tank |
| $M_{h,WB}$ | : | Linear horizontal wave bending moment calculated by hydrodynamic analysis |
| M_{SO} | : | Ship rule design still water moment |
| M_{WB} | : | Linear vertical wave bending moment calculated by hydrodynamic analysis |
| P_C | : | Resulting force due to difference between weight of cargo in tank and buoyancy along the tank length, L_t , $[W_{CT} + W_{CWBT} - 1.025 \cdot (b \cdot L_t \cdot T_{mean})] \cdot 9.81$ |
| Q | : | Shear force |
| $Q_{S,C}$ | : | Design still water shear force including shear force correction |
| $Q_{s,c}$ | : | Absolute value of maximum still water shear force at the relevant section, including shear force correction |
| Q_S | : | Uncorrected still water shear force, shear force normally found in the loading manual |
| Q_{total} | : | Total shear force |
| Q_W | : | Wave shear force |
| S | : | First moment of area |
| S_N/I_N | : | Value valid for neutral axis |
| W_{CT} | : | Cargo weight in centre tank |
| W_{CWBT} | : | Weight of ballast water in double bottom between longitudinal bulkheads |
| a_l | : | Longitudinal acceleration in the middle of the relevant tank calculated by hydrodynamic analysis with probability of exceedance same as for a_v |
| a_t | : | Transverse acceleration calculated at the relevant C.O.G. by hydrodynamic analysis with probability of exceedance same as for a_v |
| a_t | : | Transverse acceleration in the middle of the relevant tank calculated by hydrodynamic analysis with probability of exceedance same as for a_v |
| a_v | : | Vertical acceleration in the middle of the relevant tank calculated by hydrodynamic analysis |
| b | : | Breadth of centre tank |
| g | : | Acceleration of gravity |
| h_s | : | Vertical distance from the point considered to the top of the tank or air pipe |
| k_{sm} | : | Reduction factor for M_{SO} at given x values |
| m | : | Mass of the structure/equipment |
| p_d | : | Linear hydrodynamic sea pressure calculated by hydrodynamic analysis |

| | | |
|------------------------|---|---|
| p_{dyn} | : | Linear hydrodynamic sea pressure calculated by hydrodynamic analysis |
| p_e | : | External pressure amplitude (half pressure range) related to the draught of the load condition considered |
| p_l | : | Pressure due to longitudinal acceleration |
| p_s | : | Hydrostatic pressure for the relevant load condition |
| p_t | : | Pressure due to transverse acceleration |
| p_v | : | Pressure due to gravity and vertical acceleration |
| q | : | Distributed load |
| r_d | : | Reduction of pressure amplitude in the surface zone due to intermediate wet and dry surfaces |
| t | : | Thickness |
| x | : | Distance from AP |
| x_s | : | Longitudinal distance from the centre of free surface of liquid in tank to the pressure point considered |
| y_s | : | Transverse distance from the centre of free surface of liquid in tank to the pressure point considered |
| θ | : | Roll angle of ship |
| ϕ | : | Pitch angle |
| γ_{nc} | : | Non linear correction factor for vertical design bending moment |
| $\gamma_{\text{nc},s}$ | : | Non linear correction factor for global shear force |
| γ_s | : | Load factor for still water loads |
| γ_w | : | Environmental load factor |
| ρ | : | Density of ballast, bunkers and liquid cargo in tanks |

3. STILLWATER LOADS

Static loads may be referred to as *still water loads*. The still water loads acting on a hull due to buoyancy and weights will cause a global bending moment and shear force in the hull girder. The hydrostatic external sea pressure and internal tank pressure will also result in local response of the plates, stiffeners and girders.

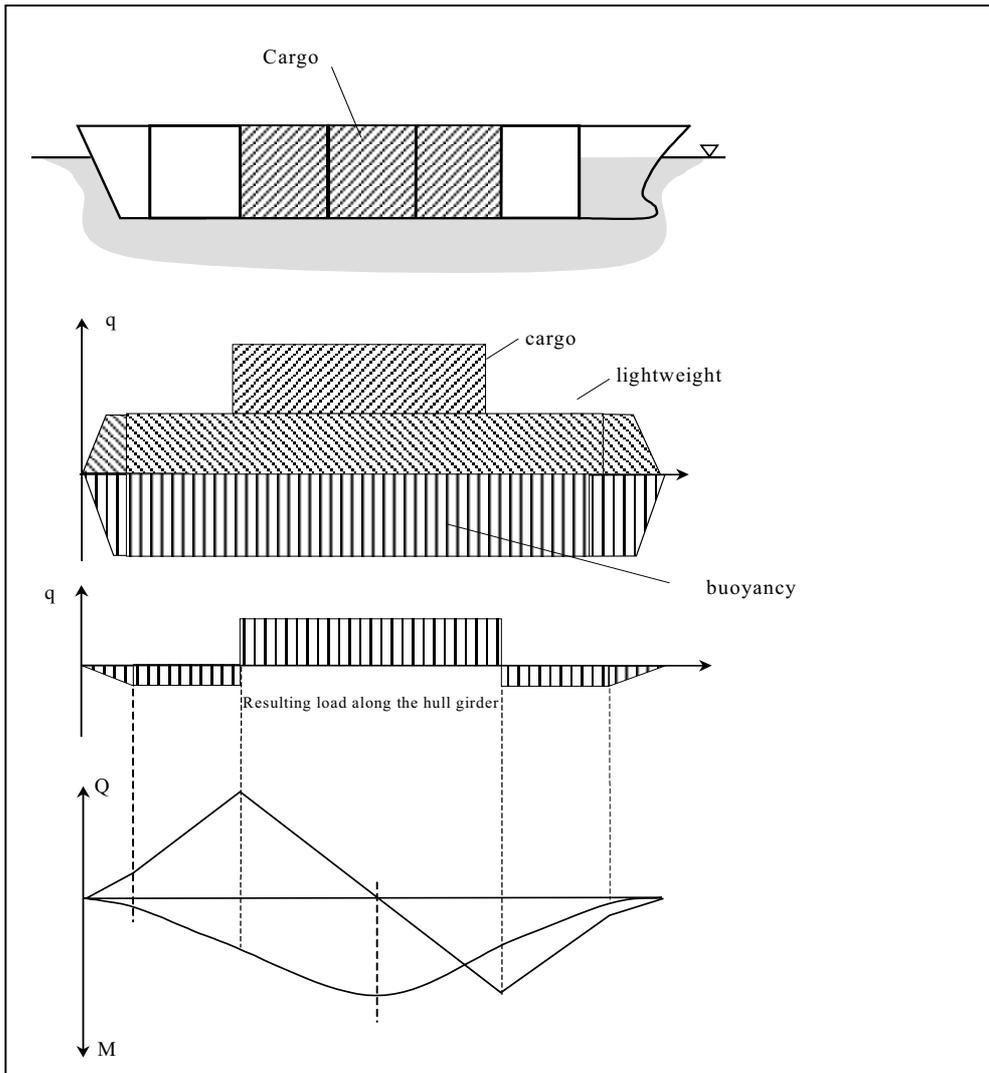


Figure B 1 Loads, bending moments and shear force distribution

3.1. Permanent Loads

Permanent loads will not vary in magnitude, position or direction during the time considered. Permanent loads relevant for offshore ships are:

- lightweight of the unit, incl. permanently installed modules and equipment, such as superstructure, propulsion (thruster), drilling/production equipment, helicopter deck, cranes, foundations
- mass of mooring lines and risers

3.2. Variable Loads

Variable functional loads may vary in magnitude, position and direction during the time considered. Typical variable functional loads are:

- buoyancy
- crude oil
- ballast water
- fuel oil
- consumables
- personnel
- general cargo and stored materials
- riser tension
- helicopter
- fendering and mooring of vessel.
-

Comment:

The domination loads for an FPSO/FSU are crude oil (deadweight) and the ballast water.

3.3. Stillwater bending moment and shear force

The hull beam is in equilibrium implying that the buoyancy forces are equal to the gravity forces from the hull lightship weight and dead weight. The static weight and buoyancy distribution along the hull girder will result in bending moments and shear forces as illustrated in principle in Figure B 1. The curves follow the sign conversions of Q_s and M_s as given in Rules for Classification of Ships, Pt.3 Ch.1. i.e.:

- surplus in weight over the tank length leads to a positive inclination of the shear force curve, while surplus of buoyancy gives a negative inclination of the curve.
- the moment curve is drawn on the negative side since the unit is shown in a sagging condition.

The shear forces and bending moments shown are derived from eq. (B 1) and (B 2), respectively.

The shear force is expressed by:

$$Q = \int_{x=0}^{x=FP} q \, dx \quad [kN] \quad (\text{B } 1)$$

q = resulting load distribution

x = distance from AP

The bending moment is expressed by:

$$M = \int_{x=0}^{x=FP} Q \, dx \quad [kNm] \quad (\text{B } 2)$$

Q = resulting shear force

x = distance from AP

All relevant loading conditions as well as permissible limits of still water bending moment and shear force and shear force correction values should be given in the loading manual. These curves should reflect all relevant modes of operation. This will be further discussed in the following sections.

3.3.1. Still water bending moment

The design still water bending moment at any position along the length of the unit is in accordance to Rules for Classification of Ships, Pt.3 Ch.1 normally not to be taken less than:

$$M_S = k_{sm} \cdot M_{SO} \quad [kN] \quad (\text{B } 3)$$

$$M_{SO} = -0.065 C_{wu} L^2 B (C_B + 0.7) \quad [kNm] \text{ in sagging}$$

$$C_{wu} L^2 B (0.1225 - 0.015C_B) \quad [kNm] \text{ in hogging}$$

$$k_{sm} = 1.0 \text{ within } 0.4 L \text{ amidships}$$

$$= 0.15 \text{ at } 0.1 L \text{ from AP or FP}$$

$$= 0.0 \text{ at AP and FP}$$

$$C_{wu} = C_w \text{ for unrestricted service}$$

$$= 0.0792 L \quad \text{for } L \leq 100$$

$$= 10.75 - [(300 - L) / 100]^{3/2} \quad \text{for } 100 < L < 300$$

$$= 10.75 \quad \text{for } 300 \leq L \leq 350$$

$$= 10.75 - [(L - 350) / 150]^{3/2} \quad \text{for } 350 < L$$

The envelope curve for still water bending moment based on Rules for Classification of Ships is shown in Figure B 2.

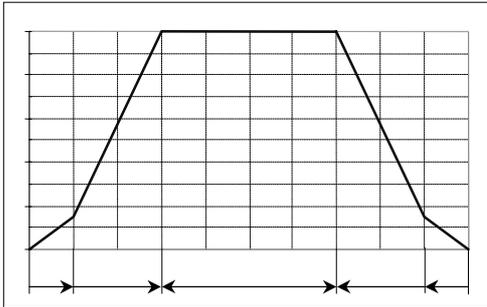


Figure B 2 Still water bending moment envelope

The limit curve for the still water bending moment may be based on the actual load conditions provided that all relevant conditions are considered and given in the loading manual. The envelope curve given by the Rules for Classification of Ships could be used in the initial design phase when no detailed information is available.

3.3.2. Still water limit curves based on actual loading conditions

The maximum still water bending moment for actual loading conditions may exceed the Rule values as given by equation (B 3). An example of these areas is indicated in Figure B 3.

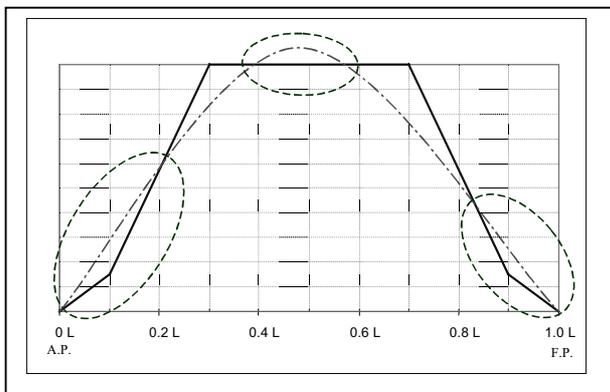


Figure B 3 Homogeneous load condition

In this example the still water bending moment curve is shown for a homogeneous load condition in hogging. The limit curve could, in this case, be taken as the envelope curve of the two given curves in the figure.

If all relevant loading conditions prove that the Rule limit curve is never reached, the Rule value within $0.4L$ may be reduced to the actual value, but by maximum 50%.

3.3.3. Still water shear force

The same principles as for the still water bending moment apply for the still water shear forces. The shear force at any position along the length of the unit is, according Rules for Classification of Ships, normally not to be taken less than:

$$Q_s = k_{sq} \cdot 5 \frac{M_{SO}}{L} \quad [kNm] \quad (\text{B } 4)$$

M_{SO} = design still water bending moment (sagging or hogging) given in equation (B 4)

k_{sq} = 0.0 at A.P. and F.P.

= 1.0 between 0.15 L and 0.3 L from A.P.

= 0.8 between 0.4 L and 0.6 L from A.P.

= 1.0 between 0.7 L and 0.85 L from A.P.

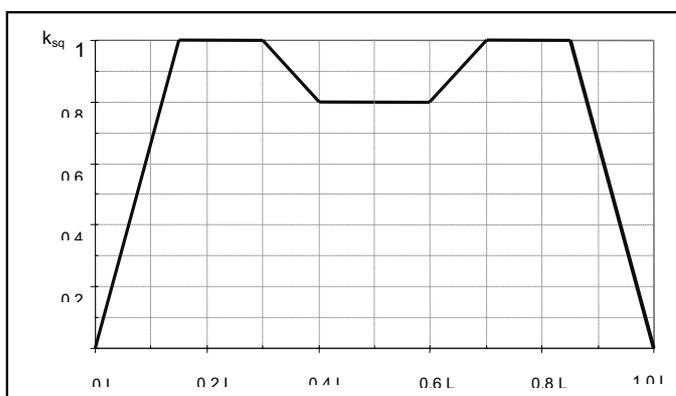


Figure B 4 Still water shear force

The still water shear force limit curve based Rules for Classification of Ships shown in Figure B 4 is normally used in the initial design phase if no detailed information of actual load conditions is available.

The limit curve for the still water shear force could be based on the actual load conditions in the same manner as described for the bending moment.

The shear force distribution along the hull girder as given in Figure B 4 would be true if the main shear panels in the hull, i.e. side, inner side and longitudinal bulkheads, experienced a uniform loads distribution from the vertical loads acting on the bottom structure. This would be the case if there were only transverse web frames/floors and no longitudinal side girders (see the left bottom arrangement in Figure B 5) as the loads on the bottom structure is carried out to the inner side/side and longitudinal bulkheads. If the bottom arrangement is similar to the arrangement shown to the right in Figure B 5, part of the bottom load will be transferred to the transverse bulkheads, and this effect must be accounted for by means of shear correction factors.

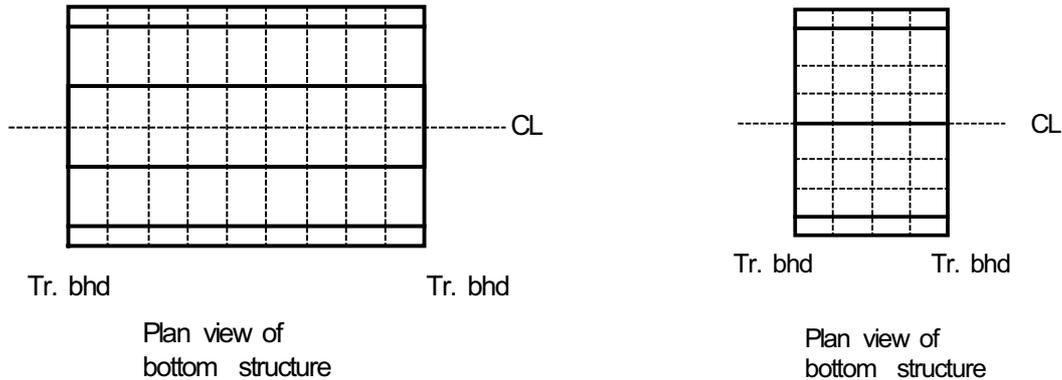


Figure B 5 Bottom structural arrangement

The shear force distribution among the shear carrying elements (side, inner side, longitudinal bulkhead and longitudinal girders) at any transverse section is normally derived from Rules for Classification of Ships. Alternatively, the shear force distribution factor can be calculated based on a shear flow analysis.

3.3.4. Shear force transverse distribution

A typical shear force distribution at a given transverse section derived from a shear flow calculation is shown in Figure B 6. The shear force distribution based on shear flow analysis does not consider the local distribution of the vertical force acting on the bottom structure between transverse bulkheads. If the bottom structure comprises longitudinal girders, some of the force will be transmitted directly to the transverse bulkheads. This effect is referred to as the 3D-effect and must be accounted for in the longitudinal shear force distribution. The theoretical shear force distribution in a transverse cross section is shown in Figure B 7.

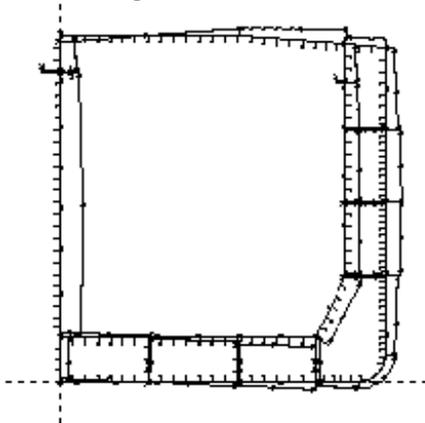


Figure B 6 Typical shear force distribution

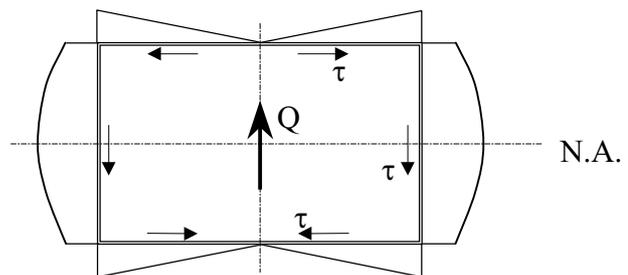


Figure B 7 Shear force distribution

3.3.5. Shear force correction

For ships with several shear carrying elements as described above the 3-D effect of the load distribution on the bottom structure must be considered in a shear flow analysis.

The definition of shear stress is described as follows:

$$\tau = \frac{Q_{total} \cdot S}{t \cdot I} \cdot 10^2 \quad [N/mm^2] \quad (\mathbf{B} \ 5)$$

where:

- Q_{total} = total shear force [kN]
- S = first moment of area [cm³]
- t = thickness [mm]
- I = moment of inertia [cm⁴]

S_N is the first moment of area of the longitudinal section above and below the horizontal neutral axis, and I_N is the moment of inertia about the horizontal neutral axis. I_N/S_N is the maximum value at the neutral axis. In the initial design stage, a value of 90D may be used.

The total shear force is the sum of the static and dynamic shear force ($Q_s + Q_w$). The total shear stress may then be written as:

$$\tau = \frac{Q_{s,c} + Q_w}{t} \cdot \frac{S_N}{I_N} \cdot 10^2 \quad [N/mm^2] \quad (\mathbf{B} \ 6)$$

where:

- $Q_{s,c}$ = design still water shear force including shear force correction [kN]
- Q_w = wave shear force [kN]
- S_N/I_N = value valid for neutral axis [cm⁻¹]

The corrected still water shear force is expressed by:

$$Q_{s,c} = Q_s \pm K \cdot P_C \quad (\mathbf{B} \ 7)$$

where:

- Q_s = uncorrected still water shear force, shear force normally found in the loading manual [kN]
- K = shear force correction factor [-]

- P_C = resulting force due to difference between weight of cargo in tank and buoyancy along the tank length, L_t , given in [kN]
 = $[W_{CT} + W_{CWBT} - 1.025 \cdot (b \cdot L_t \cdot T_{mean})] \cdot 9.81$, see Figure B 8
- W_{CT} = cargo mass in centre tank [ton]
- W_{CWBT} = mass of ballast water in double bottom between longitudinal bulkheads (for single skin vessels $W_{CWBT} = 0$) [ton]
- b = breadth of centre tank [m]
- L_t = length of centre cargo tank [m]

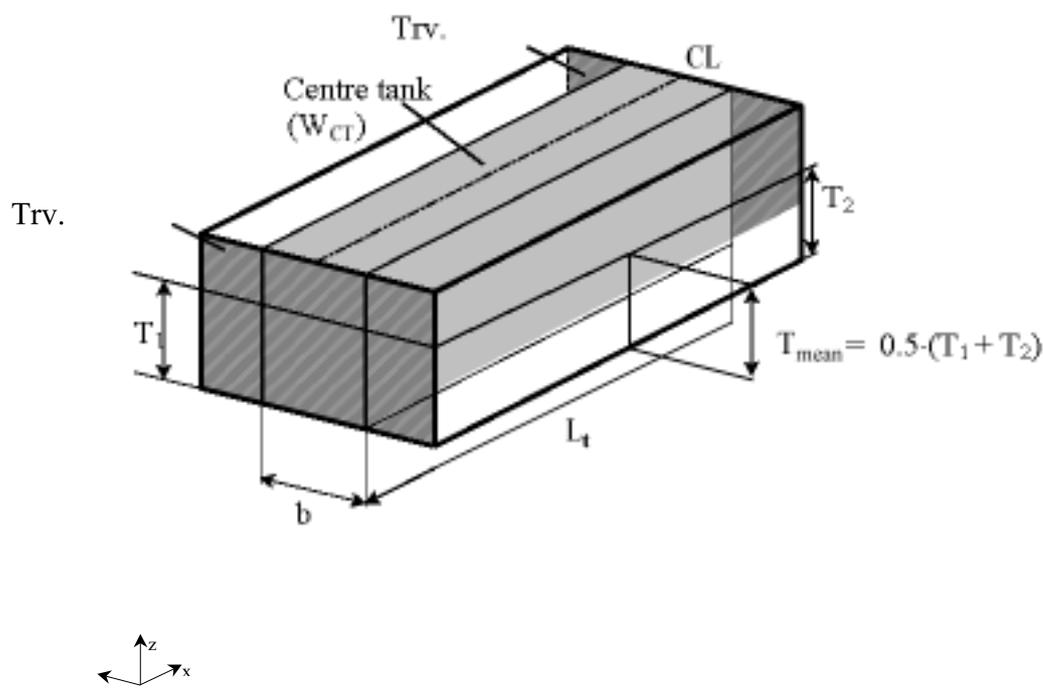


Figure B 8 Dimensions included in the calculation of P_C

The shear force distribution for effective longitudinal shear carrying elements in the hull girder (e.g. ship side and longitudinal bulkhead) is given by a shear force distribution factor, Φ , derived from *Section Scantlings*. In the initial stage the shear force distribution factor can be taken as given in Rules for Classification of Ships, Pt.3 Ch.1 Sec.5 D103, table D1.

By paraphrasing formula (B 6) the thickness requirement for side shell or longitudinal bulkhead as given in Rules for Classification of Ships, Pt.3 Ch.1 Sec.5 D103 can be derived:

$$t = \frac{|\Phi(Q_S + Q_w) + 0.5\Delta Q_S|}{\tau} \cdot \frac{S_N}{I_N} \cdot 10^2 \quad [mm] \quad (\mathbf{B\ 8})$$

where: $0.5 \cdot \Delta Q_S = K \cdot P_C \cdot \Phi$

Formula (B 6) can also be written as:

$$Q_{S,C} = \frac{t\tau}{100} \cdot \frac{I_N}{S_N} - Q_w \quad [kN] \quad (\mathbf{B\ 9})$$

By including the shear force distribution factor (Φ) and the expression for corrected shear force ($Q_{S,C} = Q_S \pm K \cdot P_C$) equation (B 6) may be written as:

$$Q_S + K \cdot P_C = \frac{t\tau}{100\Phi_{Side}} \cdot \frac{I_N}{S_N} - Q_w \quad [kN] \quad (\mathbf{B\ 10})$$

The *left hand side* of equation (B 9) and (B 10) shall be considered as “actual corrected still water shear force”, see equation (B 7).

The *right hand side* of equation (B 9) and (B 10) shall be considered as “allowable still water shear force”. This is based on the actual thickness along the side or longitudinal bulkhead and the allowable shear stresses.

By drawing one curve for the “actual corrected still water shear force” and one curve for “allowable still water shear force” along the ship length it is possible to detect if the “actual corrected still water shear force” curve exceeds the “allowable still water shear force” curve.

The following example is applicable for ship units with two longitudinal bulkheads (see Rules for Classification of Ships, Pt.3 Ch.1 Sec.5 D300). The basic principles are approximately the same for other ship types with different number of longitudinal bulkheads. See also Classification Notes 31.3 - *Strength Analysis of Hull Structure in Tankers* for further information.

The tank fillings for the load condition are shown in the example in Figure B 9.

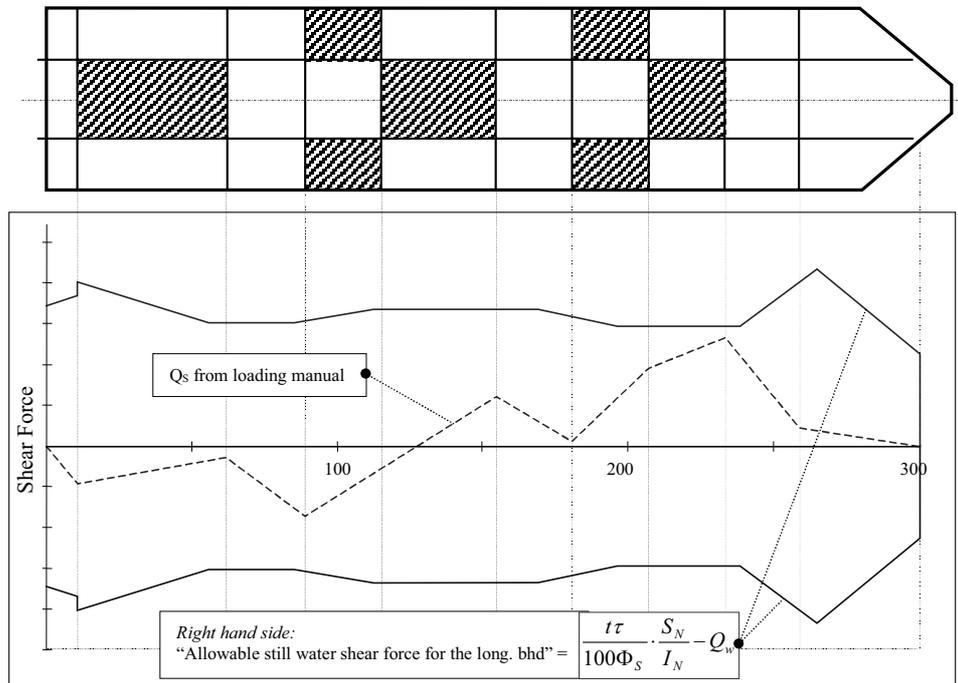


Figure B 9 Actual shear force curve from loading manual and allowable shear force curve based on thicknesses in longitudinal bulkhead

The diagram in Figure B 9 shows the still water shear force, Q_s , for one load condition taken from the loading manual. This curve does not include the shear force correction ($K \cdot P_C$). I.e. the *left hand side* of equation (B 10) is only represented by the “actual still water shear force”, Q_s , and hence the shear force correction ($K \cdot P_C$) must be included.

As may be seen from the “actual still water shear force”-curve in Figure B 9 a surplus in weight over the tank length leads to a positive inclination of the shear force curve, while a surplus of buoyancy gives a negative inclination of the curve.

The *right hand side* of equation (B 10) is represented in the diagram in

Figure B 10 as “allowable still water shear force for the longitudinal bulkhead”. I.e. that the shear force distribution factor (Φ_L) and the thickness (t) for the longitudinal bulkhead are included. The curve is drawn as an envelope curve (limit curve) on both the positive and negative side. I.e. the actual shear force given in the loading manual including the shear force correction (i.e. $Q_{s,C} = Q_s \pm K \cdot P_C$) shall not exceed these limit curves.

3.3.6. Shear force correction factor

The shear force correction factor, K, for ship units with two longitudinal bulkheads may be calculated as follows:

$$K_S = \left[0.5 \left(1 - \frac{s}{l_c} \right) (1 - C_T) \left(\frac{r}{r+1} \right) - \Phi_S \right] \frac{0.5}{\Phi_S} \quad (\text{B } 11)$$

and

$$K_L = \left[0.5 \left(1 - \frac{s}{l_c} \right) (1 - C_T) \left(\frac{1}{r+1} \right) - \Phi_L \right] \frac{0.5}{\Phi_L} \quad (\text{B } 12)$$

where:

- K_S = Shear force correction factor for ship side
- K_L = Shear force correction factor for longitudinal bulkhead
- $(1 - C_T)$ = Fraction of P_C transferred to LBHD and double side without going through the transverse girders
- $(1 - s/l_c)$ = Fraction of P_C going through the main girder system. s/l_c is the fraction of P_C that is “transferred” directly to the transverse bulkhead (i.e. carried by bottom and inner bottom stiffeners and not going through the main girder system)
- $r / (r+1)$ = Fraction of P_C to be transferred to the double side (r is the fraction of the load transferred from LBHD to double side and depends on the stiffener of wing tank structure)
- $1 / (r+1)$ = Fraction of P_C to remain in LBHD

(For information regarding other tank arrangements see Classification Notes 31.3 - *Strength Analysis of Hull Structure in Tankers*)

The shear force correction ($K \cdot PC$) is calculated for each tank. The plus or minus sign before the $K \cdot PC$ -term at the right hand side of equation (B 7) depends on whether the inclination of the shear force curve increases or decreases due to the loading in the centre tank. This relation is shown in Rules for Classification of Ships, Pt.3 Ch.1 Sec.5 D300, table D2. These principles are further explained in the table in Figure B 10.

By including the shear force correction for each tank ($K_L \cdot PC$), the “corrected shear force”-curve for the longitudinal bulkhead ($Q_{S,C} = Q_S \pm K_L \cdot PC$) can be compared with “the allowable still water shear force” curve for the longitudinal bulkhead as shown Figure B 10. In this example the corrected shear force in the longitudinal bulkhead exceeds the allowable shear force (indicated in Figure B 10). This indicates that the thickness in the bulkhead must be increased in order to satisfy the shear force capacity requirements.

The same principles shall be applied for the ship side. The “actual corrected still water shear force” curve for the ship side should then include the shear force correction factor (K_S) for each tank, and the “allowable still water shear force” curve for the ship side should include thicknesses for the ship side.

This example emphasises the importance of calculating the shear force correction based on each load condition given in the loading manual. Experience shows that the actual loading conditions quite frequently contain higher values than given for Q_S by Rules for Classification of Ships.

When the shear force correction is calculated for each load condition, the maximum shear force values at each bulkhead (longitudinal bulkhead and ship side) can be determined. A limit curve for the maximum acting shear force can then be calculated and the curve will be the basis for the design of the hull structure. The curve will inter alia be used in evaluation of the yield and buckling strength of global shear carrying elements.

| Shear force correction for longitudinal bulkhead (Rules for Classification of Ships, Pt.3 Ch.1 Sec.5 D300, table D2) | | | | | | | | | | |
|---|------------|------------------|------------|-------------|------------------|------------|-------------|-------------|------------|--------------------|
| The ship has over the length | e.o. buoy. | excess of weight | e.o. buoy. | e.o. weight | excess of weight | e.o. buoy. | e.o. weight | e.o. weight | e.o. buoy. | excess of buoyancy |
| The centre tank has over the length | e.o. buoy. | excess of weight | e.o. buoy. | e.o. buoy. | excess of weight | e.o. buoy. | e.o. buoy. | e.o. weight | e.o. buoy. | excess of buoyancy |
| Inclination of shear force curve | incr. | increases | incr. | decr. | increases | incr. | decr. | incr. | incr. | increases |

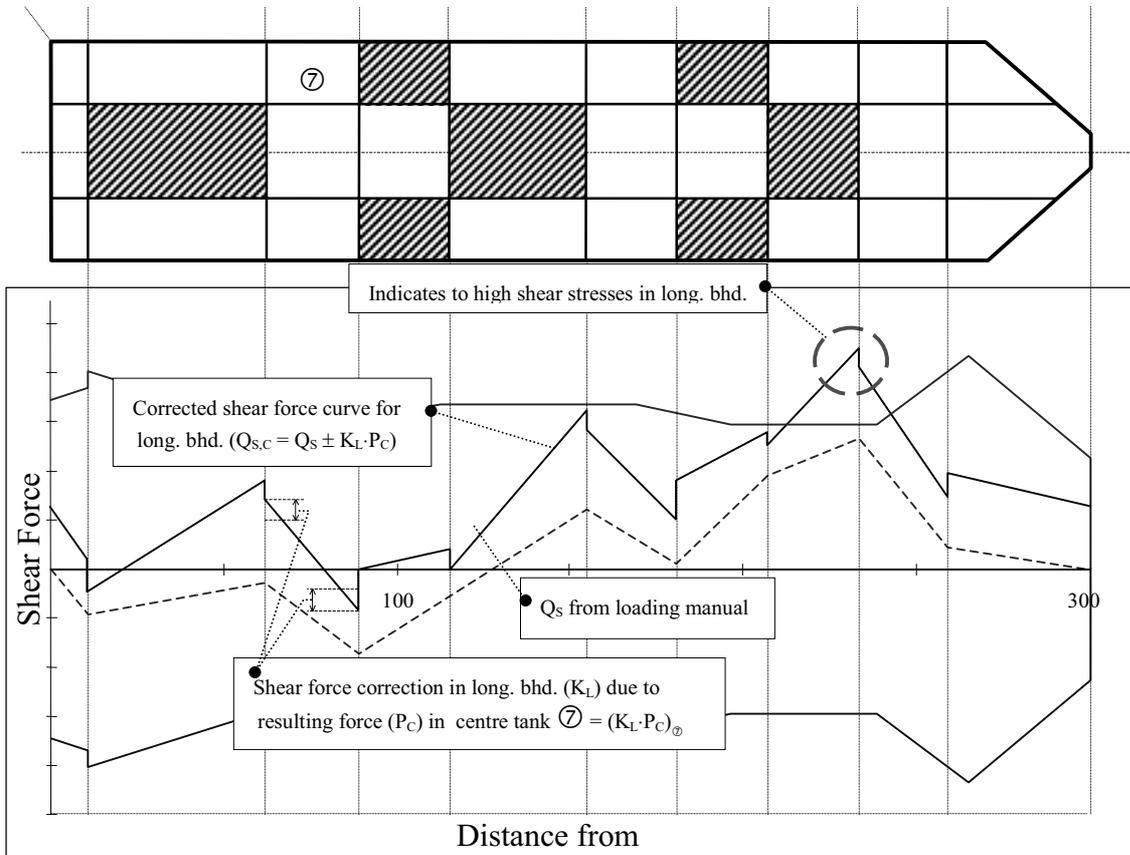


Figure B 10 Actual shear force curve from loading manual and allowable shear force curve based on thicknesses in longitudinal bulkhead

4. ENVIRONMENTAL LOADS

4.1. General

Environmental loads are loads caused by waves, wind, current, ice, etc. and covers natural phenomena which contribute to the structural response. Phenomena of general importance for offshore ships are:

- wave
- wind
- current
- sloshing in tanks
- green water on deck
- slamming (e.g. on bow and bottom in fore and aft ship)
- vortex induced vibrations (e.g. resulting from wind loads on structural elements in a flare tower)

Phenomena which may be important in certain cases are:

- temperature
- snow and ice

The long-term variation of the environmental phenomena such as wind, waves and current shall be described by statistical distributions relevant for the environmental parameters considered.

4.2. Wave induced forces

The following load responses shall usually be calculated by the hydrodynamic wave load analysis:

- vertical bending moment
- horizontal bending moment
- global shear force
- external sea pressure distribution
- accelerations
- global axial force
- torsional moment (only if relevant)

The procedure for wave load analysis is described in Appendix C.

The long-term responses are calculated by combining the transfer functions with a wave spectrum (scatter diagram). The joint probability of the significant wave heights (H_s) and mean wave periods (T_Z) (or peak periods (T_p)) is combined with the short-term prediction of the response.

A Weibull distribution is usually fitted to the resulting distribution of the response against probability of exceedance. For ULS capacity checks, the Weibull distribution is based on the environmental conditions with 100 years return period, whereas a probability of exceedance of 10^{-4} is used for fatigue capacity calculations (correspond approximately to daily return period).

Comment:

Fatigue calculations based on simplified method should preferably be based on 10^{-2} probability level of occurrence, but 10^{-4} is often used as the Ship Rule local loads and tables with allowable stresses are based on such probability level of occurrence.

Units that intend to operate worldwide are in general to be designed based on the *North Atlantic* scatter diagram for the ULS and the FLS conditions. However, if the unit is intended to stay on the same location during the whole operating life, it shall be based on a site specific scatter diagram.

4.3. Wind

Wind loads are to be determined by relevant analytical methods. A model test may also be carried out. Dynamic effects of the wind are to be considered for structures which are sensitive to wind loads. Typical structural parts for an offshore vessel is topside modules, flare boom, derrick, etc.

Reference is made to Classification Notes 30.5 - *Environmental Conditions and Environmental Loads* for further information.

4.4. Sloshing loads in tanks

In case of no restrictions on partly filling of cargo tanks, the phenomenon of resonant liquid motion (sloshing) inside the tanks shall be considered. The pressure fields created by sloshing of the cargo/ballast shall be calculated according to the requirements given in Rules for Classification of Ships Pt.3 Ch.1 Sec.4 C300 - *Liquid in tanks*.

5. COMBINATION OF STILL WATER AND ENVIRONMENTAL LOADS

5.1. General

This section intends to describe the combination of static and environmental loads according to the LRFD format.

For the fatigue evaluation the static part of the following expressions will obviously be disregarded. The FLS loads will be calculated with a probability of exceedance of 10^{-4} (approx. daily return period)

The procedure for the ultimate strength analysis (ULS) is shown in Appendix D and for fatigue analysis in Appendix E. Descriptions of the determination of the load factors are outlined in the relevant appendices.

5.2. Global bending moments, shear force and axial force

5.2.1. Vertical bending moment

The total vertical bending moment is calculated as follows:

$$M_{\text{tot}} = \gamma_s \cdot M_s + \gamma_w \cdot \gamma_{\text{nc}} \cdot M_{\text{WB}}$$

where:

- M_s = Absolute value of maximum still water bending moment at the relevant section
- M_{WB} = Linear vertical wave bending moment calculated by hydrodynamic analysis:
ULS: Annual probability of exceedance 10^{-2} (100 years return period)
FLS: Probability of exceedance of 10^{-4} (approx. daily return period)
- γ_s = Load factor for still water loads
- γ_w = Environmental load factor
- γ_{nc} = Non linear correction factor for vertical design bending moment

The values of the load factors for the different limit states shall be in accordance with Appendix F and G.

5.2.2. Horizontal bending moment

The horizontal bending moment is calculated as follows:

$$M_{\text{h,tot}} = \gamma_w \cdot M_{\text{h,WB}}$$

where:

- $M_{h,WB}$ = Linear horizontal wave bending moment calculated by hydrodynamic analysis:
ULS: Annual probability of exceedance 10^{-2} (100 years return period)
FLS: Probability of exceedance of 10^{-4} (approx. daily return period)
- γ_w = Environmental load factor, similar to the vertical bending moment

5.2.3. Global Shear force

The total shear force is calculated as follows:

$$Q_{tot} = \gamma_s \cdot Q_{s,c} + \gamma_w \cdot \gamma_{nc,s} \cdot Q_{WB}$$

where:

- $Q_{s,c}$ = Absolute value of maximum still water shear force at the relevant section, including shear force correction
- Q_{WB} = Linear wave shear force calculated by hydrodynamic analysis:
ULS: Annual probability of exceedance 10^{-2} (100 years return period)
FLS: Probability of exceedance of 10^{-4} (approx. daily return period)
- γ_s = Load factor for still water loads, similar to the still water bending moment
- γ_w = Environmental load factor, similar to the wave bending moment
- $\gamma_{nc,s}$ = Non linear correction factor for global shear force

Note that the load factors will have the same values as for the global bending moment. The acceptance criteria for global shear is calculated in the same way as the global bending moment.

5.2.4. Axial force

The same principles are applied for the axial force:

$$F_{tot} = \gamma_w \cdot F_{WB}$$

where:

- F_{WB} = Linear axial force calculated by hydrodynamic analysis:
ULS: Annual probability of exceedance 10^{-2} (100 years return period)
FLS: Probability of exceedance of 10^{-4} (approx. daily return period)
- γ_w = Environmental load factor, similar to the wave bending moment

5.3. External sea pressure and internal tank pressure

5.3.1. External sea pressure

The external sea pressure distribution is defined as normal pressure with an annual probability of exceedance equal to 10^{-2} (100 years return period) for ULS. Probability of exceedance of 10^{-4} (approx. daily return period) is used for the FLS.

The total external design pressure is calculated as follows:

$$\text{ULS: } p_{\text{tot}} = \gamma_s \cdot p_{\text{static}} + \gamma_w \cdot p_{\text{dyn}}$$

$$\text{FLS: } p_{\text{tot}} = \gamma_w \cdot r_d \cdot p_{\text{dyn}}$$

where:

| | | |
|---------------------|---|---|
| p_{static} | = | Hydrostatic pressure for the relevant load condition |
| p_{dyn} | = | Linear hydrodynamic sea pressure calculated by hydrodynamic analysis |
| γ_s | = | Load factor for still water loads |
| γ_w | = | Environmental load factor |
| r_d | = | Reduction of pressure amplitude in the surface zone due to intermediate wet and dry surfaces (ref. section 4.3 in Classification Notes 30.7 - <i>Fatigue Assessment of Ship Structure</i>) |

In case of a quasi-static response analysis for simplified approach, the external pressure distribution for all load conditions and limit states is assumed to be constant in the longitudinal direction, i.e. the actual wave profile and elevation is not taken into consideration. This assumption of constant pressure distribution in the longitudinal direction is regarded as a conservative assumption.

Any phase angles between external sea pressure and bending moments are disregarded. Hence, it is assumed that maximum external sea pressure and design bending moment occurs at the same time.

The load factors for the external pressure are the same as for the global vertical bending moment.

Linear distribution of the hydrodynamic pressure may be assumed in the structural analysis as a simplified (and conservative) method.

5.3.2. ULS Condition

The evaluation of the ultimate strength of the hull girder is based on direct calculations of the environmental loads for normally two load conditions, the fully loaded condition and the ballast condition. These two conditions result in maximum global bending moment and will be applied in buckling/yield check of longitudinal strength elements.

Sagging Condition

The wave profile is for the sagging condition assumed to be as shown in Figure B 11, with a wave trough amidship.

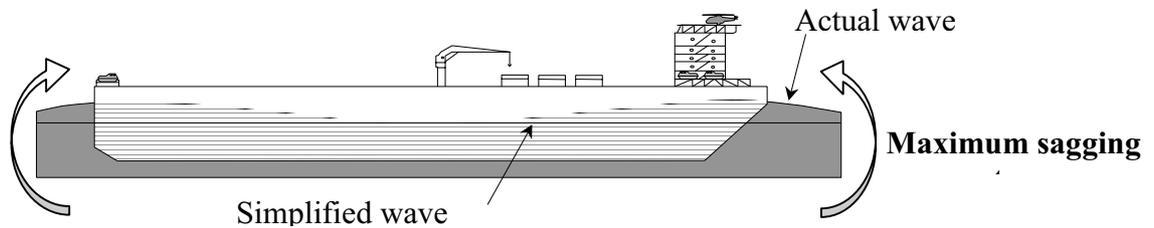


Figure B 11 Sagging condition

The dynamic pressure for a cross section amidships is usually to be obtained from a linear 3D sink-source program (e.g. WADAM). The wave amplitude is calculated based on dynamic pressure at the waterline:

$$\text{Wave amplitude} = h/2 = \frac{\gamma_w \cdot P_{dynWL}}{\rho g}$$

Pressure variation under the wave crest and the wave trough according to linear wave theory is shown in Figure B 12. The “hydrostatic” pressure “ $-\rho g z$ ” should cancel the dynamic pressure at the free surface. Under a wave trough there is a higher-order error. By “higher-order error” it means that the error is approximately proportional to z_a^n , where the order $n \geq 2$ (z_a = wave amplitude), see Figure B 12.

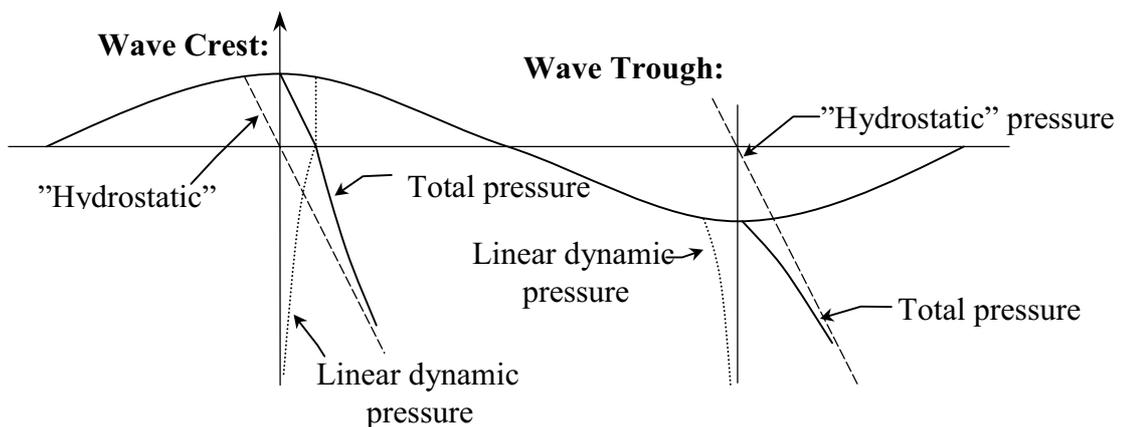


Figure B 12 Pressure variation under a wave crest and a wave trough according to linear theory

However, it is an accepted method to use the “zero point” as the free surface and calculate the total external design pressure by using the following formula:

$$p_{tot} = \gamma_s \cdot p_{static} - \gamma_w \cdot p_{dyn, 100 \text{ years}}$$

Since the area of interest is above the neutral axis, this small error will not affect the results significantly.

Figure B 13 shows how the total external pressure in the sagging condition may be calculated. Reference is also made to Figure B 17 for the distribution of the external and internal pressure in sagging condition.

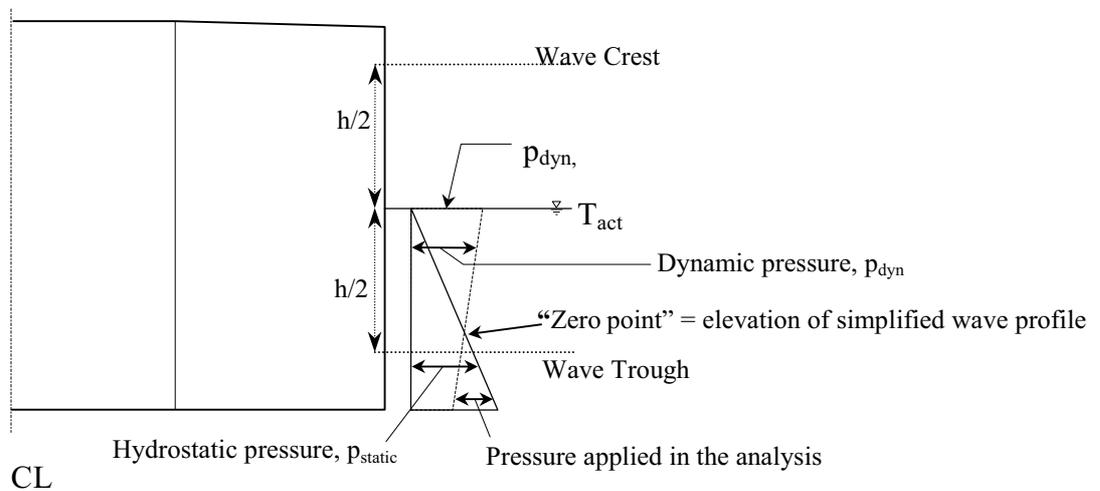


Figure B 13 External Pressures – Sagging Condition (ULS) Wave trough

Hogging Condition

The wave profile is for the hogging condition assumed to be as shown in Figure B 14, with a wave crest amidship.

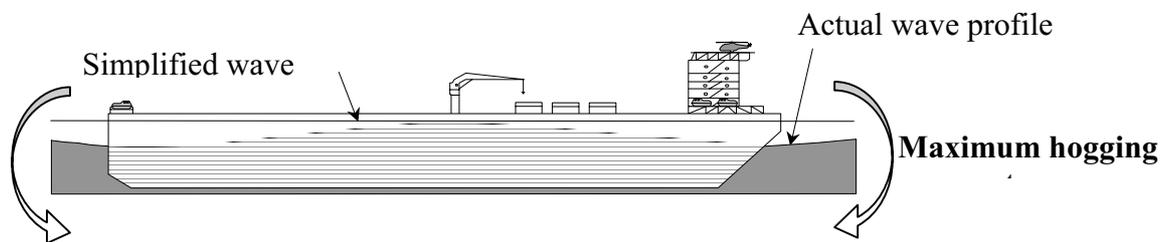


Figure B 14 Hogging condition

The dynamic pressure and the wave amplitude are calculated in the same way as for the sagging condition.

Pressure variation under a wave crest according to linear wave theory is shown in Figure B 15. The “hydrostatic” pressure “ $-\rho g z$ ” cancel the dynamic pressure at the free surface for the wave crest.

The total external design pressure for the hogging condition is calculated by using the following formula:

$$p_{tot} = \gamma_s \cdot p_{static} + \gamma_w \cdot p_{dyn, 100 \text{ years}}$$

The total external pressure in the hogging condition is shown in Figure B 15.

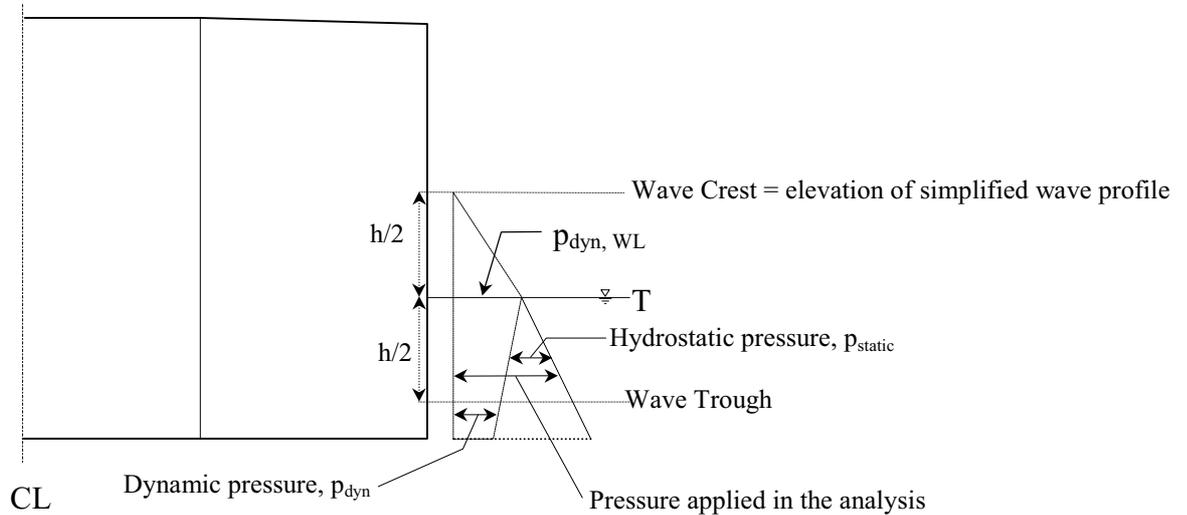


Figure B 15 External Pressures – Hogging Condition (ULS) Wave crest

The distribution of external and internal pressure for hogging condition for the ULS is shown in Figure B 18.

5.3.3. FLS Condition

As for the ULS condition, the evaluation of the FLS condition is based on direct calculations of the environmental loads for normally two load conditions, the fully loaded condition and the ballast condition. The external dynamic design pressure amplitude is to be according to section 4.3 in Classification Notes 30.7 - *Fatigue Assessment of Ship Structure* (except that pressure is calculated based on direct calculation instead of simplified calculation):

$$p_e = r_d p_{dyn}$$

where:

- p_e = External design pressure amplitude (half pressure range) related to the draught of the load condition considered
- r_d = Reduction of pressure amplitude in the surface zone due to intermediate wet and dry surfaces (see section 4.3 in Classification Notes 30.7)
- p_{dyn} = Linear hydrodynamic sea pressure calculated by hydrodynamic analysis with an probability of exceedance of 10^{-2} (approx. daily return period)

Note that the environmental load factor (γ_w) is set to 1.0. The distribution of external pressure amplitudes for the FLS condition is shown in Figure B 19.

5.4. Internal tank pressure

The internal design pressure distribution from liquid cargo or ballast water is to be based on the ship motion and are calculated as follows:

$$p_v = (\gamma_s \cdot g + \gamma_w \cdot a_v) \rho \cdot h_s$$

$$p_t = (\gamma_s \cdot g \cdot h_s + \gamma_w \cdot a_t \cdot y_s) \rho$$

$$p_l = (\gamma_s \cdot g \cdot h_s + \gamma_w \cdot a_l \cdot x_s) \rho$$

where:

- p_v = Pressure due to gravity and vertical acceleration
- p_t = Pressure due to transverse acceleration
- p_l = Pressure due to longitudinal acceleration
- γ_s = Load factor for static loads, similar to the still water bending moment
- γ_w = Environmental load factor, similar to the hydrodynamic bending moment
- g = Acceleration of gravity = 9.81 m/s^2
- ρ = Density of ballast, bunkers and liquid cargo in tanks (not to be taken less than 1.025 t/m^3 in the ULS conditions)
- a_v = Vertical acceleration in the middle of the relevant tank calculated by hydrodynamic analysis with:
ULS: Annual probability of exceedance 10^{-2} (100 years return period)
FLS: Probability of exceedance of 10^{-4} (approx. daily return period)
- a_t = Transverse acceleration in the middle of the relevant tank calculated by hydrodynamic analysis with probability of exceedance same as for a_v
- a_l = Longitudinal acceleration in the middle of the relevant tank calculated by hydrodynamic analysis with probability of exceedance same as for a_v
- h_s = Vertical distance from the point considered to the top of the tank or air pipe
- y_s = Transverse distance from the center of free surface of liquid in tank to the pressure point considered ¹⁾
- x_s = Longitudinal distance from the center of free surface of liquid in tank to the pressure point considered ¹⁾

¹⁾ The free surface is considered to be in the middle of the tank

The load factors for the internal pressure will be determined based on the same factors as the global vertical bending moment.

5.4.1. ULS Condition

The internal pressure in tanks is based on the maximum acceleration in the cargo area, with respect to longitudinal position. Any phase angles between accelerations and bending moments are disregarded. Hence, it is assumed that maximum acceleration and design bending moment occurs at the same time.

For the ULS condition it is normal practice to consider long crested head sea. The transverse acceleration will be more or less zero (no roll and sway motion). The longitudinal acceleration is also small. It is usually sufficient to only include the vertical acceleration induced by the heave and pitch motion:

$$p_v = (\gamma_s \cdot g + \gamma_w \cdot a_v) \rho \cdot h_s$$

The distribution of external and internal pressure for the ULS condition is shown in Figure B 17 and Figure B 18. Note that the internal pressure at the top of the tank may be > 0 , since the height of the air pipes should be included. The height of the air pipes should normally not be taken less than 0.76 m.

Air pipes

The following assumption is recommended if no efficient alarm system is provided and if the height of the air pipe is:

- not known, default value may be set to 0.76 m
- known, the possibility of having liquid in the air pipes should be considered. The whole height of the air pipe should then be included in the pressure height.

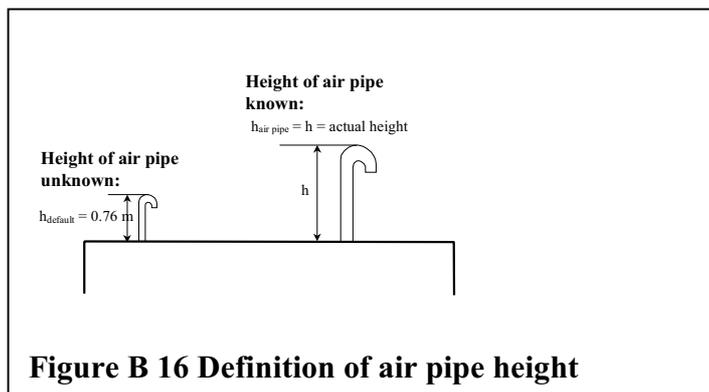


Figure B 16 Definition of air pipe height

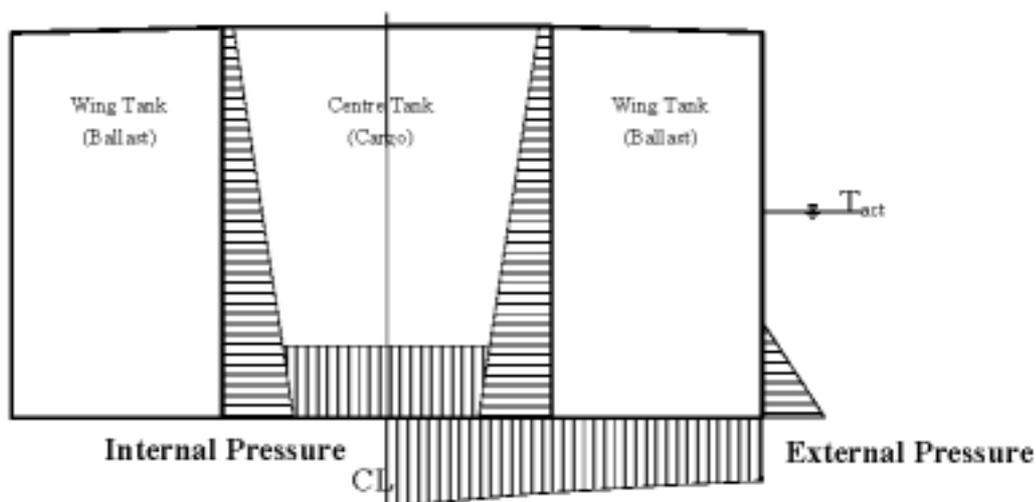


Figure B 17 External and internal pressure - Sagging Condition (ULS)

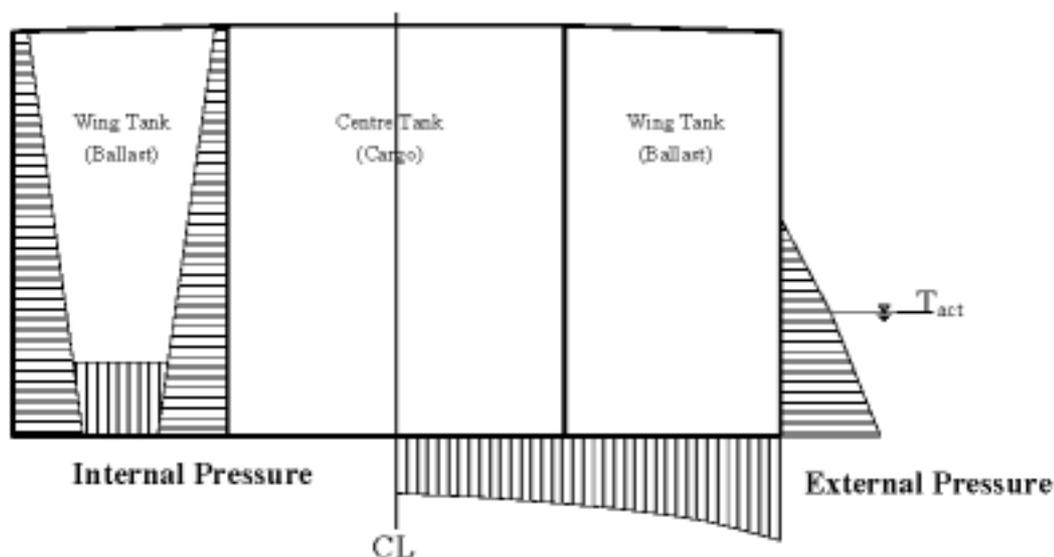


Figure B 18 External and internal pressure - Hogging Condition (ULS)

5.4.2. FLS Condition

The dynamic pressure from liquid cargo or ballast water for the FLS condition is defined in each tank according to the distribution from acceleration in longitudinal, transverse and vertical direction.

$$p_v = a_v \cdot \rho \cdot h_s$$

$$p_t = a_t \cdot \rho \cdot y_s$$

$$p_l = a_l \cdot \rho \cdot x_s$$

where:

- a_v = Unit vertical acceleration in the middle the tank
- a_t = Unit transverse acceleration in the middle of the tank
- a_l = Unit longitudinal acceleration in the middle of the tank

The environmental load factor (γ_w) is set to 1.0 in the FLS condition.

The internal pressure will be applied in the analysis. For each load case the stress per unit internal pressure will be combined with the relevant transfer function. The principles of combination of transfer function are shown in section 5.4 of Classification Notes 30.7 - *Fatigue Assessment of Ship Structure*.

Figure B 19 shows the distribution for the external and internal pressure (only internal pressure due to vertical acceleration shown) for the FLS condition.

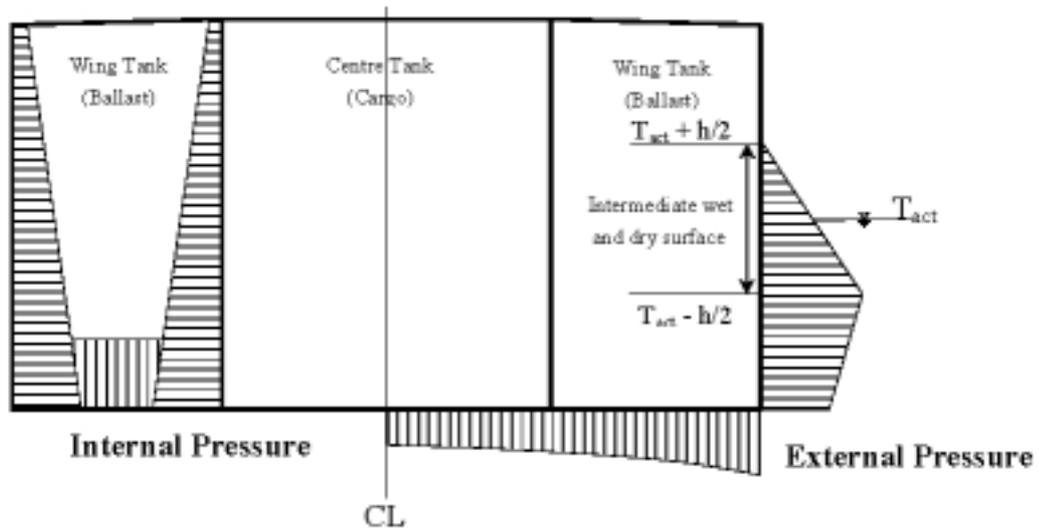


Figure B 19 External and internal pressure (FLS)

5.5. Induced Forces on Topping and Deck Equipment

The total design force for topside/deck equipment is calculated as follows:

$$F_v = (\gamma_s \cdot g + \gamma_w \cdot a_v) \cdot m$$

$$F_t = (g \sin \theta + \gamma_w \cdot a_t) \cdot m$$

$$F_l = (g \sin \phi + \gamma_w \cdot a_l) \cdot m$$

where:

| | | |
|------------|---|--|
| F_v | = | Force on the structure due to gravity and vertical acceleration |
| F_t | = | Force on the structure due to transverse acceleration |
| F_l | = | Force on the structure due to longitudinal acceleration |
| γ_s | = | Load factor for static loads |
| γ_w | = | Environmental load factor |
| g | = | Acceleration of gravity = 9.81 m/s ² |
| a_v | = | Vertical acceleration at the relevant C.O.G. calculated by hydrodynamic analysis with annual probability of exceedance: ULS: 10 ⁻² (100 years return period) FLS: 10 ^{-1.3} (20 years return period) |
| a_t | = | Transverse acceleration calculated at the relevant C.O.G. by hydrodynamic analysis with probability of exceedance same as for a_v |
| θ | = | Roll angle of ship |
| a_l | = | Longitudinal acceleration at the relevant C.O.G. calculated by hydrodynamic analysis with probability of exceedance same as for a_v |
| ϕ | = | Pitch angle |
| m | = | Mass of the structure/equipment |

Additional dynamic effects from wind loads are to be included when relevant.

APPENDIX C

HYDRODYNAMIC ANALYSIS

CONTENTS

| | | |
|-----------|---|-----------|
| 1. | INTRODUCTION..... | 3 |
| 2. | DEFINITIONS AND SYMBOLS..... | 5 |
| 3. | STEP 1 - HYDRODYNAMIC MODELLING AND CALCULATIONS OF RAO | 6 |
| 3.1. | HYDRODYNAMIC MODELLING..... | 6 |
| 3.1.1. | <i>General</i> | 6 |
| 3.1.2. | <i>Hydrodynamic models</i> | 7 |
| 3.2. | MASS MODELLING..... | 8 |
| 3.2.1. | <i>Beam Model</i> | 9 |
| 3.2.2. | <i>FE model</i> | 10 |
| 3.2.3. | <i>Relation between hydrodynamic-, mass- and structural models</i> | 11 |
| 3.3. | LOADING CONDITIONS..... | 12 |
| 3.4. | FORWARD SPEED EFFECTS AND SPEED REDUCTION ANALYSIS..... | 12 |
| 3.5. | VISCOUS EFFECTS..... | 12 |
| 3.6. | WAVE HEADINGS..... | 12 |
| 3.7. | WAVE PERIODS..... | 13 |
| 4. | STEP 2 - PREDICTION OF LONG TERM VALUES..... | 14 |
| 4.1. | SHORT TERM DESCRIPTION OF THE SEA..... | 14 |
| 4.2. | WAVE SPECTRUM..... | 14 |
| 4.3. | SHORT CRESTED WAVES..... | 15 |
| 4.4. | LONG-TERM DESCRIPTION OF THE SEA..... | 16 |
| 5. | STEP 3 - DETERMINATION OF REGULAR DESIGN WAVE (ULS ONLY)..... | 17 |
| 6. | STEP 4 - CALCULATION OF LOADS FOR STRUCTURAL CARGO HOLD MODEL..... | 18 |
| 7. | STEP 5 - DETERMINATION OF NON-LINEAR CORRECTION FACTORS..... | 19 |

1. Introduction

This appendix provides guidance and describes acceptable methods for carrying out hydrodynamic analysis according to DNV-OS-C-102 Offshore Ships of conventional shape.

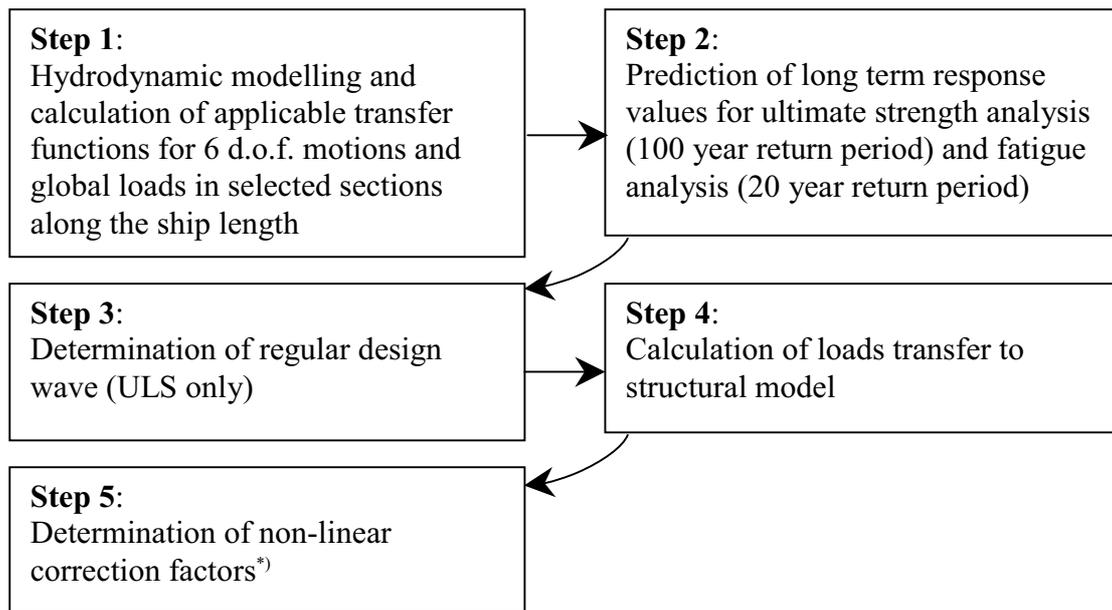
The hydrodynamic analysis serves two main objectives:

- *predict seakeeping characteristics and section loads for a selection of responses*
- *calculate the actual dynamic wave loads for transfer to structural models*

The following load responses shall usually be calculated by the hydrodynamic analysis:

- *vertical bending moment*
- *horizontal bending moment*
- *global shear force*
- *external sea pressure distribution*
- *accelerations*
- *global axial force*
- *torsional moment (only if relevant)*

The procedure for wave load analysis may be described in the steps as shown in Figure C 1.



*) The application non-linear correction factors will normally be made to the final response (i.e. global responses: global vertical bending moment and shear force)

Figure C 1 Procedure for wave load analysis

The following main model types are relevant:

- Hydro model used to calculate hydrodynamic forces, usually modelled as a panel model for calculation of hydrodynamic results based on potential theory
- Mass model to describe the mass properties of the unit, modelled as beams with concentrated mass points at each end or as a FE model
- Structural model where the hydrodynamic and hydrostatic loads are represented as finite elements loads. Direct transfer of calculated loads is only applicable for global strength analyses

Each of the steps is described in more detail in the following sections.

2. Definitions and symbols

- $S(\omega)$: Unidirectional wave spectrum with energy distributed according to wave frequency ω
- $f(\alpha)$: Represent the directional distribution of energy in the waves
- k_r : Roll radius of gyration
- x : Length of the diagonal of a panel element
- α : Angel of elementary waves relative to the main direction β .
- β : Main wave direction
- λ : Wave length
-

3. Step 1 - Hydrodynamic Modelling and calculations of RAO

The hydrodynamic load model shall give a good representation of the wetted surface of the ship, with respect to both geometry description and hydrodynamic properties. The mass model must ensure a proper description of local and global moments of inertia.

In case of a global strength analysis the structural FE model of the vessel is to provide results suitable for the objective of the analysis (i.e. buckling, yield and fatigue assessment of relevant parts of the vessel). Usually the 3D FE model of the whole ship is supported by one or more levels of sub models. Several approaches may be applied, ranging from a detailed 3D-model of the ship to a coarsely meshed 3D-model supported by more densely meshed sub models. Coarsely meshed models may be used for determining stress results suitable for buckling capacity checks and for determining deformations. These deformations may then be applied as boundary conditions for sub models with the purpose of determining the stress distribution in more detail.

3.1. Hydrodynamic modelling

3.1.1. General

The seakeeping and hydrodynamic load analysis is to be carried out by means of recognised computer programs.

More specifically, this means that either 3D linear theory or 2D linear theory (or both) may be applied:

- 3D linear sink-source theory, zero speed or forward speed codes, for ultimate strength calculations (ULS), and fatigue calculations (FLS)
- 2D linear strip theory, forward speed codes, for both ULS and FLS.
(Note that 2D codes may give higher bending moments than 3D codes. Furthermore, pressures from 2D codes are less accurate than pressures from 3D codes. It has to be demonstrated that the use of the less accurate 2D codes does not result in non-conservative fatigue life results. It should also be noted that 2D theory is not capable of computing axial forces. For ships with the neutral axis not coinciding with the still water level, this axial force should be included in the vertical bending moment computations. This can only be achieved with 3D-codes.)

Step 1 in Figure C 1 will only supply response amplitude operators (RAO's, also referred to as transfer functions) for motions and loads in long crested regular waves. These transfer functions should be further processed by means of a wave statistics post-processor in order to determine long term design values in short crested irregular waves for the ocean area of interest when required (*Step 2*, Figure C 1).

3.1.2. Hydrodynamic models

In the following some general requirements to the models applied in a direct hydrodynamic load analysis are given.

2D Strip model

At least 20-25 strips should be applied, including at least 10-14 offset points (on half ship side) for each strip. A good representation in areas with large transitions in shape (bow area, aft area, around the bilge etc.) should be ensured. This will in general require higher strip density in the bow and aft areas compared to the midship area. A higher density of offset points around the bilge and close to the still water level should also be applied.

3D sink source model - Panel model

The element size of the 3D hydrodynamic mesh has to be sufficiently small to avoid numerical inaccuracies. In general this implies at least 30-40 elements along the vessel and at least 15-20 (on half shipside) elements in transverse direction. This means a minimum of 500-800 elements in one half of the ship. If the panels in the model are constructed by drawing straight line segments between the corner nodes of the finite element sides, care must be taken to ensure a good representation in areas with skewed elements and large transitions in shape (bow area, aft area, around the bilge etc.). This will in general require higher element density in bow and aft area compared to the midship area.

In order to reduce the computer hardware resource consumption one should take advantage of the symmetry plane through the centreline (XZ-plane). The co-ordinate system used by the program may typically be as shown in Figure C 2.

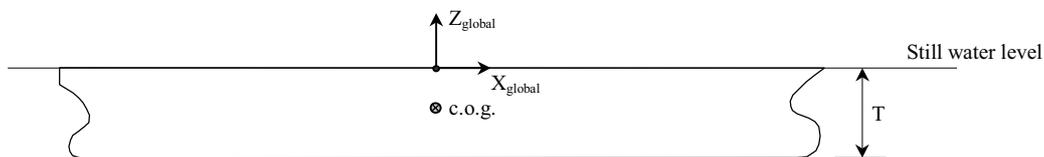


Figure C 2 - Location of global co-ordinate system for the panel model

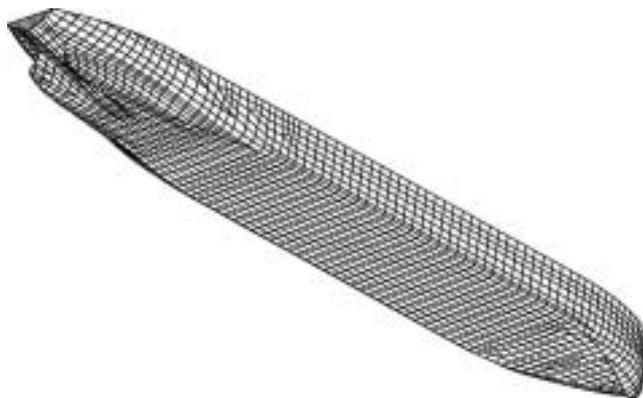


Figure C 3 - Example of total panel model of a ship (including transformed part)

A total panel model of a ship is shown in Figure C 3. The basic part of the panel model may consist of one half only and the total model is generated through transformation (“mirroring”) of the basic part.

For global structural analysis the panel model and the structural model may be based on the same model. It is important that the panels are adjusted in the water line and that this is considered when modelling the structural model, see Figure C 4.

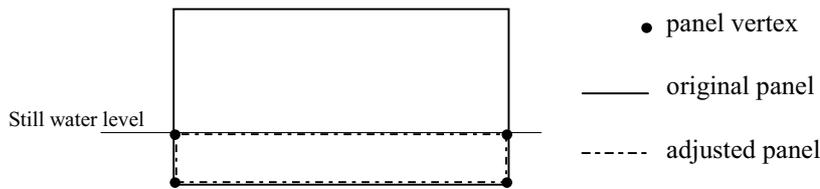


Figure C 4 - Panel adjustment

The wave lengths associated with the relevant range of wave periods to be analysed should be considered when establishing the panel model. The length (x) of the diagonal of a panel element should not be longer than a quarter of the shortest wavelength (λ):

$$x \leq \lambda/4$$

where:

| | | |
|-----------|---|---|
| λ | = | Wave length |
| x | = | The length of the diagonal of a panel element |

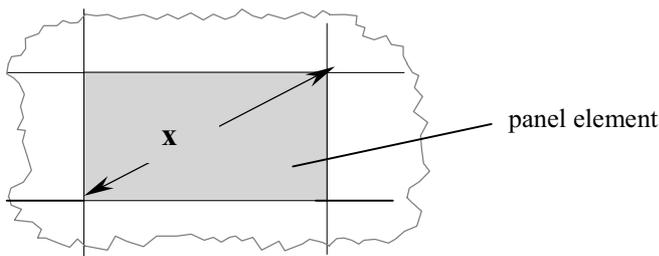


Figure C 5 Panel sizing

3.2. Mass modelling

The mass modelling should be according to the loading manual, i.e. have the same longitudinal, vertical and transverse mass distribution. This is important both for the hydrostatic/dynamic analysis and for the structural analysis. The hydrodynamic analysis requires a correct mass description in order to produce correct motions and sectional forces.

The global/local stress patterns are however affected by the mass description in the structural analysis. If the loads from the hydrodynamic analysis are transferred directly to a global structural model, the mass description will affect the global/local stress patterns computed in the structural analysis.

The mass model may be modelled by either of two different methods, based on the purpose and extent of the analyses. These are outlined in the following.

3.2.1. Beam Model

If the purpose of the hydrodynamic analysis is to calculate the correct motion and global responses (no direct transfer of loads to structural model), the local mass distribution may simply be modelled by transverse beams representing the mass of each section. This is sufficient for the calculation of sectional forces, accelerations and external sea pressures.

It should be ensured that the roll radius of gyration and the metacentric heights are correct. The roll radius of gyration may be taken as given in Rules for Classification of Ships Pt.3 Ch.1 Sec.4 B402:

$$\begin{aligned}k_r &= 0.39 B \text{ for ships with even transverse distribution of mass} \\ &= 0.35 B \text{ for ships in ballast}\end{aligned}$$

The mass distribution may be modelled as shown in Figure C 6. The transverse beams are modelled with no material property (mass of beam ≈ 0). The mass of each section (m_i) is represented as point masses at the ends of the beams. Each end has a mass equal to $m_i/2$. The mass along the vessel is usually taken from the loading manual or stability program as ton/m. With a correct representation of mass within each section the location of the longitudinal centre of gravity for the mass model compared to the loading manual will be correct. The beams are modelled at the vertical centre of gravity, see Figure C 6.

Any large additional masses (e.g. derrick, turret etc.) may be modelled as point masses in the centre of gravity for each item considered.

The balance between the mass model and panel model is important and may be obtained by adjusting the longitudinal position of the models.

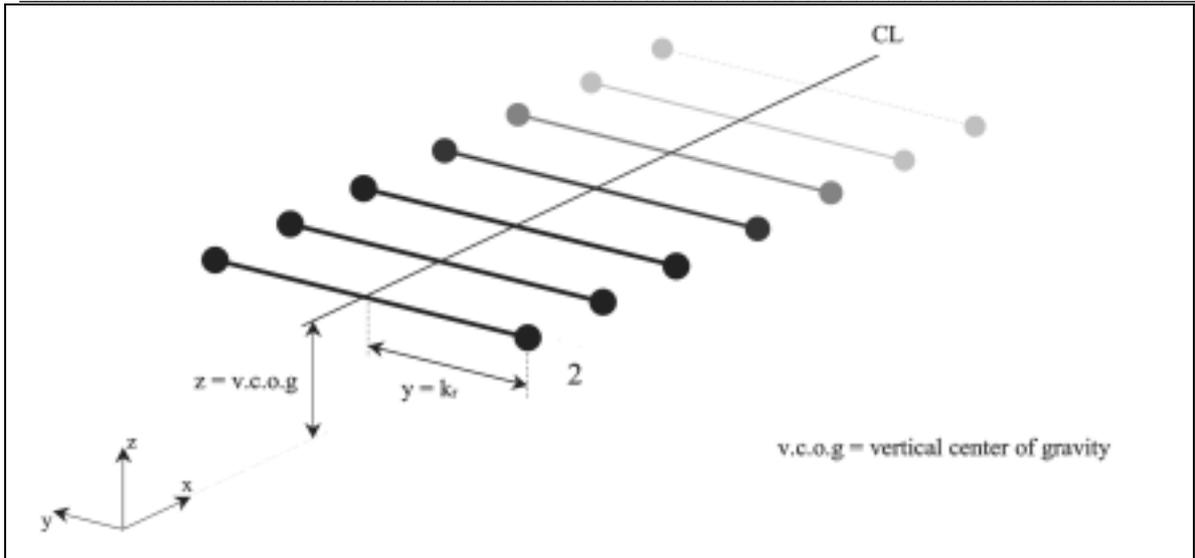


Figure C 6 - Example of mass model for a ship

3.2.2. FE model

If the loads from the hydrodynamic analysis are to be directly transferred to a structural model, the global/local stress patterns are affected by the mass description.

The mass model shall ensure a proper description of local and global moments of inertia around the longitudinal, transverse and vertical global ship axes. In particular the determination of sectional torsion loads may be sensitive to the accuracy and refinement of the mass model.

The mass should have correct longitudinal position, correct transverse position relative to the ship centreline and correct vertical position relative to base line, in order to give a proper description of both local and global moments of inertia. The hydrodynamic model and the mass model should be in proper balance, giving still water shear force distribution with zero value at fore and aft perpendiculars (FP and AP).

The still water loads resulting from the hydrodynamic and mass model should give a good representation of the still water loads from the ship loading manual for the loading condition to be considered.

Identical mass models should be used in the hydrodynamic and structural analyses. The structural model should consequently be used as mass model in the final hydrodynamic analysis, which shall supply the 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 for external and internal hydrostatic and hydrodynamic loads and point mass for non-structural

members and non-liquid cargo (this depends on type of cargo and may differ for some ship types). The point mass representation should be sufficiently distributed to give a correct representation of rotational mass and to avoid unintended results. Point masses should be located in structural intersections such that local response is minimised.

A simplified mass representation should be such that the results are not affected. This means that the global model may have a rather coarse mass description while other models where local deflections are of interest may need a more precise mass description. This depends on model size, mesh size, local loading and the results that will be produced. For some local models, the inertia load from the local model itself will be insignificant while stresses from more global actions will dominate the response.

To balance the model such that correct mass description is obtained may not be a straightforward task. The global structural model usually consists of one or, if super element technique is applied, several large super elements. The size of each super element is relatively large and correct centre of gravity within each super element need not necessarily mean that the distribution within each super element is correct. Even small inaccuracies in the mass description may lead to relatively large errors in global forces/moments.

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

3.2.3. Relation between hydrodynamic-, mass- and structural models

There should be adequate correlation between hydrodynamic model mesh and structural mesh, i.e.:

- Same buoyancy
- Coinciding mesh geometry as far as possible
- Mass models should give same resulting mass and balance with hydrodynamic displaced mass.

If there is a slight unbalance between the mass model and hydrodynamic model, the mass model should be modified.

The hydrodynamic analysis will, in many cases, be performed before the structural model is completely established. In the design phase there is often uncertainty concerning topside weights. In the structural model, topside weights will be part of the final model applied for load transfer. Therefore, topside weights on the unit should be included in the mass used in the hydrodynamic analysis, based on preliminary loading manuals. It is advisable to apply the structural mass model in the final hydrodynamic analysis for load transfer to the structural model.

The procedure of balancing the hydrodynamic and mass model will in general imply a slight modification of the mass model. This may often be achieved by adjustment of point masses close to FP and AP.

3.3. Loading Conditions

The design loading conditions for offshore vessels are described in Appendix B. In general the most unfavourable loading conditions are to be used. The load conditions are to be based on the vessels loading manual, including ballast and full load conditions. Part load conditions are to be included when relevant and/or specified by the designer/owner.

3.4. Forward speed effects and speed reduction analysis

The effect of forward speed is normally neglected for offshore vessels in the operation conditions for both ULS and FLS assessment.

However, an estimate of speed reduction in heavy weather for ULS and FLS may be performed if proper software (post-processor) is available. Such software should include effects resulting in both involuntary and voluntary speed reduction like:

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

3.5. Viscous effects

Since short crested seas and relevant wave heading angles are to be analysed for both ULS and FLS, the inclusion of appropriate roll viscous damping is important to arrive at realistic roll angles and thereby realistic transverse acceleration levels.

Viscous damping may be taken from model tests (if available) or may be estimated using for example 2D strip theory. The effects of eddy making viscosity, skin friction, bilge keels and fin stabilisers should be included if relevant.

The applied viscous damping should reflect a realistic damping for the probability level under consideration i.e. damping at probability level $\approx 10^{-2}$ for FLS, and at probability level $\approx 10^{-8.7}$ for ULS.

3.6. Wave headings

The wave heading angle and number of wave headings depend on the operational modes (operation on site, transit), type of analysis (ULS, FLS) and mooring system (turret or spread mooring). This is further discussed in Appendix D and E.

The definition of heading angles between ship and waves will depend on the computer program used and may be as shown in Figure C 7.

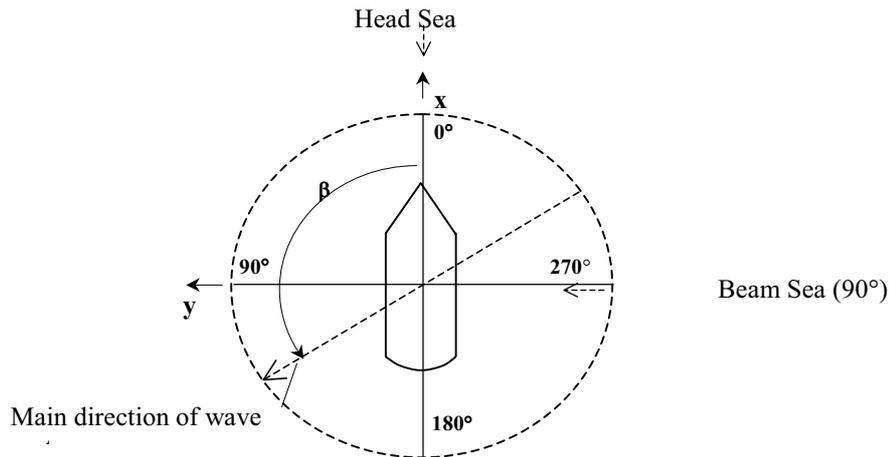


Figure C 7 – Typical definition of heading angle between ship and waves

If several wave headings are investigated, or if a spreading function is applied, the wave heading angle spacing should be equal to or less than 30 degrees. In case of “all heading included” this results in at least 12 heading angles (or 7 when symmetry between port and starboard with respect to loads).

3.7. Wave periods

The hydrodynamic analysis should consider a sufficient number of regular wave periods or frequencies, i.e. 20 or more, in order to cover the relevant range.

In addition, the following general guidelines with respect to wave periods are given:

- The *range* of wave periods should be selected to ensure a good representation of all relevant response transfer functions for the wave period range of the applicable scatter diagram, see Figure C 8.
- A proper wave period *density* should be selected to ensure a good representation of all relevant response transfer functions, including peak values.

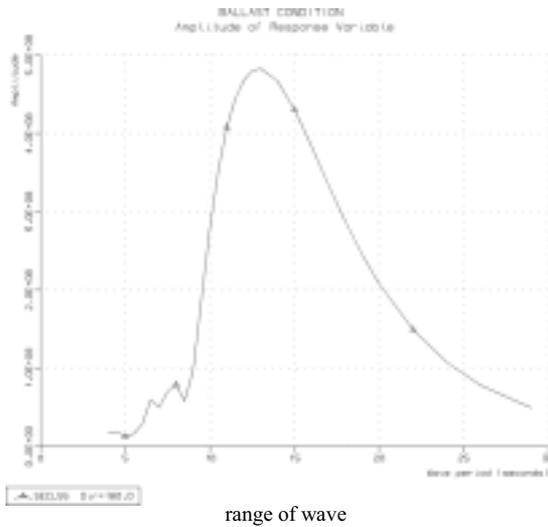


Figure C 8 - Example of transfer function for vertical bending moment

4. Step 2 - Prediction of long term values

4.1. Short term description of the sea

In order to post-process the transfer functions resulting from the linear seakeeping analysis (*Step 1*), the short term description of the sea must be evaluated. I.e. in a short time interval (hours) the statistical properties of the sea are considered as invariant and the sea is termed stationary. The theory of stationary stochastic processes is used to describe such a constant seastate.

4.2. Wave spectrum

A Pierson-Moskowitz wave spectrum representing fully developed seas is applicable when the growth of the waves is not limited by the size of the generation area. The Pierson-Moskowitz wave spectrum is usually applied for most fatigue analyses and ultimate strength analyses.

The JONSWAP wave spectrum is a peak-enhanced Pierson-Moskowitz spectrum which takes into account the unbalance of energy flow in a seastate when the waves are in process of growing under strong winds, i.e. the seas are not fully developed. This is the case for extreme wave conditions in the North Sea. The JONSWAP wave spectrum is usually applied for ultimate strength analyses for vessels operating in harsh environment (e.g. North Sea).

4.3. Short crested waves

Short-crested waves are the combination of different long-crested waves from different directions, hence other directions than the main wave direction β are taken into account. The wave energy of such a system may be described as a directional wave spectrum:

$$S(\omega, \alpha) = S(\omega) \cdot f(\alpha) \quad (\text{C } 1)$$

where:

$S(\omega)$ = unidirectional wave spectrum with energy distributed according to wave frequency ω

$f(\alpha)$ = represent the directional distribution of energy in the waves, usually defined as a cosine-square function:

$$= \int_{\alpha=-n}^{\alpha=n} \cos^2 \alpha \, d\alpha$$

The angle α represents the elementary waves relative to the main direction β .

When wave spreading is applied in the analysis, the available wave directions should cover 180° or more, i.e. preferably from -90° to 90° relative to the considered main wave direction. This depends on the position wave heading system. The wave spacing should be kept constant. In the example in Figure C 9 below the main wave direction is 150° . When short crested waves are applied, the available wave directions cover a range from 60° to 240° . The spacing between the wave heading angles are set to 30° .

Short crested waves may be applied with \cos^2 wave energy spreading for both ULS and FLS. This is further discussed in Appendix D and E.

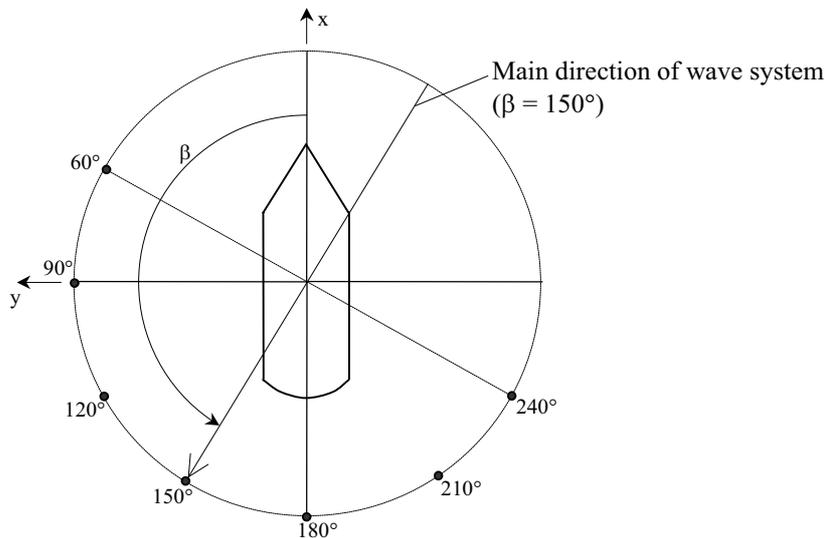


Figure C 9 - Example of minimum available wave headings when short-crested waves are applied for a wave system with a main direction of 150°

4.4. Long-term description of the sea

Whereas short term statistics relate to stationary processes over periods of only a few hours, the long term statistics are associated with non-stationary processes occurring over a period of month and years. In forming a long-term statistical description of the sea a suitable statistical model providing a joint probability distribution of wave height and wave period is required.

Long term response calculations are usually based on scatter diagrams. The cumulative distribution of the significant wave heights is normally described by a three-parameter Weibull distribution. The long-term value computations should be based on different probability levels and scatter diagrams for ULS and FLS condition. Note that site specific scatter diagrams may be used when relevant.

ULS:

North Atlantic scatter diagram should be applied as described in DNV Classification Notes 30.7.

Long-term value computations should be based on probability levels 10^{-5} , 10^{-7} , 10^{-8} .

FLS:

World Wide scatter diagram and/or the *North Atlantic* scatter diagram should be applied as described in Appendix E.

Long-term value computations should be based on probability levels 10^{-2} and 10^{-3} for harsh environments and 10^{-2} , 10^{-3} and 10^{-4} for benign environments.

For the stress component based stochastic fatigue analysis the responses will be based on the load transfer function (RAO) calculated from the hydrodynamic analysis (*Step 1*) and multiplied by the local stress response for the considered detail due to a unit load. The method is further described in Appendix E.

For a full stochastic fatigue analysis the loads are directly transferred to the global FE model. The analysis is run for all wave load cases. E.g. 2 load conditions (fully loaded and ballast), 12 headings and 20-25 periods per heading leads to 480 - 600 load cases.

5. Step 3 - Determination of regular design wave (ULS only)

The design loads for ULS and FLS analysis are determined by the use of a long-term calculation procedure as described above.

For the ULS condition the design wave approach may be applied. The design wave is defined as a regular wave that gives the same response level as the long-term design value for a critical response. The method is described in Appendix D.

6. Step 4 - Calculation of loads for structural cargo hold model

In case of a cargo hold model the calculated loads will be manually transferred to the FE model. Determination of external sea pressure and internal tank pressure based on direct calculation of environmental loads are shown in Appendix B.

In case of a global model the loads are directly transferred from the hydrodynamic analysis to the structural model. The calculated hydrodynamic loads are to be applied as a combination of pressure forces on the hull and inertia forces. For well designed models the global FE model loads will be close to equilibrium, and the reaction forces should be close to zero.

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

A check of sectional loads should be performed in order to ensure that the sectional loads in the structural FE model are similar to the sectional loads in the hydrodynamic load analysis.

7. Step 5 - Determination of non-linear correction factors

In the linear hydrodynamic load calculations (*Step 1*), the non-linear effects are not accounted for. The non-linear effects should be considered if these effects are regarded to be important based on an initial evaluation of the hull shape. For this purpose either a 2D or a 3D non-linear time simulation code may be applied. The non-linearity is to be considered originate from the integration of the hydrodynamic pressure over the instantaneous position of the hull relative to the waves, and should include effects from bow flare, bottom slamming and deck wetness (if relevant).

The non-linear corrections may be based on a set of "equivalent" regular design waves. This enables a set of consistent loads to be established (external dynamic pressure, accelerations, shear forces and bending moments) and these loads may then be applied to an FE model since they represent a set of loads in equilibrium.

The non-linear codes should preferably be time domain codes where the equations of motion are solved using a predictor corrector technique, to integrate the non-linear differential equations. This way a set of consistent loads will be ensured at the time instant of occurrence of the hydrodynamic load.

Dimensioning water heights on deck should (if relevant) be estimated for the fully loaded condition (minimum freeboard).

APPENDIX D

ULTIMATE STRUCTURAL CAPACITY

CONTENTS

| | |
|---|-----------|
| 1. INTRODUCTION | 3 |
| 1.1. GENERAL..... | 3 |
| 2. DEFINITIONS AND SYMBOLS..... | 4 |
| 3. HULL GIRDER TYPICAL CARGO AREA..... | 6 |
| 3.1. DESIGN PRINCIPLES | 6 |
| 3.2. GOVERNING ULS COMBINATION..... | 7 |
| 3.3. DESIGN WAVE | 9 |
| 3.4. GLOBAL LOAD CONDITIONS..... | 10 |
| 3.5. CONSISTENT EXTERNAL AND INTERNAL PRESSURE..... | 12 |
| 3.6. COMBINATION OF STRESSES..... | 13 |
| 3.6.1. <i>Longitudinal Stresses</i> | 13 |
| 3.6.2. <i>Transverse Stresses</i> | 17 |
| 3.6.3. <i>Shear stresses</i> | 18 |
| 4. HULL GIRDER IN WAY OF TURRET AND MOONPOOL..... | 20 |
| 4.1. GENERAL..... | 20 |
| 4.2. GLOBAL STRESS CONCENTRATIONS (SCF) | 20 |
| 5. TOPSIDE STRUCTURE AND HULL INTERFACE..... | 22 |
| 5.1. DESIGN PRINCIPLES | 22 |
| 5.2. LOADS..... | 24 |
| 5.3. STRUCTURAL RESPONSE OF SUPPORTING STRUCTURE..... | 24 |
| 6. CAPACITY CHECKS | 28 |
| 6.1. GENERAL COMMENTS TO THE ULS CAPACITY CHECKS | 28 |
| 6.2. GENERAL PRINCIPLES | 28 |
| 6.3. HULL GIRDER MOMENT CAPACITY CHECKS | 30 |
| 6.4. HULL GIRDER SHEAR CAPACITY CHECK..... | 32 |
| 6.5. CAPACITY OF SUPPORTING STRUCTURE FOR TOPSIDE FACILITIES..... | 33 |
| 6.6. MISCELLANEOUS STRUCTURES..... | 34 |
| 6.6.1. <i>Bilge Keel</i> | 34 |

1. INTRODUCTION

1.1. General

The ultimate limit state assessment comprises yield and buckling capacity checks of the structure according to the LRFD format as described in DNV-OS-C101 - *Design of Steel Structure*.

Ultimate hydrodynamic loads from the hydrodynamic analysis are to be combined with static loads in the yield and buckling checks. All relevant load conditions should be examined to ensure that all dimensioning loads are correctly included.

2. DEFINITIONS AND SYMBOLS

| | |
|---------------|---|
| A | : Total area (plate + stiffener) of element |
| H_D | : Design wave amplitude |
| A_i | : Cross section area |
| $A_{p,i}$ | : cross sectional area of plate part of panel |
| A_{ps} | : Area of panel, (plate and stiffeners) |
| A_s | : Total area of vertical stiffeners |
| B | : Breadth |
| C_B | : Block coefficient |
| C_W | : Wave coefficient |
| F_{as} | : Characteristic still water axial force |
| F_{aw} | : Characteristic wave axial force |
| L | : Length |
| M_D | : Design bending moment |
| M_g | : Characteristic bending moment resistance |
| M_S | : Characteristic still water bending moment |
| M_T | : Characteristic torsional moment |
| M_W | : Characteristic wave bending moment |
| P_v | : Vertical load |
| Q_D | : Design shear force |
| Q_g | : Characteristic shear resistance |
| Q_S | : Characteristic design still water shear force |
| Q_W | : Characteristic wave shear force |
| M_{Wv} | : Characteristic vertical bending moment |
| RAO | : Response Amplitude Operator |
| a_v | : Vertical acceleration |
| Z_i | : Section modulus |
| f_y | : Characteristic yield strength of panel |
| f_{yD} | : Von Mises yield stress for deck area elements |
| l | : Length of stool above deck measured along the deck |
| m | : Total number of elements in bottom area |
| n | : Total number of elements in deck area |
| q | : Shear flow factor |
| t | : Plate thickness |
| t_p | : Plate thickness |
| z | : Vertical distance from deck to the section considered |
| γ_m | : Material factor |
| γ_s | : Load factor for still water loads |
| γ_w | : Environmental load factor |
| σ_a | : Nominal axial stresses |
| σ_{a2} | : Nominal secondary axial stress |
| σ_h | : Nominal horizontal hull girder bending stress |
| σ_t | : Nominal torsion stress in hull girder |
| σ_v | : Nominal vertical hull girder bending stress |
| σ_{ve} | : Nominal vertical hull girder bending stress |
| σ_{v2} | : Nominal secondary vertical hull girder bending stress |
| σ_x | : Longitudinal design stress |

| | | |
|----------------------------|---|---|
| σ_y | : | Transverse design stress |
| σ_{y1}, σ_{y2} | : | Transverse stress components at lower and upper end of plate between stiffeners |
| σ_2 | : | Nominal secondary bending stress in double bottom or double side |
| τ | : | Design shear stress |
| λ_D | : | Length of design wave |

3. HULL GIRDER TYPICAL CARGO AREA

3.1. Design principles

The ULS capacity of the hull girder is primarily governed by the buckling and yield capacity of the top and bottom flanges of the hull girder, i.e. the deck and bottom structure, when the ship experiences maximum longitudinal bending stresses. The buckling capacity of the stiffened panels in a considered section are dependent on:

- structural arrangement and dimensions of plates and stiffeners
- stresses parallel to the stiffener direction (typically longitudinal stresses)
- stresses normal to the stiffener direction (typically transverse stresses due to bending of transverse frames and vertical stresses from top side loads)
- shear stresses
- lateral pressure

The hull girder bending capacity (D 1) in the operating conditions shall comply with:

$$\gamma_s M_s + \gamma_w M_w \leq M_g / \gamma_m \quad (\text{D } 1)$$

The hull girder shear capacity (D 2) in the operating conditions shall comply with:

$$\gamma_s Q_s + \gamma_w Q_w \leq Q_g / \gamma_m \quad (\text{D } 2)$$

where:

- | | |
|------------|--|
| M_g | Characteristic bending moment resistance of the hull girder |
| M_s | Characteristic still water bending moment based on actual cargo and ballast conditions |
| M_w | Characteristic wave bending moment based on an annual probability of exceedance of 10^{-2} . |
| Q_g | Characteristic shear resistance of the hull girder |
| Q_s | Characteristic still water shear force based on actual cargo and ballast conditions |
| Q_w | Characteristic wave shear force based on an annual probability of exceedance of 10^{-2} . |
| γ_m | Material factor |
| γ_s | Load factor for still water loads (permanent + variable functional loads) |
| γ_w | Environmental load factor |

The partial load factors to be used in ULS hull girder capacity checks are given in Table D 1.

| Load combination | Load category | |
|------------------|-------------------|---------------------|
| | Still water loads | Environmental loads |
| a) | 1.2 | 0.7 |
| b) | 1.0 | 1.15 |

Table D 1 Partial coefficients for the Ultimate Limit State

Therefore, for hull girder ULS checks, the design moment and shear force (left hand side of the (D 1) and (D 2) respectively) will be:

for combination a):

$$M_D = 1.2 M_S + 0.7M_W \quad (\text{D } 3)$$

for combination b):

$$M_D = 1.0 M_S + 1.15M_W \quad (\text{D } 4)$$

and

for combination a):

$$Q_D = 1.2 Q_S + 0.7Q_W \quad (\text{D } 5)$$

for combination b):

$$Q_D = 1.0 Q_S + 1.15Q_W \quad (\text{D } 6)$$

where:

M_D and Q_D are the design bending moment and design shear force, respectively.

It should be noted that the use of partial load factor 1.2 for combination a) assumes that the permanent loads can be determined with high accuracy like tank loads with known geometry.

3.2. Governing ULS combination

The ship is to comply with the ✕1A1 requirements, and there are thus requirements to both still water loads and wave loads.

The Rule still water moments amidships for unrestricted operation are:

$$M_S = - 0.065 C_W L^2 B (C_B + 0.7) \text{ (kNm) in sagging} \quad (\text{D } 7)$$

$$M_S = C_W L^2 B (0.1225 - 0.015 C_B) \text{ (kNm) in hogging} \quad (\text{D } 8)$$

The Rule wave bending moments for seagoing conditions are:

$$M_W = -0.11 C_W L^2 B (C_B + 0.7) \text{ (kNm) in sagging} \quad \text{(D 9)}$$

$$M_W = 0.19 C_W L^2 B C_B \text{ (kNm) in hogging} \quad \text{(D 10)}$$

The ratio between M_W and M_S are thus for minimum values:
for sagging:

$$\frac{M_W}{M_S} = \frac{-0.11 C_W L^2 B (C_B + 0.7)}{-0.065 C_W L^2 B (C_B + 0.7)}$$

or

$$\frac{M_W}{M_S} = 1.69$$

Similarly for hogging,

$$\frac{M_W}{M_S} = \frac{0.19 C_W L^2 B C_B}{C_W L^2 B (0.1225 - 0.015 C_B)}$$

or

$$\frac{M_W}{M_S} = 1.57 \text{ using a block coefficient of } 0.9$$

Figure D 1 shows the resulting design bending moment for ULS combinations a) and b) for different M_W/M_S ratios. The shaded area in the right hand part of the graph represents the relevant M_W/M_S ratios for normal $\times 1A1$ ships, and it can thus be concluded that the b) combination is governing for hull girder strength.

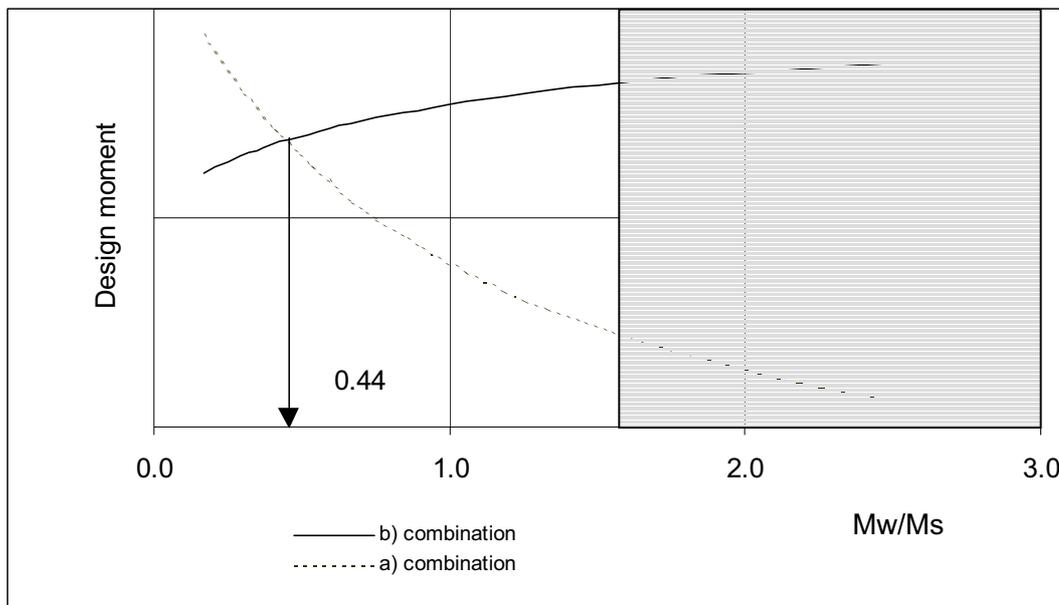


Figure D 1 Design moments for ULS a) and b) combination

3.3. Design wave

In order to establish the design loads for the hull girder section, a design wave approach may be used. The longitudinal stresses, both global and local, shall be combined with transverse stresses and shear stress. Stresses due to lateral pressure on the panel shall be included.

These stresses shall be taken from *consistent loads* using actual internal and external pressures corresponding to the worst combination of still water loads and wave position. In order to obtain *consistent loads* a design wave is defined.

The "equivalent" regular design wave is defined as the regular wave that gives the same response level as the long-term value for a specific response parameter. For the ULS the critical long-term design response level is to be determined for site specific or North Atlantic environment at 100-year return period. Units intended to stay on a specific location for a long period (typical FPSOs or FSU) are normally based on the relevant scatter diagram, whereas units designed for world wide operation with short periods on the various locations (like drilling, well intervention/service units) are based on the North Atlantic scatter diagram. The design wave is found as:

$$H_D = \frac{\text{Long term response}}{RAO}$$

where:

$$H_D = \text{Design wave amplitude}$$

In general, the loads are transferred for the wave period and direction where the transfer function has its maximum value. In extreme "head sea condition" the vertical bending moment midship with a return period of 100 years, $M_{Wv 100}$, is the most important load effect.

The regular design wave is then chosen as the wave where the transfer function of the response, M_{Wv} , has its maximum value. The design regular wave amplitude H_D is chosen to give a value of response of M_{Wv} at the design wave length, equal to the long-term extreme amplitude, for the specified return period:

$$H_D = \frac{M_{Wv}}{RAO_{M_{Wv}}}$$

In some cases this procedure may result in a design regular wave with a wave steepness that is too high:

$$S = 2 \cdot \frac{H_D}{\lambda_D} > \text{approx. } 1/7$$

This may occur in case of transfer function curves with somewhat blunt peaks. In that case it may be necessary to choose a slightly longer wavelength than the wavelength where the transfer function has its actual maximum, and to repeat the procedure for this new wave length.

Stillwater loads are to be combined with the corresponding design hydrodynamic loads such that sets of simultaneously acting loads are obtained. These will then be the set of design loads to be used in the strength evaluation of the vessel. This will ensure *consistent loads* in the design. Phase angles between the different responses are neglected and the maximum values are used as a conservative approach.

Calculations of internal tank pressure and external sea pressure are shown in Appendix B.

A summary of different characteristic responses that may be governing is shown in Table D 2 below.

| | Global Vertical Bending Moment | Vertical Acceleration | Torsional moment ³⁾ |
|--------------------|--------------------------------|-----------------------|--------------------------------|
| Head sea condition | $M_{WV, 100}^{1)}$ | $a_{v, 100}^{1)}$ | |
| Beam sea condition | | $a_{v, 20}^{2)}$ | $M_{T, 20}^{2)}$ |
| Transit condition | $M_{WV, 20}^{2)}$ | $a_{v, 20}^{2)}$ | $M_{T, 20}^{2)}$ |

1) denotes probability of exceedance of $10^{-8.7}$

2) denotes probability of exceedance of 10^{-8}

3) The torsional moment has to be considered only when the vessel has large openings in deck/bottom. A typical moonpool midship will usually not reduce the torsional stiffness significantly, and the effect of the torsional moment can be neglected.

Table D 2 Characteristic responses which may be governing for the Design Wave

3.4. Global load conditions

The global load conditions used for the ULS b) combination capacity checks of the hull girder are shown in Table D 3. These load conditions are selected as they will result in the highest longitudinal hull girder bending compression stress in bottom and deck, respectively.

| | | |
|--|---|--|
| | <p>LC. 1. Hogging condition</p> <p>T = Ballast draught Wave crest at considered section</p> | <p>Governing for bottom structure. Both still water and wave loads cause compression longitudinal stresses.</p> |
| | <p>LC. 2. Sagging condition</p> <p>T = Full load draught Wave trough at considered section</p> | <p>Governing for main deck structure. Both still water and wave loads cause compression longitudinal stresses.</p> |

Table D 3 Global load conditions for hull girder ULS checks

Load condition 1 – Maximum hogging

The purpose of the “Maximum hogging” load condition is to combine the still water condition, normally a ballast condition, and a position of the design wave that both will result in maximum longitudinal compression stress *below* the neutral axis of a transverse section. The extreme hogging condition is considered in a head sea situation, but may allow for some fluctuation. Typical values are $\pm 15^\circ$ or $\pm 30^\circ$.

The extreme load conditions may be summarised as follows:

- Head sea condition (i.e. 180°)
- Speed: 0 knots
- Extreme still water hogging condition based on loading manual or DNV hogging moment limit curve if greater. The most likely simultaneous global distribution of the topside loads to be included
- Long crested waves
- 100 years return period for environmental loads
- Environmental loads based on the scatter diagram (site specific or North Atlantic)

Load condition 2 – Maximum sagging

The purpose of the “Maximum sagging” load condition is to combine the still water condition, normally a fully loaded condition, and a position of the design wave that both will result in maximum longitudinal compression stress *above* the neutral axis of a transverse section. The extreme sagging condition is considered in a head sea situation, but may allow for some fluctuation as in LC 1.

The extreme load conditions may be summarised as follows:

- Head sea condition (i.e. 180°)
- Speed: 0 knots

- Extreme still water sagging condition based on loading manual or DNV sagging moment limit curve, if greater. The most likely simultaneous global distribution of the topside loads to be included
- Long crested waves
- 100 years return period for environmental loads
- Environmental loads based on the scatter diagram (site specific or North Atlantic)

The wave induced linear responses are normally:

- vertical bending moment
- global shear force
- external sea pressure distribution
- accelerations (induced internal tank pressure)
- global axial force
- torsional moment, if relevant

Further details regarding the wave load analysis are given in Appendix C.

3.5. Consistent external and internal pressure

The two global load conditions described in section 3.4 will result in a lateral net pressure on the hull due to double bottom bending girder system. Transverse stresses due to bending of the transverse frames and longitudinal stresses must be determined for capacity checks of the hull girder. The two load cases described above are used to determine these stresses. Normally, the same 3-tank finite element model used for designing the girder system according to §1A1 requirements is used to calculate the stresses.

Load condition 1

The pressure distribution will in principle be as shown in Figure D 2 Detailed description of calculation of the pressure is given in Appendix B.

Note that the ballast tanks are empty as this will result in an axial compression force in the double bottom floors due to the external side shell pressure acting between the tank top and bottom plate. If the analysis is carried out with full ballast tanks, a filling restriction must be given in the load manual.

The dynamic sea pressure is not calculated above the still water line, but a linear pressure distribution between the top of the wave and the still water line is assumed.

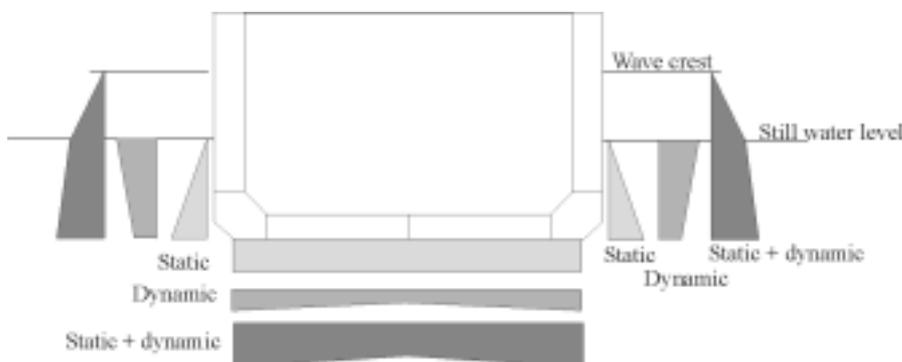


Figure D 2 Lateral pressure distribution load condition 1.

Load condition 2

The pressure distribution will in principle be as shown in Figure D 3. Detailed description of calculation of the pressure is given in Appendix B.

The load condition is used to determine the transverse and secondary longitudinal stresses in the upper deck and upper parts of the longitudinal bulkheads/side. Since the net pressure is acting from the cargo tanks and outwards, the deck transverse beams will be exposed to axial tension stresses. However, due to the horizontal deflection of the vertical web at side/longitudinal bulkheads, the fixation moment at the top of the vertical webs will result in bending compression stresses in the deck plate. The resulting normal stress (axial + bending) may be a compression stress which should be considered in the buckling checks of the deck panels.

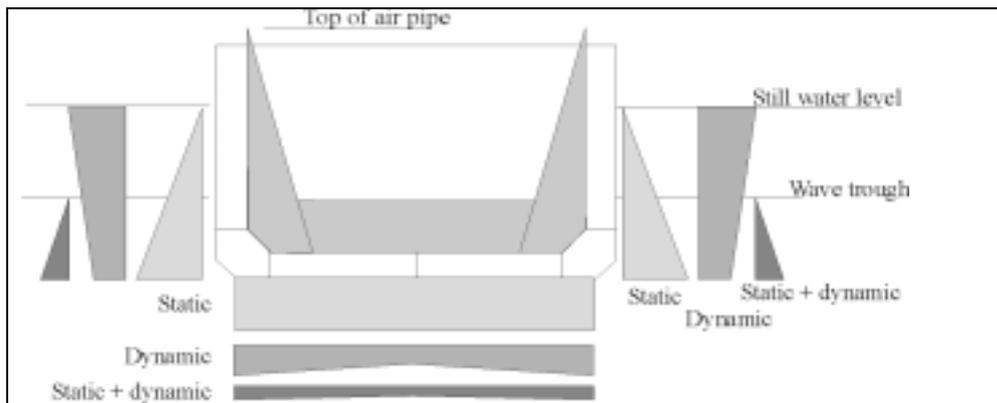


Figure D 3 Lateral pressure distribution load condition 2.

3.6. Combination of Stresses

In order to carry out ULS moment capacity checks, both global and local stresses must be combined. Note that the effect of topside loads are further described in section 4.

Generally the total longitudinal design stress may be derived as:

$$\sigma_{x, \text{Total}} = \sigma_{x, \text{Global}} + \sigma_{x, \text{Local}}$$

Total transverse design stress:

$$\sigma_{y, \text{Total}} = \sigma_{y, \text{Global}} + \sigma_{y, \text{Local}}$$

Total design shear stress:

$$\tau_{\text{Total}} = \tau_{\text{Global}} + \tau_{\text{Local}}$$

Note that the global and local stresses must be calculated for static and dynamic loads separately in order to include the ULS partial load factors in the calculation of design stresses.

3.6.1. Longitudinal Stresses

The nominal longitudinal stresses derived from the analysis will be a combination of the stress components shown in Figure D 4. The following stress components for the total longitudinal design stress are discussed:

$$\sigma_{x, \text{Total}} = (\sigma_v + \sigma_h + \sigma_t + \sigma_a + \sigma_{ve})_{x, \text{global}} + (\sigma_2 + \sigma_{v2} + \sigma_{a2})_{x, \text{local}}$$

where:

- σ_v = Nominal vertical hull girder bending stress
- σ_h = Nominal horizontal hull girder bending stress
- σ_t = Nominal torsion stress in hull girder
- σ_a = Nominal axial stresses due to hull girder end pressure
- σ_{ve} = Nominal vertical hull girder bending stress due to end pressure
- σ_2 = Nominal secondary bending stress in double bottom or double side

- σ_{v2} = Nominal secondary vertical hull girder bending stress due to lateral pressure on transverse bulkheads
- σ_{a2} = Nominal secondary axial stress due to lateral pressure on transverse bulkheads

The σ_{ve} component is usually included in the vertical hull girder bending stress σ_v . The dynamic part of σ_{ve} from the wave load analysis need therefore to be derived from bending moments about the neutral axis for the different transverse sections.

The different loads and stress components are shown in Figure D 4.

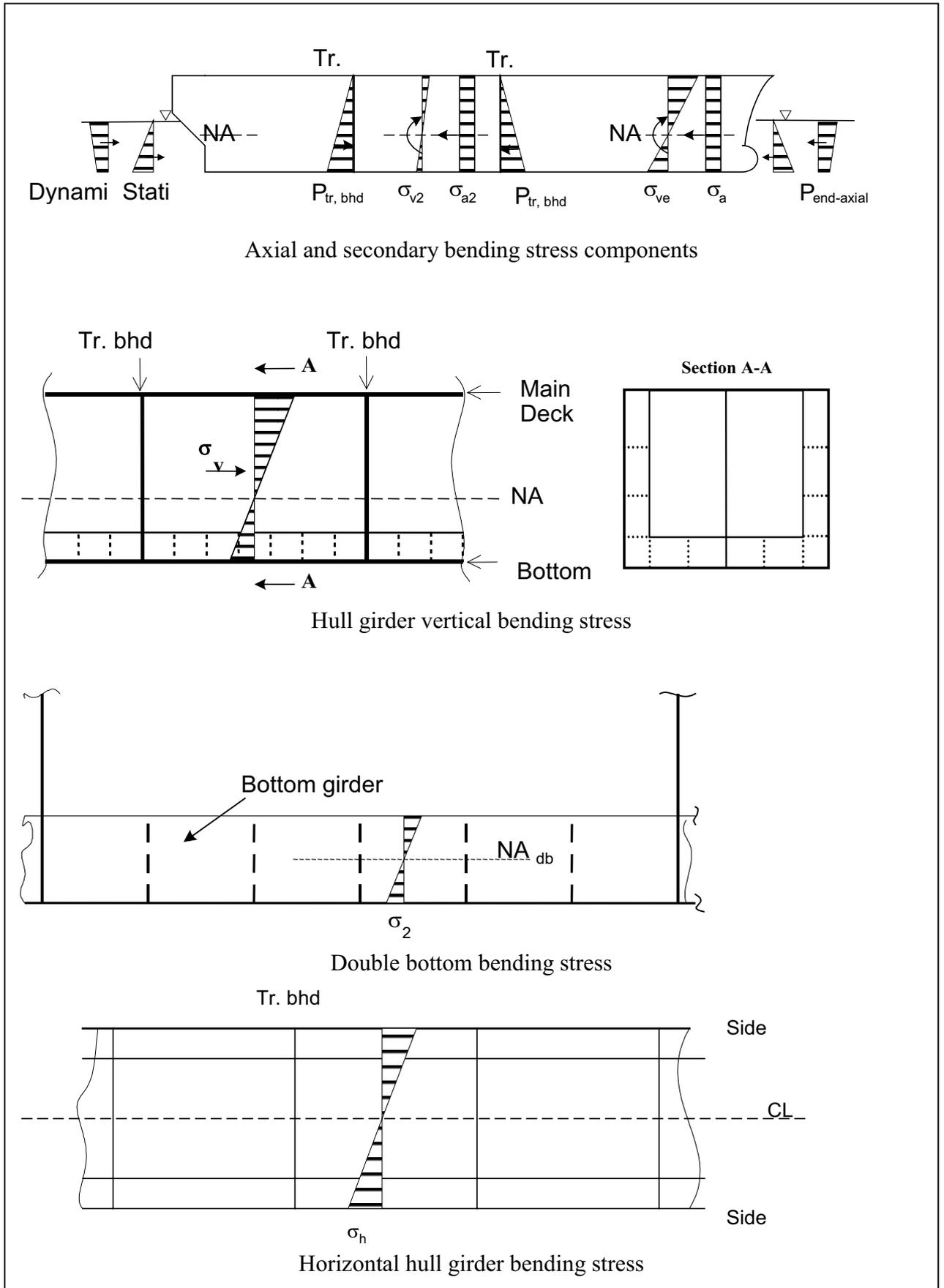


Figure D 4 Longitudinal stress components

σ_v - nominal vertical hull girder bending stress

The σ_v nominal design stress from vertical hull girder bending is normally calculated according to beam theory for the midship section. The design stress is obtained according to the following formula:

$$\sigma_v = \frac{\gamma_s \cdot M_s + \gamma_w \cdot M_w}{Z_i} \quad (\text{D } 11)$$

where:

- M_s = Characteristic still water bending moment based on actual cargo and ballast conditions
- M_w = Characteristic wave bending moment based on an annual probability of exceedance of 10^{-2} (100 years)
- γ_s = Load factor for still water loads
- γ_w = Environmental load factor
- Z_i = section modulus at the considered transverse section (i)

σ_h - nominal horizontal hull girder bending stress

The nominal horizontal bending stress should be considered. The horizontal bending stress is caused by wave loads only. The stress σ_h can be calculated as:

$$\sigma_h = \frac{\gamma_w \cdot M_w}{Z_i} \quad (\text{D } 12)$$

The symbol definition is the same as for vertical bending stress above.

σ_a - nominal axial stresses due to hull girder end pressure

The hull girder axial stress due to the static and dynamic end pressure can be considered to act uniformly on the complete cross section within the cargo area. The design stress can be calculated according to the following formulae:

$$\sigma_a = \frac{\gamma_s \cdot F_{as} + \gamma_w \cdot F_{aw}}{A_i} \quad (\text{D } 13)$$

where:

- F_{as} = Characteristic still water axial force due to hull end pressure
- F_{aw} = Characteristic wave axial force based on an annual probability of exceedance of 10^{-2} (100 years)
- γ_s = Load factor for still water loads
- γ_w = Environmental load factor
- A_i = Cross section area based on gross thickness at the considered transverse section (i)

 σ_{ve} - nominal vertical hull girder bending stress due to end pressure

The end pressure will impose a vertical bending moment due to the moment arm between the neutral axis of the hull section and the centroid of the axial load. This moment is normally included in the still water and wave bending moment. If, however, the wave vertical moments are not presented at the neutral axis, the hull girder vertical bending moment must be corrected to account for the induced end pressure moment.

 σ_2 - nominal secondary bending stress in double bottom or double side

The double bottom or side may impose significant longitudinal stresses due to bending of side/centre girders and horizontal stringers respectively. The stress component can be derived from the 3-tank FE analysis. Care should be taken to exclude the global bending of the tank model.

 σ_{v2} - nominal secondary vertical hull girder bending stress due to lateral pressure on tank boundaries

This bending stress can normally be ignored as the moment is very small due to the position of the neutral axis. If the stress is to be accounted for, it can be done by adjusting the vertical hull girder bending moment. The stress is not readily available from the tank model analysis, however, the moment can be determined by simple hand calculations.

 σ_{a2} - nominal secondary axial stress due to lateral pressure on tank boundaries

The axial stress due to lateral pressure on the tank transverse bulkheads can be accounted for by adjusting axial stress due to hull girder end pressure. The stress can be derived from simple hand calculations.

3.6.2. Transverse Stresses

The nominal transverse stresses in the hull girder panels are caused by the bending of the transverse frames and transverse axial force from external/internal pressure. The loads as given in Figure D 2 and Figure D 3 are applied in the cargo tank analysis in order to determine the transverse stresses.

The bottom plate act as flange for the transverse frame and will be exposed to transverse compression stresses in the middle of the span, as shown in Figure D 5. This will represent the transverse stresses due to bending of the transverse frames (σ_{tb}) as applied in the ULS capacity checks. The example in Figure D 5 shows maximum transverse stresses in the middle part of the tank, and considerable lower stress values near the transverse bulkheads. The reason is that there are several longitudinal side girders with relative short span compared to the span of the transverse frames. Other designs with none, or few, side girders and short span of the transverse frames will show almost the same transverse stresses for all the frames

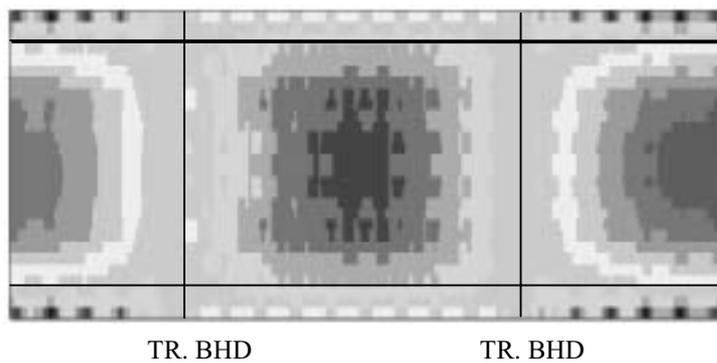


Figure D 5 Transverse stresses in bottom plate

3.6.3. Shear stresses

The wave shear forces derived from direct calculations for world wide operation has proved to be much higher (30 - 80%) than the unified IACS requirement. From experience the hull shear capacity at the “¼ lengths” of the hull must be considered early in the design, and the effect of topside loads must be included. The main reasons for this are:

- The still water shear forces and wave shear forces are usually maximum at the ends of the cargo area.
- The scantlings of the longitudinal bulkheads/sides are also often reduced in these regions or terminate completely.
- The hull girder wave bending moment and wave shear force are almost in phase
- The shear stresses and vertical stresses from the topside loads are often higher than in the midships area due to pitch accelerations. (see section 4)

The following shear stress components should be considered for the total shear stress:

$$\tau_{\text{Total}} = \tau_{\text{Global}} + \tau_{\text{Local}}$$

where:

τ_{Global} = total nominal design shear stress from global shear force

τ_{Local} = total nominal design shear stress from local effects

The global design shear force (τ_{Global}) in side, inner side, longitudinal bulkheads and other global shear carrying elements is normally obtain by the following formula:

$$\tau_{\text{Global}} = \frac{\gamma_S \cdot Q_S + \gamma_W \cdot Q_W}{t} \cdot q(Fz1) \quad (\text{D } 14)$$

where:

Q_S = Characteristic design still water shear force based on actual cargo and ballast conditions

Q_W = Characteristic wave shear force based on an annual probability of exceedance of 10^{-2} (100 years)

| | | |
|------------|---|---|
| γ_s | = | Load factor for still water loads |
| γ_w | = | Environmental load factor |
| t | = | Plate thickness of considered panel |
| $q(F_z1)$ | = | Shear flow factor [N/mm] given by the shear flow analysis (eg. in Nauticus) due to a unit vertical shear force ($F_z = 1$ N) |

The nominal shear stress from local effects (τ_{Local}) are normally derived from the cargo hold FE analysis or the local FE analysis.

However, the global hull girder loads may be included in a cargo hold model by using the method as described in Classification Notes 31.3 - *Strength Analysis of Hull Structure in Tankers*, Appendix A. The total shear stress may then be determined directly from the cargo hold analysis.

4. HULL GIRDER IN WAY OF TURRET AND MOONPOOL

4.1. General

Some offshore units have large openings in the hull girder such as openings for internal turrets or moonpool for drilling/well service operations. Some of these openings are located near midships where the wave bending moment is still very high. Other turret designs are of a submerged type located in the bow area. The different designs have different impact on the global and local strength of the ship which can be analysed by different models. Further details of structural finite element models are given in Appendix F.

4.2. Global stress concentrations (SCF)

In ships with large openings in the cargo area, e.g. to accommodate turrets, the distribution of the global longitudinal bending stresses must be determined. In Figure D 6 the longitudinal stresses in the main deck (similar for bottom structure) is shown.

In the typical cargo area away from the turret opening, the shear lag effect can usually be ignored for ships with CL bulkhead or two longitudinal bulkheads. In way of the moonpool the global stress concentration may typically vary between 1.7 and 2.9. In Table D 4 different factors that affect the global SCF are discussed. The SCFs given are only indicative and not to be used without documentation.

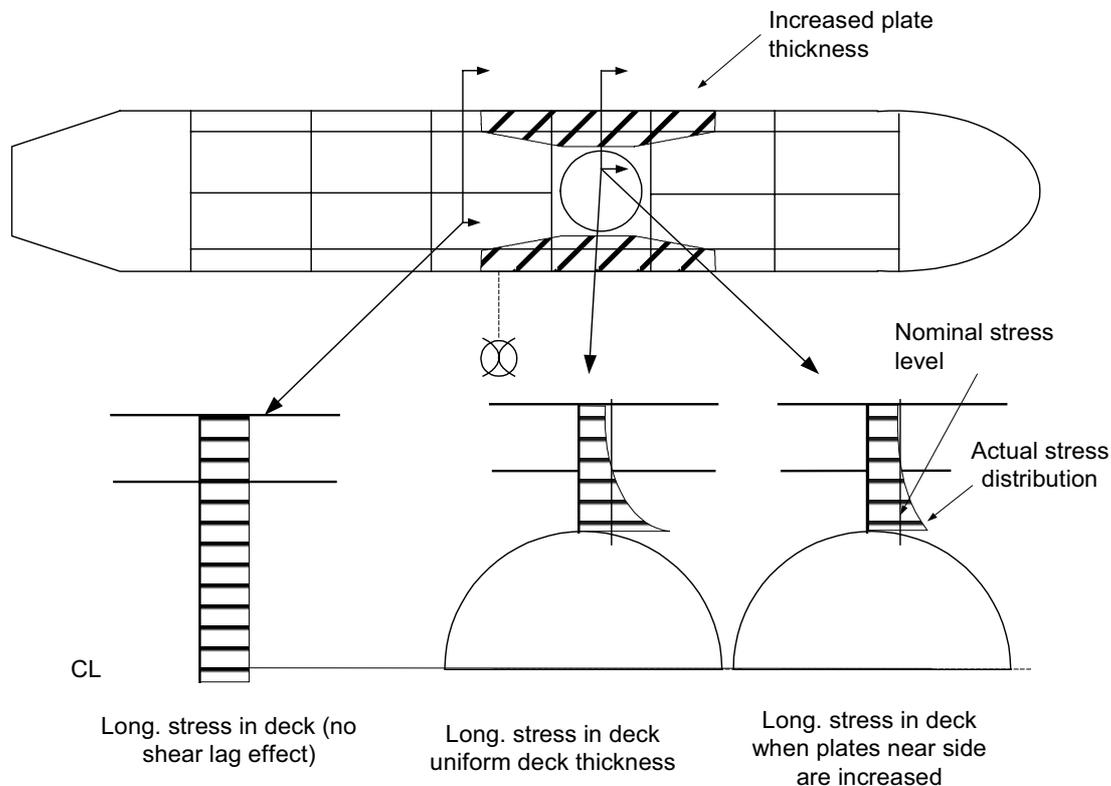


Figure D 6 Distribution of global longitudinal stresses in main deck

| Arrangement | Typical SCF | Comments |
|-----------------------------|-------------|--|
| Double side, no long. bhds | 2.0 – 2.5 | The lower SCF is obtained if thicker plates are used towards the sides, see hatched area in Figure D 6. This will counteract the desired shear lag effect of the thicker plates. |
| Double side, CL bhd | 2.2 – 2.9 | The CL bulkhead is terminated at the transverse bulkhead adjacent to the moonpool. Lower SCF is obtained by increased plate thickness towards side. |
| Double side, two long. bhds | 1.7 – 2.2 | Lower SCF is obtained if the long. bhd continue through the moonpool area. Further reduction in SCF by increased plate thickness towards side. |

Table D 4 Typical global stress concentration factors

The longitudinal global bending stresses σ_v including the global SCF shall be used in the ULS capacity checks. Transverse stresses and shear stresses are determined as described in section 2.

5. TOPSIDE STRUCTURE AND HULL INTERFACE

5.1. Design principles

Topside facilities include all types of equipment above main deck needed for the functional operation of the ship. This may typically be processing equipment on FPSOs or drillfloor and derrick for drilling vessels. The stress distribution in the topside structure is governed by the structural design of the topside, local loads and ship accelerations and deformation. The stresses are dominated by local static loads and both ULS a) and b) combination need to be analysed.

The interface structures between the deck and the topside structures shall satisfy several functional requirements:

- Carry the weight of the topside
- Possess sufficient capacity for the inertia loads induced by vessel accelerations due to waves.
- Possess sufficient flexibility to minimise the stresses induced by the hull deformation in all loading conditions and seastates.
- Be possible to install, maintain, repair and inspect.

The requirements shall be satisfied for all limit states.

Three principal different designs of the interface structures are described.

5.1.1. Plated flexible supports

A design of plated flexible supports in the longitudinal direction of the hull and shear plates in the transverse direction is shown in Figure D 7. Acceleration loads in the longitudinal direction is transferred to the deck by means of diagonals.

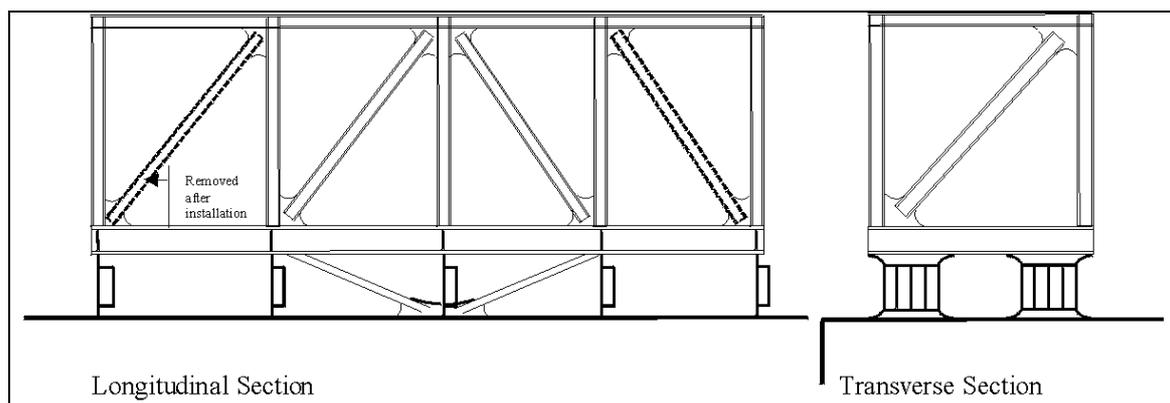


Figure D 7 Plated flexible supports

5.1.2. Portal frame

A design using a portal framed solution is shown in Figure D 8.

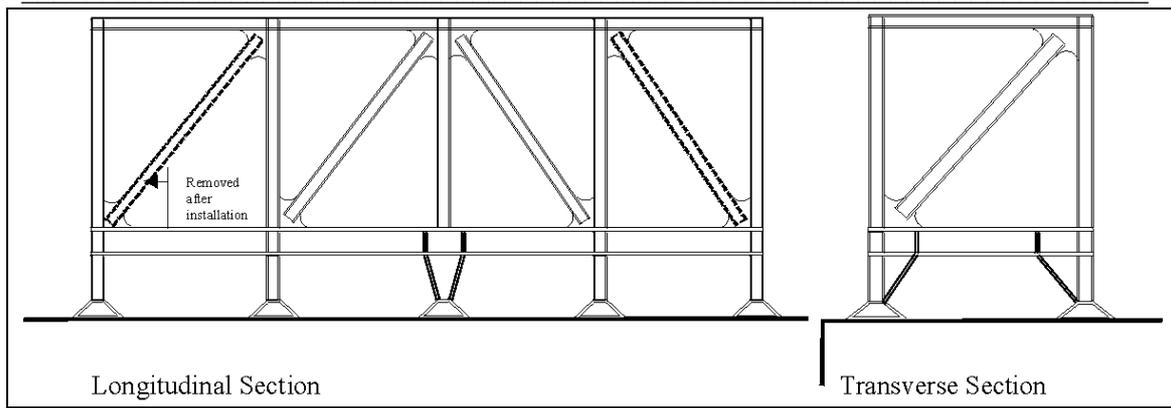


Figure D 8 Portal frame

5.1.3. Sliding support

A third design type referred to as a sliding support design is shown in Figure D 9 .The design is typically provided with bearings at the supports except for the ones that are fixed.

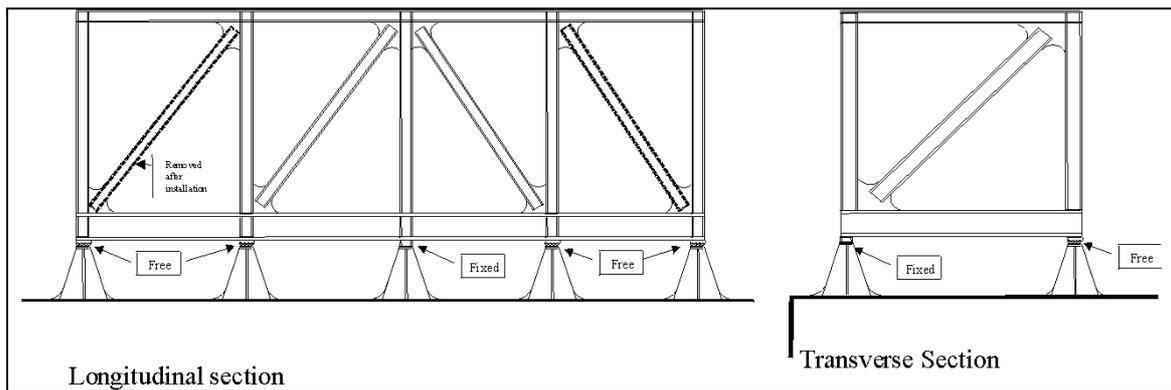


Figure D 9 Sliding support

The different designs are discussed in Table D 5

| <i>N</i> <i>o.</i> | <i>Advantages</i> | <i>Disadvantages</i> | <i>Comments/Design recommendations</i> |
|-----------------------|---|--|---|
| Plated flexible | Welded connections only Fatigue calculations can be performed for all vital components Shear panels may be positioned above web frames (No need for transverse bulkheads) The most critical connections may be located above the deck Easy to maintain and inspect | Reduce the utilisation of the deck area Exposed to green water slamming | The buckling capacity for shear panels may have to be assessed by eigen-value analysis The external water pressure on the side shell influences fatigues life. The radius of the flexible plate next to the ship-side should be designed such that this effect is minimised. |
| Portal framed | Design of structure is relatively simple in both ULS and FLS Allows for flexible use of the deck area since few and limited size of columns Simple to maintain and inspect. Possible to design such that critical welds are above main deck, i.e. possible to inspect. | | Needs a longitudinal bulkhead or large longitudinal deck stringer underneath the column in addition to a web frame due to bending moments in the frame columns. |
| Sliding supports | Reduced effect of horizontal relative displacement between main and process deck due to global hull bending. Allows for flexible use of the deck area since few and limited size of columns | Uplift forces may create major problems in the design and fabrication. Maintenance and repair of bearings is difficult. Bearings should be interchangeable. Difficult to design for fatigue due to non-linear behaviour. | Friction forces should be considered in fatigue design. |

Table D 5 Topside design principles

5.2. Loads

The design loads for local structure shall be defined. This is the maximum static loads that the local structure may experience, i.e. design loads for plates, stiffeners and girders often specified as a design load for specified areas. A topside module when loaded to the maximum local design load will be the basis for local stresses in the supporting hull structure, whilst the most likely simultaneous loads on all topside equipment will be the basis for global load condition for the hull girder.

5.3. Structural response of supporting structure

The stresses in the supporting structure depend on a number of factors:

- size of topside support stool
- location of topside support stool
- internal tank loads and external pressure
- stiffness distribution

Figure D 10 shows schematically the distribution of vertical in plane stresses from a topside support stool located in line with a longitudinal bulkhead midway between two transverse bulkheads. The distribution of vertical stresses will depend on the internal tank loading and the position of the wave. In the case of empty cargo tank, a wave crest

under the topside structure will provide good vertical support for the longitudinal bulkhead. Hence the vertical stress may be significant all the way down to the tank top. On the contrary, a wave trough will give little (less) support for the longitudinal bulkhead. This implies that the vertical stresses are mainly present in the area just below the support as the main part of the load is seen as shear stresses in the longitudinal bulkhead being supported by the transverse bulkheads.

The structural response is normally determined by means of a local finite element model. The extent of the model must be sufficient to ensure proper boundary conditions.

As a first estimate the vertical stresses at any section below the deck can be determined according to:

$$\sigma_y = \frac{P_v}{(l + 2z \tan 45) t_p + A_s} \quad \text{(D 15)}$$

where:

- P_v = Vertical load on the bulkhead considered
- l = Length of stool above deck measured along the deck
- z = Vertical distance from deck to the section considered
- t_p = Plate thickness
- A_s = Total area of vertical stiffeners within the $(l + 2z \tan 45)$ section. $A_s = 0$ for horizontally stiffened bulkheads.

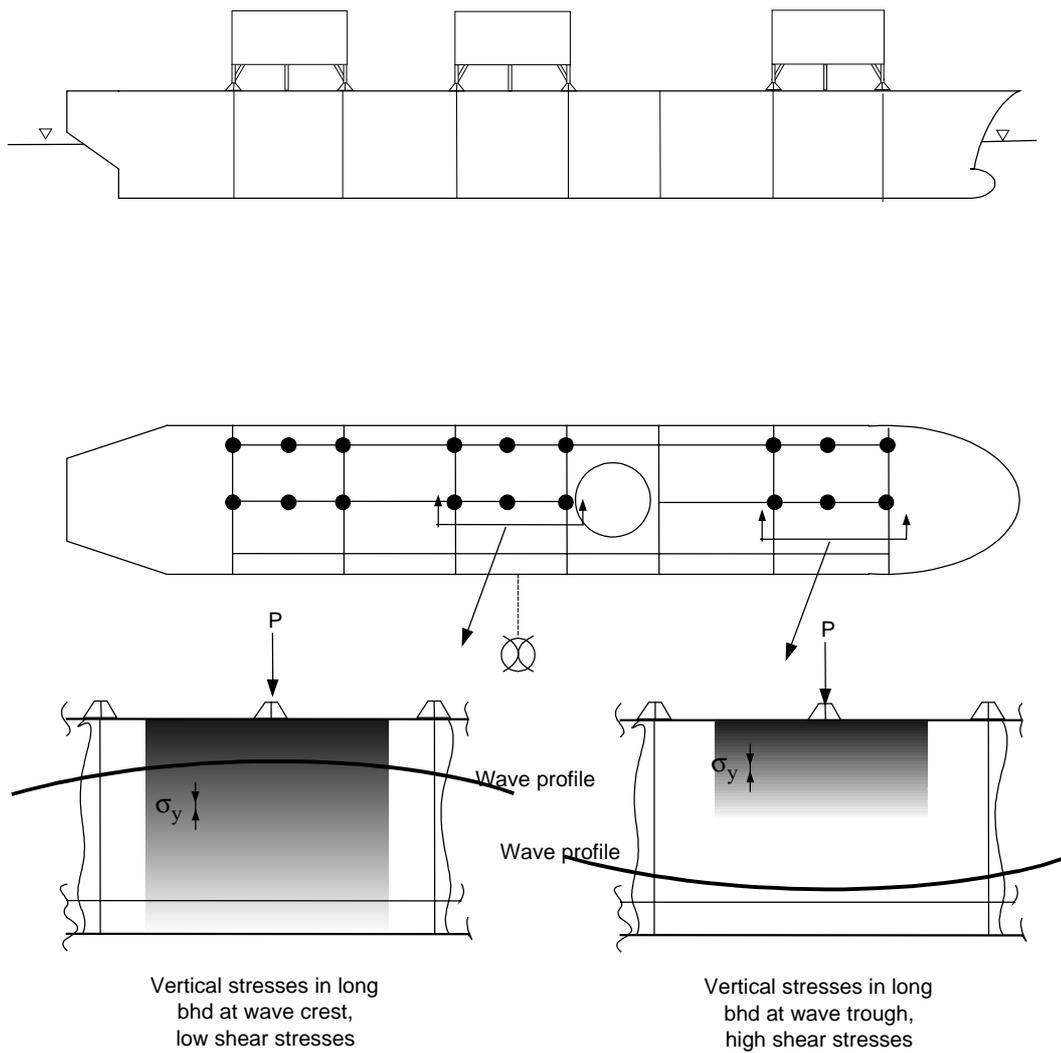


Figure D 10 Distribution of vertical stress in longitudinal bulkhead due to topside load.

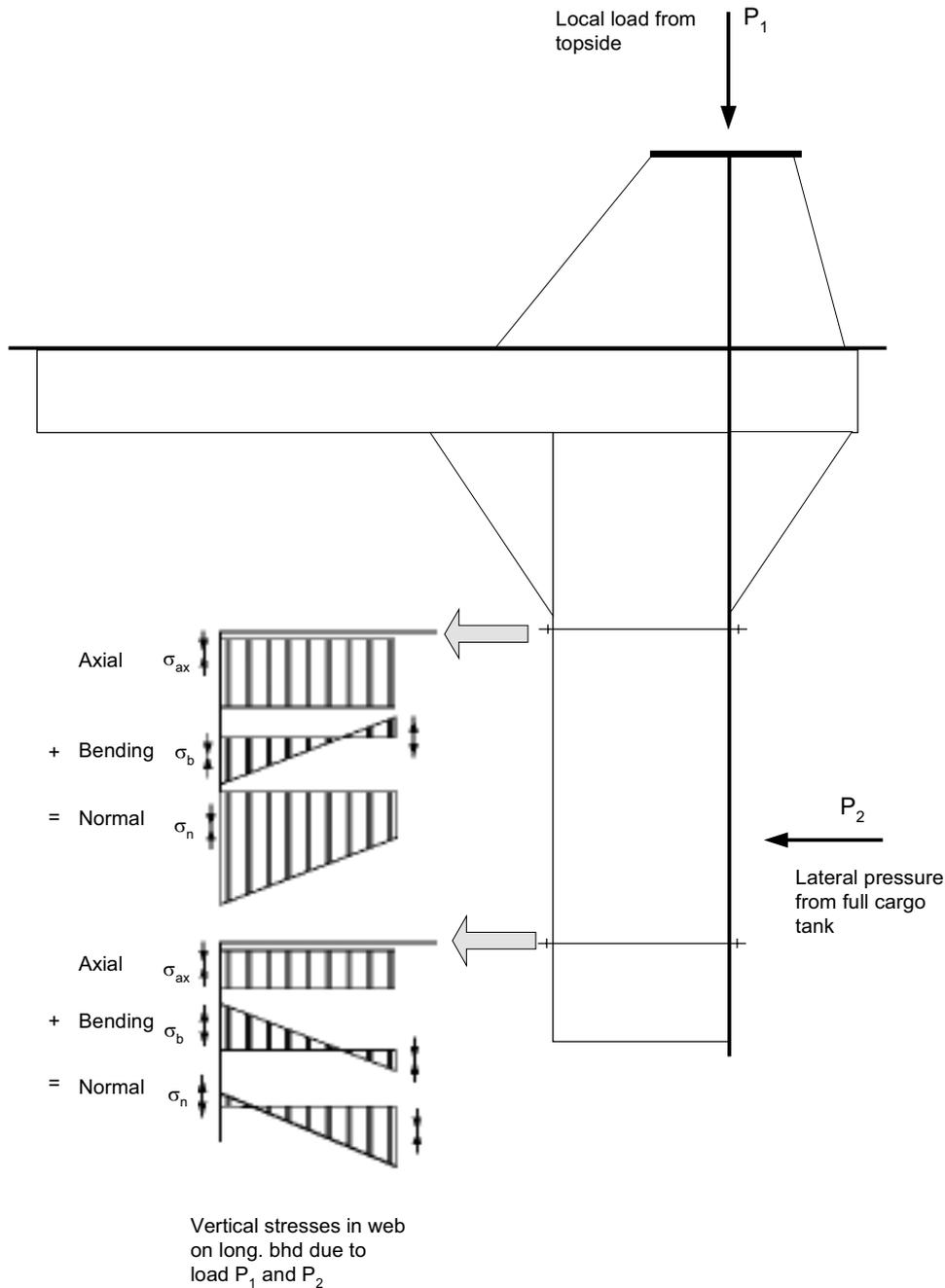


Figure D 11 Stresses in transverse web frame due to local topside load

If the topside support stools are located over transverse frames or transverse bulkheads, see Figure D 11, the loads may result in bending of the transverse deck girder which may add transverse compression stress in the deck. The vertical web on the bulkhead and the bulkhead plates will experience vertical stresses and bending stresses from the tank load. The distribution of the axial stress in the web will also depend on the position of the wave as explained above, ref. Figure D 10. Note that the figure only shows the effect of the vertical force. The local moment from the topside structure at top of the stool, the horizontal force and the moment from the horizontal force must also be considered.

6. CAPACITY CHECKS

6.1. General comments to the ULS capacity checks

The ULS capacity checks as specified in the DNV-OS-C102 Structural Design of Offshore Ships October 2000 is presently under modification. Since Classification Notes 30.1 is being revised and will also include one part employing the DNV PULS program, the description of the ULS capacity checks as given in the following sections correspond to the new requirements that will be given in DNV-OS-C102 Structural Design of Offshore Ships.

6.2. General principles

The ULS capacity checks include both checks of yield and buckling resistance. The yield check reads :

$$\sigma_{ed} \leq \frac{1}{\gamma_m} f_y$$

where:

σ_{ed} Design Von Mises equivalent stress (including load factors)
=

γ_m Material factor = 1.15
=

f_y Characteristic yield strength of the material
=

Stresses in areas with local concentrations like bracket toes and other limited areas within brackets, may significantly exceed the yield limit. This means that yielding will occur, but if the extent of yielding is governed by the forced deformation from the surrounding structure, it is considered to be acceptable. Therefore, in general it is assumed that local linear peak stresses in areas with pronounced geometrical changes may exceed the yield stress criterion given above, provided that plastic mechanisms are not developed in the adjacent structural parts, and that local buckling is avoided.

The buckling resistance of the different plate panels is calculated according to Classification Notes 30.1 - *Buckling Strength Analysis*. The stiffened plate panels shall be checked for the effect of bi-axial stresses and lateral pressure.

Stiffened flat plates should be checked for buckling. The stiffeners are typically aligned in the longitudinal direction, which is the most dominant direction with respect to compressive loads. Heavier and more widely spaced transverse girders support the stiffeners. Large girder webs may also be considered and designed as a stiffened plate.

The longitudinal girders and bulkheads may also provide support for topside equipment, such as topside modules, crane pedestals, helicopter deck, derrick, etc. In such cases the structures will be exposed to both longitudinal and transverse compression stresses.

The following definitions with respect to buckling control of stiffened plates are made:

| | |
|----------------------------------|--|
| <i>Plate buckling</i> | Local buckling of plate panels between stiffeners. The plate may be subjected to in-plane loading only, or it may be subjected to distributed lateral loads alone or combined with in-plane loading. |
| <i>Panel buckling</i> | Buckling of stiffened panels between girders. This buckling mode includes plate-induced and stiffener-induced failure, ref. Figure D 12 |
| <i>Plate-induced failure</i> | Deflection away from the plate associated with yielding in compression at the connection between plate and stiffener. The characteristic material resistance for the plate is to be used. |
| <i>Stiffener-induced failure</i> | Deflection towards the plate associated with yielding in compression in top of the stiffener or torsional buckling of the stiffener. |

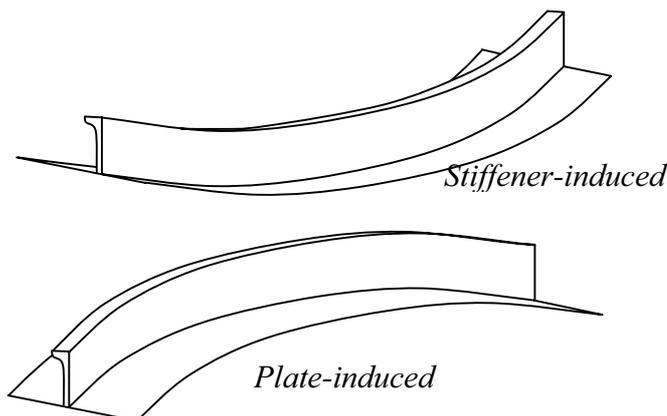


Figure D 12 Plate-induced and stiffener-induced failure

Stiffened panels shall be designed to resist the acting loads with required load and material factors. Stiffened panels which are asymmetric in geometry about the plate-plane must be checked for both plate-induced failure and stiffener-induced failure.

Girders:

Overall flexural buckling of girders may usually be disregarded. Otherwise, strength check for orthogonal stiffened panels has to be carried out. The girder strength can be assessed as a stiffened panel. Girders that are subjected to high stresses due to topside loads can be assessed as a stiffened panel.

Brackets:

Buckling capacity of large stiffened brackets is calculated according to Classification Notes 30.1 - *Buckling Strength Analysis*, section 3.7.

Non-linear strength assessment methods using recognised programs may alternatively be used. In such cases, geometrical imperfections must be included, residual stresses and boundary conditions need careful evaluation. The model should be capable of capturing all relevant buckling modes and detrimental interactions between them.

6.3. Hull girder moment capacity checks

Hull girder capacity checks are carried out for load condition 1 and 2 as defined in 3.4. In order to calculate the buckling capacity of each panel in the hull girder section, the stresses defined in section 3.6.1, 3.6.2 and 3.6.3 shall be used. This implies that the hull cross section shall be based on stresses within the elastic range of the material. In the moonpool, or similar, area, , the global stress concentration factor must be included.

Each longitudinal panel in the cross section shall be checked for permissible yield (von Mises equivalent stress) and buckling capacity.

Example procedure:

An example is given below showing the typical steps in the ULS midship section capacity checks. The capacity checks are based on a linear distribution of bending stresses over the cross section. The bending stresses from horizontal bending is in this example neglected. Normally it is assumed that the horizontal and vertical bending stresses are in phase and thus the maximum values of the bending moments are used. For simplicity in this example only a few plates are shown in the table.

Load conditions and stresses

For the bottom area, the hogging condition is used, i.e. LC.1 in section 3.4, to calculate the stillwater bending stresses and as basis for the wave load analysis. The longitudinal stresses are calculated using maximum wave bending moment. Double bottom stresses are given in the middle of the tank as the outer bottom plates and profiles are checked (compression stresses). The other local stresses as described in section 3.6.1 are neglected. The transverse stresses are due to bending of transverse frames.

For the deck area, the sagging condition is used, i.e. LC.2 in section 3.4. The longitudinal stresses are calculated using maximum wave bending moment as described in section 3.6.1. The other local stress components are negligible.

The design stresses are tabulated in Table D 6.

Capacity curves

A convenient way of checking the capacity of a stiffened panel is illustrated in Figure D 13. The capacity curves are produced for a given geometry, lateral pressure and shear stress. The panel has acceptable buckling capacity for all buckling modes when the design stresses in the transverse and longitudinal directions are below the capacity curves. For stiffened plate panels, buckling of the plate itself is accepted.

BUCKLING DESIGN STRESSES FOR STIFFENED PLATE FILEDS

| GENERAL DATA: | | DESIGN GLOBAL BENDING MOMENTS: | | | | | |
|-------------------------------|-----------------------------|--------------------------------|-----------|----------------|----------------|---------------------|------------|
| Neutral Axis | NA 10,038 [m] | Still water | Wave | Non lin. corr. | Wave corrected | Design moment M_D | |
| Moment of inertia | I 626,127 [m ⁴] | Ms [kNm] | Mw [kNm] | Nfac [-] | Mw*Nfac | Ms + LF*Mwcorr | |
| Material factor | MF 1,150 [-] | Hogging | 3 678 750 | 5 763 000 | 0,85 | 4 898 550 | 9 312 083 |
| Load factor | LF 1,15 [-] | Sagging | 2 940 057 | 7 005 000 | 1,00 | 7 005 000 | 10 995 807 |
| Allowable usage factor 1 / MF | 0,87 | | | | | | |

(Sagging moment used above NA, hogging moment below)

| PRINCIPAL DATA | | Main Deck Center Line | Inner Side | Bottom Stiff. #11 |
|--|------------------------------|--------------------------|------------|----------------------|
| Position of point, above BL | [m] | 22,8 | 21,396 | 0 |
| Modulus of Elasticity | E= [N/mm ²] | 2,10E+05 | 2,10E+05 | 2,10E+05 |
| Poissons Ratio | v= [-] | 0,3 | 0,3 | 0,3 |
| Shear Modulus | G= [N/mm ²] | 80769 | 80769 | 80769 |
| PLATE | | | | |
| Thickness | t= [mm] | 26 | 20 | 27 |
| Yield stress | Sigkb= [N/mm ²] | 315 | 315 | 315 |
| Breadth of plate | S= [mm] | 2490 | 3266,5 | 5810 |
| Length of plate (stiffener length) | L= [mm] | 4000 | 4000 | 4000 |
| Intermediate plate stiff. spacing | Ls= [mm] | 4000 | 4000 | 4000 |
| STIFFENER | | | | |
| Yield stress | Sigks= [N/mm ²] | 315 | 315 | 315 |
| Stiffener type | Type= (T,FB or A) | a | a | a |
| Height of section | he= [mm] | 430 | 340 | 430 |
| Web thickness | tw= [mm] | 21 | 14 | 21 |
| Flange breadth | bf= [mm] | 75 | 56 | 75 |
| Flange thickness | tf= [mm] | 54 | 42 | 54 |
| Stiffener spacing | s= [mm] | 830 | 818 | 830 |
| Lateral support length (tripping) | Lt= [mm] | 4000 | 4000 | 4000 |
| LOAD DATA (Uniform along edges assumed) | | | | |
| Lateral pressure (p) | Plate = [N/mm ²] | 0 | 0 | 0,271 |
| | Stiff.= [N/mm ²] | 0 | 0 | 0,145 |
| STRESSES | | | | |
| Shear Stress | Tau= [N/mm ²] | 50,00 | 60,00 | 26 |
| Long. Stresses (Compr. Neg.) | Sigma-x [N/mm ²] | -224,1 | -199,5 | -149,3 |
| Double Bottom Stresses (Compr. Neg.), Si-x | [N/mm ²] | 0 | 0 | -20,8 |
| Transv. Stresses (Compr. Neg.) | Sigma-y [N/mm ²] | -4 | -13 | -45 |
| USAGE FACTORS | | | | |
| Plate Induced | | 0,81 | 0,85 | 0,72 |
| Stiff. Induced | | 0,79 | 0,76 | 0,77 |
| Plate | | 0,82 | 0,84 | 0,59 |
| | | OK | OK | OK |
| Local Buckling of Stiffeners | | OK | OK | OK |

Table D 6 Example values for ULS buckling checks

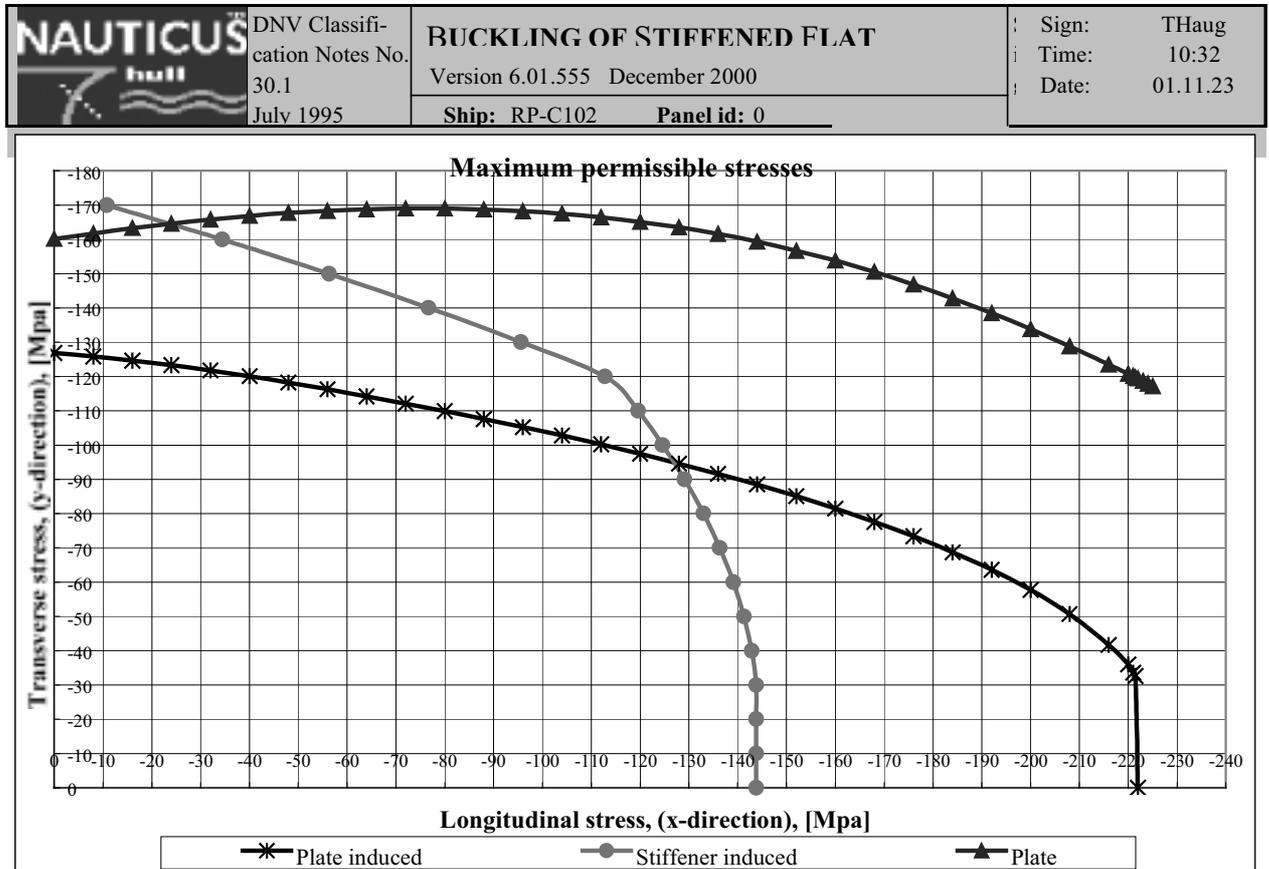


Figure D 13 Capacity curves for a stiffened panel

6.4. Hull girder shear capacity check

The global shear capacity at any section is found from the following expression:

$$Q_g = \sum_j \tau_{cr_j} A_{p_j} \quad \text{(D 16)}$$

where:

A_p = Area of panel in the shear element (plate area only)

τ_{cr} = The smaller of:

- Characteristic shear stress in panel corresponding to critical buckling capacity
- Characteristic shear stress in panel corresponding to the yield capacity of the panel

j = includes all panels in the longitudinal shear element

The global shear capacity is considered slightly different from the moment capacity checks. Each global shear element like ship side, inner side and longitudinal bulkheads is considered separately. It is not assumed any redistribution of shear forces between the

global elements, but each global element may be fully utilised for the total shear force in the element. The total shear force taken by a global shear element can be derived from a shear flow analysis of the transverse section. The shear capacity can be calculated as illustrated in the table below:

| Panel no | Fixed input parameters | | | Results | |
|----------|------------------------|--------------------|-----------------------|---|--|
| | Vertical stress | Total long. stress | Net pressure on panel | Maximum shear stress associated with yield or critical buckling | Shear capacity contribution $\tau_{cr} A_p$ |
| 1 | | | | | |
| 2 | | | | | |
| etc | | | | | |

Note that the panels must be checked for both yield and buckling. The lower of the two values is used in the “Maximum shear stress associated with yield or critical buckling”. Most buckling codes (like Classification Notes 30.1) include both a yield check and buckling check when both in-plane stress components are in compression. However, when only one stress component is in tension the yield check must be carried out separately and also a buckling check setting the tension component to zero.

6.5. Capacity of supporting structure for topside facilities

Each panel in the supporting structure for topside equipment is checked separately for yield and buckling. When the supporting structure also contributes to the hull section modulus, the global stresses in the longitudinal element is normally calculated for the ULS b) combinations only. This may be used both in the a) and b) combination as a conservative assumption.

In cases where the buckling capacity of the panels near the neutral axis is insufficient due to high vertical stresses and thinner plates, redistribution of forces may be considered according to the procedure below:

- Neglect the contribution of the thinner plates.
- Calculate new vertical stresses and shear stresses with the thinner plates “removed” from the finite element model.
- Calculate the buckling and yield capacity of the upper part of the bulkhead based on new vertical stresses and shear stresses.

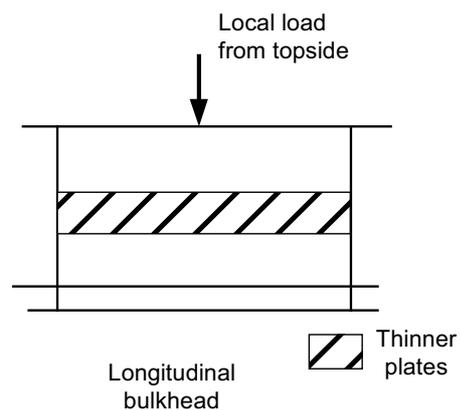


Figure D 1 Support structure for topside

6.6. Miscellaneous structures

6.6.1. Bilge Keel

Bilge keels on offshore ships are often much larger than conventional trading vessels. This is particularly the case for storage and production units which is stationary moored on the same site for many years. Bilge keels should be considered both with respect to structural capacity and fatigue.

Large bilge keels should be welded to the shell plating without the use of doubling plates. Transverse brackets or similar arrangements that provide transverse strength are often fitted.

Typical size of the bilge keel is 1m width extending over $0.4L_{pp}$. The bilge keel is often designed to be little affected by the hull girder stresses. The grade of materials and weld types are to be in accordance with Appendix A. The loads due to ship motions shall be determined from the wave load analysis. For fatigue capacity checks a viscous damping coefficient representative for seastates which contribute most to fatigue damage shall be used. This imply that the damping coefficient should be determined at 10⁻² probability of exceedance.

Fatigue calculations should be carried out by means of a spectral analysis. A component stochastic analysis as described in the Classification Notes 30.7 is acceptable. If the bilge keel is of a closed construction type, the transfer functions for stress responses from the wave dynamics and motion induced drag forces shall be determined separately. The transfer functions shall be combined in the cumulative damage calculations.

Appendix E

FATIGUE ASSESSMENT

CONTENTS

| | | |
|-----------|--|-----------|
| 1. | INTRODUCTION | 3 |
| 2. | DEFINITIONS AND SYMBOLS | 4 |
| 3. | PROCEDURE | 5 |
| 4. | BASIC FATIGUE KNOWLEDGE | 7 |
| 4.1. | SCANTLINGS | 7 |
| 4.2. | CUMULATIVE DAMAGE | 7 |
| 4.3. | S-N CURVES | 7 |
| 4.4. | MEAN STRESS EFFECTS | 7 |
| 4.5. | FATIGUE AREAS | 8 |
| 4.6. | LOAD CONDITIONS | 9 |
| 4.7. | EFFECT OF CORROSION PROTECTION | 10 |
| 4.8. | EFFECT OF INTERMITTENT WET SURFACES | 11 |
| 4.9. | COMBINED WAVE FREQUENCY AND SWELL RESPONSE | 12 |
| 4.10. | SIMPLIFIED METHOD - COMBINED WAVE FREQUENCY AND SWELL RESPONSE | 12 |
| 4.11. | SHIP RULES BASED METEOCEAN CRITERIA | 13 |
| 4.12. | WORKMANSHIP (GEOMETRICAL IMPERFECTIONS) | 14 |
| 4.13. | EFFECT OF GRINDING OF WELDS..... | 15 |
| 4.14. | THICKNESS EFFECTS..... | 15 |
| 4.15. | STRESS CONCENTRATION FACTORS | 16 |
| 4.16. | STRESS DEFINITIONS | 17 |
| 4.17. | DERIVATION OF HOT SPOT STRESS..... | 19 |
| 4.18. | FILLET WELDS AND PARTIAL PENETRATION WELDS..... | 19 |
| 4.18.1. | <i>Example of doubling plate</i> | 19 |
| 4.18.2. | <i>Example of fillet welds at scallops</i> | 21 |
| 4.18.3. | <i>Stress from 3 D FE analysis</i> | 23 |
| 5. | FATIGUE ANALYSIS METHODOLOGY AND PROCEDURES | 24 |
| 5.1. | GENERAL | 24 |
| 5.2. | SIMPLIFIED FATIGUE ANALYSIS..... | 24 |
| 5.3. | STRESS COMPONENT BASED STOCHASTIC FATIGUE ANALYSIS..... | 25 |
| 5.3.1. | <i>Determination of stress factors</i> | 26 |
| 5.3.2. | <i>Check list for the component stochastic method</i> | 28 |
| 5.4. | FULL STOCHASTIC FATIGUE ANALYSIS | 29 |
| 5.4.1. | <i>Check list for the full stochastic method</i> | 33 |
| 6. | SELECTION OF FATIGUE APPROACH | 35 |

1. Introduction

This appendix intends to give general background and provide guidance for fatigue control for Offshore Ships of conventional shape according to DNV-OS-C-102. The procedure is also applicable for fatigue control based on the CSA-2, described in Rules for Classification of Ships Pt.3 Ch.1 sec.16. The guidance is adjusted to fit offshore vessel design, however, the main principles are basically the same.

Ship designers have known fatigue cracks and fatigue damages for several decades. Initially the obvious remedy was to improve detail design. With the introduction of higher tensile steels (HTS-steels) in hull structures, at first in deck and bottom to increase hull girder strength, and later in local structures, the fatigue problem became more imminent.

The fatigue strength does not increase according to the yield strength of the steel. In fact, fatigue strength is found to be independent of the yield strength. The higher stress levels in modern hull structures have therefore led to a growing number of fatigue crack problems.

To ensure that the structure will fulfil its intended function fatigue assessment, supported where appropriate by a detailed fatigue analysis, should be carried out for each individual type of structural detail that is subjected to extensive dynamic loading. Every welded joint and attachment or other form of stress concentration is potentially a source of fatigue cracking and should be individually considered.

This appendix gives an overview of the necessary analysis to be performed such that fatigue strength of offshore vessels can be documented satisfactorily. A more detailed description of the different fatigue procedures, S-N curves, stress concentration factors, etc. is given in Classification Notes 30.7 - *Fatigue Assessment of Ship Structures*.

2. Definitions and symbols

| | | |
|-------------------------------|---|--|
| A_1 | : | Stress per unit axial force |
| A_2 | : | Stress per unit vertical bending moment |
| A_3 | : | Stress per unit horizontal bending moment |
| A_4 | : | Stress per unit torsional bending moment, if relevant |
| A_5 | : | Stress per unit relative lateral external pressure load |
| A_6 | : | Stress per unit relative lateral internal pressure load |
| D | : | Total damage for that loading condition |
| D_{air} | : | Calculated damage for the entire design life, T_{ds} , with effective corrosion protection |
| D_{corr} | : | Calculated damage for the entire design life, T_{ds} , without effective corrosion protection |
| $H_{\sigma}(\omega \theta)$ | : | Transfer function for combined local stress. |
| $H_a(\omega \theta)$ | : | Transfer function for axial force at a representative section. |
| $H_v(\omega \theta)$ | : | Transfer function for vertical bending moment at a representative section. |
| $H_h(\omega \theta)$ | : | Transfer function for horizontal bending moment. |
| $H_t(\omega \theta)$ | : | Transfer function for torsional bending moment. |
| $H_p(\omega \theta)$ | : | Transfer function for external pressure in centre of the considered panel. |
| $H_c(\omega \theta)$ | : | Transfer function for liquid cargo pressure in centre of the considered panel |
| K_g | : | Stress concentration factor due to the gross geometry of the detail considered |
| K_n | : | Additional stress concentration factor for un-symmetrical stiffeners on laterally loaded panels |
| K_w | : | Stress concentration factor due to the weld geometry |
| K_{te} | : | Additional stress concentration factor due to eccentricity tolerance |
| $K_{t\alpha}$ | : | Additionally stress concentration factor due to angular mismatch |
| T_{ds} | : | Design life in years |
| T_c | : | Coating duration |
| T_{c+5} | : | Effective corrosion protection period |
| p_d | : | Dynamic pressure amplitude below the waterline |
| r_p | : | Reduction of pressure amplitude in the surface zone |
| z_{wl} | : | Distance in m measured from actual water line. |
| σ_{xx} | : | Normal stress in stiffener direction |
| σ_{yy} | : | Normal stress normal to stiffener direction |
| τ_{xy} | : | Shear stress in plate |

3. Procedure

Fatigue analyses can be performed based on simplified analytical expressions for fatigue lives or on a more refined analysis where the loading and the load effects are calculated by numerical analysis. The fatigue analysis may also be performed based on a combination of simplified and refined techniques.

The load effects shall generally be based on the hydrodynamic loads and response from the structural analysis.

Different procedures for fatigue calculations are shown in Figure E 1. These procedures are described in more detail in sections 3, 4 and 5.

Which approach to use for different areas of the ship is discussed in section 6.

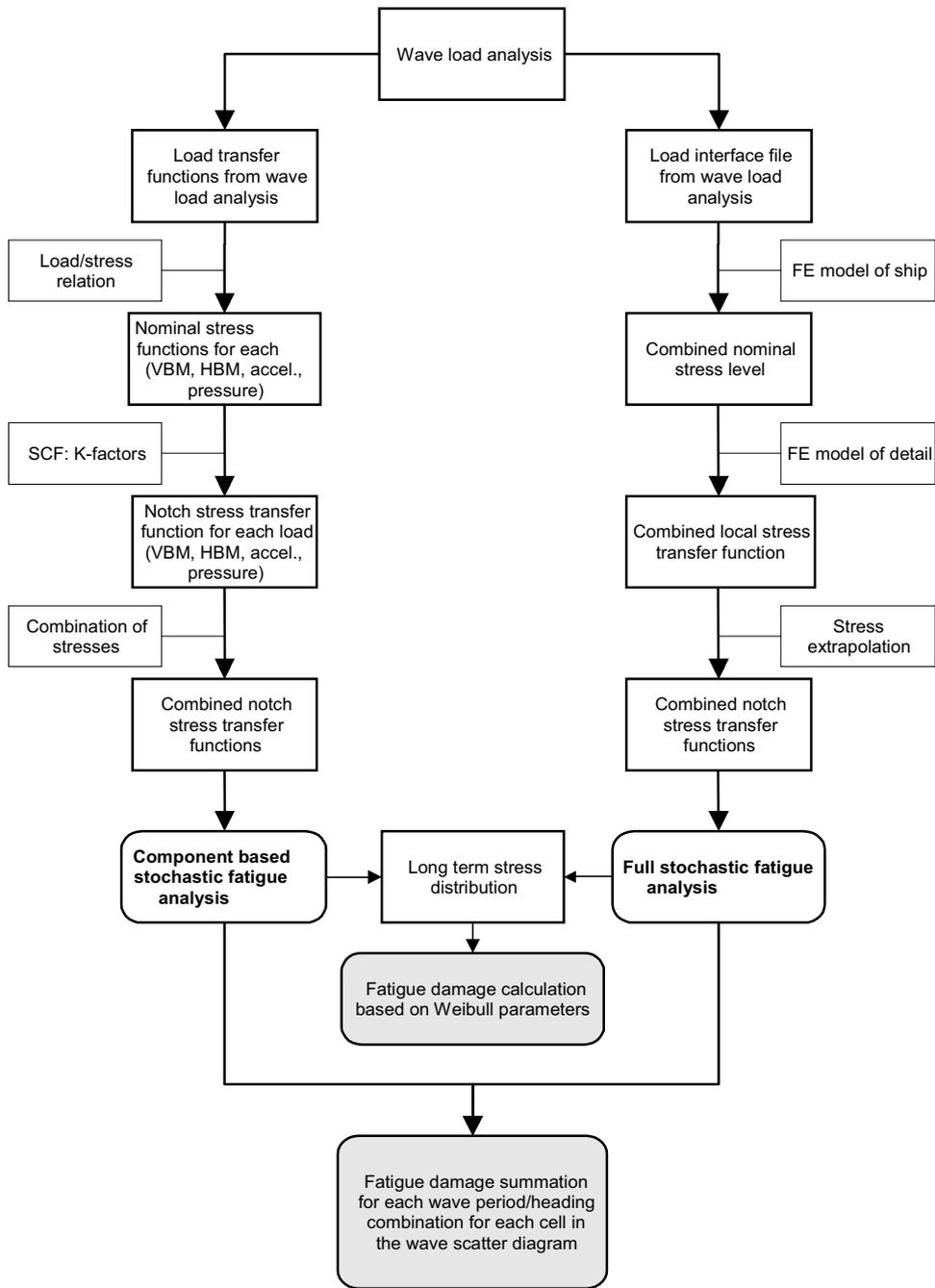


Figure E 1 - Flow diagram showing different fatigue analysis procedures

4. Basic fatigue knowledge

This section gives an overview of conditions influencing on the fatigue life of ship structures and basic knowledge of fatigue calculations. Only a brief explanation is made for each part. Reference is given to relevant sections of the Classification Notes 30.7 - *Fatigue Assessment of Ship Structures*. Only contents valid for global structural analysis are included. It is recommended to familiarise with the contents of Classification Notes 30.7 in order to get a better understanding of the fatigue concept.

4.1. Scantlings

Calculations carried out in connection with the fatigue limit state may be based on gross thicknesses (i.e. without deducting the corrosion additions), provided a corrosion protection system in accordance with DNV-OS-C101 Design of Steel Structures is maintained.

In other cases all structural calculations are to be carried out on reduced scantlings, i.e. corrosion addition according to the Rules for Classification of Ships Pt.3 Ch.1 Sec.2 is to be deducted from the actual scantlings. This applies for both global and local stresses.

4.2. Cumulative damage

Fatigue damage ratio is calculated according to the linear cumulative damage approach using the Palmgren-Miner hypothesis. The damage may either be calculated on basis of the long-term stress range distribution using Weibull parameters, or on summation of damage from each short-term distribution in the scatter diagram. Reference is given to Classification Notes 30.7, sections 2.1.1 to 2.1.3.

4.3. S-N Curves

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

See Classification Notes 30.7 - *Fatigue Assessment of Ship Structures*, section 2.3 for definition of S-N curves and DNV-OS-C102 for selection of S-N curves.

4.4. Mean stress effects

The procedure for the fatigue analysis is based on the assumption that it is only necessary to consider the ranges of cyclic principal stresses in determining the fatigue endurance. However, in design of conventional trading vessels some reduction in the fatigue damage accumulation has been credited when parts of the stress cycle range are in compression. Reference is made to Classification Notes 30.7 - *Fatigue Assessment of Ship Structures*, section 2.2.

According to DNV-OS-C101/102, fatigue life improvements based on mean stress level should not be applied for welded joints.

4.5. Fatigue areas

The potential for fatigue damage is dependent on weather conditions, corrosion protection, location on ship, structural detail, weld geometry and workmanship. The potential danger of a fatigue damage will also vary according to crack location and number of potential damage points.

Most of the existing FPSOs have not been in service for a sufficient period of time in order to verify the design premises for the unit. Therefore, the experience from tanker structures showing similar details is of importance when hot spot areas to be analysed are considered. Fatigue damages are known to occur frequently for some ship types and categories of hull structure elements. The fatigue life, is in particular, 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, which all vary depending on ship type and structure considered. The importance of a possible fatigue damage is related to the number of potential damage points of the considered type for the ship or structure in question and to its consequences.

A major fraction of the total number of fatigue damages on ship structures occurs in panel stiffeners on the ship side and bottom and on the tank boundaries of ballast- and cargo tanks. However, the calculated fatigue life depends on the type of stiffeners used, and the detail design of the connection to supporting girder webs and bulkheads. In general asymmetrical profiles will have a reduced fatigue life compared to symmetrical profiles unless the reduced efficiency of the asymmetrical profile is compensated for by an improved design for the attachment to transverse girder webs and bulkhead structures.

Structural elements in the cargo area being of possible interest for fatigue evaluation are listed in Table E 1. More information about fatigue sensitive locations can be found in the Rules for Classification of Ships Pt.3 Ch.1 Sec.16 C700 and in CN 30.7.

Table E 1 - Example of hot spot areas to be considered

| Structural member | Structural detail | Load type |
|---|--|---|
| Side-, bottom- and deck plating and longitudinals | Butt joints, doubling plates, module support stools, deck openings and attachment to transverse webs, transverse bulkheads, hopper knuckles and intermediate longitudinal girders | Hull girder bending, stiffener lateral pressure load and support deformation from topside inertia loads |
| Transverse girder and stringer structures | Bracket toes, girder flange butt joints, curved girder flanges, knuckle of inner bottom and sloped hopper side and other panel knuckles including intersection with transverse girder webs. Single lug slots for panel stiffeners, access and lightening holes | Sea pressure load combined with cargo or ballast differential pressure load |
| Longitudinal girders of deck and bottom structure | Bracket terminations of abutting transverse members (girders, stiffeners) | Hull girder bending, and bending / deformation of longitudinal girder and considered abutting member |
| FPSO specific details | Moonpool and turret Riser hang off platforms Crane pedestals Topside modules including flare tower and helideck | Anchoring forces and FPSO accelerations |

4.6. Load conditions

Vessel response may differ significantly for different loading conditions. It is therefore of major importance to include response for actual loading conditions. Since fatigue is a result of numerous cyclic loads, only the most frequent loading conditions are included in the fatigue analysis. These will normally be ballast, intermediate and full load condition. Under certain circumstances, other loading conditions may be used.

The fraction of design life in the fully loaded cargo and ballast conditions, p_n , may be taken according to the DNV-OS-C-102, Section 7. Unless otherwise documented, 50% in full loaded, 25% in intermediate condition and 25% in ballast condition may be used. If the difference in draught in ballast and loaded conditions is less than 4-5 metres one may consider only full load and ballast condition. Typical values would be 60% in full load and 40% in ballast.

According to Rules for Classification of Ships a tanker is analysed for fatigue in loaded and ballast conditions. It is assumed that the tanker is in a loaded condition 45 % of the lifetime and 40 % in ballast condition. The remaining 15 % are considered to be proportion of time spent in port or dock. This may apply to drilling and service types of vessels.

4.7. Effect of corrosion protection

Depending on the required accuracy of the fatigue evaluation it may be recommended to divide the design life into several time intervals due durability of the corrosion protection.

It is recognised that the fatigue life of steel structures is considerably shorter in freely corroding condition submerged in sea water than in air, i.e. in dry indoor atmosphere such as common laboratory air. For steel submerged in sea water and fully cathodic protected, approximately the same fatigue life as in dry air is obtained for small stress ranges.

An intact coating system will also protect the steel surface from the corrosive environment, so that the steel can be considered to be as in dry air condition.

The basic S-N curve for welded regions in air is only to be applied for joints situated in dry spaces or joints effectively protected against corrosion.

Estimating the efficient lifetime of coating- and cathodic protection systems, due consideration is to be given to specification, application and maintenance of the systems. Normally a durable corrosion protection system is used for FPSOs that should be efficient for the considered service life.

The design life may be divided into one interval with good corrosion protection and one interval where the corrosion protection is more questionable for which different S-N data should be used, see Section 3.2. Each of these intervals should be divided into loaded and ballast condition, or other conditions if relevant.

Damage in one loading condition may then be calculated according to equation (E 1) as:

$$D = D_{air} \cdot \frac{T_C + 5}{T_{ds}} + D_{corr} \cdot \frac{T_{ds} - (T_C + 5)}{T_{ds}} \quad \text{if } T_C > 0 \quad \text{(E 1)}$$

otherwise $D = D_{corr}$

where D : total damage for that loading condition
 D_{air} : calculated damage for the entire design life, T_{ds} , with effective corrosion protection
 D_{corr} : calculated damage for the entire design life, T_{ds} , without effective corrosion protection
 T_C : coating duration
 T_C+5 : effective corrosion protection period

T_{ds} : design life of ship in years

See also Classification Notes 30.7 - Fatigue Assessment of Ship Structures.

4.8. Effect of intermittent wet surfaces

Due to intermittent wet and dry surfaces, the range of the pressure is reduced above $T_{act} - z_{wl}$, see Figure E 2. The dynamic external pressure amplitude (half pressure range), p_e , related to the draught of the load condition considered, may be taken as:

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

where ;

p_d : dynamic pressure amplitude below the waterline

r_p : 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} : Relative wave motion in m measured from a the actual still water draught at a probability level 10^{-4} . In the area of side shell above $z : T_{act} + z_{wl}$ it is assumed that the external sea pressure will not contribute to fatigue damage.

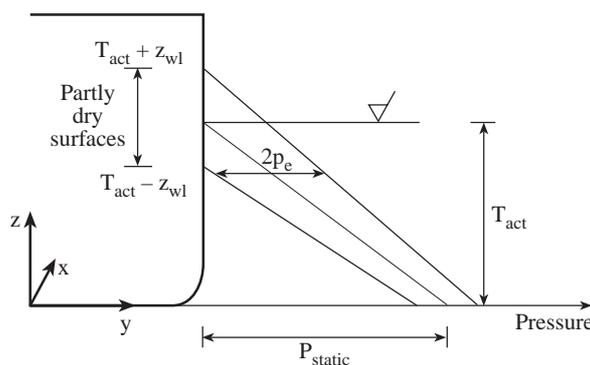


Figure E 2 - Reduced pressure ranges in the surface region

4.9. Combined wave frequency and swell response

The wave frequency response is calculated as described in Classification Notes 30.7 with a $\cos^n\theta$ spreading function, where n normally is 2.

The response due to swell may be calculated similarly to the response to wind-waves in Appendix C “Hydrodynamic Analysis” using the Jonswap spectrum with peak enhancement factor equal to 5 and a $\cos^8\theta$ spreading. The responses to wind-waves and to swell are independent, and hence the combined effect may be obtained by adding the variances. The up-crossing period of the combined response through the mean level may be computed using the sums of the respective spectral moments.

A combined response is calculated as described under the fatigue limit state in DNV-OS-E301 Position Mooring (Chapter 2 Section 2).

Alternatively a simplified calculation of fatigue damage can be carried out as described 4.10

4.10. Simplified method - combined wave frequency and swell response

In some situations it is convenient to use a simplified method in the calculation of fatigue damage from the combined wave frequency and swell responses.

For example:

- Calculate the fatigue damage D1 for wave frequency response only.
- Calculate the fatigue damage D2 for the swell response only.

It should be noted that adding these fatigue damages linearly to a resulting fatigue damage is non-conservative and is normally not accepted.

A conservative value for the combined fatigue damage can be calculated as:

$$D = D_1 \left(1 - \frac{v_2}{v_1}\right) + \left\{ \left(\frac{v_2}{v_1} D_1\right)^{1/m} + (D_2)^{1/m} \right\}^m \quad (\text{E } 3)$$

where:

- D1 : Fatigue damage due to wave frequency response only
- D2 : Fatigue damage due to swell response only
- v_1 : Mean zero up crossing frequency for wave frequency response
- v_2 : Mean zero up crossing frequency for swell response
- M : Inverse slope of the S-N curve for number of cycles less than 107

4.11. Ship Rules Based Metocean Criteria

The long-term distribution of load responses for fatigue analyses may be estimated using the wave climate represented by the distribution of Hs and Tz. Accumulated fatigue damage for the period spent on a specific location should preferably be based on a scatter for that location. If no site specific scatter diagram is used for the operational phase, the North Atlantic scatter diagram should be used. If the average sailing in transit is mainly in harsh environments, North Atlantic scatter diagram should be used, otherwise World Wide scatter diagram can be used as basis. The scatter diagrams are given in Table E 2 and Table E 3.

Table E 2 - Scatter diagram for the North Atlantic

| TZ(s) | 3.5 | 4.5 | 5.5 | 6.5 | 7.5 | 8.5 | 9.5 | 10.5 | 11.5 | 12.5 | 13.5 | 14.5 | 15.5 | 16.5 | 17.5 | Sum |
|--------|-----|-----|------|------|-------|-------|-------|-------|------|------|------|------|------|------|------|--------|
| Hs (m) | | | | | | | | | | | | | | | | |
| 1.0 | 0 | 72 | 1416 | 4594 | 4937 | 2590 | 839 | 195 | 36 | 5 | 1 | 0 | 0 | 0 | 0 | 14685 |
| 2.0 | 0 | 5 | 356 | 3299 | 8001 | 8022 | 4393 | 1571 | 414 | 87 | 16 | 3 | 0 | 0 | 0 | 26167 |
| 3.0 | 0 | 0 | 62 | 1084 | 4428 | 6920 | 5567 | 2791 | 993 | 274 | 63 | 12 | 2 | 0 | 0 | 22196 |
| 4.0 | 0 | 0 | 12 | 318 | 1898 | 4126 | 4440 | 2889 | 1301 | 445 | 124 | 30 | 6 | 1 | 0 | 15590 |
| 5.0 | 0 | 0 | 2 | 89 | 721 | 2039 | 2772 | 2225 | 1212 | 494 | 162 | 45 | 11 | 2 | 1 | 9775 |
| 6.0 | 0 | 0 | 1 | 25 | 254 | 896 | 1482 | 1418 | 907 | 428 | 160 | 50 | 14 | 3 | 1 | 5639 |
| 7.0 | 0 | 0 | 0 | 7 | 85 | 363 | 709 | 791 | 580 | 311 | 131 | 46 | 14 | 4 | 1 | 3042 |
| 8.0 | 0 | 0 | 0 | 2 | 27 | 138 | 312 | 398 | 330 | 197 | 92 | 35 | 12 | 3 | 1 | 1547 |
| 9.0 | 0 | 0 | 0 | 1 | 8 | 50 | 128 | 184 | 171 | 113 | 58 | 24 | 9 | 3 | 1 | 750 |
| 10.0 | 0 | 0 | 0 | 0 | 3 | 17 | 50 | 80 | 82 | 59 | 33 | 15 | 6 | 2 | 1 | 348 |
| 11.0 | 0 | 0 | 0 | 0 | 1 | 6 | 18 | 33 | 37 | 29 | 17 | 8 | 3 | 1 | 0 | 153 |
| 12.0 | 0 | 0 | 0 | 0 | 0 | 2 | 7 | 13 | 15 | 13 | 8 | 4 | 2 | 1 | 0 | 65 |
| 13.0 | 0 | 0 | 0 | 0 | 0 | 1 | 2 | 5 | 6 | 6 | 4 | 2 | 1 | 0 | 0 | 27 |
| 14.0 | 0 | 0 | 0 | 0 | 0 | 0 | 1 | 2 | 2 | 2 | 2 | 1 | 1 | 0 | 0 | 11 |
| 15.0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 1 | 1 | 1 | 1 | 0 | 0 | 0 | 0 | 4 |
| 16.0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 1 | 0 | 0 | 0 | 0 | 0 | 1 |
| Sum | 0 | 77 | 1849 | 9419 | 20363 | 25170 | 20720 | 12596 | 6087 | 2465 | 872 | 275 | 81 | 20 | 6 | 100000 |

Table E 3 - Scatter diagram for the World Wide operation

| TZ(s) | 3.5 | 4.5 | 5.5 | 6.5 | 7.5 | 8.5 | 9.5 | 10.5 | 11.5 | 12.5 | 13.5 | 14.5 | 15.5 | 16.5 | 17.5 | Sum |
|--------|-----|------|-------|-------|-------|-------|-------|------|------|------|------|------|------|------|------|--------|
| Hs (m) | | | | | | | | | | | | | | | | |
| 1.0 | 311 | 2734 | 6402 | 7132 | 5071 | 2711 | 1202 | 470 | 169 | 57 | 19 | 6 | 2 | 1 | 0 | 26287 |
| 2.0 | 20 | 764 | 4453 | 8841 | 9045 | 6020 | 3000 | 1225 | 435 | 140 | 42 | 12 | 3 | 1 | 0 | 34001 |
| 3.0 | 0 | 57 | 902 | 3474 | 5549 | 4973 | 3004 | 1377 | 518 | 169 | 50 | 14 | 4 | 1 | 0 | 20092 |
| 4.0 | 0 | 4 | 150 | 1007 | 2401 | 2881 | 2156 | 1154 | 485 | 171 | 53 | 15 | 4 | 1 | 0 | 10482 |
| 5.0 | 0 | 0 | 25 | 258 | 859 | 1338 | 1230 | 776 | 372 | 146 | 49 | 15 | 4 | 1 | 0 | 5073 |
| 6.0 | 0 | 0 | 4 | 63 | 277 | 540 | 597 | 440 | 240 | 105 | 39 | 13 | 4 | 1 | 0 | 2323 |
| 7.0 | 0 | 0 | 1 | 15 | 84 | 198 | 258 | 219 | 136 | 66 | 27 | 10 | 3 | 1 | 0 | 1018 |
| 8.0 | 0 | 0 | 0 | 4 | 25 | 69 | 103 | 99 | 69 | 37 | 17 | 6 | 2 | 1 | 0 | 432 |
| 9.0 | 0 | 0 | 0 | 1 | 7 | 23 | 39 | 42 | 32 | 19 | 9 | 4 | 1 | 1 | 0 | 178 |
| 10.0 | 0 | 0 | 0 | 0 | 2 | 7 | 14 | 16 | 14 | 9 | 5 | 2 | 1 | 0 | 0 | 70 |
| 11.0 | 0 | 0 | 0 | 0 | 1 | 2 | 5 | 6 | 6 | 4 | 2 | 1 | 1 | 0 | 0 | 28 |
| 12.0 | 0 | 0 | 0 | 0 | 0 | 1 | 2 | 2 | 2 | 2 | 1 | 1 | 0 | 0 | 0 | 11 |
| 13.0 | 0 | 0 | 0 | 0 | 0 | 0 | 1 | 1 | 1 | 1 | 0 | 0 | 0 | 0 | 0 | 4 |
| 14.0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 1 | 0 | 0 | 0 | 0 | 0 | 0 | 1 |
| Sum | 331 | 3559 | 11937 | 20795 | 23321 | 18763 | 11611 | 5827 | 2480 | 926 | 313 | 99 | 29 | 9 | 0 | 100000 |

The actual orientation of the vessel has to be taken into account relative to the directionality of the waves and swell for calculation of response. The mooring system (fixed or weathervaning)

needs to be considered. I.e. the orientation of the vessel has to be taken into account relative to the directionality of the waves and number of headings to be used in the analysis are to be chosen based on this.

The scatter diagrams are equal for all wave directions and specified at class midpoint values. Wave spreading shall normally be included, see Appendix C. Other regulations than DNV may specify other wave spreading functions.

4.12. Workmanship (geometrical imperfections)

The fatigue life of a welded joint is much dependent on the local stress concentrations factors arising from surface imperfections during the fabrication process, consisting of weld discontinuities and geometrical deviations

Surface weld discontinuities are weld toe undercuts, cracks, overlaps, porosity, slag inclusions and incomplete penetration. Geometrical imperfections are defined as misalignment, angular distortion, excessive weld reinforcement and otherwise poor weld shapes.

Embedded weld discontinuities like porosity and slag inclusion are less harmful for the fatigue strength when kept below normal workmanship levels.

Classification Notes 30.7 gives equations for calculation of K_g -factors due to fabrication tolerances for alignment of butt joints and cruciform joints, and the local weld geometry. Normally the default values given in the tables in Classification Notes 30.7 should be used if not otherwise defined. These normal default values are estimated assuming geometrical imperfections within limits normally accepted according to good shipbuilding practices. The S-N curves given in this classification note are assumed to include the effect of surface weld discontinuities representative for normal, good workmanship.

In special cases, K-factors may be calculated based on a specified, higher standard of workmanship. However, care should be taken not to underestimate the stress concentration factors by assuming a quality level that is difficult to achieve and follow up during production.

4.13. Effect of grinding of welds

For welded joints involving potential fatigue cracking from the weld toe an improvement in strength by a factor of at least 2 on fatigue life can be obtained by controlled local machining or grinding of the weld toe.

The benefit of grinding may be claimed only for welded joints which are adequately protected from sea water corrosion.

In the case of partial penetration welds; when failure from the weld root is considered, grinding of the weld toe will not give an increase in fatigue strength.

Note that grinding of welds should not be used as a “design tool”, but rather as a mean to lower the fatigue damage when special circumstances has made it necessary. This should be used as a reserve if the stress in special areas turn out to be larger than estimated at an earlier stage of the design. More details are given in Classification Notes 30.7.

4.14. Thickness effects

The fatigue strength of welded joints is to some extent dependent on plate thickness and on the stress gradient over the thickness. Thus for thickness larger than 25 mm, the S-N curve in air reads

$$\log N = \log \bar{a} - \frac{m}{4} \log \left(\frac{t}{25} \right) - m \log \Delta \sigma \quad (\text{E } 4)$$

where t is thickness (mm) through which the potential fatigue crack will grow. This S-N curve in general applies to all types of welds except butt-welds with the weld surface dressed flush and small local bending stress across the plate thickness. For fatigue analysis of details where the stress concentration factor is less than 1.3, the thickness effect can be neglected and the basic S-N curve can be used. Such stress concentration factors are normally only achieved through grinding or machining of the weld/base material transition.

The above expression is equivalent with a reduction in the allowable stress range of

$$\Delta \sigma_{allow} = \left(\frac{25}{t} \right)^{\frac{1}{4}} \quad (\text{E } 5)$$

or an increase of the response with:

$$\Delta \sigma_{resp} = \left(\frac{t}{25} \right)^{\frac{1}{4}} \quad (\text{E } 6)$$

4.15. Stress concentration factors

Stress concentration factors or K-factors may be determined according to Classification Notes 30.7, based on fine mesh FE analyses or obtained from the selection of factors for typical details in ships.

The notch stress range governs the fatigue life of a detail. For components other than smooth specimens the notch stress is obtained by multiplication of the nominal stress by K-factors. The K-factors in this document are thus defined as

$$K = \frac{\sigma_{notch}}{\sigma_{nominal}}$$

The S-N curves in Section 3.2 are given for smooth specimens where the notch stress is equal to the nominal stress: $K = 1.0$.

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

$$\Delta\sigma = K \cdot \Delta\sigma_{nominal}$$

All stress risers have to be considered when evaluating the notch stress. This can be done by multiplication of K-factors arising from different causes. The resulting K-factor to be used for calculation of notch stress is derived as

$$K = K_g \cdot K_w \cdot K_{te} \cdot K_{t\alpha} \cdot K_n$$

where

- K_g : Stress concentration factor due to the gross geometry of the detail considered
- K_w : Stress concentration factor due to the weld geometry. K_w : 1.5 if not stated otherwise
- K_{te} : Additional stress concentration factor due to eccentricity tolerance (normally used for plate connections only)
- $K_{t\alpha}$: Additionally stress concentration factor due to angular mismatch (normally used for plate connections only)
- K_n : Additional stress concentration factor for un-symmetrical stiffeners on laterally loaded panels, applicable when the nominal stress is derived from simple beam analyses.

The different K-factors for typical details in ships may be found in Classification Notes 30.7.

4.16. Stress definitions

The stress level obtained from a structural FE analysis will depend on the fineness of the model. Effects from all stress raisers that are not implicitly included in fatigue test data and corresponding S-N curves must be taken into account in the stress analysis. In order to correctly determine the stresses to be used in fatigue analyses, it is important to note the definition of the different stress categories:

- Nominal stresses:* stress calculated in the sectional area under consideration, disregarding the local stress raising effects of the structural detail, i. e. the stress increase due to the structural joint configuration (structural effect) as well as that of the weld toe (notch effects) for ship structure normally subdivided into:
- global stresses
 - local stresses acting secondary structural elements
-

Nominal stress is usually derived from coarse FEM models and are combined with SCF's in fatigue analysis

Geometric stresses: includes nominal stresses and stresses due to structural discontinuities and presence of attachments, but excluding stresses due to presence of welds. Stresses derived from fine mesh FE models are geometric stresses. Effects caused by fabrication imperfections as misalignment of structural parts, are normally not included in FE analyses, and must be separately accounted for. The greatest value of the extrapolation to the weld toe of the geometric stress distribution immediately outside the region effected by the geometry of the weld, is commonly denoted hot spot stress.

Notch stress: is the total stress at the weld toe (hot spot location) and includes the geometric stress and the stress due to the presence of the weld. The notch stress may be calculated by multiplying the hot spot stress by a stress concentration factor, or more precisely the theoretical notch factor, K_w . FE may be used to determine the notch stress. However, because of the small notch radius and the steep stress gradient at a weld, a very fine mesh is needed.

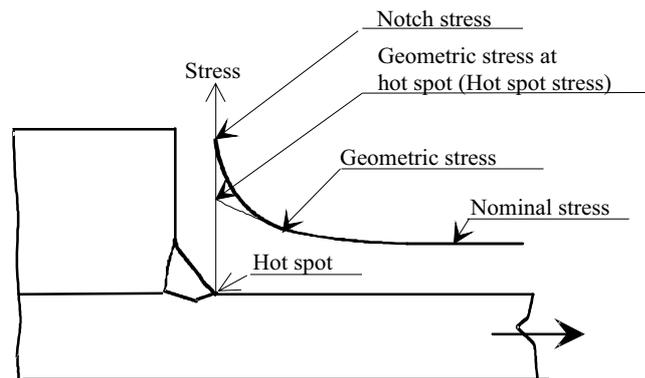


Figure E 3 - Definition of stress categories

Fatigue cracks are assumed to be independent of principal stress direction within 45° of the normal to the weld toe. Maximum principal stress in this sector, see below, is to be used in the fatigue analysis.

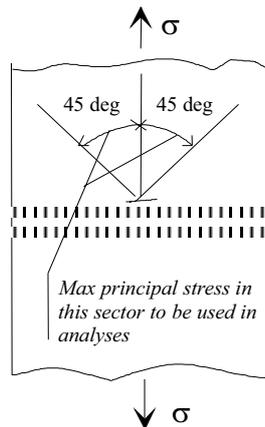


Figure E 4 - Stress directions to be considered

Principal stress is the maximum/minimum tensile/compressive stresses in the element acting in a plane with no shear stress and is defined as the following for shell elements:

$$\begin{Bmatrix} \sigma_1 \\ \sigma_2 \end{Bmatrix} = \frac{\sigma_{xx} + \sigma_{yy}}{2} \pm \sqrt{\left(\frac{\sigma_{xx} - \sigma_{yy}}{2}\right)^2 + \tau_{xy}^2} \quad (\text{E } 7)$$

where:

- σ_{xx} : normal stress in stiffener direction
- σ_{yy} : normal stress normal to stiffener direction
- τ_{xy} : shear stress in plate

For solid elements, the formula has to be extended to three dimensions.

4.17. Derivation of hot spot stress

Stresses derived from stress concentration models are to be extrapolated to the hot spot according to the procedure given in Classification Notes 30.7

Nominal stresses found from other models should be multiplied with appropriate stress concentration factors as described in Section 4.15.

4.18. Fillet welds and partial penetration welds

Design should be performed such that fatigue cracking from the root is less likely than from the toe region. The reason for this is that a fatigue crack at the toe can be found by in-service inspection while a fatigue crack starting at the root can not be discovered before the crack has grown through the weld. Thus the design of the weld geometry should be performed such that the fatigue life for cracks starting at the root is longer than the fatigue life of the toe. The evaluation of required throat thickness and penetration may be based on DNV-RP-C203.

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

$$\Delta\sigma_w = \sqrt{\Delta\sigma_{\perp}^2 + \Delta\tau_{\perp}^2 + 0.2\Delta\tau_{\parallel}^2} \quad (\text{E } 8)$$

The total stress fluctuation (i.e. the maximum compression and the maximum tension) should be considered to be transmitted through the welds for fatigue assessments. The S-N curve W3 should be used for calculation of the fatigue damage.

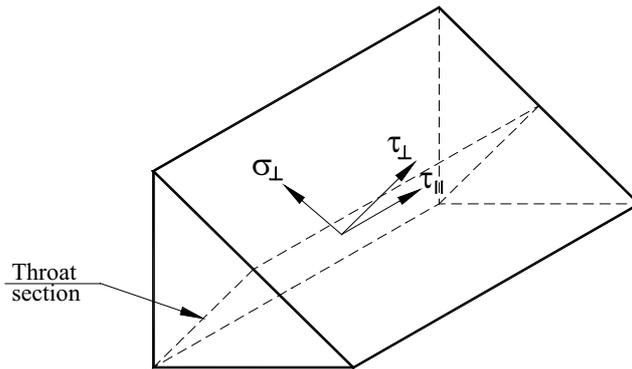


Figure E 5 – Stresses on the throat section of a weld

4.18.1. Example of doubling plate

The nominal stress 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 E 6, then the nominal stress, σ_d , should be derived from the 3 dimensional FE analysis. The nominal stress, σ_d , can then be used to calculate the stress in the weld based on considerations of equilibrium as follows:

Equilibrium in the transverse direction:

$$\sigma_{\perp} = \tau_{\perp} \quad (\text{E } 9)$$

Equilibrium in the longitudinal direction:

$$\sigma_d t_d = \frac{1}{2} \sqrt{2} (\sigma_{\perp} + \tau_{\perp}) a \quad (\text{E } 10)$$

From the equations (E 10 and (E 11, the stress to be used with the W3 curve is:

$$\sigma_w = \sigma_d \frac{t_d}{a} \quad (\text{E } 11)$$

For a bi-axial stress field, the hot spot with the largest principal stress range should be analysed.

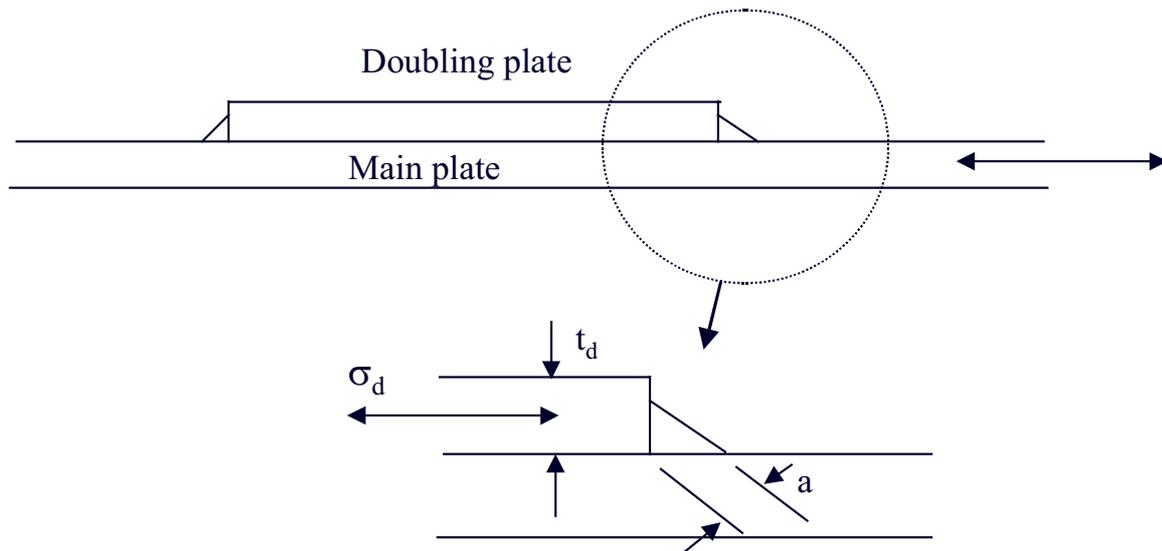


Figure E 6 – Example with fatigue design of a fillet weld

4.18.2. Example of 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 E 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 E 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:

$$A_w = 2da + at + 2a^2 \quad (\text{E } 12)$$

The corresponding nominal specimen section area is calculated as:

$$A_s = dt \quad (\text{E } 13)$$

From the requirement for equilibrium:

$$\sigma_n A_s = \sigma_w A_w \quad (\text{E } 14)$$

giving:

$$\sigma_w = \frac{A_s}{A_w} \sigma_n \quad (\text{E } 15)$$

The nominal stress in the bracket is now calculated based on the stress at the Gaussian points 3 and 4 as shown in Figure E 9. The nominal stress is calculated as:

$$\sigma_n = \frac{1}{4} \left((\sigma_3 + \sigma_4)_{\text{At element surface}} + (\sigma_3 + \sigma_4)_{\text{At middle of element}} \right) \quad (\text{E } 16)$$

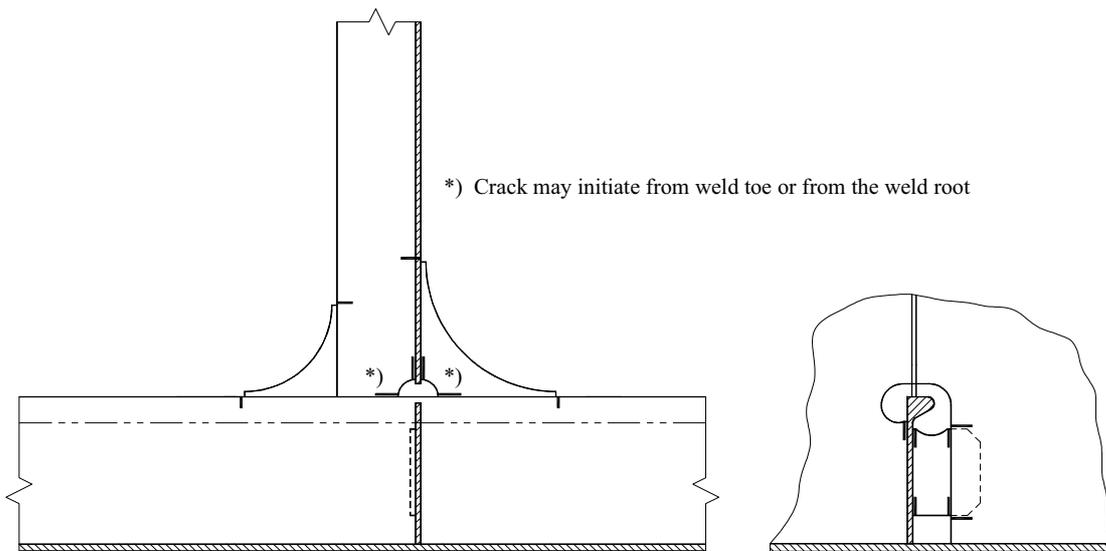


Figure E 7 – Potential fatigue crack locations at a connection between a side longitudinal and transverse web frame

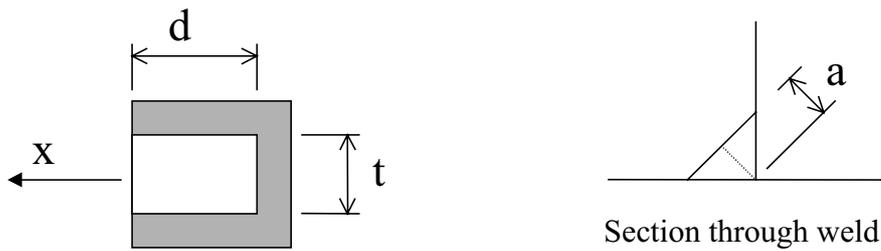


Figure E 8 – Section through weld at bracket scallop considered

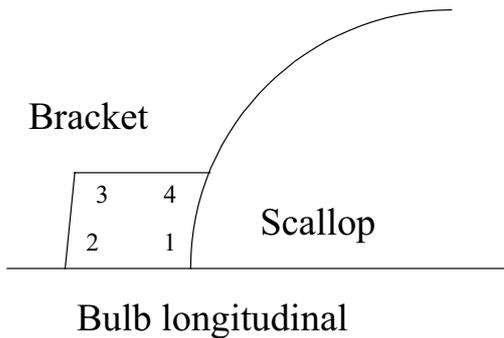


Figure E 9 – Element at scallop showing numbering of Gaussian points

4.18.3. Stress from 3 D FE analysis

If a 3 D 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. The mean stress over the throat thickness should be used for design purposes.

5. Fatigue Analysis Methodology and Procedures

5.1. General

The fatigue capacity is documented according to the principles given in Classification Notes 30.7 Fatigue Assessment of Ship Structures (denoted Classification Notes 30.7). The fatigue capacity is calculated assuming that the linear accumulated damage (Palmgrens – Miner rule). The following methods of fatigue calculations are referred to in DNV-OS-C102:

- Simplified fatigue analysis
- Component stochastic fatigue analysis
- Full stochastic fatigue analysis

The different methods are suitable for different areas and used at different stages in the design loop. This is further discussed in section 6.

5.2. Simplified fatigue analysis

The simplified fatigue analysis is based on the assumption that the long term distribution of stresses can be described by the maximum dynamic stress amplitude and a Weibull shape parameter. The method is described in Classification Notes 30.7 section 2. and 3. For offshore ships the loads and long term statistics (Weibull parameter and number of load cycles) should be determined by means of direct load calculations based on a given scatter diagram, see appendix C.

The simplified fatigue analysis may be used in the initial design for longitudinal structural members (plating and longitudinals) according to the procedures described in Classification Notes 30.7.

A simplified fatigue analysis may also be performed in order to assess the fatigue strength of transverse structural elements where the stress response in is governed by the external and internal pressure loads. Typical details are:

- web frame scallops
- web frame bracket terminations
- hopper knuckles

The stress response may be determined based on cargo hold models and local stress concentration models by applying external and internal pressures calculated at a probability level of 10^{-4} to the FE models. The internal pressure distribution should be applied according to Classification Notes 30.7 based on direct calculated accelerations, and the external pressure distribution should be modified due to the effect of intermittent wet and dry surfaces according to section 4.8. The combined geometric stress amplitude at a probability level of 10^{-4} will thus be directly obtained from the FE results by combining the stress response due to external and internal pressure loads according to Classification Notes 30.7.

5.3. Stress component based stochastic fatigue analysis

The idea of the stress component-based fatigue analysis is to make direct use of the load transfer functions from a hydrodynamic wave load analysis in the fatigue calculations. The procedure is described in Classification Notes 30.7 section 5.4 and 5.6. The main advantage of this procedure compared to simplified fatigue analysis is that the correlation between the different stress/load components is automatically included in the calculations. The stochastic fatigue evaluation may be performed by calculating the part fatigue damage for each cell in the scatter diagram based on the applied wave spectrum, wave spreading, ship heading and S-N data. A flow diagram of the procedure is shown in Figure E 1.

The load transfer functions from a hydrodynamic wave load analysis normally include:

- Global hull girder sectional forces and bending moments
- External pressures
- Vessel motions in 6 degrees of freedom

The load transfer functions are multiplied by a stress per unit load ratio (stress factors) to establish stress transfer functions, and the combined stress response is determined by a linear complex summation of the stress transfer functions. Since the procedure is based on a summation/superposition of stresses due to different load components, it is suitable for joints where the stress response from the different load effects have a common and well-known/defined principal stress direction.

The relevant stress factors to be considered for various structural details in the design of offshore ships are:

Stress due to global loads:

- axial loading
- vertical bending moment
- horizontal bending moment
- torsional bending moment

Stress due to external pressures:

- local bending of plate and stiffeners
- bending of the main girder system (secondary bending and relative deflections)

Stress due to vessel accelerations:

- internal pressures
 - topside loads
-

5.3.1. Determination of stress factors

Stress factors may be determined by global or cargo hold FE analysis and tabulated values of stress concentration factors or stress concentration models used as sub-models in global/cargo hold model. For details with a known stress concentration factor and a well defined load/stress response, stress factors may be determined based on simplified calculations according to CN30.7 chapter 3, replacing the described loads by unit loads.

Global Loads:

Stress factors due global wave loads may be determined by applying unit loads at one end of the FE model and a full fixation at the other end (cantilever beam). The topside framing system and PAU's should be represented in the model since global deflections of the hull girder will cause local bending and axial loads at the supports. Stress factors for topside support should in general be calculated based on stress concentration models using sub-modelling technique.

Axial loads should be applied at the cross-sectional area centre of gravity, hull girder bending moments at the neutral axis and torsional moments at the shear centre. All nodes of the cross-section should be coupled to the node where the load is applied by rigid body dependency. It should be noted that torsional moments are only relevant for vessels with large deck openings and may require a global FE model to determine the response correctly.

Stress factors due to axial loading, horizontal and vertical bending may as an alternative be determined by simplified calculations.

External pressure:

The stress response due to external sea pressure may be separated in stresses due to local bending of shell plating and stiffeners and stresses due to bending of the main girder system (secondary bending and relative deflections). This separation is required since the local bending results from the local loading of the plate/stiffener whereas the bending of the main girder system results from a "global" pressure distribution. In general stresses due to bending of the main girder system should be determined based on FE analyses, while stress factors due to local bending of plating stiffeners may be based on simplified calculations.

The external pressure distribution, calculated at a 10^{-4} probability level and modified due to the effect of intermittent wet and dry surfaces, should be applied to the FE model. The pressure distribution is normally assumed to be constant along the considered part of the hull. The longitudinal pressure distribution may however influence the results for all stresses where this variation is of importance. This may be areas like; hopper knuckle, transverse bulkheads, double bottom, etc. Vertical restraints may be applied at transverse bulkheads to avoid global bending of the hull girder.

Stress factors due to secondary bending and relative deflections in the bottom should be normalised based on the pressure at the centroid of the bottom pressure distribution and combined with the load transfer function for the corresponding panel of the hydrodynamic

model. Accordingly, stress factors due to secondary bending and relative deflections in the side should be normalised based on the pressure at the centroid of the side pressure and combined with the load transfer function for the corresponding panel of the hydrodynamic model. When the stress factors are determined by local FE models, it is not necessary to separate between secondary bending and relative deflections. The effect of local bending may be isolated by a separate FE analysis with additional restraints on the main girder system.

The stress factors for local bending of details subjected to the effect of intermittent wet and dry surfaces are to be multiplied by the r_p -factor, see section 4.8. The load transfer function for the panel at the water line thus applies for details located above the still water line. Since the location of the hot spot of interest does not exactly correspond to the hydrodynamic point, the stress factors should also be modified by linear interpolation between the hydrodynamic points and extrapolation to the still water line as shown in Figure E 10. The correction factor is obtained as:

$$C = \frac{P1}{P2} \quad (\text{E } 17)$$

where C = Correction factor
P1 = Interpolated pressure at considered detail
P2 = Pressure at nearest hydrodynamic point

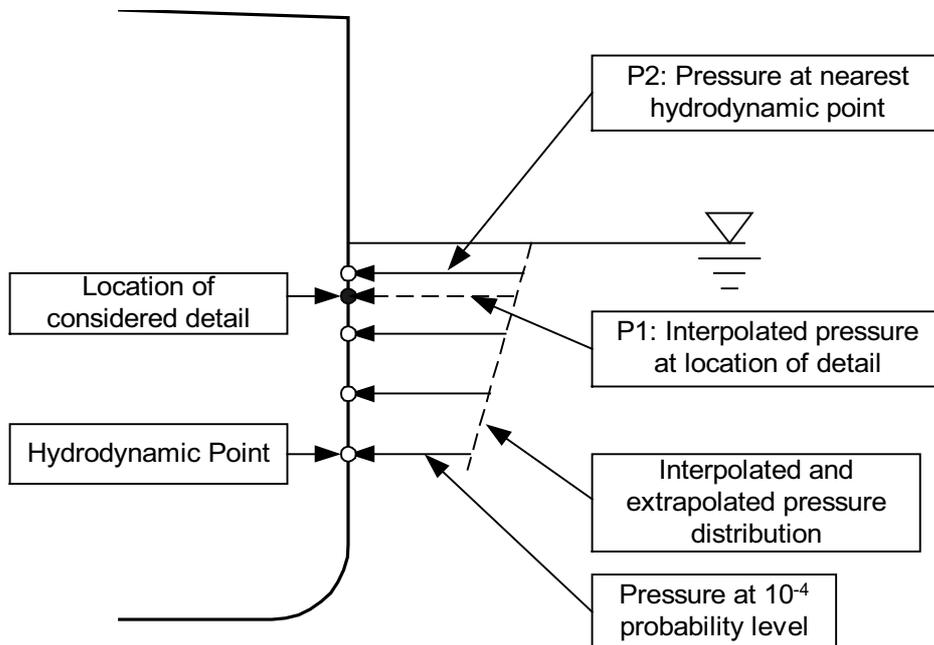


Figure E 10 – Correction factors for local bending due to external pressure

Vessel Motions:

Vessel motions will cause dynamic pressures in tanks and inertia loads from heavy equipment such as topsides. The g-component of roll and pitch should be included in the calculations.

Stresses due to internal pressures should be calculated independently for accelerations in longitudinal, transverse and vertical direction. Normally the pressures are based on the accelerations in the tank mass centre. The internal pressure distribution may be calculated in accordance with Classification Notes 30.7 chapter 4, and the stress factors may be evaluated as for external pressures.

Stresses due to inertia loads from topside modules are normally to be determined based on stress concentration models applying unit accelerations at the centre of gravity of the topside module.

5.3.2. Check list for the component stochastic method

As a verification tool a checklist for the different steps in the component stochastic analysis is given in Table E 4 below.

Table E 4 - Component stochastic fatigue analysis checklist

| Item | Action | Comments |
|------|---|---|
| 1 | Decide on which areas to perform component based fatigue analysis | See flow chart Figure E 13 |
| 2 | Perform hydrodynamic analysis | Typical loading conditions to be used. Normally full load and ballast. |
| 3 | Apply unit loads on structural model | |
| 4 | Calculate ratio between unit load and nominal stress | See if sub-models are used. |
| 5 | Extract load transfer functions from hydrodynamic analysis | Both real and imaginary parts to be included |
| 6 | Transform load transfer functions from hydrodynamic program into stress transfer functions. | Ratios calculated in 4) to be used in the transformation |
| 7 | Reduce local pressure loading due to effect of intermittent wet and dry surfaces if appropriate | Global effects of this should be taken care of in 4). Local effects (stiffener bending) should, however, be treated separately. |
| 8 | Apply stress concentration factors :> notch stress transfer functions | The following factors to be included if appropriate: $K_g, K_w, K_{tc}, K_{t\alpha}, K_n$ |
| 9 | Combine notch stress transfer functions | In order to include the phasing between the different stress transfer functions, both real and imaginary parts of the transfer functions must be included. |
| 10 | Reduce wave exposure time according to "time in sea" | |
| 11 | Select SN-curve to be used in the fatigue calculations | If the detail is in corrosive environment, one analysis may be necessary for the protected time interval and for non-protected interval |
| 12 | Include thickness effect if plating thickness is larger 25mm ($t > 25\text{mm}$) | Thickness of the plating where the crack starts. Not included if stress concentration is below 1.3. |
| 13 | Apply correct wave scatter diagram and wave spreading | |
| 14 | Perform fatigue analysis | To be performed by appropriate method. For instance: <ul style="list-style-type: none"> • miner summation of damage for each sea state • damage calculations according to long term stress distribution (Weibull) and allowable stress range |

5.4. Full stochastic fatigue analysis

The full stochastic analysis is an analysis where all load effects from global and local loads are included by use of stress concentration models and direct transfer of loads from the hydrodynamic analysis to the structural model in equilibrium. Hence, all stress components are combined using the correct phasing and without simplifications or omissions of any stress component. This method will thus usually be the most exact for determination of fatigue damage.

For fatigue analysis of structural details where the stress level is significantly affected by lateral sea pressure on the ship side, the linear loads transferred from the hydrodynamic analysis should be adjusted according to section 4.8. In order not to disturb the equilibrium of forces, the correction should be performed only in the region of interest and on both sides. Typical areas where correction of the side pressure due to intermittent wet and dry surfaces should be considered are:

- Side longitudinals
- Side plating
- Hopper knuckle
- Transverse bulkheads/frames

A full stochastic fatigue analysis may be performed based on a global FE model of the entire vessel or a part-ship model as described in Appendix F. Flowcharts for full-ship stochastic analysis and part-ship stochastic analysis are shown in Figure E 11 and Figure E 12 respectively.

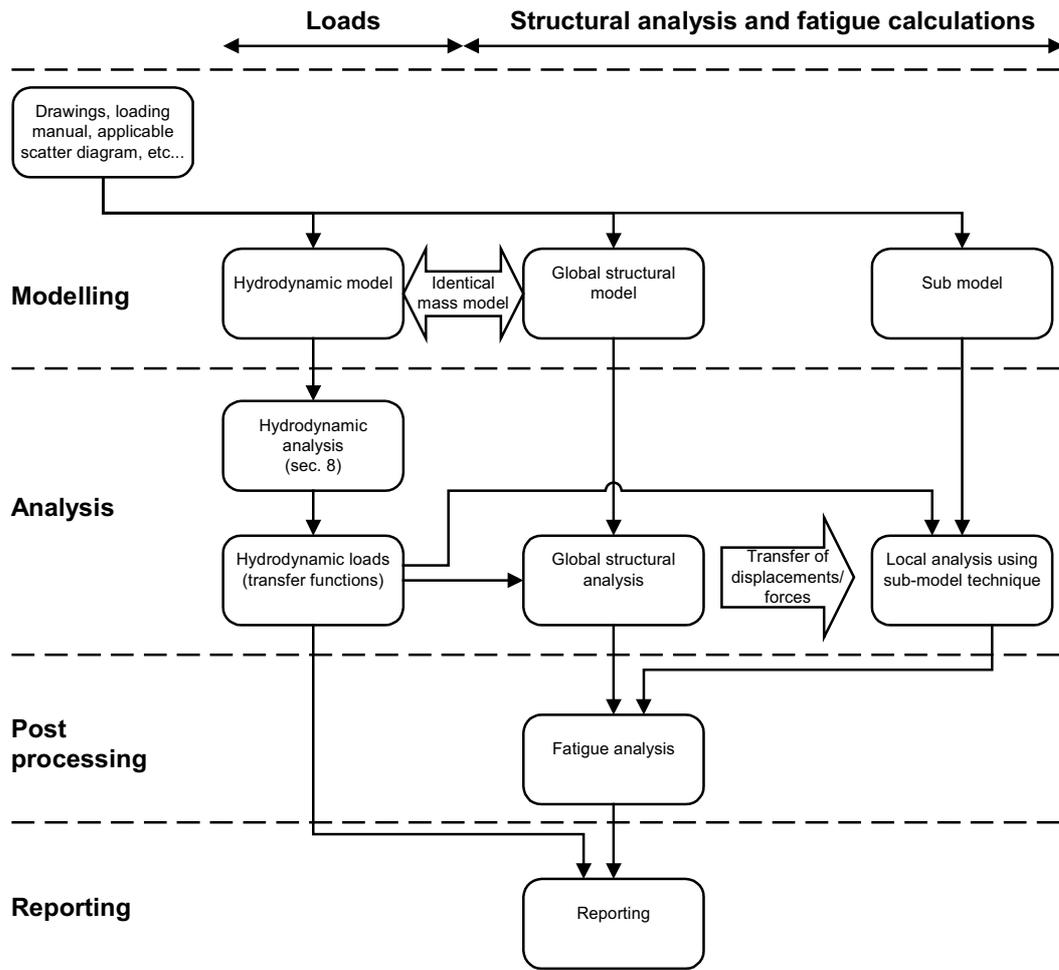


Figure E 11 - Full-ship direct stochastic analysis procedure flowchart

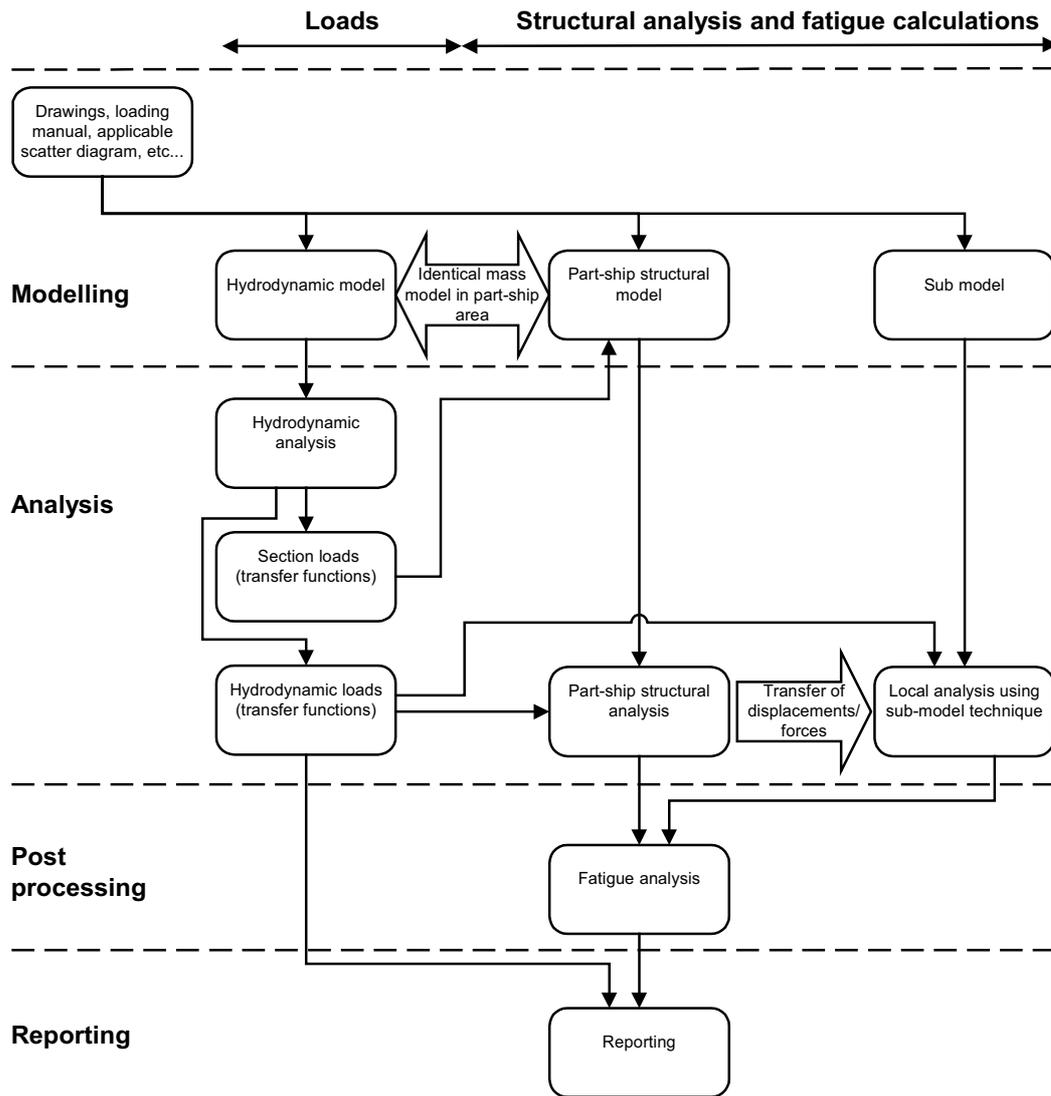


Figure E 12 - Part-ship direct stochastic analysis procedure flowchart

5.4.1. Check list for the full stochastic method

As a verification tool a checklist for the different steps in the full stochastic analysis is given in Table E 5 below.

Table E 5 - Hull stochastic fatigue analysis check list

| Item | Action | Comments |
|------|--|---|
| 1 | Decide on which areas to perform full stochastic analysis | See flow chart in Figure E 13 |
| 2 | Establish stress concentration factor | |
| 3 | Perform hydrodynamic analysis <u>The following items to be performed for all relevant loading conditions.</u> | Typical loading conditions to be used. Normally full load, intermediate and ballast. Sufficient wave periods and headings to describe the transfer function properly to be used. Normally 20-25 periods · 3-36 headings (depending on mooring system) |
| 4 | Transfer hydrodynamic loads to the global structural model | The following loads to be applied: internal and external pressures, inertia and viscous damping (if applicable), spring forces from mooring (if present). |
| 5 | Perform structural analysis of global structural model | |
| 6 | Transfer displacements/forces from the global model to the edges of the local structural model | Both real and imaginary parts to be included |
| 7 | Transfer hydrodynamic loads to the local structural model | Loads as for global structural model. May be omitted stresses due to local loads are insignificant. |
| 8 | Perform structural analysis of local structural model. | |
| 9 | Extract stresses and perform stress extrapolation | Stress extrapolation to be performed for each stress component |
| 10 | Apply stress concentration factors :> notch stress transfer functions | The following factors to be included if appropriate: K_w , K_{te} , K_{ta} (to be applied on principal stresses, or all stress components) |
| 11 | Reduce wave exposure time according to "time at sea", if applicable | |
| 12 | Select SN-curve to be used in the fatigue calculations | If the detail is in corrosive environment, one analysis may be necessary for the protected time interval and for non-protected interval |
| 13 | Include thickness effect if plating thickness is above 25mm ($t > 25\text{mm}$) | Thickness of the plating where the crack starts. Not to be included if stress concentration is below 1.3. |
| 14 | Apply correct wave scatter diagram and wave spreading (Appendix D) | |
| 15 | Perform fatigue analysis | To be performed by appropriate method. For instance: - Miner summation of damage for each sea state - damage calculations according to long term stress distribution (Weibull) and allowable |

| Item | Action | Comments |
|-------------|---|-----------------|
| | | stress range |
| 16 | Add results from different loading conditions (and different corrosion protection periods if relevant). | |

6. Selection of fatigue approach

Several fatigue approaches exist, they all have advantages and disadvantages. The different approaches are therefore suitable for different areas. This is described below. Load effects, accuracy of the analysis, computer demands, etc. should be evaluated before one of the approaches is chosen.

Full stochastic analysis

Advantages:

- All linear effects automatically included. Both for global and local loading
- Phasing between responses automatically included
- Can be used for all geometry's, even if geometric stress concentration factors, K_g , are not available
- Shear lag effects included

Disadvantages:

- A large number of load cases (periods x headings) have to be analysed using the global structural model. This may demand large CPU and storage capacity
- Difficult to include non-linearity's for one load component as all load components are mixed into one stress response
- Partly a black box procedure (program dependent). This makes it difficult to check intermediate results.

Suitable areas:

Areas where the stress concentration factors are unknown. Typical areas are:

- Discontinuous panel knuckles (hopper knuckles)
- Bracket and flange terminations of main girder systems
- Stiffeners subjected to large relative deformations

Component based stochastic analysis

Advantages:

- Possible to use separate load factors on each load component and thus include effects as reduced (modified) dynamic pressures around still water line and non-linear variation of tank pressures
- Only a few load cases have to be analysed using the global structural model.
- Effect from different loads on the results can be found.
- Possible to perform analysis using simplified formulas for the area of interest.

Disadvantages:

- Errors are easily made in the combination of stresses
 - Manual definition of extra load cases. This may cause errors.
 - Simplifications are usually made in load calculation:
 1. Constant pressure loading over the length of the 3-tank model. This means that the relative deformation of the transverse frames will be overestimated. Rotations of the longitudinals may be slightly underestimated for the wave periods contributing most
-

significantly to the fatigue damage. In general the procedure is considered to be to the safe side.

2. The same load/profile is used for each wave heading/period. However, the load profile is based on 10^{-4} probability level (referred to a Weibull long term distribution) which is not far from the load level that contributes most significantly to fatigue damage. Normalisation of the pressure should be performed close to the detail to be analysed. This will ensure that a good representation of the local load with respect to heading and period.

Suitable areas:

- All areas where geometric stress concentration factors, K_g , are available:
 - Longitudinals
 - Plating
 - Cut-outs
 - "Standard" hopper knuckles
- Areas where side pressure is of importance
- Can also be used for areas where the stress concentration factors are unknown

Figure E 13 shows a schematic overview of the descriptions given above.

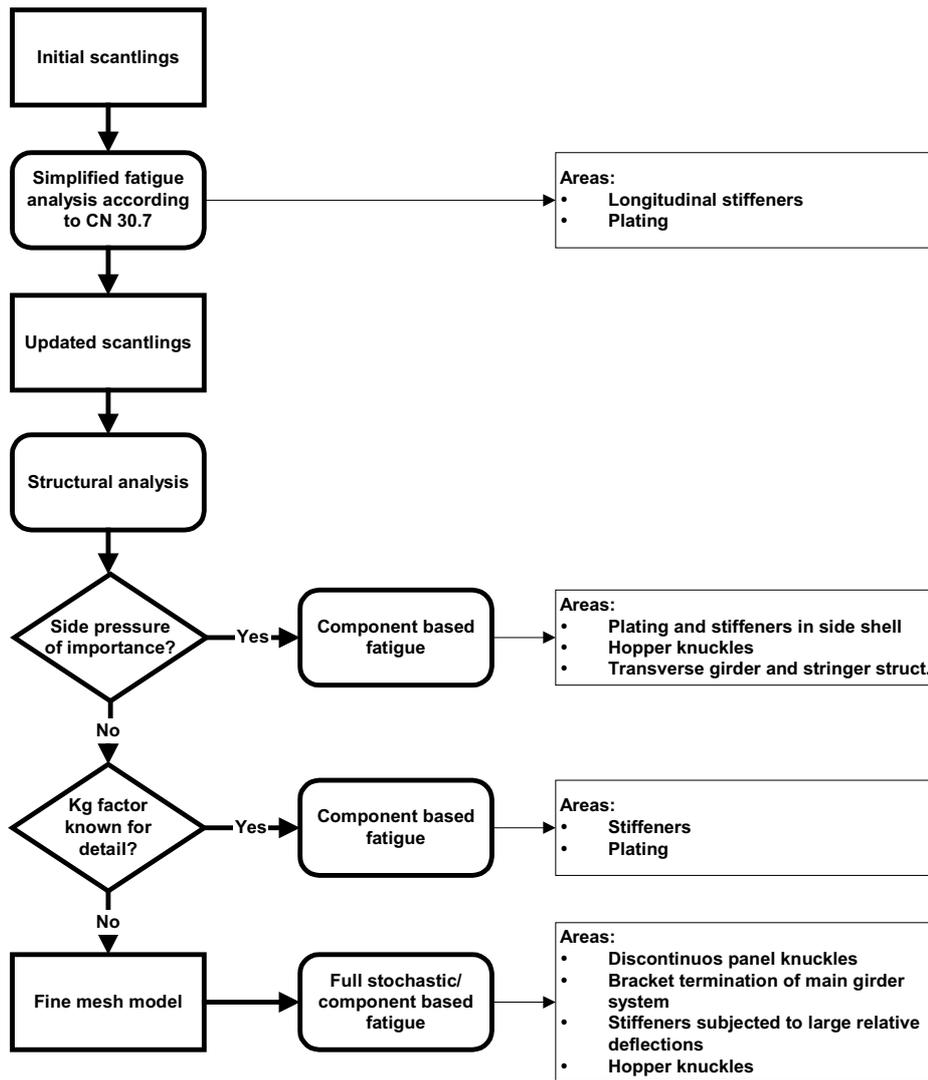


Figure E 13 - Flow diagram showing different fatigue approaches

Appendix F

ANALYTICAL MODELS

CONTENT

| | |
|---|----------|
| 1. INTRODUCTION..... | 3 |
| 2. SYMBOLS AND DEFINITIONS | 4 |
| 3. CARGO TANK ANALYSIS FOR STRENGTH ASSESSMENT | 5 |
| 3.1. GENERAL | 5 |
| 4. FATIGUE ANALYSES | 6 |
| 4.1. SCANTLINGS | 6 |
| 4.2. OVERVIEW OF STRUCTURAL FE MODELS | 6 |
| 4.3. STRUCTURAL MODELLING PRINCIPLES THE WHOLE SHIP | 7 |
| 4.3.1. <i>Model idealisation</i> | 7 |
| 4.3.2. <i>Extent of model</i> | 8 |
| 4.3.3. <i>Modelling of girders</i> | 9 |
| 4.3.4. <i>Modelling of stiffeners</i> | 9 |
| 4.3.5. <i>Elements and mesh size</i> | 9 |
| 4.3.6. <i>Mass modelling</i> | 10 |
| 4.3.7. <i>Modelling of cargo and ballast water</i> | 11 |
| 4.3.8. <i>Boundary conditions</i> | 11 |
| 4.3.9. <i>Application of hydrodynamic loads</i> | 12 |
| 4.4. PRINCIPLES FOR STRUCTURAL MODELLING OF PART-SHIP..... | 12 |
| 4.4.1. <i>Model idealisation</i> | 12 |
| 4.4.2. <i>Extent of Model</i> | 13 |
| 4.4.3. <i>Modelling of Girders, stiffeners</i> | 13 |
| 4.4.4. <i>Elements and Mesh Size</i> | 13 |
| 4.4.5. <i>Mass modelling</i> | 13 |
| 4.4.6. <i>Modelling of cargo and ballast water</i> | 14 |
| 4.4.7. <i>Boundary conditions</i> | 14 |
| 4.4.8. <i>Application of hydrodynamic loads</i> | 15 |
| 4.5. STRUCTURAL MODELLING PRINCIPLES FOR CARGO HOLD (3 TANK)..... | 16 |
| 4.5.1. <i>Model idealisation</i> | 16 |
| 4.5.2. <i>Extent of model</i> | 16 |
| 4.5.3. <i>Component loads</i> | 17 |
| 4.5.4. <i>Finite element mesh</i> | 17 |
| 4.5.5. <i>Boundary conditions</i> | 17 |
| 4.5.6. <i>Balance of loads</i> | 17 |
| 4.6. STRUCTURAL MODELLING PRINCIPLES FOR LOCAL DETAILS..... | 18 |
| 4.6.1. <i>Application of Mooring and Riser Forces</i> | 18 |

1. Introduction

This Appendix describe typical finite element models used in the structural and fatigue capacity checks. A finite element analysis may comprise the following tasks:

- Description of Geometry, mesh, element properties, boundary conditions
- Loads
- Response calculations
- Post-processing of results

Loads are addressed in Appendix B and C. Application of the results are described in Appendix D regarding structural capacity and Appendix E regarding fatigue capacity.

2. Symbols and definitions

| | | |
|----------|---|------------------|
| t_w | : | Web thickness |
| l_{co} | : | Length of cutout |
| h_{co} | : | Height of cutout |
| h | : | Web height |

3. Cargo Tank Analysis for strength assessment

3.1. General

The structural analysis of a typical cargo tank is carried out according to Classification Notes 31.3 “Strength Analysis of Hull Structures in Tankers”. In addition to the load conditions given in the Classification Notes the load conditions required to determine the transverse stresses for the hull girder ultimate capacity check must be included, ref. Appendices B and D.

4. Fatigue analyses

4.1. Scantlings

The unit is assumed to be provided with an effective corrosion protection system. All finite element models may therefore be based on gross scantlings (i.e. nominal plate and bracket thickness).

4.2. Overview of structural FE models

Use of a finite element model in the fatigue analysis aims to obtain a more accurate assessment of the stress response in the hull structure. The analysis can be by several types or levels of finite element models. The following three different levels of finite element models are referred to in this RP.

- 1) *Global structural model*: A relatively coarse mesh model used to represent the overall stiffness and global stress distribution of the primary members of the hull. A typical model is shown in Figure F 1
- 2) *Cargo hold model (3-tank model)*: A model used to analyse the deformation response and nominal stresses of the primary members of the midship area. The model will normally cover $\frac{1}{2} + 1 + \frac{1}{2}$ cargo hold length in the midship region (exception for turrets and similar structural configurations $1 + 1 + 1$). Typical models are shown in Figure F 2
- 3) *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 F 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..

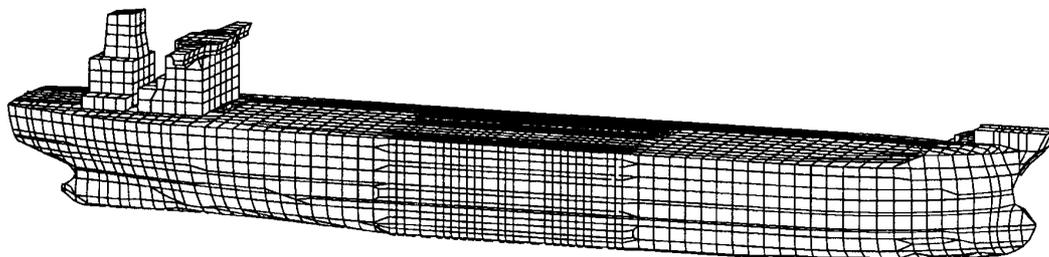


Figure F 1 Global hull model of shuttle tanker

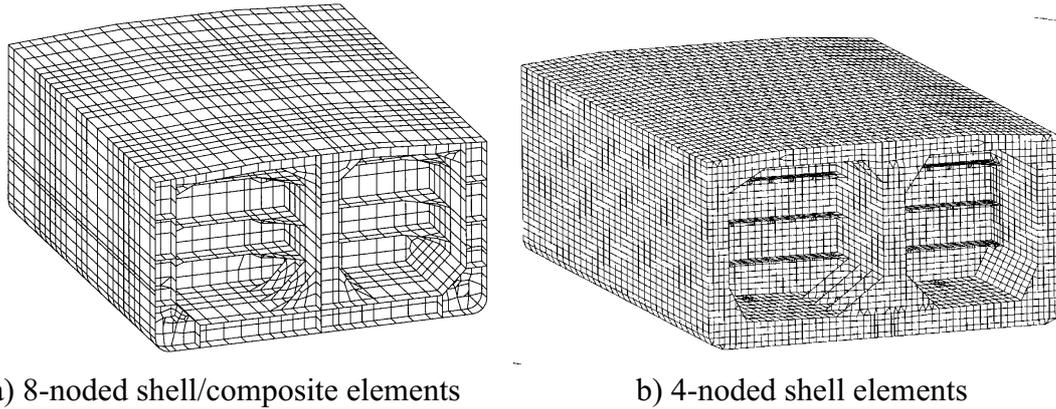


Figure F 2 Cargo hold models (midship area)

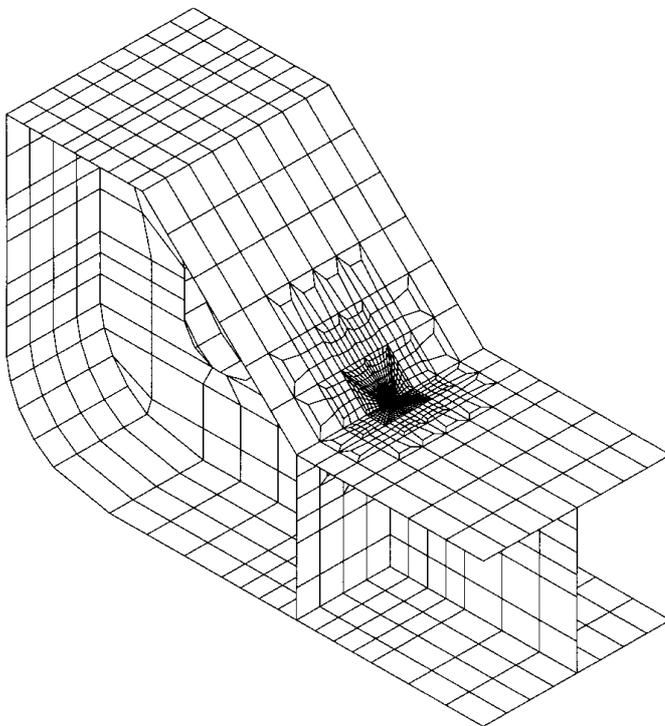


Figure F 3 SCF model of hopper tank knuckle

4.3. Structural Modelling Principles the whole ship

4.3.1. Model idealisation

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.

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 give a good representation of the overall membrane panel stiffness in the longitudinal/transverse and shear directions.

An example global finite element model of an oil tanker is shown in Figure F 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, 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 must 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 must 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.

4.3.2. Extent of model

The full structure of the vessel 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 vessel may be disregarded. The mass of disregarded elements shall be included in the model.

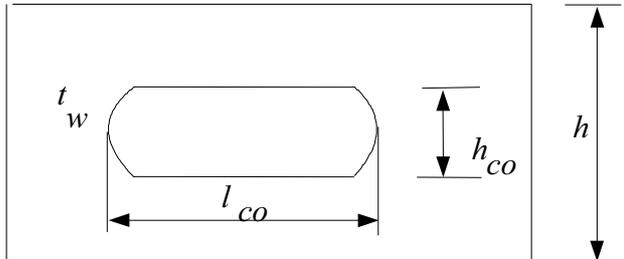
Structural components not contributing to the global stiffness, such as superstructure, topsides, topsides support, etc., are not normally included in the global analysis. However, the mass of these elements should be correctly included in the model. It should be emphasised that these structures can lead to local/global stress concentrations and it should be checked that omission of these parts does not lead to non-conservative results.

The omission of minor structures may be acceptable provided that such omission does not either significantly change the deformation of the structure or give favourable results, i.e. too low stress, to the structural analysis.

4.3.3. 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.

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



$$t_{mean} = \frac{h - h_{co}}{hr_{co}} t_w$$

where

t_w = web thickness

$$r_{co} = 1 + \frac{l_{co}^2}{2.6(h - h_{co})^2}$$

l_{co} = length of cut - out

h_{co} = height of cut - out

h = girder web height

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

4.3.4. 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).

4.3.5. 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 requires 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.

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 F 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.

4.3.6. 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 in structural intersections such that local response is minimised.

The mass from topside structures should be included in the model.

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.

4.3.7. Modelling of cargo and ballast water

Modelling of the cargo/ballast water should be included in global analysis models used for detailed calculations of side shell details or details affected by liquid pressures. The phase difference between internal and external pressure will be accounted for automatically.

4.3.8. 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 F 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 in the centre line close to the aft and the forward ends of the vessel.

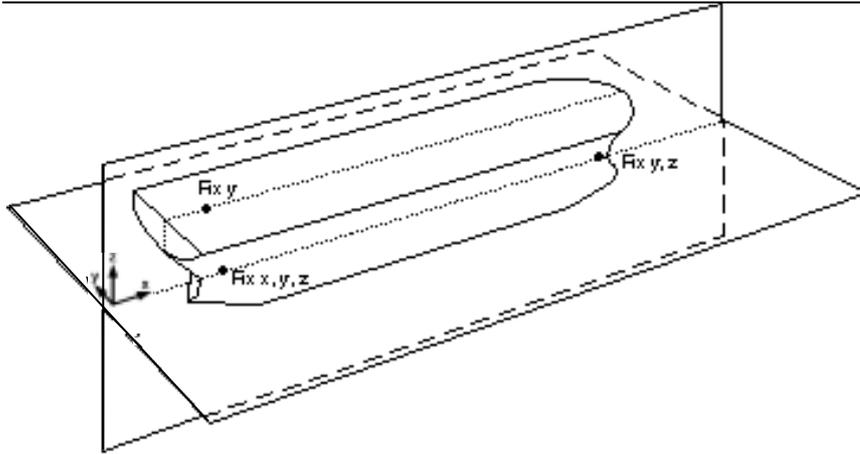


Figure F 4 Example of boundary conditions

4.3.9. Application of hydrodynamic loads

Hydrodynamic loads for use in the structural analysis shall be directly transferred from the hydrodynamic load analysis. Direct transfer will ensure that all loads maintain the correct phase relationship for further post processing. 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 must be connected to the structural model).

Viscous damping forces can be important for some vessels, particularly those vessels where roll resonance is in an area with substantial wave energy, i.e. roll resonance periods of 6-15 seconds.

4.4. Principles for Structural Modelling of Part-ship

4.4.1. Model idealisation

The purpose of the partial hull analysis is to perform a stochastic fatigue analysis based on direct load transfer on a selected part of the vessel without having to create a finite element model or analyse the entire vessel. Apart from the size of the finite element model, the part-ship approach is similar to the full-ship approach with exception of the load application.

External items such as cranes, derrick, topside support etc. should be included in the model at their respective positions. The main objective is to transfer the inertia loads from the appendages into the hull and can therefore be represented with the same mesh detail as the rest of the hull.

The units used in the finite element model should be consistent with what is used for the hydrodynamic analysis.

4.4.2. Extent of Model

The procedure described assumes that the partial model has a bulkhead at the aft and forward end. Typically the model will consist of 3 complete compartments, i.e. 1+1+1, where the primary fatigue calculations will be completed for the middle compartment. The other compartments are to be included to limit the influence of the boundary conditions on the results.

Variations of this may occur towards the ends of the vessel where a additional compartment is not available, e.g. external bow mounted turret. In this instance the model is to extend completely to the end of the vessel and include one compartment on the opposite side of the location being considered, as shown in Figure F 5.

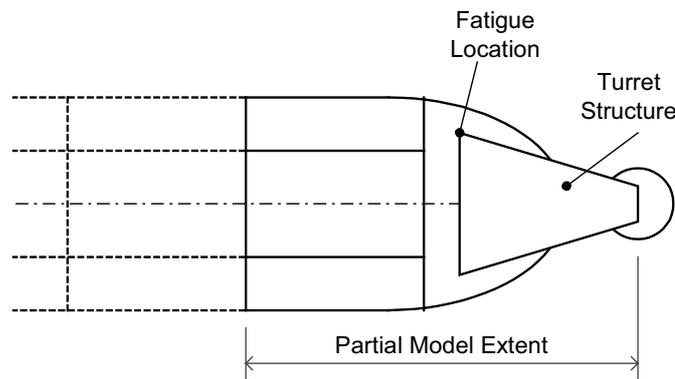


Figure F 5 Plan View of Partial Model Extent

4.4.3. Modelling of Girders, stiffeners

Refer to Sections 4.3.3 and 4.3.4.

4.4.4. Elements and Mesh Size

Refer to Section 4.3.5.

4.4.5. Mass modelling

The mass modelling should be according to the loading manual, i.e. have the same longitudinal, vertical and transverse mass distribution. This is important both for the hydrostatic/dynamic analysis and for the structural analysis. The hydrodynamic analysis needs a correct mass description in order to produce correct motions and sectional forces, while global/local stress patterns are affected by the mass description in the structural analysis.

Identical mass models should be used in hydrodynamic analysis and structural analysis for the considered part of the vessel. 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 for external and internal hydrostatic and hydrodynamic loads and point mass for non-structural members. The point mass representation should be sufficiently distributed to provide a correct representation of rotational mass and to avoid unintended results. Point masses should be located in structural intersections such that local response is minimised.

The mass from topside structures and other appendages should be included in the model.

If the program system supports the use of non-structural members, this may be a suitable method for modelling the cargo, which may otherwise be 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.

To balance the model such that correct mass description is obtained may not be a straightforward task. The partial structural model usually consists of one, or if super element technique is available, several super elements. The size of each super element may be relatively large and correct centres of gravity within each super element need not necessarily mean that the distribution within each super element is correct. Even small inaccuracies in the mass description may lead to relatively large errors in global forces/moments.

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

4.4.6. Modelling of cargo and ballast water

If the global analysis shall be used for detailed calculations of side shell details 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.

4.4.7. Boundary conditions

The boundary conditions for the partial structural model will be springs at both ends of the model. The spring stiffness should be small (1/1000 of actual spring stiffness) and the springs are only used to avoid singularities from small unbalances in the applied loads. Springs, see Figure F 6, should be applied in all three translation degrees of freedom and the ratio between them should be such that no unwanted moments are introduced.

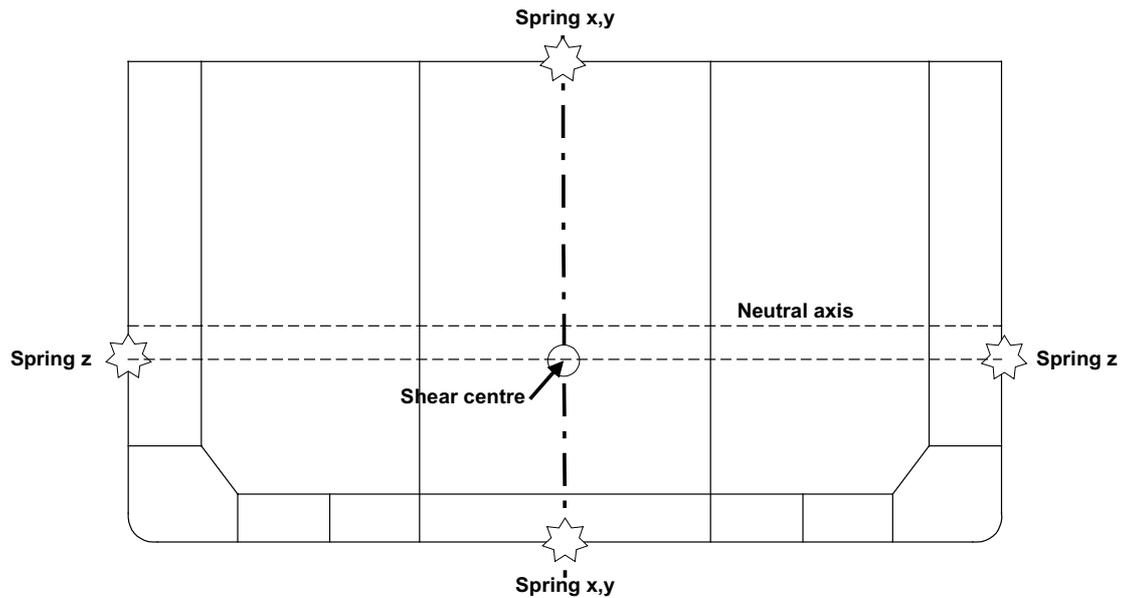


Figure F 6 Position of Applied Springs on a Part-Ship Model

4.4.8. Application of hydrodynamic loads

The hydrodynamic loads are to be taken as calculated in 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 etc. *

Inertia loads from equipment etc.

Viscous damping forces (see below).

* The internal pressure loads may be exchanged with mass of the liquid (with correct centre of gravity) if stresses in areas of interest are not changed significantly (The mass must be connected to the structural model).

As a substantial part of the vessel is not included in the partial model concept, sectional loads must 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 partial model and one at the forward end.

The section forces and moments at these sections must be transferred to the partial model in order to account for the missing parts. In order to ensure that the model is in 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.

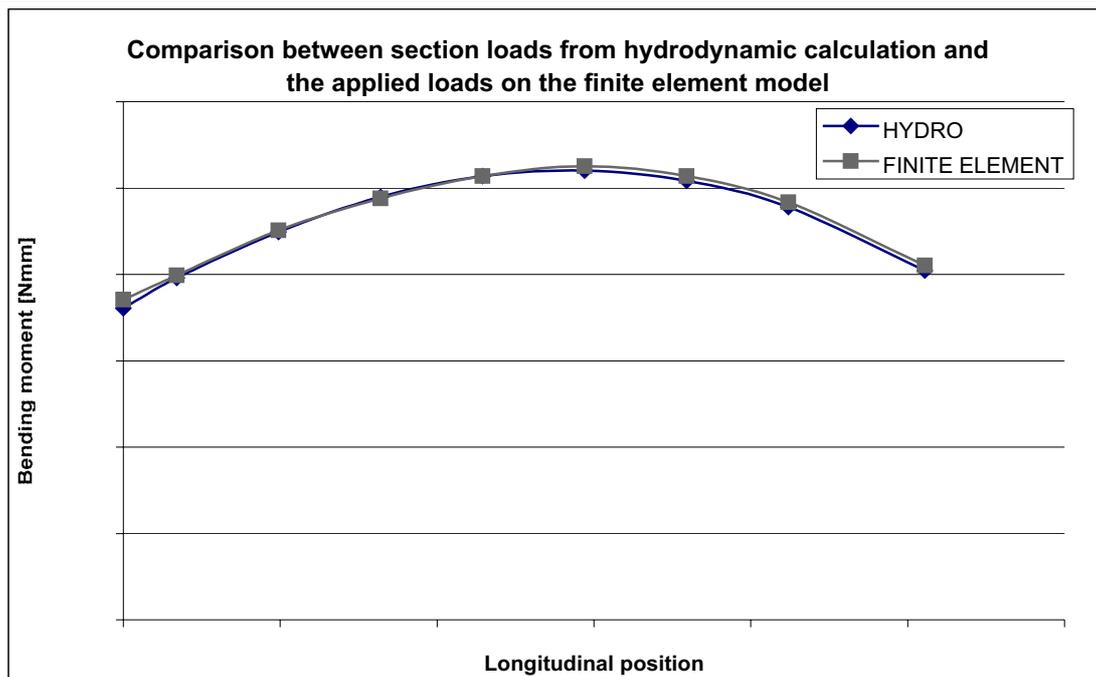


Figure F 7 Comparison of Sectional Loads between Hydrodynamic and FE Calculations

As seen from Figure F 7, 10 cuts are usually sufficient to get a good description of the bending moment and shear force distribution along the hull. The first and last must correspond with the ends of the finite element model.

Viscous damping forces may be important for some vessels. This is typically the case for vessels where the roll resonance is in an area with substantial wave energy, i.e. roll resonance periods of 6-15 seconds.

4.5. Structural Modelling Principles for Cargo Hold (3 Tank)

4.5.1. Model idealisation

The cargo hold/tank 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 cargo hold/tank analysis is a requirement for DNV ✕1A1 main class.

4.5.2. 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 $\frac{1}{2} + 1 + \frac{1}{2}$ holds or tanks. A model covering the half breadth of the ship may be used provided that there is symmetry in 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 vessel types.

The model for analysis of the moonpool area should normally include an adjacent tank on each side, i.e. 1 + 1 + 1 tank.

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

4.5.3. Component loads

Lateral loads from sea pressure, cargo etc. shall be applied to the model. The applied loads shall be normalised 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 vessel 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 combined the hull girder response with the response from the lateral load distribution.

4.5.4. 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 F 2 illustrates these two alternatives for possible mesh subdivisions in a double skin UNIT.

For the tanker model shown in Figure F 2a, 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 e. g. 4. The modelling of the frames should also be seen in connection with extent of the local model, see section 4.6.4.

The mean girder web thickness in way of cut-outs may generally be taken as in section 4.4.3.

4.5.5. Boundary conditions

In order to address the lateral load response, 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,

the intersection of the transverse bulkheads with bottom/inner bottom,

the intersection of the transverse bulkheads with deck, and

the intersection of the transverse bulkheads with 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.

4.5.6. 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. Ref. Also Appendix E.

4.6. Structural Modelling Principles for Local Details

4.6.1. Application of Mooring and Riser Forces

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, bending stiffeners, etc. Local structural models of these regions are required in order to determine the hot-spot stresses. The load effects should be combined with other hull girder loads. 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 both for tensions and for bending moments. This influence coefficient is frequency independent and therefore differs from the transfer functions discussed elsewhere in this 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 equation (F 1).

$$s_c = \sqrt{s_a^2 + 2 \cdot s_a \cdot s_b + s_b^2} \quad (\text{F 1})$$

The standard deviations of the two contributing stresses are s_a and s_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 and hence need not be combined conservatively. The procedure for determining the combined fatigue damage effect of the two frequency ranges shall be applied as described in Appendix F.

APPENDIX G

MOORING AND RISER SYSTEMS

Content

| | |
|--|-----------|
| 1. INTRODUCTION..... | 3 |
| 2. TYPES OF MOORING AND RISER SYSTEM..... | 4 |
| 2.1. SPREAD MOORING SYSTEM..... | 4 |
| 2.2. TURRET SYSTEMS..... | 4 |
| 2.2.1. <i>Overview of Turret Designs</i> | 5 |
| 2.2.2. <i>Impact of Turret Location</i> | 7 |
| 2.2.3. <i>Turrets located near the bow or stern</i> | 8 |
| 2.2.4. <i>Turrets located near amidships</i> | 8 |
| 2.3. RISERS SYSTEM..... | 9 |
| 3. HULL /SPREAD MOORING INTERFACE DESIGN | 11 |
| 3.1. HULL-MOORING SYSTEM INTERACTION..... | 11 |
| 3.2. DESIGN PROCEDURE..... | 11 |
| 3.3. LOADS | 11 |
| 3.4. LOCAL STRUCTURAL RESPONSE ANALYSIS..... | 11 |
| 4. TURRET AND HULL/TURRET INTERFACE DESIGN..... | 13 |
| 4.1. GENERAL | 13 |
| 4.2. MANAGING THE HULL / TURRET INTERFACE ANALYSIS | 13 |
| 4.3. DESIGN PROCEDURE..... | 14 |
| 4.4. ULS LOADS | 16 |
| 4.4.1. <i>ULS load conditions</i> | 17 |
| 4.5. FATIGUE LOADS | 18 |
| 4.5.1. <i>Mean Response</i> | 18 |
| 4.5.2. <i>Wave-Frequency Response</i> | 18 |
| 4.5.3. <i>Low-Frequency Response</i> | 18 |
| 4.6. HULL / TURRET INTERFACE CALCULATION METHODS | 18 |
| 4.6.1. <i>Coupled stochastic analysis based on riser/mooring RAO's</i> | 19 |
| 4.6.2. <i>Coupled analysis based on linear spring representation of each riser and mooring line</i> .. | 20 |
| 4.6.3. <i>Simplified fatigue using maximum mooring / riser dynamic range</i> | 21 |
| 4.7. FINITE ELEMENT MODEL OF INTERNAL TURRETS..... | 22 |

1. INTRODUCTION

This section of the RP is applicable to units that are designed to be stationary on a location by means of mooring lines. For FPSOs and FSUs this is the common practice today.

The mooring system principles used is either a Turret moored system or a spread moored system.

The aim of this appendix is to provide guidelines for the structural design of these two types of mooring system including the interface with the hull structure.

2. TYPES OF MOORING AND RISER SYSTEM

2.1. Spread mooring system

Spread mooring systems are typically used in benign waters. The vessel is stationary with limited or none possibility for weather vaning. There are however, new designs of spread mooring systems that allow significant vessel rotation. These systems could prove to extend the use of the spread mooring systems into more harsh environments. Spread mooring systems comprises a number of (8-12) mooring lines fixed to the main deck by means of a chain stopper. The mooring lines are evenly spread in order to keep the ship in the same position for all environmental conditions. The system is simple and relatively inexpensive.

A example of a spread moored design is shown Figure G 1.

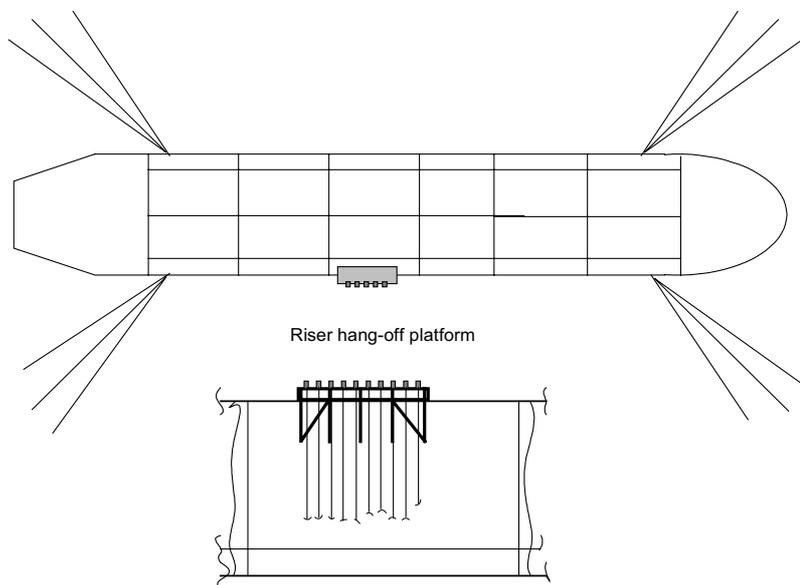


Figure G 1 Spread mooring system and riser hang-off platform

The unit is typically moored such that the dominating wind and/or current direction is towards the bow.

2.2. Turret Systems

The turret serves three main functions:

- To maintain the vessel at the desired location.
- To allow the vessel to rotate about the vertical turret axis.
- To provide support for the risers allowing fluid transfer to and from the fixed piping system onboard.

•

The turret is moored to the seabed by means of a number of mooring lines and is connected to the vessel by means of bearings 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 vessel and mooring system.

In order to achieve the required directionality, the vessel can rotate 360° about the turret. Typically this will keep the vessel's orientation correct relative to the incoming waves. The vessel may freely rotate about the turret or it may rely on assistance from thrusters, often depending on the operating environment. The turret location in the vessel, bearing design, loading condition, environmental conditions and the hull and topside design will govern the weathervaneing characteristics. However completely passive systems may utilise thrusters to minimise fishtailing effects and control heading during offloading to 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 vessel.

2.2.1. Overview of Turret Designs

Vessel turret design arrangements may be divided into two primary categories: 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 of these are further described in the following sections.

External Turrets

External turrets are attached to the hull at the bow or stern and are often used for conversions in either benign waters or tropic storm locations where disconnection is necessary. The interface with the hull is generally governed by the local forces from the mooring and riser system. An external turret configuration is presented in Figure G 2.

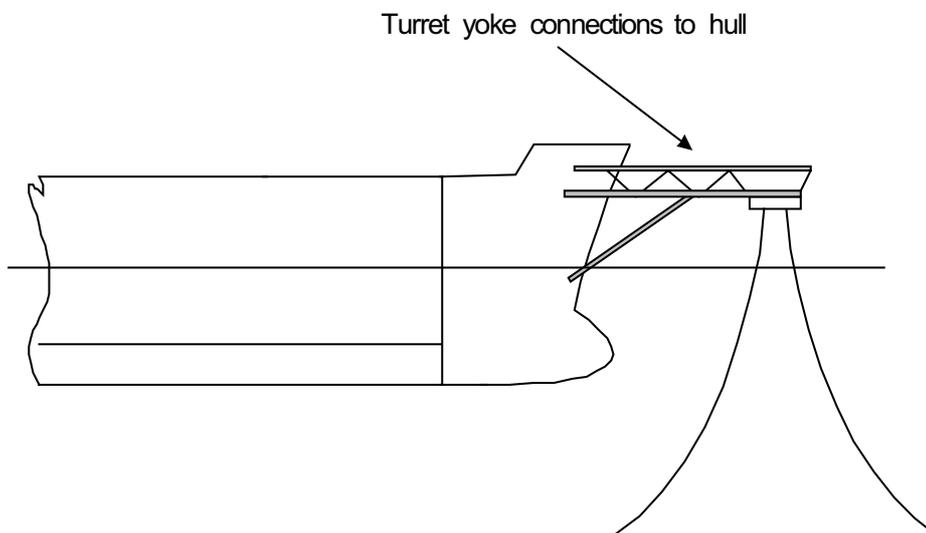


Figure G 2 External turret yoke arrangement

Internal Turrets

Internal turrets are generally categorised into permanent or disconnectable types. Internal permanent turrets are designed to be an integrated part of the hull and do not allow the release of the vessel from the mooring and riser system.

The connection design for an internal permanent turret to the hull may significantly differ. Some designs use upper supports to take both the vertical and horizontal forces from the mooring and riser system. In addition a lower bearing supporting structure may also be provided. In this situation the upper support structure bearings resist 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. A typical turret design with both upper and lower support are shown in Figure G 3.

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.

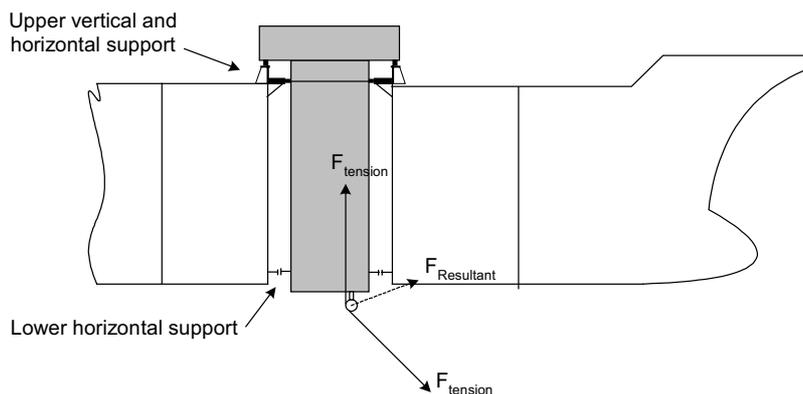


Figure G 3 Internal permanent turret with upper and lower support

The supporting structure for the bearings may also vary between designs. Some designs utilise a number of radial brackets above main deck to transfer the vertical forces directly to the moonpool cylinder, transverse web frames and vertical shear panels. Other turret designs achieve a uniform load distribution from the turret through the bearings to the interface structure via a torsion box located at the upper support as shown in principle in Figure G 4. The uniform stiffness for the bearings can be less sensitivity to moonpool ovalization.

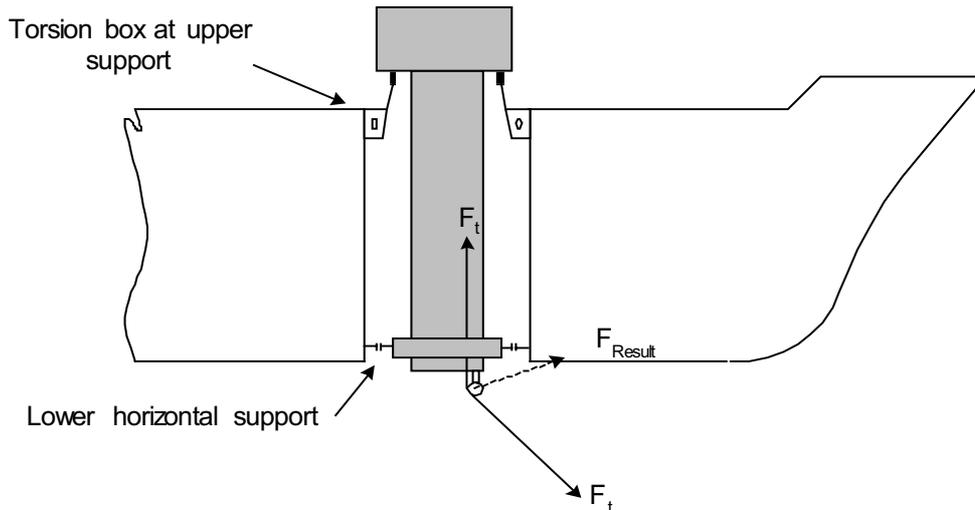


Figure G 4 Internal permanent turret with torsion box

Internal disconnectable turrets are connected to the hull such that the whole 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 an equilibrium in forces is achieved; allowing the vessel to sail away. The system is commonly used where quick release is needed in case of an emergency event, such as possible collision with icebergs or extreme weather conditions. The concept is also used for storage vessels and/or shuttle tankers. The turret can be re-connected to the vessel by hoisting the turret up into a recess located in the bottom of the hull structure. The turret design principle is shown in Figure G 5.

The horizontal load components are mainly taken by the lower bearing.

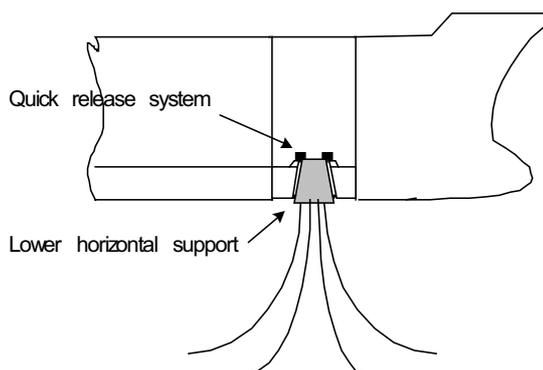


Figure G 5 Internal disconnectable turret

2.2.2. Impact of Turret Location

The location of the turret will influence the weathervaneing properties of the vessel. 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 vessel 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 vessel will mainly orient itself into the head sea direction. Since the turret is frequently located towards one end and on the centreline of the vessel pitch and heave typically dominate the design.

In a collinear environment the roll and transverse accelerations are normally small, however these can 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.

vessels 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.

Turrets located closer to amidships can also provide a passive system but require a bearing system that is very sensitive to external forces on the vessel, e.g. low friction. 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 vessel hull or the rotational stiffness of the mooring / riser system. In this situation the vessel may not fully orientate 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 vessel directionality. Thrusters are sometimes installed also in passive systems to improve directional control in all sea states.

Occasionally thrusters may be used to orientate the vessel so the weather is slightly off the bow to control motions from a crew comfort consideration.

2.2.3. Turrets located near the bow or stern

This applies to both internal disconnectable and 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. 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 calculated in an integrated hull-turret FE analysis.

2.2.4. Turrets located near amidships

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. Typical values of the global SCF are from 1.7 to 2.4. In such locations the wave bending moment may be almost 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 ovalization due to hull bending must 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 ovalization of the moonpool and thus affect the clearance in the lower bearing, which again will affect the dynamic load distribution on the lower bearing. Due to the interaction between the hull and the turret structure it is essential that a close co-operation for information exchange exists between the turret and hull designer.

| No. | Turret type | Location | Typical areas of concern |
|-----|---|--------------------|---|
| 1. | Internal permanent, no lower horizontal support. Supported on deck by means of radial brackets. | Cargo area | <ul style="list-style-type: none"> - ULS capacity of turret, global response. - Presence of turret have impact on global SCF. - Supporting structure above deck exposed to local loads. Fatigue critical. - Supporting structure below deck exposed to hull global and local loads. Buckling and fatigue. - Supporting structure for fairleads and risers. Fatigue critical. - Relative stiffness of hull supports and turret bearing structural support affect local peak loads. |
| | | Bow region | <ul style="list-style-type: none"> - As for the cargo area, but little or none global impact. Hull shear capacity may be important. |
| 2. | As no. 1., but with lower bearing horizontal support. | Cargo and bow area | <ul style="list-style-type: none"> - As for no. 1., but vertical local loads on deck support are much less. - Local structure in turret and hull at lower support. Hull longitudinal stresses are negligible near centre line due to the shadow effect of the moonpool - Necessary clearance between hull and turret horizontal bearing affect distribution of interface loads. |
| 3. | Internal disconnectable. | Bow area | <ul style="list-style-type: none"> - Little impact on global stress distribution in deck. Global SCF in bottom area may be of concern if turret located near cargo area. - Yield and buckling of local support structure. - Support structure for horizontal forces may be fatigue sensitive. |
| 4. | External | Bow | <ul style="list-style-type: none"> - Local supporting structure for turret |

Table G 1 Areas of concern for different turret types

2.3. Risers system

The type of connection between the risers and the ship depend on the mooring system. For spread mooring system the risers are typically fixed to a hang-off platform on top of

the main deck. The riser platform itself are 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 from each riser will govern the local supporting structure. The riser forces will have negligible effect on the global response of the ship.

For turret moored vessels 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. Typical riser systems are shown in Figure G 6.

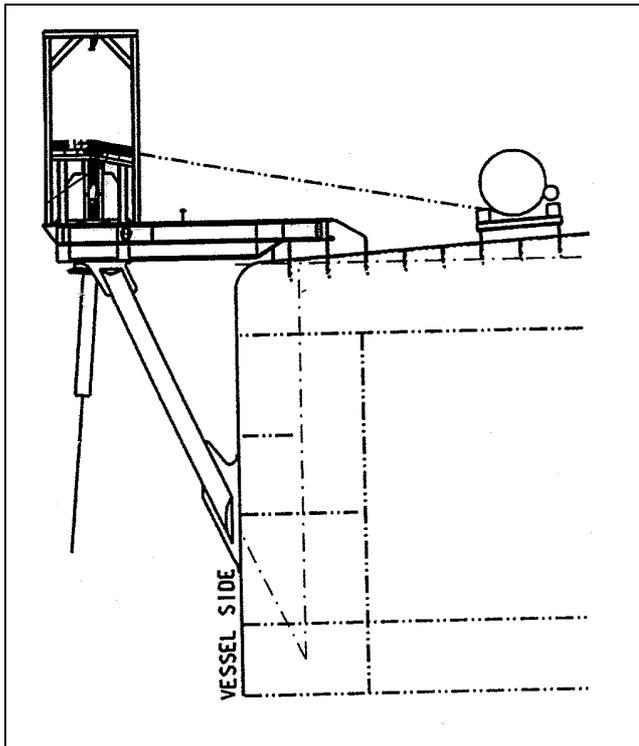


Figure G 6 Typical riser hang-off platform

3. HULL /SPREAD MOORING INTERFACE DESIGN

3.1. Hull-mooring system interaction

The spread mooring system used in benign waters have little impact on the hydrodynamic motions and acceleration of the hull but for the mass representation of the of the mooring lines in the hydrodynamic analysis. For extreme deep water operations, the mooring system and hull motions may be analysed in a coupled analysis.

3.2. Design procedure

A typical design procedure for spread mooring systems is be as shown in Figure G 7. The motions and accelerations are calculated for the 100 year return period for ULS capacity checks, and for the 20 year return period for simplified fatigue calculations.

3.3. Loads

The tension forces in the mooring lines are determined according to the procedure given in DNV offshore standard OS-E301. If a simplified fatigue analysis is to be carried out, the Weibull shape parameter for definition of the long term distribution of the mooring line(s) force must be defined.

Hull girder stresses are derived from the wave load analysis. As a conservative assumption, the extreme longitudinal stress may be assumed to act in phase with the extreme tension force in a mooring line. In benign waters, the non linear effects on the hull girder bending moment may be ignored.

A set of ULS load conditions are combining design values (including partial load factors for the ULS b) combination) of the tension forces in the mooring line(s) from mooring analysis and the hull girder longitudinal stresses. A special load condition is defined based on the breaking strength of one chain for strength capacity check. No other loads are considered in this special case.

The loads for the fatigue capacity check are the mooring line tension forces and the hull girder longitudinal wave bending stresses.

3.4. Local structural response analysis

The local structural response of the supporting structure for the mooring line is normally determined by means of a finite element model covering the local structure only.

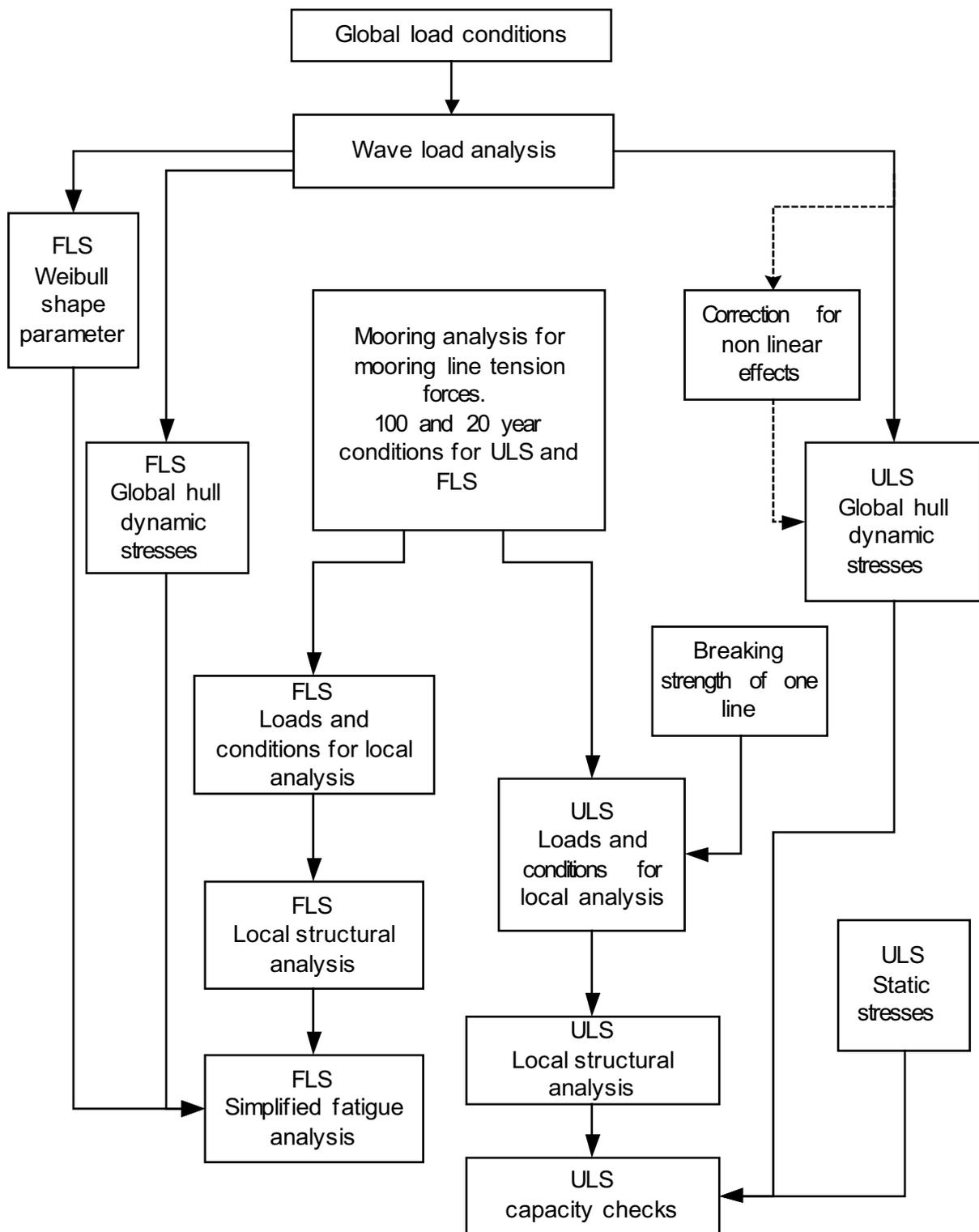


Figure G 7 Design procedure for spread mooring systems

4. TURRET AND HULL/TURRET INTERFACE DESIGN

4.1. General

The design of the turret and turret/hull interface structure need to be considered with respect to ultimate strength and fatigue capacity. The necessity for an integrated hull and turret analysis must be evaluated for each case. As an example the global buckling capacity of large permanent turrets would in most cases be sufficient to assess by applying the resulting mooring and riser force at the fairlead assuming rigid support at the hull interface structure. Fatigue calculations of riser supports must however in most cases account for the motion of the hull.

4.2. Managing the hull / turret interface analysis

Turrets are connected to the hull by different means depending on the turret design. Common to all designs is that the supporting structure must have sufficient capacity to take the vertical and horizontal reaction forces imposed by the mooring and riser forces. The hull deformations due to global hull girder bending response and local structural response will have an impact on the design forces for the turret. The importance of hull deformation for the turret design will differ between the turret designs, but need 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, imperative that compatibility of loads and displacements is achieved throughout the design process in a timely manner. An example of the information that may flow from both the hull and turret contractor to the interface structure responsible is presented in Figure G 8. This figure clearly indicates the criticality for establishing close working relationships between the respective parties and familiarity with their counterparts requirements so as to avoid any misunderstanding. A project specific procedure describing the analysis approach, flow of information between relevant parties, holding points and milestones is to be developed.

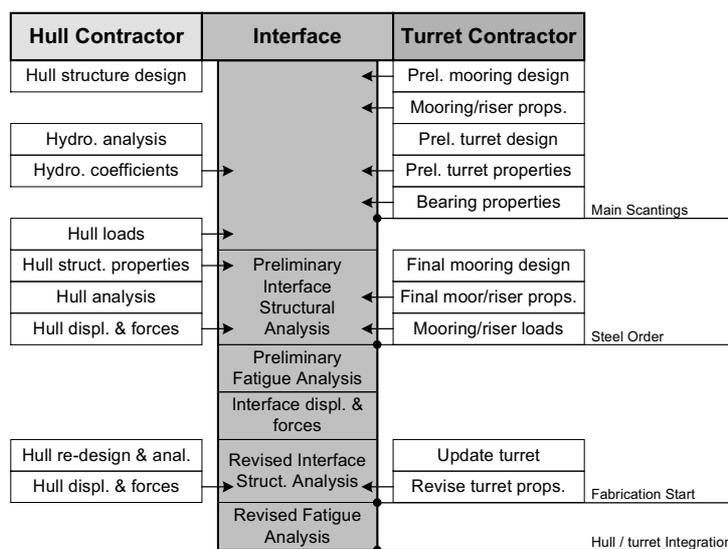


Figure G 8 Example Hull / Turret Information Flow

As shown in Figure G 8 the turret designer must 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. After the hull designer must receive, from the turret designer, scantlings and arrangement of the turret, bearing system and mooring line and riser stiffness and response.

The amount and type of required information will depend on the analysis method and approach followed. Examples of different methods are presented in this Appendix. These portray the required information exchange between hull and turret designer throughout the complete project.

4.3. Design procedure

A typical design procedure for turret mooring systems is be as shown in Figure G 9. The motions and accelerations are calculated for the 100 year return period for ULS capacity checks, and for the 20 year return period for simplified fatigue calculations. Typical areas of consideration are given in Table G 1 for different turret types and location.

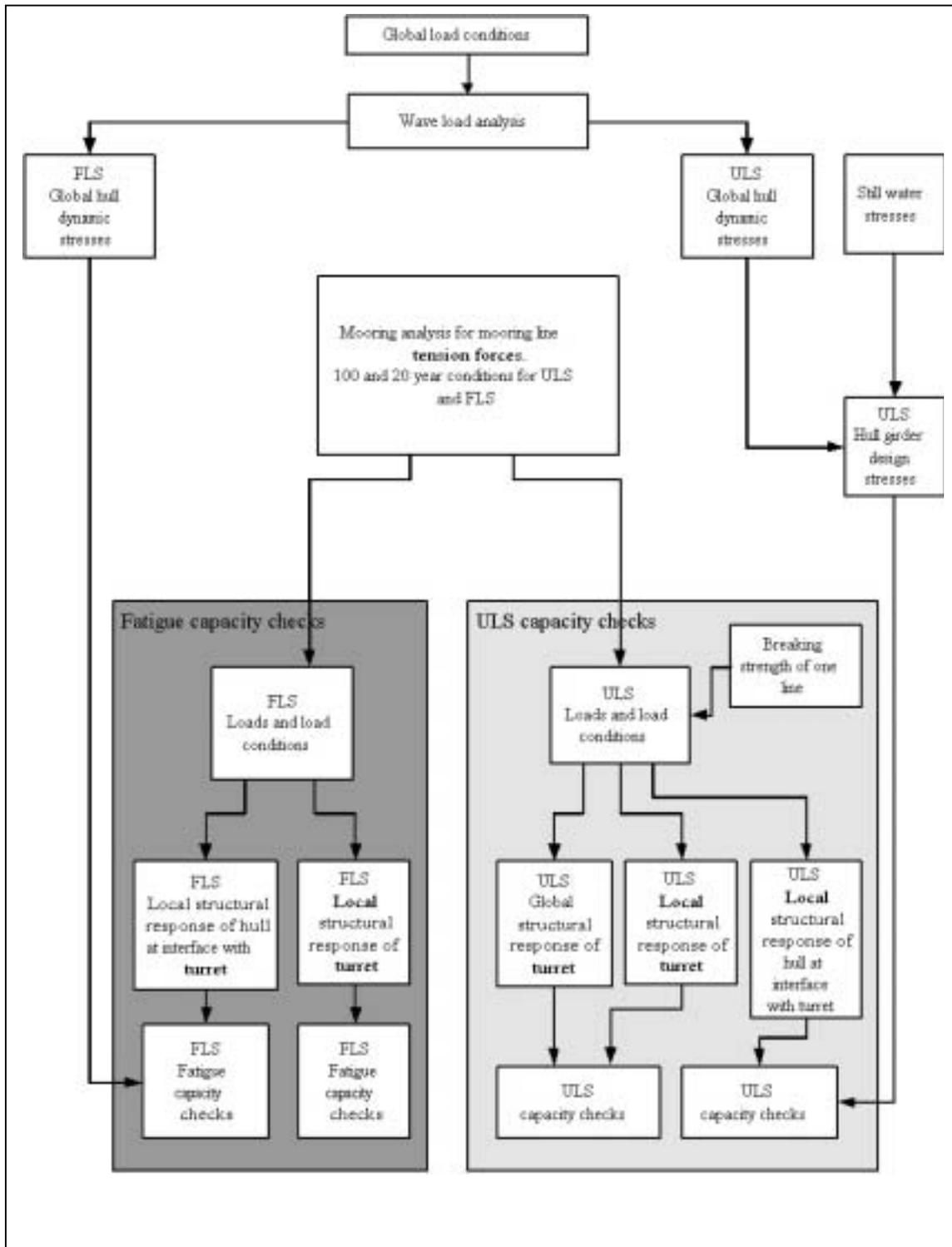


Figure G 9 Design procedure for turret type mooring systems

4.4. ULS loads

ULS loads for the structural response analysis of the turret consist of the tension forces in the mooring lines, the static force from the chains and the gravity force from the turret. Mooring line tensions are determined according to the procedure given in DNV-OS-E301.

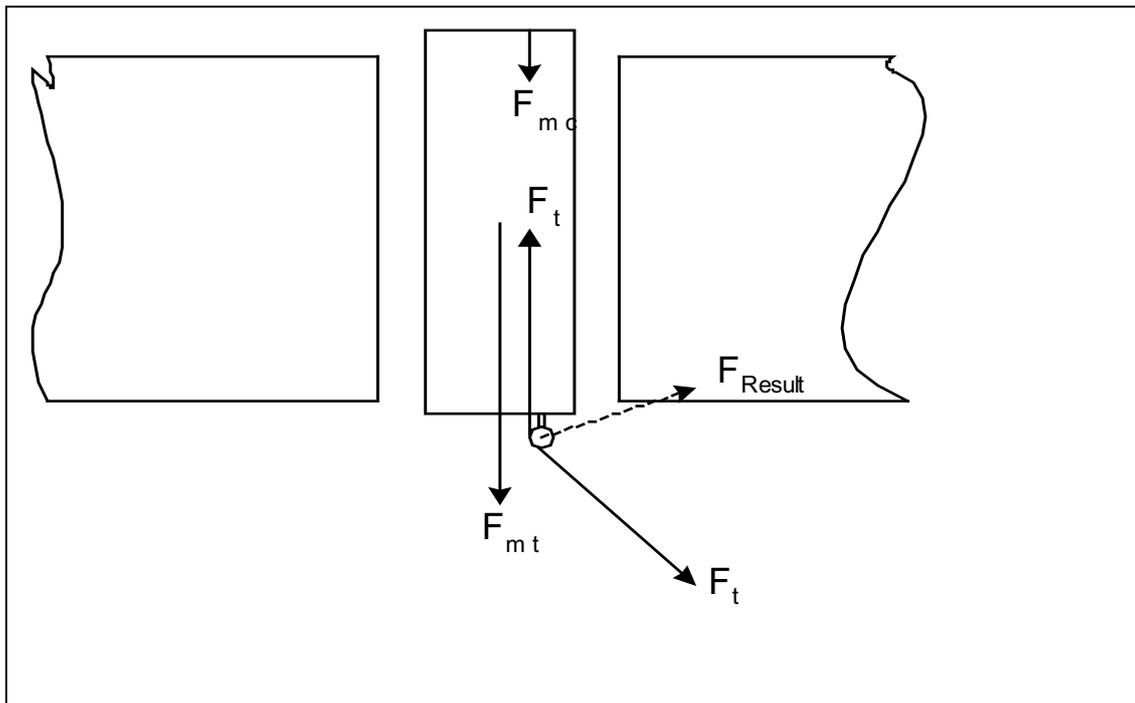


Figure G 10 Longitudinal CL section. Applies mooring forces

A special load case for strength assessment is the force equivalent to the breaking strength of one chain.

The loads are applied to the fairleads for the ship in head sea direction and may be assumed to act on the hull in a sector of typically $\pm 30^\circ$.

Figure G 10 shows a principal sketch of forces from the mooring lines and static forces applied to the turret model. Possible buoyancy forces must be included. The figure also shows the resulting force F_{Result} from the line tension. The “breaking chain” strength load is assumed to act horizontally, or maximum 10° down from the horizontal plane, at the fairlead.

4.4.1. ULS load conditions

The local loads from internal tank pressure or external sea water on the hull structure must be considered in the design of the turret – hull interface. The governing combinations of hull loads and turret loads must be established. In Figure G 11 an example is shown for an internal, disconnectable turret located in the forward part of the hull with one or two adjacent cargo tanks. The transverse bulkheads are assumed to have vertical webs as primary girders.

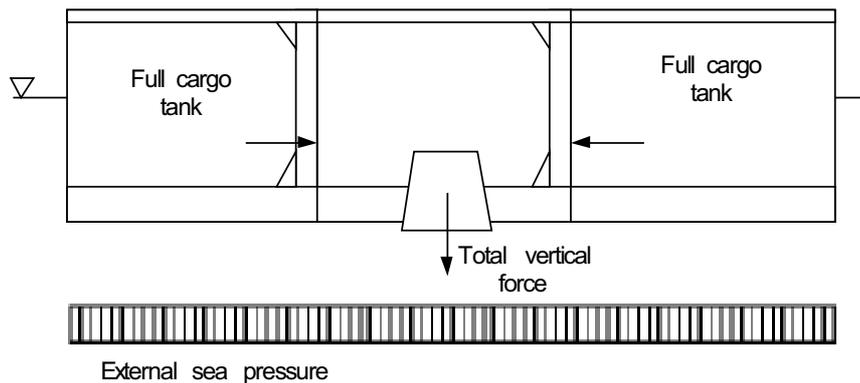


Figure G 11 Typical load condition

As a minimum, the following main load conditions should be considered for ULS b) combination:

| LC no. | Description | Comments |
|--------|--|---|
| 1. | Full load condition, wave crest . Max vertical mooring forces. Full adjacent tanks. | The vertical force from turret will cause local sagging of double bottom. Full draught + wave crest will counteract this deformation, but the rotation of the DB girders from lateral pressure on the transverse bulkheads may be governing. |
| 2. | Ballast condition, wave trough . Max vertical mooring forces. Full adjacent tanks. | The vertical force from turret will cause local sagging of double bottom. Ballast draught + wave trough minimise the counteracting force. Rotation of the DB girders from lateral pressure on the transverse bulkheads will be less, but than may not be governing. |
| 3. | Full load condition, wave crest . Breaking force of on line applied horizontally. Empty adjacent tanks | This is a special load case. Hull loading selected to maximise upwards displacement at turret |

Several variations of the main load conditions may be relevant as the mooring forces are assumed to act in different directions to the supporting structure.

4.5. Fatigue loads

The fatigue loads are the mooring line tension forces maximised in different directions, and the hull girder longitudinal and transverse dynamic stresses.

4.5.1. Mean Response

The mean loads adequately determine the mean position of the ship 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 may be large locally, 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 draught, trim or heel and are static in nature.

4.5.2. Wave-Frequency Response

The wave frequency tensions are also small in comparison with the first order wave loads on an hull and can be neglected in the determination of the motion response. One possible exception may be roll response. If the transverse metacentric height is low, there is a relatively small roll restoring coefficient. A spread mooring system may contribute somewhat to this restoring moment and affect the roll response. A turret mooring will not have the same effect, since the line tensions are applied closer to the roll axis. However, difficulty in modelling the damping of the roll motion leads to relatively lower accuracy for the predicted roll response, compared to the other modes of motion. Hence, it is inappropriate to model the effect of the mooring on the roll response.

It is possible to integrate the wave-frequency response of the mooring system into a hull structural analysis, if the restoring characteristics of the individual lines are linearised. Such a linearisation is to some extent dependent on the mean position of the system and the direction of the environmental actions; i.e. some compromise has to be made in selecting a linearisation point.

4.5.3. Low-Frequency Response

The low-frequency motions of ship 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 might be neglected if the resonant system were not sensitive to these exciting frequencies. Although the exciting forces are small, the resonant vessel excursions may be large, and give rise to significant variations in line tension. Drag forces on the mooring lines and risers tend to damp the low-frequency vessel motions. Hence, the number of risers that are present may be important for the response.

The second order wave exciting forces are nonlinear and the restoring forces from the mooring system are nonlinear, so that these effects cannot easily be integrated into a hull structural analysis that is primarily designed to cater for linear response and excitation.

4.6. Hull / turret interface calculation methods

As discussed in section 4.2 there are several ways in which the fatigue capacity of the interface structure may be established. The calculation methods should be able to capture stresses from all relevant loads, including both wave-frequency and low-

frequency loads. To simplify and manage the process separate analyses may be used for wave-frequency and low-frequency responses. However, the combination of stresses is important for fatigue damage calculations. Acceptable methods for fatigue calculations of the turret/hull interface structure are summarised below. All loads should be calculated in a hydrodynamic analysis where the location and mass of the turret is taken into account.

4.6.1. Coupled stochastic analysis based on riser/mooring RAO's

The flowchart for a typical coupled stochastic analysis is shown in Figure G 12 . The different steps are as follows:

- Hydrodynamic loads, excluding riser loads are transferred from the hydrodynamic analysis to the structural model. The hydrodynamic analysis may be performed without stiffness from the riser/mooring system.
- Linearised riser/mooring loads from the mooring analysis are transferred to the structural model.
- Loads/stresses from the two load types are combined either before or after the structural analysis.
- Full stochastic analysis is performed with the calculated stresses.

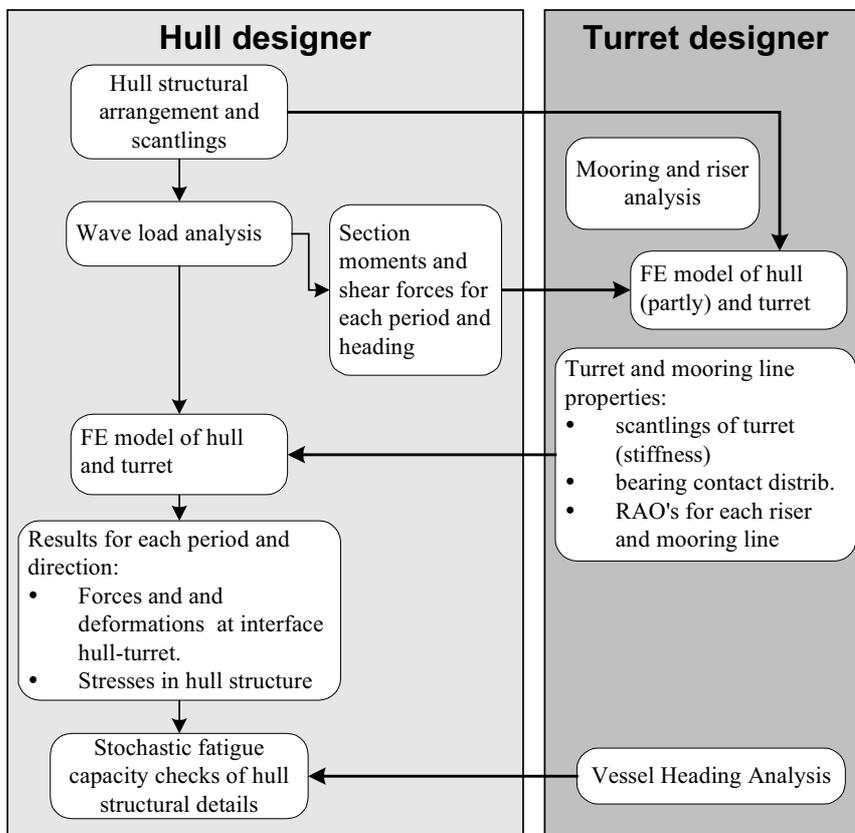


Figure G 12 Coupled analysis based on RAOs for each riser and mooring line

4.6.2. Coupled analysis based on linear spring representation of each riser and mooring line

The flowchart for a typical coupled stochastic analysis based on linear spring representation of the mooring lines and risers is shown in Figure G 13. The different steps are as follows:

- Hydrodynamic loads, excluding riser loads are transferred from the hydrodynamic analysis to the structural model. The hydrodynamic analysis may be performed without stiffness from the riser/mooring system.
- Spring representation of mooring lines and risers are included in the wave load analysis.
- Full stochastic analysis is performed with the calculated stresses

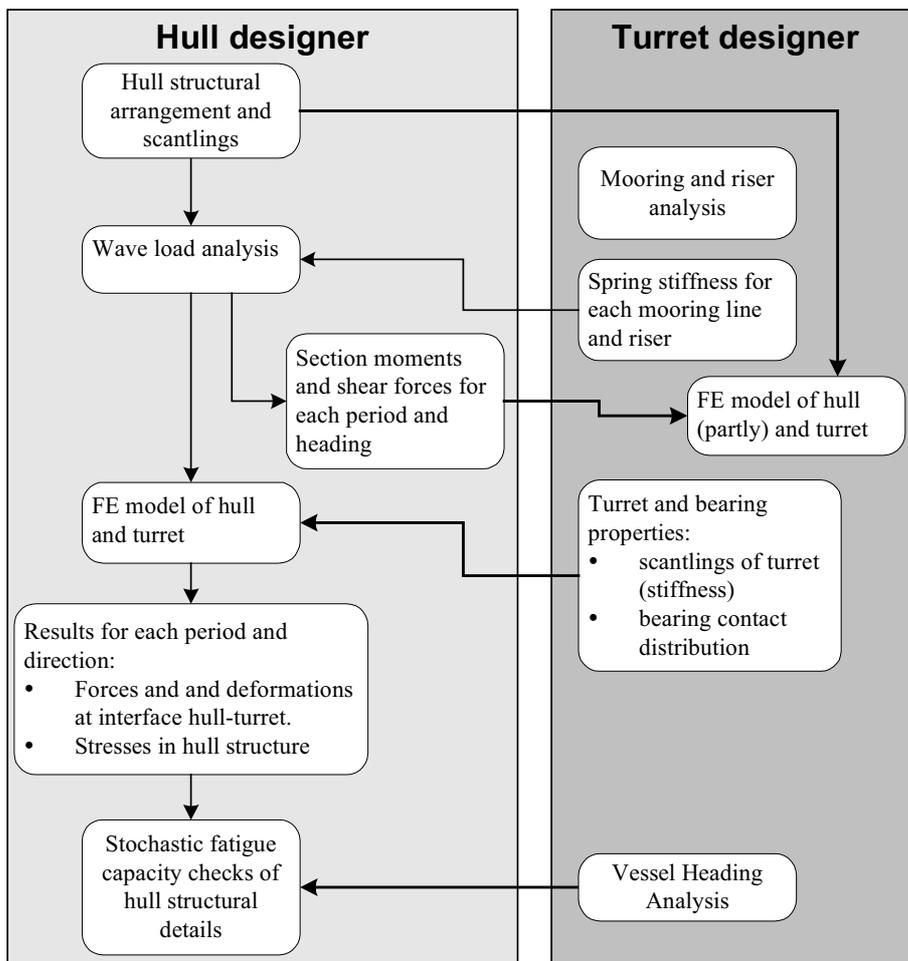


Figure G 13 Coupled analysis based on linear spring representation of each riser and mooring line

4.6.3. Simplified fatigue using maximum mooring / riser dynamic range

The design steps in the simplified fatigue analysis are shown in Figure G 14. The fatigue approach is based on analysis of the 20 years dynamic stress levels in combination with a Weibull parameter for description of the long term distribution of stress or load amplitudes. The Weibull parameter should be estimated for all important load effects for the given design. The dominating load effects should be used for description of the long term distribution of stresses in the turret section.

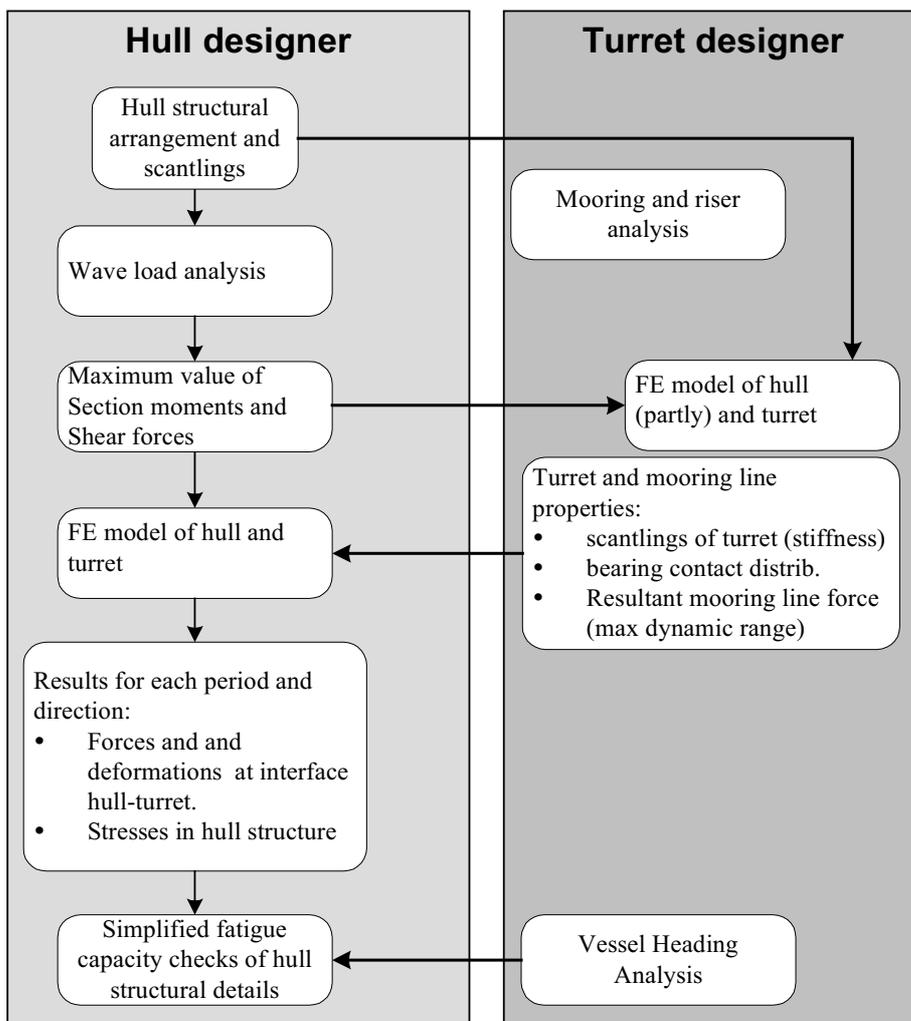


Figure G 14 Simplified fatigue analysis based on maximum dynamic range of the resultant riser and mooring line force

4.7. Finite element model of internal turrets

The element analysis shall represent the global structural response as well as the local response of supports for mooring lines and risers. This is often done by sub-modelling techniques where the boundary conditions for the local fine mesh model are automatically transferred from the global model in terms of forced nodal displacements. The complete turret is modelled. The connection to the hull is modelled such that the total spring stiffness of the bearing and hull supporting structure. This may be achieved by including the support structure of the ship sufficiently to represent the local relative stiffness, and to represent the bearing system as spring elements. An example of a

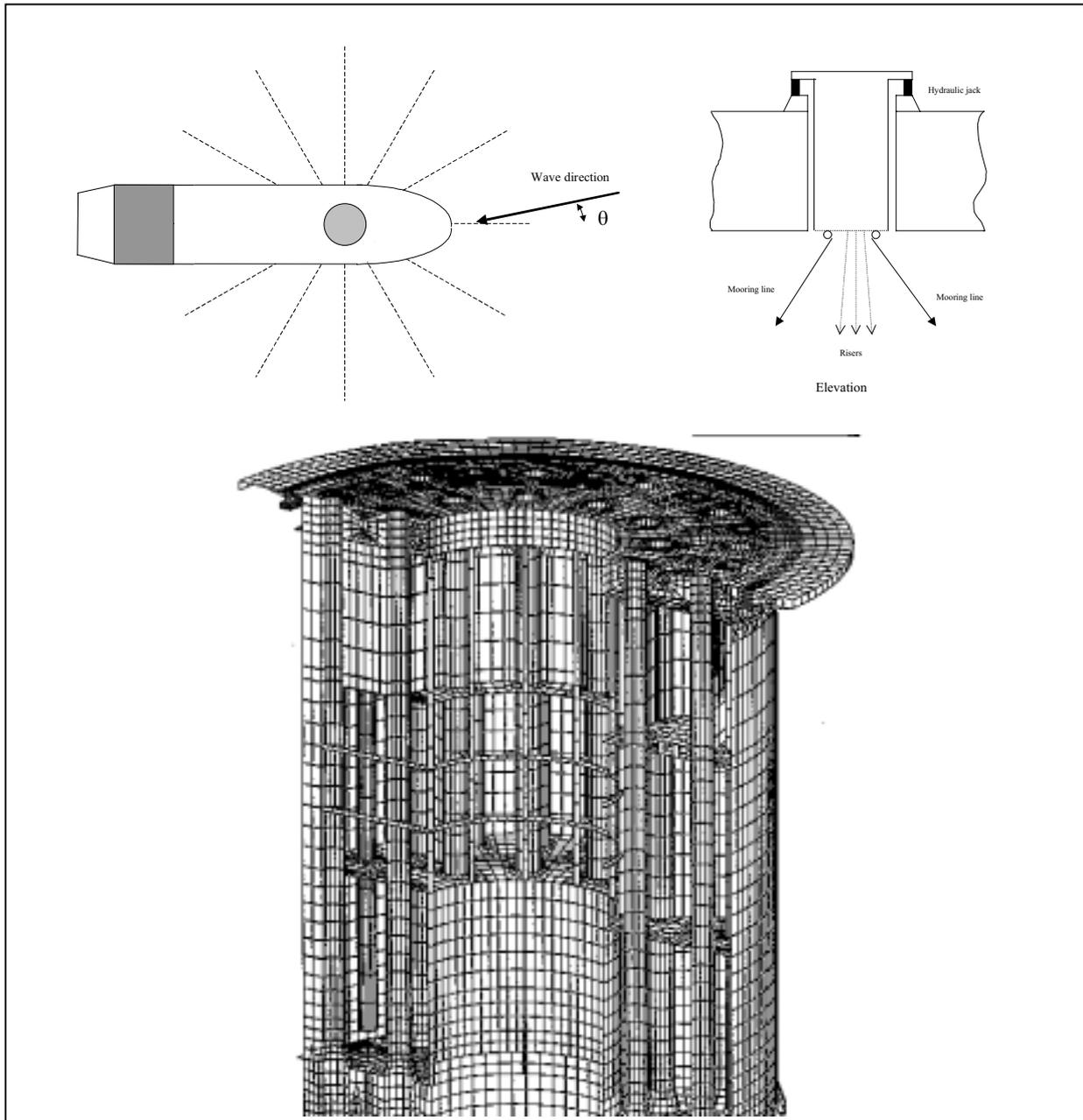
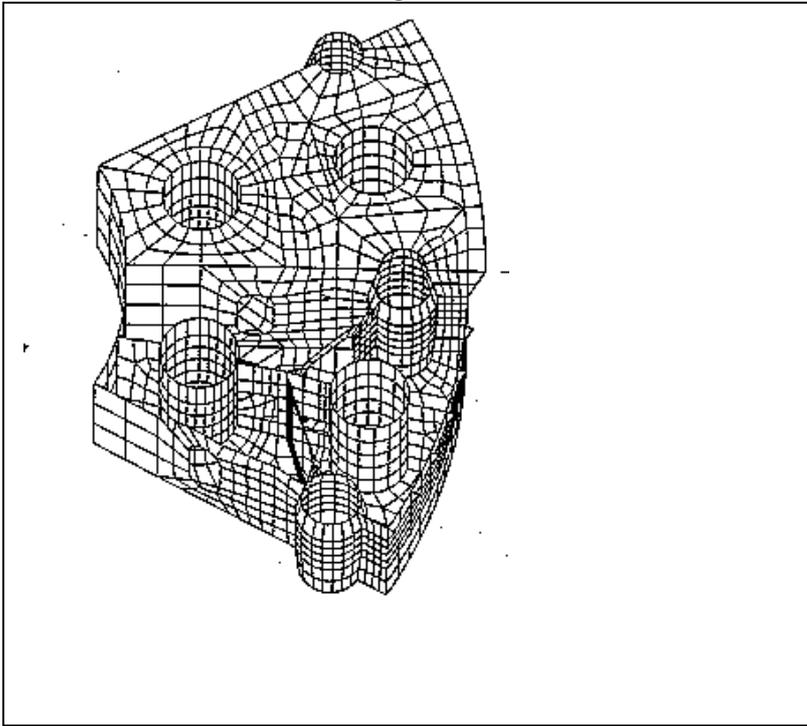


Figure G 15 Finite element model of internal permanent turret, global model

global element model of an internal permanent turret is shown in Figure G 15. Only the turret super-element is shown, not the surrounding structure. In this case, there were no lower bearing support. The hull structure between adjacent transverse bulkheads is modelled with a relative coarse mesh, as the only purpose is provide correct stiffness. The local analysis of the risers and fairleads are in this case done by means of a fine mesh sub-model as shown in Figure G 16. The element mesh is not fine enough to



determine the local geometrical stress concentration factors for fatigue analysis.

Figure G 16 Sub-model of riser and fairlead connections

Appendix H

CRANE PEDESTALS

| | | |
|-----------|---|-----------|
| 1 | CRANE PEDESTAL STRUCTURAL EVALUATION | 3 |
| 1.1 | SCOPE AND RULE REFERENCE. | 3 |
| 2 | INTRODUCTION | 5 |
| 3 | DOCUMENTATION | 6 |
| 4 | DESIGN LOADS | 7 |
| 4.1 | CRANE PEDESTAL..... | 7 |
| 4.2 | CRANE PEDESTAL SUPPORT | 7 |
| 5 | DETERMINATION OF CRANE GROUP | 9 |
| 6 | GENERAL FATIGUE ASPECTS | 11 |
| 7 | CAPACITY ASSESSMENT OF PEDESTAL | 13 |
| 7.1 | GENERAL | 13 |
| 7.2 | BUCKLING CHECKS OF CRANE PEDESTAL | 13 |
| 7.3 | FATIGUE CHECKS OF CRANE PEDESTAL | 14 |
| 7.4 | FATIGUE EVALUATION OF PEDESTAL ACCORDING TO THE LIFTING APPLIANCES..... | 16 |
| 8 | CAPACITY ASSESSMENT OF SUPPORTING STRUCTURE | 18 |
| 8.1 | YIELD CHECKS OF CRANE PEDESTAL SUPPORT STRUCTURE | 18 |
| 8.2 | BUCKLING CHECKS OF CRANE PEDESTAL SUPPORT STRUCTURE | 19 |
| 8.3 | FATIGUE CHECKS OF CRANE PEDESTAL SUPPORT STRUCTURE | 19 |
| 9 | DETERMINATION OF CRANE GROUP | 21 |
| 10 | EXAMPLES OF STRESS PLOTS | 23 |

1 Crane Pedestal Structural Evaluation

1.1 Scope and rule reference.

This appendix covers the crane pedestal and the supporting structure. The appendix is primarily covering crane pedestals on offshore ships, but the principles it may also be utilised for other type of units. The purpose is to give an understanding of the procedure for evaluation of yield, buckling and fatigue capacities of the crane pedestal and the supporting structure. According to main class a crane pedestal is considered to extend from the deck, upon which it is mounted, to the slewing ring. The extent of the support structure includes the structural elements affected by the crane load and will typically include:

- Part of deck plate and stiffeners
- Bracket welded to deck
- Longitudinal or transverse bulkhead below the pedestal
- All weld within these elements

Relevant rule references, regulations and guidelines are:

- a) Rules for Classification of Ships Pt.3 Ch.1
- b) Rules for Classification of Mobile Offshore Units Pt.3 Ch.1
- c) Seats for additional lifting, towing or mooring equipment.
Rules for Classification of Ships Pt.3 Ch.3 Sec.5
- d) Structural design and strength of cranes
Rules for Certification of Lifting Appliances, 1994, Section 3.
- e) Kraner og løfteutstyr, Stålkonstruksjoner, Beregninger
Norsk Standard 5514, Second edition, Sept. 1978
- f) Rules for the design of hoisting appliances,
Federation Europeenne de la Manutention, FEM 1.001, third edition Oct.1987
- g) DNV-OS-C101 – Design of Steel Structures
- h) DNV-OS-C102 – Structural Design of Offshore Ships
- i) DNV-OS-C103 – Column Stabilized Units (LRFD)
- j) DNV-OS-C201 – Design of Offshore Units (WSD)
- k) DNV-RP-C202 Buckling Strength of Shells

- l) DNV-RP-C203 Fatigue Strength Analysis of Offshore Steel Structures
- m) Classification Notes 31.4 Column Stabilized Units, Appendix D, “Typical fatigue parameters”
- n) Classification Notes 30.7 Fatigue Assessment of Ship Structures
- o) Classification Notes 30.1 Buckling Strength Analysis

2 Introduction

The crane pedestal is normally categorised as a major structural equipment. It is considered to be exposed to heavy dynamic loading. On a typical crane pedestal structure including the support, there are a number of structural details, which requires thorough evaluation, in particular with respect to fatigue life.

3 Documentation

General documentation requirements are given in Rules for Classification of Ships Pt.3 Ch.3 Sec.5 A200. The following documentation is normally carried out:

- Detailed drawings of the crane pedestal structure, including exact location on the hull.
- Structural design report documenting yield, buckling and fatigue capacity.
- Crane specifications including:
 - Design loads, normally given at slewing ring, dynamic coefficient included. Alternatively, as a minimum, the safe working load shall be given.
 - Crane group, which depends on the number of load cycles and the corresponding working loads.

The level of documentation needed depends on the complexity of the structure, but in general the capacity of the support structure shall be documented by Finite Element analysis. The basic circular crane pedestal can in principal be evaluated using simple hand-calculations, but F.E.M analysis may be required for details where the local stress concentration factor are not given in the literature. Such typical details that may require finite element analyses are cutouts, knuckles and transitions.

4 Design loads

4.1 Crane pedestal

In the Rules for Lifting Appliances Sec 3C, the various loading conditions to be considered for a *crane* are given. The corresponding acceptance criteria for the strength evaluations are given in Sec. 3D. For strength evaluations of the crane pedestal the governing criteria are given in Rules for Classification of Ships Pt.3 Ch.3 Sec.5.

Normally the crane manufacturer has performed an analysis for the crane itself with an envelope of loadcases as given in Rules for Lifting Appliances Sec.3C. The output of the analysis should be a set of forces and moments to be applied to the crane pedestal. These forces normally act at the slewing ring. Particular care must be made to ensure that a sufficient dynamic coefficient, ψ , has been applied (See /c/ A500, and /d/ B302-307).

Offshore crane: Crane which loads/unloads off the edge of the unit from supply vessel, barge or similar units. For offshore cranes a minimum dynamic factor of 2.0 shall be applied.

Shipboard crane: All cranes that only do internal lifting operations.

If no set of maximum forces and moments are given, the loads are to be determined using Rules for Classification of Ships Pt.3 Ch.3 Sec.5 A501-502. The safe working load (SWL) shall, as a minimum, be specified.

Please note that the same load is applicable for both yield, buckling and fatigue evaluations. All loads are considered to be dynamic.

Note:

In the Offshore Standards (and NPD-regulations) which follows the Load Resistance Factor Design (LRFD) principles, all functional loads are to be multiplied by an additional load factor. The acceptance criteria are also different from the ship rules. Experience suggests that an equivalent safety level is found by using the design loads and acceptance criteria from the Rules for Classification of Ships compared to the LRFD method. Hence the design load need not be multiplied by the additional load factor if acceptance criteria from the Rules for Classification of Ships are employed.

4.2 Crane pedestal support

The support structure will be exposed to the following loads:

- Functional loads from crane action
- Inertia loads (accelerations) from crane/crane pedestal due to movement of the vessel.

– Hull deflections

Functional loads and inertia loads (accelerations) will for most offshore vessels be given explicitly by the designer as a result of direct calculations.

If no set of design loads is available from crane manufacturer, then the procedure as given in Rules for Classification of Ships Pt.3 Ch.3 Sec.5 A501-502 may be applied.

Accelerations to be taken from Rules for Classification of Ships Pt.3 Ch.1 Sec.4 or direct calculations as applicable. Accelerations shall be realistically combined. As a general rule for ship shaped units they can be combined as indicated in Rules for Classification of Ships Pt.3. Ch.1 Sec.4 C500.

Bending moments giving hull deflections are to be taken from Rules for Classification of Ships Pt.3 Ch.1 Sec.5 or direct calculations as applicable. Note that only the wave bending moment shall be applied in fatigue evaluations. For units other than ships, hull deflections need normally not be considered.

Realistic combinations of functional loads, inertia loads and hull deflections to be considered.

Please note that in fatigue evaluations inertia loads and wave bending moments are to be considered at the same probability level.

5 Determination of crane group

Crane group is normally specified by crane manufacturer and is a function of the number of load cycles and the load spectrum.

The following is an extract from NS5514 with guidelines for determination of crane group. This can be employed if the crane group is not given by crane manufacturer.

Table 1.1 Class of utilization

| Class of utilization | Frequency of operation | No. of load cycles |
|----------------------|---|--------------------|
| A | Irregular operation followed by long rest periods | 63000 |
| B | Regular operation at normal load level | 200000 |
| C | Continuous operation at normal load level | 630000 |
| D | Continuous operation at high load level | 2000000 |

Table T-1.122 State of Loading

| State of loading | Definition | Curve |
|-------------------|---|-----------|
| 0 (very light) | Component rarely exposed to maximum stresses. Small stresses normal. | $p = 0$ |
| 1 (light) | Component seldom exposed to maximum stresses. Stresses at 1/3 of maximum is normal. | $p = 1/3$ |
| 2 (moderate) | Component often exposed to maximum stresses. Stresses between 1/3 and 2/3 of maximum is normal. | $p = 2/3$ |
| 3 (heavy) | Component regularly exposed to maximum stresses. | $p = 1$ |

Table T-1.13 Crane group

| State of loading | Class of Utilization | | | |
|-----------------------------|----------------------|---|---|---|
| | A | B | C | D |
| 0 (very light, $p = 0$) | 1 | 2 | 3 | 4 |
| 1 (light, $p = 1/3$) | 2 | 3 | 4 | 5 |
| 2 (moderate, $p = 2/3$) | 3 | 4 | 5 | 6 |
| 3 (heavy, $p = 1$) | 4 | 5 | 6 | 6 |

Typical group classification of lifting appliances as found in Section 3 of the Rules for Lifting Appliances:

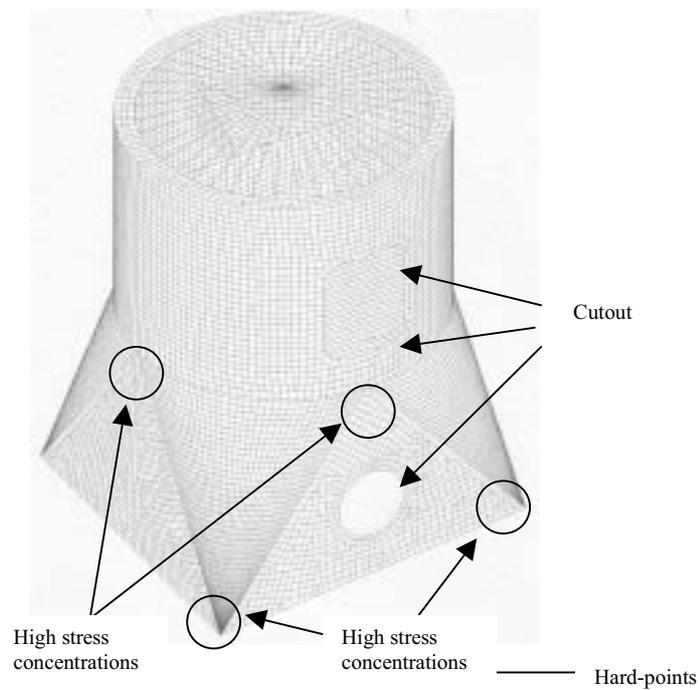
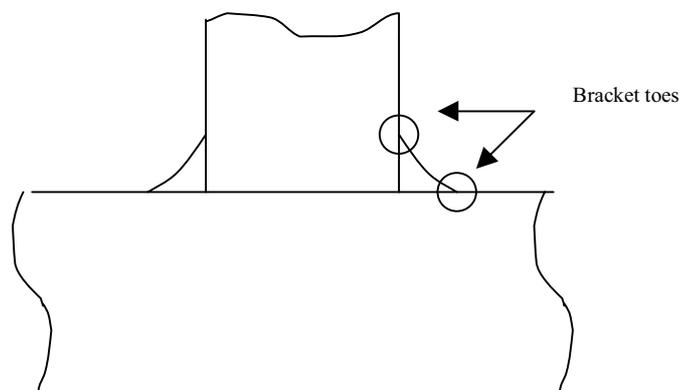
| Definition of crane | Class of utilization | State of loading | Group |
|---|----------------------|------------------|-------|
| Cranes for exceptionally low service time, e.g. BOP crane | A | 0 | 1 |
| Workshops and similar industrial cranes with hook | B | 1 | 3 |
| Overhead travelling cranes for maintenance purpose | A | 1 | 2 |
| Pipe rack cranes | A | 1 | 2 |
| Store room cranes | A | 2 | 3 |
| Jib or gantry cranes for container service | B | 2 | 4 |
| Dock side and shipyard jib cranes | B | 2 | 4 |
| Shipboard cranes | B | 2 | 4 |
| Offshore cranes, whip hoist | B | 2 | 4 |
| Offshore cranes, main hoist | A | 1 | 2 |
| Hose handling cranes | A | 2 | 3 |
| Provision cranes | A | 2 | 3 |

A more extensive list can be found in NS5514, table T-1.15.

6 General fatigue aspects

Typical details to be checked for fatigue, and which may require detailed S.C.F calculations are:

- Bracket toes
- Knuckles/transitions
- Cut-outs
- Deck connections in general (Possible misalignment problems).



Improving fatigue capacity

Depending on where the problem exists, the following measures can be taken:

At bracket toe:

- Grind the weld flush.
- Soften the bracket to reduce S.C.F at toe.

At knuckle:

- Reduce nominal stresses and possibly stress concentrations by introducing additional structure such as vertical stiffeners or brackets.

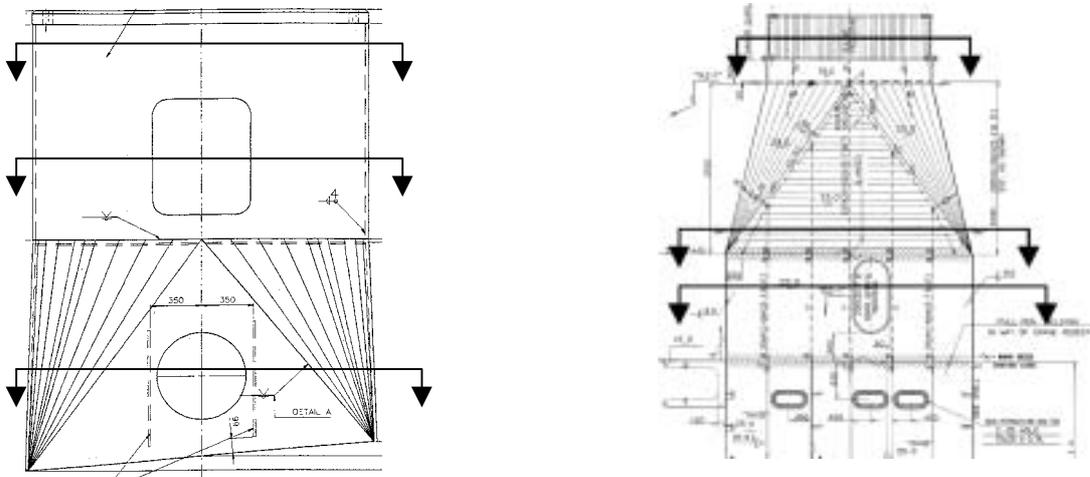
At cut-out:

- Grinding of weld/Change the corner radius to reduce S.C.F.
- Change location of the cutout.

7 Capacity assessment of pedestal

7.1 General

Equivalent stress and shear stress in pedestal to be evaluated at representative sections along the pedestal. Typical sections are shown in the figures below. Bending stresses are typically the governing components. Simple hand calculations may be sufficient depending on the structural arrangement. Since load magnitudes are given at the slewing ring, the horizontal (shear) force will set up an additional moment along the length of the pedestal to be added to the overturning moment.



Acceptance criteria as given in the Rules for Lifting Appliances Sec 3.D. table D1 for elastic analysis.

- Loadcase I (without wind) the safety factor is 1.5:

$$\sigma_p = \sigma_y / 1.5$$

- Loadcase II (with wind) the S.F is 1.33:

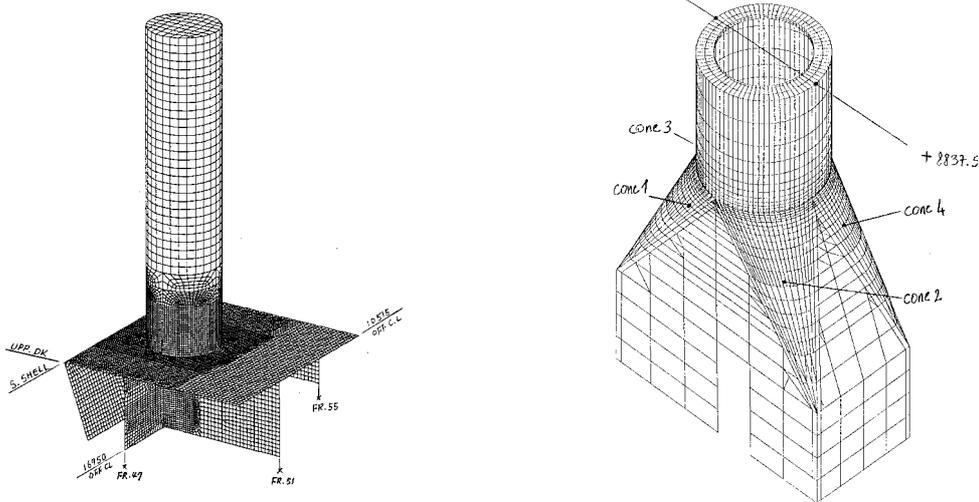
$$\sigma_p = \sigma_y / 1.33$$

etc.

Yield check of hot-spot areas may also be required. This may require an FEM-analysis. See section 1.12 for typical hot-spot areas.

7.2 Buckling checks of crane pedestal

A typical crane pedestal typically comprises either a cylindrical shell structure, or a conical and a rectangular part structure.



Buckling is to be evaluated using DNV-RP-C202 or Classification Notes.30.1 Sec.4, 5 or 6, using acceptance criteria according to the Rules for Lifting Appliances Sec.3D.

(Note that the DNV computer program PILOT NV5080 can be used to evaluate buckling capacities)

In addition, if applicable, the buckling capacity of stiffened/unstiffened plane plates within the pedestal must be evaluated based Classification Notes 30.1 Ch.3.

Acceptance criteria as given in the Rules for Lifting Appliances Sec 3.D table D2. Elastic-plastic buckling redistribution is not considered. Similar for plates and shells.

- Loadcase I (without wind) the safety factor is 1.69:
 $\sigma_p = \sigma_y / 1.69$ ($\eta = 1/1.69$)
- Loadcase II (with wind) the S.F is 1.51:
 $\sigma_p = \sigma_y / 1.51$ ($\eta = 1/1.51$) etc.

7.3 Fatigue checks of crane pedestal

7.3.1 General

Fatigue capacity may be based on the Norwegian standard NS5514 which is based on the FEM standard of 1970. However the FEM standard has later been revised, and the procedure for evaluating fatigue parameters has changed slightly. So far, experience shows that the old standard (NS5514 equivalent) is more conservative than the new edition. While the new standard evaluates fatigue based on component classification, and not crane classification, the correlation is apparent.

The above standards give procedures to calculate the maximum allowable stress for particular structural details on the basis of certain given crane specifications to be supplied by crane vendor/designer.

The relevant sections in the NS5514 standard are:

- Ch.1.1: Tables T-1.11, “Bruksklasse/Class of Utilization” & T.121, “Lastklasse/State of Loading”.
- Tables T-1.122, “Spenningsstilstand/Spectrum Class” & T-1.13, “Krangruppe/Crane group”
- Ch.1.45, including appendix A-1.45

The allowable stress depends on the crane group and type of construction. Construction type is determined using table T-A-1.45 (I) page 67.

Construction type for some typical welds in a pedestal:

- Buttweld without eccentricity; K_0
- Buttweld with eccentricity (angle $1/3$); K_2
- Buttweld with eccentricity (angle $1/2$), welds between ring stiffener and pedestal, bracket toe (not ground), connection between main girders in supporting structure and pedestal; K_3 .

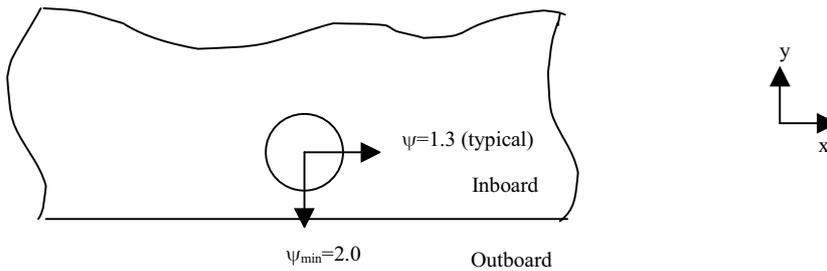
If the weld is ground, the construction type can be changed one step. E.g. from K_2 to K_1 . The allowable stress is a hot-spot stress, which implies it shall be compared to an actual stress including geometric stress concentration factors (K_g).

The K_g is determined either from FEM analyses directly or from libraries of typical details that are available in for example Classification Notes 30.1 & Classification Notes 30.7.

In the cases where the Classification Notes give only a total S.C.F ($=K_g * K_w$) then this may be applied in combination with construction type K_0 .

When evaluating fatigue capacities be aware of any additional stress concentration due to eccentricities at the connections. Small eccentricities can give large decreases in fatigue life.

Note also the importance of the dynamic factor, ψ . When evaluating fatigue capacity of particular details for an offshore crane such as cutouts and knuckles, the actual position of the detail is important in order to determine the dynamic coefficient to be applied, and evaluate the stress. A detail positioned in the y-direction (see figure below) will experience the full loading, minimum dynamic coefficient of 2.0, as the crane lifts off the edge of the unit (outboard lift). A detail positioned in the x-direction of the unit will be most heavily



stressed when an internal lift is performed, and hence a smaller dynamic factor can be applied.

When an F.E.M analysis is applied in order to determine hot-spot stresses, principal stresses are to be used.

7.4 Fatigue evaluation of pedestal according to the Rules for Lifting Appliances

The fatigue capacity is determined based on the following parameters:

- The total number of load cycles and the load spectrum the pedestal is exposed to.
- Material and geometry (stress concentration, represented by K) in the point of consideration.
- The absolute maximum stress σ_{\max} , tension or compression, which can possibly occur in the pedestal.
- Relationship κ between maximum and minimum stress in a stress cycle. Conservative to assume $\kappa = -1$. I.e. maximum tension equals maximum compression.

From the below table the maximum allowable stress can then be found for $\kappa = -1$. The stresses have been determined based on mechanical testing. Please note that the allowable stress can never exceed 0.75 times the tensile strength. Underlined values in the table exceed these values and shall not be given any consideration.

Table T-A-1.45 Values for σ_w depending on crane group and geometry (stress concentration) $\kappa = -1$

| Crane group | Non-welded connections | | | | | | Welded connections for steel St37 to St52 | | | | |
|-------------|------------------------|------|---------|------|---------|------|---|------------|------------|-------|-------|
| | w_0 | | w_1 | | w_2 | | K_0 | K_1 | K_2 | K_3 | K_4 |
| | St37/42 | St52 | St37/42 | St52 | St37/42 | St52 | | | | | |
| 1 | <u>279</u> | 346 | 238 | 294 | 196 | 237 | <u>466</u> | <u>416</u> | <u>349</u> | 247 | 150 |
| 2 | 235 | 279 | 200 | 237 | 165 | 195 | <u>330</u> | <u>294</u> | 247 | 177 | 106 |
| 3 | 198 | 225 | 169 | 191 | 138 | 157 | 233 | 208 | 175 | 125 | 74,9 |
| 4 | 167 | 181 | 141 | 154 | 117 | 127 | 165 | 147 | 124 | 88,3 | 53 |
| 5 | 140 | 146 | 119 | 125 | 98,1 | 102 | 117 | 104 | 87,3 | 62,4 | 37,4 |
| 6 | 118 | 118 | 100 | 100 | 82,4 | 82,4 | 82,4 | 73,5 | 61,8 | 44,1 | 26,5 |

w_0 – base material

w_1, w_2 – bolted connections

K_0 to K_4 is determined from table T-A 1.45 (I) in NS5514. See also section 1.8 of this guidance.

A Similar table is given in NS5514 for $\kappa \neq -1$. These tables are based on the following formulae, which can be used to calculate the allowable stress directly for any value of κ .

a) $\kappa \leq 0$

$$\text{tension: } \sigma_t = \sigma_w \frac{5}{3 - 2\kappa}$$

$$\text{compression: } \sigma_c = \sigma_w \frac{2}{1 - \kappa}$$

b) $\kappa > 0$

$$\text{tension: } \sigma_t = 1.66\sigma_w \frac{1}{1 - \left(1 - \frac{1.66\sigma_w}{0.75\sigma_R}\right)\kappa}, \text{ where } \sigma_R \text{ is the tensile strength of the material.}$$

$$\text{compression: } \sigma_c = 1.2\sigma_t$$

The reason for the factor of 0.75 on the maximum allowable stress is to account for a safety factor of 4/3. For more details on the basis for the fatigue evaluation procedure please refer to NS5514.

Comparison of weld classes (approximate):

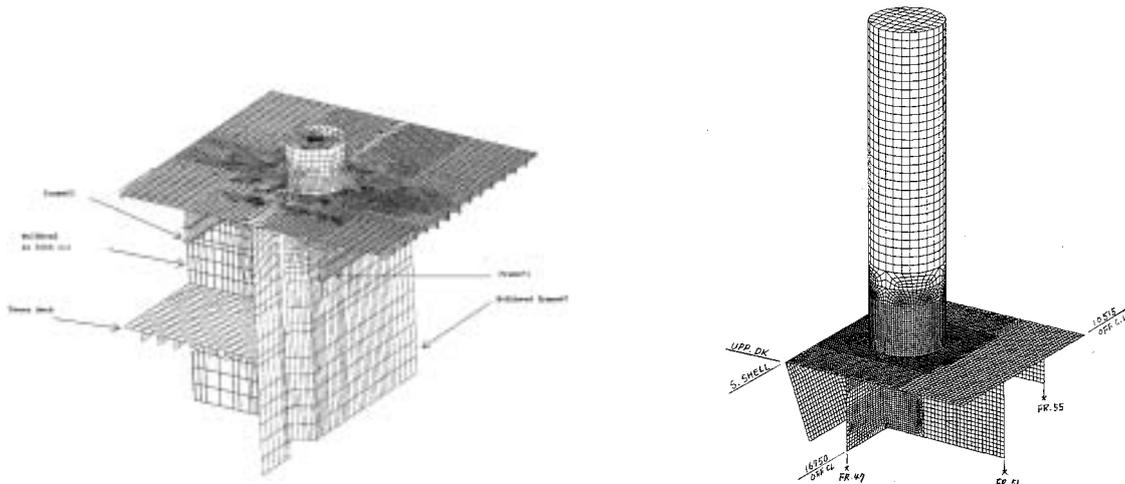
F2 → K4
F → K3
E → K2
D → K1
C → K0

8 Capacity assessment of supporting structure

8.1 Yield checks of crane pedestal support structure

Although dependent on the complexity of the support structure, an F.E.M analysis is normally applied to evaluate the strength of the support structure.

Examples of FEM-models:



Acceptance criteria as given in DNV Ship Rules Pt.3 Ch.3 Sec.5.

$$\sigma_p = \frac{\sigma_y}{1.5} \qquad \tau_p = \frac{\sigma_y}{(1.5\sqrt{3})}$$

Note that this criterion does not apply to absolute peak stresses, but to the overall stress level. Stresses are to be presented in the middle plane.

8.2 Buckling checks of crane pedestal support structure

The support structure shall comply with the requirements given in DNV-OS-C-102 for buckling. The capacity is evaluated based on Classification Notes 30.1 Sec.3. Global and local stresses are to be combined. The global bending moment may be reduced to daily value in case of maximum force acting on the crane.

8.3 Fatigue checks of crane pedestal support structure

Fatigue capacity to comply with the requirement of the project. In general a minimum fatigue life of 20 years is required with reference to main class.

The fatigue loads include hull deflections, inertia loads and functional loads. In most cases the functional loads will have little influence on the overall fatigue capacity of the structure. The procedure as given in Classification Notes 30.7 can be applied in such cases. Fatigue assessment is based on linear cumulative damage (Miner-Palmgren summation) to estimate the fatigue life.

In cases where the crane loads are large, and it operates at high loads at a significant part of the lifetime, then it may be necessary to evaluate the combined damage of functional loads, inertia loads and hull bending. Since the functional loads have less cycles than the bending and inertia loads, and are not coherent, the following simplified procedure can be followed:

Functional loads and hull bending/inertia loads are considered mutually exclusive. It is assumed full life in transit condition (maximum hull bending/inertia loading) with crane out of service, referred to as loadcase 1. Loadcase 2 is with the crane operating in calm seas. Damage ratios are then calculated separately for the two conditions.

Damage ratio D_1 for inertia/hull bending to be calculated according to procedure in CN.30.7.

$$D_1 = \frac{v_0 T_d p_s}{a} \sum_{n=1}^{N_{load}} p_n q_n^m \Gamma\left(1 + \frac{m}{h_n}\right) \leq \eta$$

Damage ratio due to functional loads is calculated based on NS5514 or FEM1.001 and Classification Notes.30.7.

Damage ratio is given as:

$$D_2 = \frac{n_H}{N_i}$$

where:

n_H = Number of hoisting cycles

N_i = Number of cycles obtained from S-N curves in relation to hot-spot stress

and

$$\log N_i = \log a - \frac{m \log(t/22)}{4} - m \log \Delta\sigma$$

a, m = are parameters for the relevant S-N curve taken from table 2.10/2.11
Classification Notes 30.7

$\Delta\sigma$ = is the hot-spot stress range

Total damage $D_T = D_1 + D_2$

For further description of symbols, see Classification Notes 30.7.

Please again note the importance of applying the correct SCF's and dynamic factors, and the effect of misalignments/eccentricities.

9 Determination of crane group

Crane group is normally specified by crane manufacturer and is a function of the number of load cycles and the load spectrum.

The following is an extract from NS5514, showing how the a crane group is identified. This can be employed if the crane group is not given by crane manufacturer.

Table 1.1 Class of utilization

| Class of utilization | Frequency of operation | No. of load cycles |
|----------------------|---|--------------------|
| A | Irregular operation followed by long rest periods | 63000 |
| B | Regular operation at normal load level | 200000 |
| C | Continuous operation at normal load level | 630000 |
| D | Continuous operation at high load level | 2000000 |

Table T-1.122 State of Loading

| State of loading | Definition | Curve |
|-------------------|---|-----------|
| 0 (very light) | Component rarely exposed to maximum stresses. Small stresses normal. | $p = 0$ |
| 1 (light) | Component seldom exposed to maximum stresses. Stresses at 1/3 of maximum is normal. | $p = 1/3$ |
| 2 (moderate) | Component often exposed to maximum stresses. Stresses between 1/3 and 2/3 of maximum is normal. | $p = 2/3$ |
| 3 (heavy) | Component regularly exposed to maximum stresses. | $p = 1$ |

Table T-1.13 Crane group

| State of loading | Class of Utilization | | | |
|-----------------------------|----------------------|---|---|---|
| | A | B | C | D |
| 0 (very light, $p = 0$) | 1 | 2 | 3 | 4 |
| 1 (light, $p = 1/3$) | 2 | 3 | 4 | 5 |
| 2 (moderate, $p = 2/3$) | 3 | 4 | 5 | 6 |
| 3 (heavy, $p = 1$) | 4 | 5 | 6 | 6 |

Typical group classification of lifting appliances as found in Section 3 of the Rules for Lifting Appliances:

| Definition of crane | Class of utilization | State of loading | Group |
|---|----------------------|------------------|-------|
| Cranes for exceptionally low service time, e.g. BOP crane | A | 0 | 1 |
| Workshops and similar industrial cranes with hook | B | 1 | 3 |
| Overhead travelling cranes for maintenance purpose | A | 1 | 2 |
| Pipe rack cranes | A | 1 | 2 |
| Store room cranes | A | 2 | 3 |
| Jib or gantry cranes for container service | B | 2 | 4 |
| Dock side and shipyard jib cranes | B | 2 | 4 |
| Shipboard cranes | B | 2 | 4 |
| Offshore cranes, whip hoist | B | 2 | 4 |
| Offshore cranes, main hoist | A | 1 | 2 |
| Hose handling cranes | A | 2 | 3 |
| Provision cranes | A | 2 | 3 |

A more extensive list can be found in NS5514, table T-1.15.

10 Examples of Stress plots

The following figures show stress distribution at typical stress concentration areas.

