ShipRight
Design and Construction

Structural Design Assessment

Global Design Loads of Container Ships and Other Ships Prone to Whipping and Springing

January 2018
<table>
<thead>
<tr>
<th>Date</th>
<th>Notes</th>
</tr>
</thead>
<tbody>
<tr>
<td>January 2018</td>
<td>New released.</td>
</tr>
</tbody>
</table>

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INTRODUCTION

Section 1: Introduction

This document provides guidance and recommendations on the assessment of non-linear effects and hull girder flexibility, so called whipping and springing actions, on the vertical wave bending moment and other global loads. This document also provides appropriate methods for the ultimate strength assessment including whipping effects and the fatigue assessment including springing effects.

The results of the wave induced non-linear ship motion analysis, the whipping and springing assessment may need to be applied to structural design assessment (SDA) and fatigue design assessment (FDA) procedures. Their application will need to be specially considered for ships that are sensitive to springing, have large variation in hull shape above and below the waterline or are subject to significant whipping loads. For these cases it may be necessary to review the structural integrity using response based analysis (RBA) or similar, i.e. direct application of dynamic loads from a ship motion analysis to the finite element analysis (FEA). The method of application of dynamic loads to FEA is to be discussed and agreed with Lloyd’s Register (LR) prior to commencement.

1.3. The ShipRight notations WDA1 (whipping design assessment level 1) or WDA2 (whipping design assessment level 2) are assigned for container ships when an appraisal has been made of the vibrational response of the hull structure due to whipping. The WDA2 notation may be voluntarily applied to other ship types or may be mandated by LR for specific cases.

1.4. The ShipRight notation FDA SPR (springing fatigue assessment) is assigned for container ships when an appraisal has been made of the fatigue performance of the hull structure taking into account the effects due to springing. The FDA SPR notation may be voluntarily applied to other ship types or may be mandated by LR for specific cases.

1.5. The proposed methods that the designer intends to adopt are to be discussed and agreed with LR prior to commencement of the assessment. Detailed reports of the assessment are to be submitted to LR for design review.
Section 2: Application

2.1. General

2.1.1. This document is applicable to ultra large container ships, ultra large ore carriers, ultra large bulk carriers as well as other ships which are likely to be prone to whipping or springing due to high length/depth ($L/D$) ratios, high speed, high bow or stern flare.

2.1.2. This document is applicable to Great Lakers while operating outside of limits of the Great Lakes and River St. Lawrence.

2.1.3. This document is applicable to the assessment of the vertical wave bending moment (and other global loads induced by waves) arising from non-linear effects of hull form shape including whipping due to slamming and operating worldwide in any sea condition. The document includes details on how to assess the hull girder strength against whipping induced hull girder loads.

2.1.4. This document is applicable to assess the dynamic responses of the hull girder vibration (springing) induced by waves.

2.1.5. This document is also applicable to other hull girder loads such as the wave torsional moment and horizontal wave bending moment; these issues are not explicitly covered in this document but the proposed methods are equally applicable.

2.1.6. The assessment of non-linear ship motion and wave induced loads, whipping effects or springing effects are to be in accordance with the Rule requirements. In addition, any of these may be performed voluntarily or may be required when LR considers it necessary.

2.2. Application to container ships

2.2.1. A summary of direct calculation analysis requirements for container ships is given in Table 1.2.1 Summary of direct calculation analysis requirements for container ships.
Table 1.2.1  Summary of direct calculation analysis requirements for container ships

<table>
<thead>
<tr>
<th>Rule requirement See Note 1</th>
<th>Rule reference</th>
<th>ShipRight notation</th>
<th>Application criteria</th>
<th>Length criteria</th>
<th>Breadth criteria</th>
</tr>
</thead>
<tbody>
<tr>
<td>LR’s ShipRight SDA Procedure for container ships</td>
<td>Pt 4, Ch 8, 1.3.3</td>
<td>SDA</td>
<td>Any of $</td>
<td>f_{IS}</td>
<td>&gt; 1.55$ or $f_{bow} &gt; 1.85$</td>
</tr>
<tr>
<td>Non-linear ship motion analysis to calculate combined vertical, horizontal and torsional loads</td>
<td>Pt 4, Ch 8, 14.1.2</td>
<td>—</td>
<td>$L &gt; 425$</td>
<td>—</td>
<td>—</td>
</tr>
<tr>
<td>Fatigue assessment</td>
<td>Pt 4, Ch 8, 14.3.2</td>
<td>FDA</td>
<td>$L &gt; 350$</td>
<td>$L &gt; 250$</td>
<td>Yes</td>
</tr>
<tr>
<td>Whipping assessment Level 1 See Note 4</td>
<td>Pt 4, Ch 8, 14.3.1</td>
<td>WDA1</td>
<td>—</td>
<td>—</td>
<td>—</td>
</tr>
<tr>
<td>Whipping assessment Level 2 See Note 4</td>
<td>Pt 4, Ch 8, 14.3.1</td>
<td>WDA2</td>
<td>$L &gt; 350$</td>
<td>$L &gt; 275$</td>
<td>—</td>
</tr>
<tr>
<td>Springing assessment See Note 2</td>
<td>Pt 4, Ch 8, 14.3.2</td>
<td>FDA SPR</td>
<td>$L &gt; 350$</td>
<td>$L &gt; 250$</td>
<td>Yes</td>
</tr>
</tbody>
</table>

Notes:

1. The stated Rule requirements may be deemed applicable to ships that do not meet the application criteria but where the structural configuration is such as to necessitate them.

2. The results of the springing assessment may also need a fatigue assessment procedure to be undertaken.

3. If ShipRight notation FDA is to be assigned, the requirements of LR’s ShipRight FDA Procedure are to be complied with; this may require calculations additional to those implied by Pt 4, Ch 8, 14.3 Procedures for verification of structural response due to whipping, springing and fatigue 14.3.2.

4. A Level 2 whipping assessment may be carried out instead of a Level 1 whipping assessment at the Owner’s request.

5. This is applicable only when a bottom steel grade greater or equal to HT36 is needed to satisfy the requirements of Pt 4, Ch 8, 16 Longitudinal strength calculations.
2.3. Application to other ship types

2.3.1. An analysis of the non-linear ship motion and wave loads may be required when one or more of the following criteria are applicable:

(a) \(|f_{s}| > 1.4\) or
(b) \(R_{A_{BF}} > 0.2\) or
(c) \(R_{A_{BFU}} > 0.2\) or
(d) \(C_{b} < 0.6\)

2.3.2. An analysis of the whipping responses may be required when one or more of the following criteria are applicable:

(a) \(L > 350\) m
(b) \(L > 275\) m and one or more of
   (i) \(|f_{s}| > 1.4\) or
   (ii) \(R_{A_{BF}} > 0.2\) or
   (iii) \(R_{A_{BFU}} > 0.2\) or
(c) \(L/D > 15\)
(d) \(C_{b} < 0.6\)
(e) \(C_{b} < 0.75\) and deep ballast draught forward \(< 0.025\) \(L\) or 7.5 m whichever is less

2.3.3. An analysis of the hull girder springing responses may be required when one or more of the following conditions apply:

(a) \(L > 250\) m and \(f_{c} > f_{sp}\)
(b) \(L > 190\) m and \(I \leq 0.8 I_{\text{min}}\)
(c) Use of HT47 or above for the deck or hatch side coaming
(d) Use of HT36 or above for the bottom shell for hull girder strength reasons

2.3.4. If the springing assessment shows that there is a significant springing response, then it may be necessary to undertake a more detailed fatigue analysis including springing effects. The fatigue assessment is to be carried out in accordance with the appropriate ShipRight FDA procedure with the results adjusted to take account of springing, see PART D.

2.3.5. LR should be consulted for further guidance and prior to commencement of any analysis.

**Section 3: Symbols**

- \(f_{s}\) is the sagging moment correction factor
  - For container ships, see the Ship Rules, Pt 4, Ch 8, 16.6
• For passenger and RoRo ships, see the Ship Rules Pt 4, Ch 2, 2.4
• For naval ships, see Naval Ship Rules Pt 5, Ch 4, 3.3
• For other ship types, see the Ship Rules Pt 3, Ch 4, 5.2

\( f_m \) is the hogging moment correction factor

• For container ships, see the Ship Rules, Pt 4, Ch 8, 16.6
• For passenger and RoRo ships, see the Ship Rules Pt 4, Ch 2, 2.4
• For naval ships, see Naval Ship Rules Pt 5, Ch 4, 3.3
• For other ship types, see the Ship Rules Pt 3, Ch 4, 5.2

\( RA_{BF} \) is the bow flare area ratio for the lower region just above the still waterline

\[
RA_{BF} = \left( \frac{A_{BF}}{0.1LB_{WL}} \right)
\]

\( RA_{BFU} \) is the bow flare area ratio for the upper region near the deck

\[
RA_{BFU} = \left( \frac{A_{BFU}}{0.1LB_{WL}} \right)
\]

\( A_{BF} \) is the bow flare area in m² for the region from the still waterline to a waterline of \( T_{C,U} \), see Pt 4, Ch 2, 2.4.3

\( A_{BFU} \) is the bow flare area in m² for the region from a waterline of \( T_{C,U} \) to \( T_{C,2U} \), calculated in the same way as \( A_{BF} \) see Pt 4, Ch 2, 2.4.3

\( B_{WL} \) is defined in Pt 4, Ch 2, 2.4.1

\( T_{C,2U} = T + C_1 \) m or the local deck edge height if this is lower

\( C_1 \) is defined in Table 4.5.1 in Pt 3, Ch 4, 5.2

\( f_{\text{bow}} \) is the bow flare shape coefficient for container ships and is to be taken as:

\[
f_{\text{bow}} = \frac{A_{DK} - A_{WL}}{0.2Lz_f}
\]

\( A_{DK} \), \( A_{WL} \) and \( z_f \) are defined in the Ship Rules, Pt 4, Ch 8, 16.6.3

\( f_{\text{bow,md}} \) is the bow flare shape coefficient for the main deck of container ships and is to be taken as:

\[
f_{\text{bow,md}} = \frac{A_{MD} - A_{WL}}{0.2Lz_{MD}}
\]

\( A_{MD} \) is the projected area in the horizontal plane of the main deck, in m². \( z_{MD} \) is the vertical distance, in m, from the waterline at draught \( T \), to the main deck, see Figure 1.3.1 Projected area \( A_{MD} \) and vertical distance \( z_{MD} \).
$f_c$ is a wave encounter frequency at which it is expected that whipping or springing will become important

$$f_c = \frac{1.3}{2\pi} \left( 1 + \frac{1.3V}{2g} \right) \text{Hz}$$

$f_{sp}$ is the natural frequency of the 2-node hull girder vertical bending mode in Hz. For ships of normal dimensions, this can be very approximately calculated as:

$$f_{sp} = \left( \frac{0.348}{\pi L^2} \right) \sqrt{\frac{E I}{f_{am} B T_d C_b}} \text{Hz}$$

$V$ = required operational speed in knots, usually this will be as defined in Pt 3, Ch 1, 6.1.10

$E$ = Young’s modulus in N/m²

$= 2.06 \times 10^{11}$ N/m² for steel

$I$ is the hull midship moment of inertia in m⁴, see Pt 3, Ch 4

$I_{min}$ is the minimum required hull midship section moment of inertia in m⁴, see Pt 3, Ch 4, 5.8

$LCB$ is the longitudinal centre of buoyancy forward of the AP, in metres.

$LCG$ is the longitudinal centre of gravity forward of the AP, in metres.

$T_d$ is the required operating draught, in metres.

$f_{am} = 1.8$ for container ships. For other ship types, then this value will be specially considered. Typically it will be in the range 1.8 to 2.1.

$C_b, L, B$ and $D$ are given in Pt 3, Ch 1, 6.1

$C_w$ is given in Pt 4, Ch 8, 16.2.
Section 4: Non-linear effects, whipping, springing and hull girder vibration

4.1. Non-linear effects

4.1.1. For large ships and ships with unusual hull form proportions, then the standard rule values of global hull girder loads are not 100 per cent reliable and direct calculation of these responses is required. As part of this assessment, the variation of the above and below water plane hull shape needs to be addressed. Hence a non-linear ship motion analysis is necessary to determine the appropriate design loads. This will provide revised hogging and sagging correction factors for application to the linear wave bending moments and shear forces. It also allows the effects of hull shape and non-linear wetted area to be taken into account in the calculation of the horizontal bending and torsional moments.

4.1.2. A time history of the vertical wave bending moment of a 6500 TEU container ship in a design wave from a flexible body model test is shown in Figure 4.1.1 Vertical wave bending moments showing the effects of non-linearity due to hull shape (model test). This illustrates the typical non-linear effects due to the hull shape and the high frequency vibrational response due to the flexibility of the hull girder.
4.2. Whipping phenomena

4.2.1. Whipping of a ship is the rapid flexing of the hull girder as a consequence of wave impacts on the hull. These high frequency oscillations of the hull girder result in increased vertical wave bending moments and shear forces. The dominant oscillation mode is the first (2-node) vertical vibration mode of the hull girder. Other hull girder vibrational modes may also be excited, such as the first (1-node) torsional mode or the coupled 1-node torsional and 2-node horizontal bending mode but they are not critical for the hull girder strength issue in severe sea conditions.

4.2.2. High whipping responses are usually driven by bow flare impacts due to large bow flare angles and high speed or by bottom slamming. Occasionally stern counter slamming can lead to high whipping responses. The high oscillations of the whipping response usually decay over several wave periods due to damping effects. Ships which have hull girder natural frequencies close to the wave energy region may experience prolonged springing phenomena after an impact event as there is little damping resistance close to the dominant hull girder vibration mode and the continuous wave excitation maintains the springing vibration.

4.2.3. The increase in vertical bending moments due to whipping is primarily a global strength issue. It is not usually a fatigue issue as the whipping induced vertical bending moment oscillations decay quickly and hence the number of cycles is small. A full scale measurement time history of vertical bending moment of an 8500 TEU container ship with a typical whipping response in waves is shown in Figure 1.4.2 Vertical wave bending moments showing the effects of whipping (full scale measurement).
4.3. Springing phenomena

4.3.1. Springing of a ship is the continual vibration of the hull girder as a consequence of the waves exciting resonant hull girder frequencies. This flexing of the hull girder due to springing may continue for several hours once initiated.

4.3.2. Springing is an issue for ships which have low natural vibration frequencies of bending or torsional modes, typically when the lowest natural frequency is less than 3 rad/s (≈ 0.5 Hz) and the ship speed is above 20 knots. This is the case for large container ships due to their high speed and open cross-sections. Great Lakers are also prone to springing due to their low moment of inertia and very high \( L/D \) ratio, which result in low natural frequencies of the hull girder vertical vibration modes. Full scale measurements have shown that large bulk carriers are also sensitive to springing.

4.3.3. Springing is not normally a strength issue as the magnitude of the springing responses and hence bending stresses is usually low, but it can have an important effect on the fatigue life of a structure as the number of cycles is relatively large (typically 4–8 times the number of wave cycles). Studies have shown that the fatigue life for the most sensitive details may be reduced by a factor of 30 per cent or more for container ships of around 6000 TEU and this will be more critical for larger ships. Hence it is necessary to review the effects of springing for large container ships and other ships with lower hull girder natural frequencies to ensure adequate fatigue life.

4.4. Hull girder natural vibration frequencies

4.4.1. All structural systems have numerous natural vibration frequencies and any cyclic load and its harmonics that are near to these natural frequencies will result in these natural vibration modes being excited. In addition, these natural vibration modes may be excited by impulsive loads such as bottom slamming and bow flare impacts which have a wide excitation range in the frequency spectrum.
4.4.2. For a ship with a full closed deck structure, e.g. a tanker, the 2-node vertical bending vibration mode, which has the lowest natural frequency, is the most important mode to be considered in the whipping/springing analysis. This vibration mode has a small damping ratio and is closest to the sensitive frequencies of waves in the real sea state hence this vibration mode is the one most often observed in service.

4.4.3. For ships with open deck structures, such as container ships and bulk carriers, the 1-node torsional mode may have a lower natural frequency than the 2-node vertical bending mode. Hence this torsional vibration mode can be important.

4.4.4. Other possible important vibration modes for these ships are the higher order (3-node) vertical bending and combined horizontal and torsional modes. These higher order modes have higher natural frequencies and high damping effects and are therefore rarely observed in full scale measurement.

4.4.5. The natural frequencies of the ship hull girder are governed by the ship length \((L)\), hull girder stiffness \((EI)\) and mass/added mass. For vertical bending modes, the natural frequencies are typically given by the following relationship:

\[
f \propto \frac{1}{L^2} \sqrt{\frac{EI}{M}} \quad \text{in Hz}
\]

4.4.6. Hence high ship length \((L)\) leads to lower natural vibration frequencies and increases the susceptibility to springing generally.

Section 5: Assessment procedures

5.1. The process to be followed for the assessment of non-linear design wave bending moment and other global loads due to non-linear effects and whipping response is outlined in Figure 1.5.1 Derivation of the non-linear and WDA2 whipping design loads.

5.2. Long term design linear wave induced loads

5.2.1. PART A provides the conventional method to determine the linear hull girder design loads along the ship length for ships. This involves calculating the ship motions and global loads for a range of ship loading conditions, speeds, heading angles and wave periods. The ship motions are calculated using a linear potential flow hydrodynamic program and statistical/probabilistic (short term and long term) results using the North Atlantic wave scatter diagram.

5.3. Non-linear ship motion wave induced loads

5.3.1. PART B provides a method to determine the non-linear hull girder design loads due to hull shape issues. The method provides revised non-linear hogging and sagging factors which can be applied in the rule assessment process.
5.3.2. A consistent method is provided that is based on calculating suitable equivalent design sea states (EDS) appropriate for severe sea states and undertaking non-linear time domain simulations for the ship motions and vertical wave bending moments. The method will also allow the calculation of the effects of non-linearities on vertical shear forces, horizontal bending and torsional moments. An equivalent design wave (EDW) approach is also possible, but it is recommended that the EDS approach is adopted. EDW can be employed based on LR’s agreement and the process has to be agreed with LR prior to commencement.

5.4. Hull girder loads including whipping loads

5.4.1. PART C provides methods to determine hull girder design loads taking account of non-linear issues, wave impact loads and hull girder flexibility. The methods provide a whipping enhancement factor for the rule vertical wave bending moment which can be applied in the ultimate strength assessment.

5.4.2. WDA1 uses parametric based formulae to predict whipping enhancement factors. These are given in PART C, Ch 1.

5.4.3. WDA2 uses an advanced direct calculation method to derive the whipping responses. The methodology is given in PART C, Ch 2. This is based on calculating a suitable EDS appropriate for severe sea states and calculation of the hull girder natural vibration modes. This is followed by the undertaking of non-linear time domain simulations using a hydroelastic ship motion code which can also calculate wave impact loads and hence the resulting vertical wave bending moments including whipping effects.

5.4.4. The ultimate strength assessment required to assess the whipping responses is given in PART C, Ch 3.
5.5. Fatigue including hull girder springing loads

5.5.1. PART D provides a method to evaluate the springing response of a ship’s hull girder for any sea state. The resulting response can then be used in a fatigue analysis to predict the fatigue life including springing.

5.5.2. The proposed method is based on assessment of springing in sea states appropriate for moderate wave environments where springing responses are expected to be significant and for sea states which are expected to be dominant for the predicted fatigue life.

5.5.3. PART D presents a consistent method which is frequency domain simulations based on a hydroelastic ship motion code to predict the springing responses. As the magnitudes of the springing responses are difficult to predict, the assessment of springing needs to be undertaken on the basis of a comparative assessment where the springing response of the new design is compared with the springing response of a known benchmark design.

Section 6: Documentation and plans to be submitted

6.1. Reports are to be submitted to LR for the approval of the hull girder structure with regard to whipping and springing and are to contain:

- list of plans used, including dates and versions;
- detailed description of structural modelling, including all modelling assumptions;
- plots to demonstrate correct structural modelling and assigned properties;
- details of material properties used;
- details of selected loading conditions;
- details and plots of the hydrodynamic model including mesh density, mass buoyancy balance parameters, calculated still water bending moment (SWBM) distribution;
- plots and tables of eigenvector results that demonstrate the correct behaviour of the structural model;
- details of the dry and wet natural frequencies for the critical vibration modes, demonstration that all the critical vibration modes have been captured;
- details of the linear ship motion analysis and the selection of the EDS;
- selected time domain traces of the linear, non-linear and whipping analyses;
- statistical analysis process and results;
- ultimate strength assessment and results;
- fatigue assessment procedure, assumptions and results;
- springing stress response amplitude operators (RAOs) or BMs plots to demonstrate the springing effects have been captured;
- demonstration that the structure complies, or otherwise, with the design criteria; and
• proposed amendments to the structure where necessary, including revised assessment of results.
PART A: LINEAR SHIP MOTION ANALYSIS

Chapter 1: Assessment of Linear Design Wave Loads

Section 1: General

1.1. In this PART, detailed requirements are given to derive the long-term design linear wave loads using a first principles direct calculation procedure.

Section 2: Loading conditions

2.1. Loading conditions for the non-linear and whipping assessments

2.1.1. For a conventional container ship which always has a large hogging SWBM, the following loading condition should be assessed for the strength assessment:

- scantling draught with container loadings that maximises the hogging SWBM amidships.

2.1.2. In some cases it may be necessary to also review the following loading conditions:

- maximum normal operational draft (usually the design draught) with container loadings that maximises the hogging SWBM amidships;
- normal ballast loading condition;

Section 10: Long term calculation
• typical container loading condition at a shallower draught that maximises the sagging SWBM amidships (or minimises the hogging SWBM).

2.1.3. For other ship types, the following loading conditions should be assessed:
• a loading condition with scantling draught that maximises the hogging SWBM;
• a loading condition that maximises the sagging SWBM;
• loading conditions with draughts that are likely to result in maximum bow flare impact pressures may also need to be reviewed.

2.1.4. It is not necessary for the permissible SWBM to be matched in any of the above loading conditions if it is not realistically possible to achieve the permissible value at the applicable draught.

2.1.5. Please consult LR with details of the required operational profile and loading conditions that are applicable to assess whether additional loading conditions need to be evaluated.

2.2. Loading conditions for the springing assessment

2.2.1. The loading condition to be used for the springing fatigue assessment is a typical full loading condition at the design draught.

2.2.2. In some cases it may be necessary to also review a normal ballast loading condition or a typical container loading condition at a shallower draught depending on the expected operational profile of the ship.

2.2.3. For other ship types, then a typical full load condition and a lightly loaded or ballast condition are to be reviewed. The loading conditions to be chosen are to be representative of those most likely to be used during everyday operation.

Section 3: Hydrodynamic modelling

3.1. Wave loads and the ship’s motion responses to unit regular waves are to be calculated using a linear potential flow hydrodynamic program.

3.2. The linear ship motion analysis is fully supported by LR’s software tools as follows:
• HydroModeller Creates the hydrodynamic mesh from offsets, FE models, STL (Standard Triangle Language data format) models or similar;
• WAVELOAD-FD Linear rigid body frequency domain 3D diffraction ship motion code.

3.3. The mesh size should be based on the highest wave encounter frequency considered. Ideally, the length of the panels should be such that one obtains at least five panels per wave length for the highest wave encounter frequency \( \omega_{\text{max}} \), i.e.

\[
L_{\text{panel}} \leq \frac{2\pi g}{5\omega_{\text{max}}} \]

3.4. Table 1.3.1 Recommended size of hydrodynamic panels for whipping and springing ship motion analyses gives the minimum recommended panel size for various ship speeds and maximum frequencies to be assessed. This illustrates the severe problem with satisfying the optimum panel size criteria. Hence consideration will be given to relaxing this requirement for very large hull forms at high speed, but sensitivity studies on panel size will be necessary to confirm that the results are converged.
Table 1.3.1 Recommended size of hydrodynamic panels for whipping and springing ship motion analyses

<table>
<thead>
<tr>
<th>Speed knots (V)</th>
<th>Maximum wave frequency $\omega_{\text{max}}$ to be assessed in rad/s, see Note 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>5</td>
<td>1,4 rad/s</td>
</tr>
<tr>
<td>15</td>
<td>1,8 rad/s</td>
</tr>
<tr>
<td>20</td>
<td>2,2 rad/s and above</td>
</tr>
<tr>
<td>25</td>
<td>3,4 m (see Note 1)</td>
</tr>
</tbody>
</table>

Notes:

1. Due to current computer hardware limits, the optimum size for the hydrodynamic panels will not be achievable for large container ships. Hence a panel size of 1 m is acceptable.

2. For container ships, the maximum wave frequency $\omega_{\text{max}}$ to be assessed should be greater than the wave frequency that corresponds to encounter wave frequency $\omega_e = \omega_n$, where $\omega_n$ is the 2-node vertical bending vibration frequency. The minimum value of this wave frequency is given by:

$$\omega_{\text{max}} = \frac{-2g}{V} \left( \frac{2g}{V} \right)^2 + \frac{4}{2} \left( \frac{2g}{V} \omega_n \right)$$

3. For other ship types, when selecting the maximum frequency other vibrational modes may need to be considered.

4. It is not normally necessary to consider a maximum wave frequency greater than 2.5 rad/s due to the very low wave energy content above this frequency.

Section 4: Verification of hydrodynamic models

4.1. Hydrostatic calculations based on the hydrodynamic mesh should be performed for each of the required loading conditions to validate the hydrodynamic model(s).

4.2. The following hydrostatic parameters are to be calculated and verified against the values given in the ship’s loading manual or calculations obtained from the ship’s loading computer software. The following tolerances are to be met:

- Displacement $\pm$ 0.5 per cent but ideally this should be less than 0.15 per cent
- LCB – LCG $\pm$ 0.1 m but ideally this should be less than $\pm$ 0.025 m
- Draught (forward; aft) $\pm$ 0.02 m
- SWBM $\pm$ 2 per cent

Notes:

It is advisable to ensure the difference between the summations of the mass distribution and the hydrodynamic buoyancy is negligible. If the ship motion software does adjust the mass or buoyancy to achieve a perfect balance, then the user should understand that this could result in a large error in the hydrodynamic results.

The methods of ensuring a balance between the hydrodynamic mesh, in preferred order, are as follows:

1. Adjust the hull form or draft and trim to achieve the mass and LCG.
2. Adjust the mass distribution to achieve the hydrodynamic displacement and LCB.
3. Scale the mass and move all mass items to achieve the hydrodynamic displacement and LCB.
Section 5: Roll damping coefficient

5.1. Viscous roll damping has a great influence on the ship’s roll motion at near roll resonance frequency. The viscous damping effect is to be considered in the hydrodynamic analysis as follows:

- The non-linear roll damping effect is to be considered in critical sea conditions where the encountered wave frequencies are close to the roll resonance frequency.
- The effects of bilge keel and other appendages are to be considered.
- The effects of active roll stabilisers or roll damping tanks are to be ignored.

5.2. The values of roll damping factors are to be suitable for the analysis being undertaken. For the whipping analysis due to hull girder vertical bending, roll damping is not an important issue. For fatigue analysis, the selected roll damping value is to be applicable for moderate sea states.

Section 6: Ship operational conditions

6.1. The ship speed is to be adjusted according to the significant wave height. For container ships the speed versus wave height relationship is to be assumed as given in Table 1.6.1 Ship speed versus wave height relationship for container ships.

<table>
<thead>
<tr>
<th>Significant wave height ($H_s$)</th>
<th>Ship speed</th>
</tr>
</thead>
<tbody>
<tr>
<td>$H_s \geq 10.5$ m</td>
<td>25% $V_s$ or 5 knots</td>
</tr>
<tr>
<td>10.5 m &gt; $H_s \geq 7.5$ m</td>
<td>50% $V_s$</td>
</tr>
<tr>
<td>7.5 m &gt; $H_s \geq 4.5$ m</td>
<td>75% $V_s$</td>
</tr>
<tr>
<td>$H_s &lt; 4.5$ m</td>
<td>$V_s$</td>
</tr>
</tbody>
</table>

Notes:
1. $V_s$ is a service speed.
2. The service speed to be used is taken as 90 per cent of the maximum service speed for a springing analysis based on a worldwide trade pattern or as specified in the FDA procedure.
3. 5 knots is to be used for the whipping analysis.

6.2. A different relationship may need to be applied if the ship is of unusual form or has a very high installed power or if specified by the owner/designer.

6.3. The speed/wave height relationship table is to be applied for the selection of the appropriate ship speed for the non-linear, whipping and springing assessments.

Section 7: Ship motion and load responses in frequency domain

7.1. Ship’s motion and load responses to sinusoidal waves are to be calculated to determine the RAOs and the associated phase angles for each of the ship’s loading conditions specified.

7.2. The RAOs are to be calculated for the following parameters specified in Table 1.7.1 Ship motion analysis parameters.
Table 1.7.1 Ship motion analysis parameters

<table>
<thead>
<tr>
<th>Comment</th>
</tr>
</thead>
<tbody>
<tr>
<td>Loading conditions</td>
</tr>
<tr>
<td>As specified in Pt A, Ch 1, Section 2 Loading conditions</td>
</tr>
<tr>
<td>Speed</td>
</tr>
<tr>
<td>As specified in Pt A, Ch 1, Section 6 Ship operational conditions</td>
</tr>
<tr>
<td>Wave/ship heading</td>
</tr>
<tr>
<td>Following seas ($\theta = 0^\circ$) to head seas ($\theta = 180^\circ$)</td>
</tr>
<tr>
<td>depending on the source of wave environment data. In general, at 15°</td>
</tr>
<tr>
<td>increments.</td>
</tr>
<tr>
<td>Wave frequency</td>
</tr>
<tr>
<td>Upper limit of frequency range depends on the maximum ship's speed and</td>
</tr>
<tr>
<td>ship's length and wave environment.</td>
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<tr>
<td>In general, 0.2-1.5 rad/s at a maximum increment of 0.05 rad/s.</td>
</tr>
</tbody>
</table>

7.3. For each ship loading condition, the following wave load responses are to be calculated:

- VWBM, VWSF, HWBM, HWSF from 0.1L to 0.9L in 0.1L increments;
- Torsion moment about shear centre from 0.1L to 0.9L in 0.1L increments.

7.4. The position of the shear centre will need to be specially considered. The geometric 2D shear centre is only an approximation and a more representative shear centre that takes account of 3D effects should be considered. This could be derived after assessment of the torsional responses using a whole ship 3D FE or LRPass 20204 or similar.

Section 8: Wave environment

8.1. The standard wave environment to be used for the assessment of the long-term loads acting on ships which are operating worldwide is the North Atlantic all-directions scatter diagram as specified in IACS Rec. No. 34, see Table 1.8.1 IACS Rec.34 – North Atlantic wave environment scatter diagram. This wave environment specifies the probability of occurrence of individual sea states, each sea state being defined by a significant wave height ($H_s$) and a zero up-crossing period ($T_z$). Each sea state is to be characterised using the modified Pierson–Moskowitz wave energy spectrum with an assumed duration of 3 hours.

8.2. The modified Pierson–Moskowitz wave energy spectrum, also called the ISSC spectrum, is to be used to model the wave environment in the North Atlantic or any open ocean area. The assumption of short crested seas with a cosine squared spreading function should be applied.

8.3. The wave environment for a restricted service requirement will be subject to special consideration.

8.4. The wave environment for springing fatigue assessment is to be derived in accordance with the FDA procedure including adoption of the appropriate ship speed to wave height relationship.
Table 1.8.1 IACS Rec.34 – North Atlantic wave environment scatter diagram

<table>
<thead>
<tr>
<th>$H_s/T_z$</th>
<th>3.5</th>
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</table>

Note: Total 100,000 wave observations.

Section 9: Linear design wave bending moment

9.1. If required, the linear long term vertical wave bending moment, $VBM_{LT}$, is to be calculated by the long-term analysis method using the wave scatter data described in Pt A, Ch 1, Section 8 Wave environment with a $10^{-8}$ probability of exceedance, which is considered equivalent to a return period of 25 years.

9.2. For container ships, the standard Rule value of the linear design vertical wave bending moment $M_{Linear}$ is to be taken as:

$$M_{Linear} = 1.5 f_i L^3 C_3 C_w \left( \frac{B}{L} \right)^{0.8}$$

where, $L$ and $B$ are given in the Ship Rules, Pt 3, Ch 1, 6.1

$f_i$ is given in the Ship Rules, Pt 4, Ch 8, 16.6

$C_3$ is given in the Ship Rules, Pt 4, Ch 8, 16.6

$C_w$ is given in the Ship Rules, Pt 4, Ch 8, 16.2.

9.3. For other ship types, the standard Rule value of the linear design wave bending moment $M_{Linear}$ is to be taken as:

$$M_{Linear} = M_{wo}$$

where, $M_{wo}$ is given in the Ship Rules, Pt 3, Ch 4, 5.2.1.
Section 10: Long-term calculation

10.1. The long-term approach is an accumulation of the statistics for all short-term sea conditions taking into account the frequency of occurrence of each short-term sea state. When a ship travels at constant speed, the long-term probability of exceedance \( \bar{Q}(x > X_0) \) for \( x > X_0 \) is

\[
\bar{Q}(x > X_0) = \int_{0}^{\infty} \int_{0}^{360} \int [Q(x > \zeta|H_s, T_z)] f(H_s, T_z) \, dH_s \, dT_z \, d\theta
\]

where,

\( f(H_s, T_z) \) is the joint probability density function of significant wave height, \( H_s \), and average zero-crossing wave period, \( T_z \), as given by the wave scatter diagram of IACS Rec.34, see Table 1.8.1 IACS Rec.34 – North Atlantic wave environment scatter diagram. Note the IACS standard assumption is to assume equal probability of heading.

\( Q(x > X_0|H_s, T_z) \) is the joint probability of exceedance for \( x > X_0 \) in a single short-term sea state with significant wave height and averaged zero-crossing wave period \( (H_s, T_z) \).

\( \theta \) is the heading angle.

10.2. It is generally assumed that the short-term response is a narrow band random process (with bandwidth parameter \( \epsilon = 0 \)) and its probability of exceedance follows the Rayleigh distribution, i.e.

\[
Q(x > X_0|H_s, T_z) = \exp \left\{ -\frac{1}{2} \left( \frac{X_0}{m_0} \right)^2 \right\}
\]

where,

\( m_0 \) is the response variance of the single sea state from short-term analysis in short crested waves.

10.3. Alternatively the long-term values may be derived considering suitable bandwidth correction factors.
PART B:
NON-LINEAR SHIP MOTION ANALYSIS

Chapter 1:
Assessment of non-linear ship motion wave loads

Section 1: General

1.1 In this Chapter, requirements are given to derive the design wave bending moment along the ship length to cover non-linear effects induced by hull shape using a first principles direct calculation procedure. Please note that the assessment of non-linear wave loads is not required for the assessment of the WDA notations and the FDA SPR notation.

1.2 The non-linear ship motion analysis is supported by LR's software tools as follows:

- HydroModeller Creates the hydrodynamic mesh from offsets, FE models, STL models or similar;
- HydroE-TD Non-linear rigid/flexible body time domain 3D diffraction ship motion code.

1.3 The process to derive the non-linear response of vertical shear force, horizontal bending moment and shear force and torsional moment is the same as that used to derive the non-linear vertical bending moment response.

1.4 The procedure derives the design wave bending moment based on the Equivalent Design Sea states (EDS) concept. The Equivalent Design Wave (EDW) method is useful for initial checking and review, but is not recommended as it is harder to achieve consistent results.

1.5 The design values are then given by applying this EDS to a non-linear time domain ship motion analysis. It should be noted that different longitudinal locations and different hull girder responses may well require different EDSs.

1.6 The procedure to derive the non-linear response for other load values such as horizontal bending moments and torsional moments is similar.

1.7 Figure 1.1.1 Vertical bending moment amidships, \( \lambda = 0.91Lpp \) shows the calculated time domain responses to a regular wave of length 0.91L for the following:
- linear wave bending moment response, curve labelled “linear”
- non-linear ship motion response due to hull shape and actual wave profile, curve labelled “non-linear”
- vertical bending moment including non-linear responses and whipping actions, curve labelled “whip”
- For this example, the effect of non-linearities and whipping actions have little effect on the hogging bending moments (positive values) but result in significantly increased sagging bending moments.

![Figure 1.1.1](image)

**Section 2: Determination of the equivalent design sea state (EDS)**

2.1. The EDS is defined as the sea state which has the maximum contribution to the $10^{-8}$ probability of exceedance long term (LT) vertical wave bending moment amidships. The LT value is calculated in accordance with PART A.

2.2. The basic steps to determine the EDS are to be follow:

(1) carry out the long-term statistical analysis for the vertical wave bending moment amidships, determine the specific vertical wave bending moment amplitude $X_m$ at a probability of exceedance $\widetilde{Q}(x > X_m)$ equal to $10^{-8}$, see Pt A, Ch 1, Section 10 Long term calculation.

(2) calculate the probability of exceedance $Q(x > X_m|H_s, T_1)f(H_s, T_1, \theta)$ for every single sea state of the wave scatter diagram based on the specific VBM value $X_m$. For vertical wave bending moments, then it is adequate to only consider the head sea heading.

(3) the EDS is the sea state with maximum probability value; see an example in Table 1.2.1 Example of individual probability contribution of each sea states to the long term $10^{-8}$ value. For this case, the
EDS is \( H_s = 11.5 \) m and \( T_z = 9.5 \) s, as \( H_s \) is greater than 10 m, see Pt A, Ch 1, Table 1.6.1 Ship speed versus wave height relationship for container ships, then the ship speed is to be taken as 5 knots.

(4) if the freeboard height at amidships is less than 67 per cent of \( H_s \) (calculated significant wave height of EDS) or the draught at amidships is less than 50 per cent of \( H_s \), special consideration of the EDS wave height may be necessary. Please consult LR for more guidance.

2.3. The calculation of the non-linear responses is then undertaken using a non-linear rigid body time domain ship motion analysis with this irregular EDS.

### Table 1.2.1 Example of individual probability contribution of each sea states to the long term \( 10^{-8} \) value

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<th>( H_s )</th>
<th>( T_z )</th>
<th>7.5</th>
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### Section 3: Computation of vertical wave bending moments in time domain

3.1. The predicted vertical wave bending moments are to be calculated using a non-linear time domain rigid body hydrodynamic program with the capabilities to calculate the hydrostatic forces and wave incident forces (Froude–Krylov forces) taking into account the change in hull shape above and below the waterline.

3.2. The time domain simulation for the non-linear response calculation should be run for at least 300 minutes. This 300 minutes is recommended to be split into several segments, e.g. 5 runs of 60 minutes each. It is to be verified that the data obtained is sufficient to obtain a statistically converged set of data that is suitable for prediction of the probable maximum value in 3 hours. A ramp up period of around 25 seconds should be specified and the results for the first 50 seconds should be ignored in the post processing in order to allow the response to stabilise. The time interval in the time domain simulation should not be greater than one tenth of the lowest period of the encounter waves.

3.3. The following approach is recommended:

- Run a linear rigid body time domain analysis.
- Run a non-linear rigid body time domain analysis.

3.4. The EDS approach allows the statistical analysis for the linear and non-linear responses (and also the whipping responses) to be calculated based on exactly the same wave data. This approach gives the same number of hogging and sagging events and ensures a consistency of approach with regard to the statistical analysis. The results displayed in Pt C, Ch 2, Figure 2.8.2 Weibull fitting of the linear, non-linear and whipping vertical sagging bending moment distributions and Pt C, Ch 2, Figure 2.8.3 Weibull fitting of the linear, non-linear and whipping vertical hogging bending moment distributions are based on this approach.
3.5. Wave components of the EDS approach

3.5.1. The amplitude of the each wave component making up the selected EDS is to be calculated using the spectral energy of the EDS. The wave amplitude at each wave frequency $\zeta(\omega)$ is given by:

$$\zeta(\omega) = \sqrt{2S(\omega)\delta \omega}$$

where,

$S(\omega)$ is the spectral value of EDS.

$\delta \omega$ is the wave frequency interval. The first and last wave frequencies are to be chosen such that the following relationship is satisfied.

$$S(\omega) \geq 0.001 \times S_{\max}$$

$S_{\max}$ is the maximum value of $S(\omega)$.

3.5.2. The time history of the wave elevation $\zeta$ at location $x_w$ is defined as:

$$\zeta(x_w, t) = \sum_{i=1}^{N_w} \zeta_\omega \cos(k_i x_w - \omega_i t + \epsilon_i)$$

where,

$k_i = \frac{\omega_i^2}{g}$ is the wave number.

$\epsilon_i$ is the phase shift of each wave component. This is to be randomly assigned. Note it is suggested that some pre-selected sets of random phase angles are used so that repetition of the analysis is possible with the same random phase shifts and hence the same wave elevation time trace.

$N_w$ is the number of wave components. $N_w$ should be at least 150 for a one hour time domain simulation run. The number of wave components is to be sufficient to ensure that the time trace of the wave elevation does not repeat or show similar repetitive patterns throughout the duration of the simulation.

Section 4: Derivation of hogging and sagging non-linear factors for the vertical wave bending moment

4.1. The probability distributions of the hogging and sagging bending moment distributions of the time domain simulation are derived using a peak counting method after subtraction of the SWBM. Both peaks and troughs (corresponding to the hogging and sagging wave bending moments, based on the usual sign convention of the vertical bending moment) of the time domain signal should be counted separately as one peak or trough per mean zero crossing period, see Pt C, Ch 2, Figure 2.8.1 Counting method for extremes of the linear, non-linear and whipping response time domain traces.
4.2. The probable maximum hogging and sagging bending moments in three hours are to be obtained by fitting a 3 parameter Weibull distribution. The method to do this is described in *Introduction, Section 6 Documentation and plans to be submitted*.

4.3. The final non-linear hog and sag ratios are derived as follows:

\[ f_{3h}^S = \frac{M_{3h}^{VS}}{M_{3h}^{VL}} \]

for sagging, but is not to be taken less than the standard rule value, see *Introduction, Section 3 Symbols*.

\[ f_{3h}^H = \frac{M_{3h}^{VH}}{M_{3h}^{VL}} \]

for hogging, but is not to be taken less than the standard rule value, see *Introduction, Section 3 Symbols*.

where,

- \( M_{3h}^{VS} \) is the “probable maximum sagging bending moment in 3 hours”.
- \( M_{3h}^{VH} \) is the “probable maximum hogging bending moment in 3 hours”.
- \( M_{3h}^{VL} \) is the “probable maximum linear bending moments in 3 hours”. This can be derived from short-term statistical analysis using the EDS or from a linear time domain simulation using the same wave elevation time trace as the non-linear time simulation but ignoring the non-linear hydrostatic restoring forces and wave incident forces due to the hull shape.
PART C: WHIPPING ASSESSMENT

Chapter 1: WDA1 – Assessment of hull girder loads including whipping

Section 1: General

1.1. In this Chapter, requirements are given to assess the global wave bending moment including whipping effects for the WDA1 notation. The hogging and sagging whipping enhancement factors to be applied to the design wave bending moment are derived using parametric equations.

1.2. The application of the WDA1 notation is primarily for container ships; see Introduction, Table 1.2.1 Summary of direct calculation analysis requirements for container ships. It may also be applied on a voluntary basis to other ship types as specified in the Introduction, 1.3 and after consultation with LR as to the applicability of the parametric WDA1 equations.

1.3. The WDA1 parametric equations may be used for the initial design assessment of container ships where the WDA2 notation is mandatory. However the final assessment has to use the global wave bending moment including whipping effects derived in accordance with the WDA2 requirements.

1.4. The parametric WDA1 equations are validated for the following range of container ships:

- $200 \text{ m} < L < 400 \text{ m}$
- $24 \text{ m} < B < 60 \text{ m}$
- $0,6 < C_b < 0,75$
- $1,5 < f_{\text{bow}} < 2,5$

For ships with a $C_b < 0,6$ and $f_{\text{bow}} > 1,5$, then WDA2 will need to be applied.

For ships with a $f_{\text{bow}} > 2,5$, then WDA2 will need to be applied.

For ships outside of this range, then LR may require a WDA2 analysis to be undertaken.
Section 2: Loading condition

2.1. The loading conditions to apply are specified in Pt A, Ch 1, 2.1.

Section 3: Derivation of hogging and sagging whipping factors for the vertical wave bending moment

3.1. The whipping enhancement factors amidships are to be taken as follows:

\[ f_{fS-W} = -5.7 \left(1 + 0.2 \frac{f_{\text{bow,md}}}{L^{0.3}} - \frac{325[0.4 + f_{\text{bow,md}}^2]}{L^{1.9}}(A_{MD} - A_{WL})^{0.5} \right) \]

\[ f_{fH-W} = 0.27 \frac{C_b}{C_w} \sqrt{T} + \frac{300[0.4 + f_{\text{bow,md}}^2]}{L^{1.9}}(A_{MD} - A_{WL})^{0.5} \exp\left(-0.4 \pi \left(\frac{\omega_n}{2 \omega_c} + 0.5\right)\right) \]

where,

- \( f_{\text{bow,md}} \) is the bow flare shape coefficient for the main deck of container ships, see Introduction, Section 3 Symbols.
- \( \omega_n \) is the natural frequency of the 1st (2-node) vertical bending mode of the hull girder, if not available, can be estimated as \( \omega_n = 2 \pi f_{sp} \).
- \( E \) is Young’s modulus in N/m²
- \( \omega_c = 8.06 L^{-0.5} + 17.03 L^{-1} \) is the critical wave frequency in rad/s

where, variables are as defined in the Introduction, Section 3 Symbols.

Section 4: Whipping vertical wave bending moments

4.1. The hogging and sagging whipping vertical wave bending moments including the effects of non-linear hull shape and whipping actions along the length of the ship are to be taken as follows:

\[ VBM_{\text{WH-S}} = f_{fS-W} f_{\text{WDA-1}} M_{\text{Linear}} \] for sagging

\[ VBM_{\text{WH-H}} = f_{fH-W} f_{\text{WDA-1}} M_{\text{Linear}} \] for hogging

where,

- \( M_{\text{Linear}} \) is the linear design wave bending moment, see Pt A, Ch 1, Section 9 Linear design wave bending moment.
- \( f_{\text{WDA-1}} \) is the longitudinal distribution factor, see Figure 1.4.1 Longitudinal distribution factors, \( f_{\text{WDA-1}} \).
Figure 1.4.1
Longitudinal distribution factors, $f_{WDA-1}$
PART C: WHIPPING ASSESSMENT

Chapter 2: WDA2 – Assessment of hull girder loads including whipping

Section 1: General

In this Chapter, requirements are given to assess the global wave bending moment including whipping effects for the WDA2 notation.

1.1. The application of the WDA2 notation is primarily for container ships, see Introduction, Table 1.2.1 Summary of direct calculation analysis requirements for container ships. It may also be applied on a voluntary basis to container ships when a WDA1 notation is required or to other ship types, see also Introduction, 1.3.

1.2. For the WDA2 notation, the hogging and sagging whipping enhancement factors to be applied to the design wave bending moment are derived by employing a first principles direct calculation procedure. It is also necessary to derive the longitudinal distribution of the wave induced bending moment including whipping effects in order to ensure that the hull girder strength of the ship is valid along its length.

1.3. If required, the same process can be applied to derive the whipping response of vertical shear force, horizontal bending moment and shear force, and torsional moment.

1.4. The procedure derives the design wave bending moment based on the Equivalent Design Sea states (EDS) and a non-linear time domain hydroelastic ship motion analysis.

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1.6. The derivation of the bending moments including whipping is supported by LR's software tools as follows:

- **Trident or Nastran** FE package to carry out eigenvalue analysis;
- **HydroModeller** Creates the hydrodynamic mesh from offsets, FE models, STL models or similar. Applies the eigenvector modal shapes to the hydrodynamic mesh;
- **HydroE-TD** Non-linear flexible body time domain 3D diffraction hydroelastic ship motion code;
- **Weibull Fit** Spreadsheet to derive non-linear and whipping hogging/sagging factors.

1.7. *Figure 2.1.1 Typical time series of VBM amidships* shows part of a time trace from full scale measurements of an 8500 TEU container ship and the time trace from a whipping calculation after a whipping impact occurred. Note that the time series of the wave elevation for the measurements was not known and hence a time trace for the calculation had to be assumed. In view of this it is only possible to compare the trends that the two motions traces display; the statistical results from the two time traces can also be compared.

![Time series of VBM amidships](image)

**Figure 2.1.1**

**Typical time series of VBM amidships**

**Section 2: Derivation of the EDSs**

2.1. The EDS derived in *Pt B, Ch 1, Section 2 Determination of the equivalent design sea state (EDS)* is to be used to calculate the whipping responses and hence the whipping hogging and sagging factors for the design vertical wave bending moment.

2.2. Whipping is more susceptible to draft and speed that the extreme wave bending moment and hence it may be necessary to consider other EDSs to account for different loading conditions and speeds to ensure that the maximum dynamic bending moments including whipping have been derived.

2.3. The derivation of the wave components of the EDS approach is given in *Pt B, Ch 1, 3.5.*
Section 3: Loading condition

3.1. The loading conditions to apply are specified in Pt A, Ch 1, 2.1.

Section 4: Hydrodynamic modelling

4.1. The mesh size, model verification and loading conditions for the hydrodynamic model are specified in Pt A, Ch 1, Section 3 Hydrodynamic modelling to Pt A, Ch 1, Section 6 Ship operational conditions.

Section 5: Structural model of the ship

5.1. The structural flexibility should be taken into account for the derivation of the whipping enhancement factor of the design wave bending moment by adding the flexible natural modes of ship hull structure to the rigid body modes. These flexible natural modes are computed using the FE method based on either a 2D model (beam model) or a 3D model of the ship structure. At least the first 5 modes of hull girder vertical vibration should be taken into account in the analysis.

5.2. 2D structural modelling

5.2.1. While a 2D FE model is adopted in the modal analysis of ship hull structure, the hull girder should be modelled as a non-uniform beam according to the so called Timoshenko beam theory and should be modelled using at least 50 and preferably more than 100 beam elements.

5.2.2. The total length of the beam model should match the total length of the ship and the sectional properties will need to be calculated at the ends to ensure that the hull girder flexibility is correctly modelled.

5.2.3. If a 3D FE model is available, but is not being used for the whipping/springing analysis, then the principal hull girder bending dry mode eigenvectors of the 2D model should give similar results to the 3D dry mode eigenvectors.

5.2.4. The calculation of the hull girder inertia and section modulus is to be done in accordance with Pt 3, Ch 3, 3.4 of the Rules of Ships, which defines the material that is longitudinally effective and the shadow areas. It is to be assumed that all hull girder section properties transform smoothly along the length of the ship and hence there are no abrupt discontinuities in the properties due to changes in cross-sections between closed sections and sections with large deck openings. This applies to all hull girder properties including the shear centre for torsional moment.

5.3. 3D structural modelling

5.3.1. When a 3D FE model is adopted for the modal analysis of ship, the ship's structure should be modelled using shell elements and bar elements to represent the whole ship. This model should extend over the full breadth and depth of the ship and represent, with reasonable accuracy, the actual geometric shape of the hull.

5.3.2. All primary longitudinal structures (i.e. all longitudinal bulkheads, decks, stringers) are to be modelled using shell elements. Similarly, all transverse primary structures (i.e. watertight bulkheads, open bulkheads (mid-hold support structure), web frames and cross-deck structures) are to be represented in the model.

5.3.3. The size and type of shell elements selected are to provide a satisfactory representation of the deflections and stress distributions within the ship's structure. As a minimum, the mesh of the shell element is to follow the primary stiffening arrangement. Hence, it is anticipated that there will be:
• transversely, one element between longitudinal girders;
• longitudinally, one element between double bottom floors; and
• vertically, one element between stringers or decks.

5.3.4. The ship’s superstructure or deckhouse is to be included in the model. This is to be represented using shell elements with a mesh arrangement similar to that used for the hull and which adequately represent the overall structural arrangement of the deckhouse.

5.3.5. Secondary stiffening members and effective longitudinal members may be modelled using line (bar) elements, grouped at plate boundaries, positioned in the plane of the plating having an axial property with the cross-sectional area representing the stiffener area.

5.4. Eigenvector analysis techniques

5.4.1. Special care is to be taken to ensure that the structural idealisation of the FE model does not introduce vibrational responses (natural modes) which are not realistic or not applicable for the assessment of global hull responses. In some cases, additional artificial constraints may need to be introduced to ensure that these unnecessary modes are suppressed.

5.4.2. With the exception of the structural mass, other mass from deadweight items may be placed at intersections of primary longitudinal structures and primary transverse structures to avoid unnecessary local vibration modes.

5.4.3. The free-free boundary condition should be used for the modal (eigenvector) analysis.

5.4.4. It is acceptable to use dynamic or static reduction techniques for the 3D whole ship model to:

• remove unwanted vibration modes of localised structural responses;
• remove spurious vibration modes due to localised poor structural modelling in areas of no interest;
• reduce the number of possible vibration modes to be analysed;
• this is applicable to even the coarse mesh FE model specified above.

5.4.5. For detailed whole ship FE models, it is recommended that, as a minimum, the following dynamic reduction node set (the A-Set in NASTRAN terms) is applied:

• at key points of the cross-section at each transverse bulkhead;
• at key points of the cross section at each mid hold or compartment length.

5.4.6. The above recommendation is applicable for a whipping analysis that is focussed on the calculation of the global hull responses. If a more detailed analysis of a structural area is required, then it is necessary to add the following node points to the dynamic reduction node set:

• at all intersections of primary members in way of all areas of interest.

5.4.7. Typically the eigenvector analysis should assess at least the first 25 modes for assessment of global membrane stresses. For a springing analysis based on global membrane stresses, it is necessary to select all major hull girder vibration modes from this set. For assessment of vertical bending whipping responses in head seas, the first five vertical hull girder vibration modes should be included as a minimum requirement.
5.4.8. The stress solution based on the modal superposition method requires at least 10 to 14 modes to be included to achieve good results. This is especially true at the low frequency range (critical wave frequency range) where more global modes included in the computation will result in better calculations of the traditional hull girder bending stress. The original eigenvector analysis and dynamic reduction node set have to be suitable for the locations where stresses are required.

5.5. Loading a whole ship 3D FE model

5.5.1. It is necessary to apply all the lightship and deadweight items as structural element mass items (QUADS, BARS, etc.) or as MASS elements. It is important to ensure that the required displacement, Δ, longitudinal centre of gravity, LCG, vertical centre of gravity, VCG, and roll radius of gyration, k_{xx}, are achieved in the whole ship FE model. If only vertical bending responses are required, then matching the correct VCG and k_{xx} is not essential. However, if horizontal bending and torsion response is required, then matching the loading condition VCG and k_{xx} is critical.

5.5.2. Liquids in tanks need to be converted to nodal masses around the tank boundary that give:

- The correct mass centroid properties for the tank
- The correct mass inertia. For small tanks, this latter requirement is not that critical. For large tanks then it is critical. In this case, use of nodal masses around the tank boundary is not adequate. It is then necessary to model the liquid as mass items that are linked to the tank boundary using NASTRAN RBE3 elements or similar in a comparable way to that used to define container stacks.

5.5.3. Containers in stacks need to be modelled in such a way that the vertical and transverse distributions of container masses are accurately modelled. Failure to do this will mean that the required loading condition VCG and k_{xx} are not maintained. The recommended way of doing this is as follows:

- model each container stack as a separate mass item and its correct radii of gyration, roll and pitch.
- link this mass item to the relevant part(s) of the structure via the use of NASTRAN RBE3 elements (or equivalent in other FE packages), see Figure 2.5.1 Example of modelling containers and Figure 2.5.2 Full breadth FE model of a container ship.

5.5.4. For containers on deck, define a MASS item for a group of container stacks then link each MASS item to the hatch cover or coaming at each corner of each stack using RBE3 elements. Figure 2.5.1 Example of modelling containers shows an example of on-deck container modelling, here the deck containers have not been split into groups. The RBE3 link elements are shown in purple with the mass element at the VCG of the container stack.

5.5.5. Similarly for containers in holds, for these it may be necessary to represent the attachment to the stack guides using RBE3 elements. Figure 2.5.1 Example of modelling containers also shows the containers in the hold, in case each stack of 20 ft boxes has been modelled and RBE3 linked to the inner bottom longitudinal girder locations.
5.6. Structural damping

5.6.1. There are many uncertainties regarding the structural damping and it is not easy to predict or measure. The modal damping coefficient of the first hull vertical vibration mode is typically between 2 per cent and 3 per cent depending on the ship type and size. Generally the damping ratio grows with increasing natural frequency.

5.6.2. It is recommended that the following formula is used to estimate structural modal damping coefficients for the whipping and springing analyses if the damping values are not available from full scale or model tests:
\[ \eta_i = 0,005 \cdot \omega_i + 0,02 \]

where,

- \( \eta_i \) is the modal damping coefficient for the \( i \)th hull girder natural frequency
- \( \omega_{si} \) is the frequency of the \( i \)th hull girder natural vibration mode, in rad/s

5.6.3. If possible the values of structural damping are to be derived on the basis of achieving a comparable springing response to that obtained for a similar reference ship for which full scale measurements are available.

**Section 6: Hydrodynamic definition of the bow flare and slamming regions**

6.1. Impact load evaluation

6.1.1. For container ships, the main factor affecting the whipping responses of the hull girder is bow flare slamming due to their highly flared bow shape. For other ship types, bottom slamming would become important if the draught is low. Stern slamming is also an issue for container ships but is not usually as important as bow flare slamming.

6.1.2. The wave impact forces due to bow flare impacts, bottom slamming or stern slamming are to be derived by using software linked to the time domain non-linear ship motion code. Suitable methods include:

- Momentum slamming theory;
- Generalised Wagner theory;
- VOF CFD (volume of fluid computational fluid dynamics) codes.

6.1.3. In general only 2D methods are applied due to computational time issues with 3D approaches. Methods such as Generalised Wagner theory or Momentum slamming theory are usually restricted to solving impact loads for bow flare shapes only, in addition they also require that cross-sections must have increasing waterline offsets, hence they cannot model bulbous bows. For bulbous bows the decreasing waterline offsets should be ignored; hence the resulting bow shape is modified to have a vertical side. This issue also means that these methods fail for large roll angles and hence calculating the whipping response in oblique seas can be problematic.

6.2. Definition of the bow flare impact sections

6.2.1. The region forward of the parallel mid-body with significant bow flare, typically forward of 0.8L, is to be defined for the bow flare impact calculation. At least 15 2D transverse cross-sections are to be generated over the forward region.

6.2.2. These 2D cross-sections are to be inclined to the vertical in such a way that the water flow over the hull will be approximately parallel to the cross-section plane, see Figure 2.6.1 Example of cross-sections defined for bow flare slamming. Typically an angle of 25 to 30 degrees will need to be applied.

6.2.3. The following angle to the vertical is suggested as a guideline:

\[ \varphi = \tan^{-1} \left( \frac{R_{hx}}{R_{cv}} \right) \] degrees

where,
\[ R_{hv} = \left( \frac{L_c}{\pi T_c} \right) \cdot a\omega_e + 0.515V \]
\[ R_{vv} = 0.5L\phi\omega_e + a\omega_e \]
\[ a = 10 \]
\[ \phi = 0.14 \]
\[ \omega_e = \omega_{cr} \left( 1 + \frac{0.515V\omega_{cr}}{g} \right) \]
\[ L_c = \text{approximate critical wavelength in m; to be taken as 0.9L} \]
\[ T_c = \sqrt{\frac{2\pi L_c}{g}} \text{ seconds} \]
\[ \omega_{cr} = \sqrt{\frac{2\pi g}{L_c}} \]
\[ V = \text{ship speed in knots} \]

**Figure 2.6.1**
Example of cross-sections defined for bow flare slamming

**Section 7: Calculation of whipping vertical bending moments in time domain**

7.1. The whipping responses of the vertical wave bending moments in the time domain are to be calculated using a time domain hydroelastic hydrodynamic program with capabilities to deal with the non-linear hydrostatic forces due to ship motions and wave incident forces (Froude–Krylov forces) due to the hull shape as well as the impact forces.

7.2. The time domain simulation is to be run using the EDS approach described in *Pt B, Ch 1, Section 2 Determination of the equivalent design sea state (EDS)*.

7.3. The time step in the time domain simulation should not be greater than one tenth of the period of the dynamic system in the time domain simulation which considers both encounter waves and ship vibrational modes. Hence if the highest natural mode of vibration (e.g. the 5th mode) in the simulation has a (wet mode) frequency of 2 Hz, then the time step should be less than 0.05 s (20 Hz).
7.4. For the EDS assessment process, the time domain simulations for the whipping response calculation should be run for at least 300 minutes in order to be able to predict the probable maximum value in 3 hours based on a statistically converged set of whipping response data. In some cases, it might be advisable to run for 600 minutes to achieve convergence. This can be achieved by performing several shorter runs of say 60 or 30 minutes and combining the results. See also Pt B, Ch 1, Section 3 Computation of vertical wave bending moments in time domain.

7.5. The whipping response is a very stochastic process and as a consequence it is necessary to ensure that the bending moment response including whipping has converged.

Note:

Convergence studies have shown that there is still a significant variation of probable maximum in 3 hours for a total simulation time of 180 minutes. Full convergence could generally be achieved after a total simulation time of 300 minutes.

Section 8: Derivation of hogging and sagging whipping factors for the vertical wave bending moment

8.1. The whipping hogging and sagging factors are to be derived for the critical locations specified in Pt C, Ch 3, Section 1 Structural assessment using the method described below.

8.2. The probability distributions of the hogging and sagging bending moment distributions of the time domain simulation are derived using a peak counting method after subtraction of the SWBM. The peaks and troughs (corresponding to the hogging and sagging wave bending moments, based on the usual sign convention of the vertical bending moment) of the time domain signal should be counted separately as one peak or trough per mean zero crossing period of the equivalent rigid-body time domain signal, see Figure 2.8.1 Counting method for extremes of the linear, non-linear and whipping response time domain traces. This can be achieved by:

- Applying this counting approach to a non-linear rigid body time domain analysis run using the same wave-train, see Pt B, Ch 1, Section 4 Derivation of hogging and sagging non-linear factors for the vertical wave bending moment. This is the recommended method.

Figure 2.8.1
Counting method for extremes of the linear, non-linear and whipping response time domain traces
8.3. The maximum expected hogging and sagging bending moments including whipping effects in a three hour return period are obtained by fitting a 3 parameter Weibull or a 3 parameter log-normal or similar distribution to the probability distribution of the peak values, see Pt C, Ch 2, Section 10 Statistical post processing. Examples of Weibull curve fitting and the derivation of the probable maximum 3 hours values are shown in Figure 2.8.2 Weibull fitting of the linear, non-linear and whipping vertical sagging bending moment distributions and Figure 2.8.3 Weibull fitting of the linear, non-linear and whipping vertical hogging bending moment distributions.

![Weibull fitting of the linear, non-linear and whipping vertical sagging bending moment distributions](Figure 2.8.2)

![Weibull fitting of the linear, non-linear and whipping vertical hogging bending moment distributions](Figure 2.8.3)
8.4. The whipping factors along the length of the ship are to be taken as follows:

\[ f_{S-W} = \frac{M_{WS}^{3h}}{M_{VL}^{3h}} \text{ for sagging} \]

\[ f_{H-W} = \frac{M_{WH}^{3h}}{M_{VL}^{3h}} \text{ for hogging} \]

where,

\( M_{WS}^{3h} \) is the “probable maximum whipping sagging moment in 3 hours” from the time domain simulations calculated at the required longitudinal location.

\( M_{WH}^{3h} \) is the “probable maximum whipping hogging moment in 3 hours” from the time domain simulations calculated at the required longitudinal location.

\( M_{VL}^{3h} \) is the “probable maximum linear bending moment in 3 hours” from the time domain simulations calculated at the midship section.

This can be derived from short-term statistical analysis using the EDS, as shown in Pt B, Ch 1, Section 2 Determination of the equivalent design sea state (EDS) or from a linear time domain simulation using the same EDS but ignoring the non-linear hydrostatic forces and wave incident forces due to the hull shape as well as the impact forces.

**Section 9: Whipping vertical wave bending moments**

9.1. The hogging and sagging vertical wave bending moments along length of the ship including the effects of non-linear hull shape and whipping actions are to be taken as follows:

\[ VBM_{WH-S} = f_{S-W} M_{Linear} \text{ for sagging} \]

\[ VBM_{WH-H} = f_{H-W} M_{Linear} \text{ for hogging} \]

where,

\( M_{Linear} \) is the linear design wave bending moment, see Pt A, Ch 1, Section 9 Linear design wave bending moment.

**Section 10: Statistical post processing**

10.1. The non-linear hogging/sagging factors and the whipping hogging/sagging factors are derived by statistically post-processing the time domain results. The factors are specified as the ratio of the probable maximum non-linear and whipping values to the linear values.

10.2. The probable maximum values in three hours are to be obtained by fitting a 3 parameter Weibull or similar distribution to the probability of exceedance of the peak values.

10.3. The Weibull curve fitting routine should be chosen such that it concentrates on a best fit over the tail of the distribution.
10.4. As the best fit over the tail of the distribution is required, then it is recommended that the first few probability bins are excluded from the curve fitting process if necessary to improve the fit over the tail. See Figure 2.10.1 Example of good and poor fitting of Weibull distributions.

10.5. This method ensures that the fitting process concentrates on the tail of the distribution. This is illustrated in Figure 2.10.1 Example of good and poor fitting of Weibull distributions, a) shows an example of fitting based on the distribution being weighted by the number of responses in each bin and using all data points; b) shows an example of fitting based on each response bin being given equal weighting and the lowest response values ignored.

10.6. It should be noted that the final non-linear hog and sag ratios (see Pt B, Ch 1, Section 4 Derivation of hogging and sagging non-linear factors for the vertical wave bending moment) and the whipping hog and sag ratios (see Pt C, Ch 2, Section 8 Derivation of hogging and sagging whipping factors for the vertical wave bending moment) are very strongly affected by the curve fitting process and hence great care and consistency of approach is required. One of the critical issues is the amount of data available and the longer the duration of the time simulation the more data is available (see Pt B, Ch 1, 3.2 and Pt C, Ch 2, Section 7 Calculation of whipping vertical bending moments in time domain). As noted earlier whipping is a stochastic process and hence it is likely that the tail of the distribution will be sparsely populated, hence the results are heavily affected by the number of data points.

10.7. Typically for a 3 parameter Weibull distribution, it is suggested that the location parameter (γ) is adjusted incrementally and the shape (β) and scale (α) parameters derived from the curve fit. The error of the curve fit, ER, is to be derived for a range of location values until the ER is minimised. Note that ER is not a true standard deviation parameter but is useful to demonstrate best fit for this application.

10.8. 3 parameter Weibull distribution parameters are as follows:

Cumulative probability (Q):

\[ Q = \exp \left( -\left( \frac{X - \gamma}{\alpha} \right)^{\beta} \right) \]

Probable maximum value (Mf) for a cumulative probability of Q:

\[ M_f = \gamma + \alpha \left( -\ln(Q) \right)^{1/\beta} \]
PART C: WHIPPING ASSESSMENT

Chapter 3: Assessment of hull girder loads including whipping

Section 1: Structural assessment

1.1. The ultimate strength of the hull girder is to be assessed against the whipping vertical wave bending moments using the following approach:

- Ultimate strength assessment based on the same approach as that adopted by the IACS Unified Requirement S11A for container ships or the IACS Common Structural Rules for Bulk Carriers and Oil Tankers (CSR BC & OT) for other ship types except as noted below.

1.2. It is necessary to review all critical locations along the length of the ship. For container ships this will typically include the following locations:

- Midship section;
- Fore and aft ends of closed and open cross sections, so in way of:
  - the forward end of the closed deck area of an aft accommodation block or engine room;
  - the aft end of the container hold open section just forward of the accommodation block;
  - the aft end of the closed deck area of a mid-length accommodation block;
  - the forward end of the container hold open section just aft of a mid-length accommodation block.
- Cross-sections where there is a significant change in longitudinally effective scantlings.

1.3. The ultimate strength of the hull girder at all critical locations along the length of the ship is to be evaluated. The incremental-iterative method specified below is to be applied:

- For container ships: use the Ship Rules, Part 4, Ch 8, 16.11 (IACS UR S11A).
- For Naval ships: use Naval Ship Rules, Vol 1, Part 6, Ch 4, Section 3 Extreme Strength Assessment and the user guide to the Naval Ship Rules software.
- For other ship types: use CRS BC & OT Ch 5, Appendix 2, see Note 1.
- Alternatively the LR ultimate strength procedure encompassed within LRPASS software LR20202 and LR20203 or equivalent procedures may be adopted, see Note 2.
Notes:

1. The single step method in the CSR BC & OT is not applicable.

2. LR20202 and LR20203 are included as the ultimate strength procedure in LR’s RulesCalc software.

1.4. The ultimate strength criteria to apply are as follows:

\[ \gamma_S M_{SS} + \gamma_W VBM_{WH-S} \leq \frac{M_{US}}{\gamma_R} \] for sagging

\[ \gamma_S M_{SH} + \gamma_W VBM_{WH-H} \leq \frac{M_{UH}}{\gamma_R} \] for hogging

where,

- \( M_{SS} \) is the permissible sagging SWBM that is practical at the draft being considered, see Pt A, Ch 1, 2.1 Loading conditions for the non-linear and whipping assessments

- \( M_{SH} \) is the permissible hogging SWBM that is practical at the draft being considered, see Pt A, Ch 1, 2.1 Loading conditions for the non-linear and whipping assessments

- \( M_{US} \) is the ultimate bending capacity in sagging based on deducting one half the standard deduction for corrosion

- \( M_{UH} \) is the ultimate bending capacity in hogging based on deducting one half the standard deduction for corrosion.

1.5. The following corrosion deductions are to be made for the assessment of the ultimate strength:

1.5.1. The CSR BC & OT or IACS UR S11A requirement is based on \( t_{net50} \) thickness values. On this basis an allowance of 50 per cent of the standard corrosion for buckling in the Ship Rules is to be applied when deriving the ultimate strength.

1.5.2. See the Ship Rules, Pt 4, Ch 8, 16.3 for container ships, and Pt 3, Ch 4, Table 4.7.1 for other ship types, and Pt 1, Ch 3, Sec 2, Table 1 for CSR BC & OT.

1.6. The following partial safety factors are to be applied.

- \( \gamma_S \) is the partial safety factor for SWBM, to be taken as 1.0

- \( \gamma_W \) is the partial safety factor for wave bending moment and is to be taken as:
  - 1.1 for container ships
  - 1.2 for other ship types

- \( \gamma_R \) is the partial safety factor for ultimate capacity and is to be taken as 1.1.

1.7. Alternatively the ultimate strength assessment criteria can be rewritten into a load utilisation factor (LUF) format as follows:

\[ \text{LUF}_{\text{sag}} = \frac{(\gamma_S M_{SS} + \gamma_W VBM_{WH-S})}{M_{US}/\gamma_R} \leq 1 \] for sagging
\[ \text{LUF}_{\text{Hog}} = \left( \gamma_s M_{\text{SH}} + \gamma_w VBM_{\text{WH-H}} \right) \frac{M_{\text{UH}}}{M_{\text{R}}} \leq 1 \quad \text{for hogging} \]
PART D:
SPRINGING ASSESSMENT

Chapter 1:
Fatigue assessment including hull girder springing

Section 1: General

1.1. In this Chapter, requirements are given to calculate the springing effects on the predicted fatigue life. Ideally the effects of springing should be directly included in the spectral fatigue calculation but this is very complex when the non-linearities of the dynamic load response, e.g. intermittent wetting of the hull in way of the static waterline or non-linear ship motion response in large waves, are also included. In view of this the proposed method derives a springing fatigue reduction factor based on a linear spectral fatigue assessment procedure. This springing fatigue reduction factor can then be applied to the non-linear "rigid body" spectral fatigue calculation (FDA3) to obtain a predicted fatigue life including springing effects.

1.2. The first part of the process is to determine the springing response; this is followed by a fatigue assessment using the stress RAOs including the springing responses and also a fatigue assessment excluding the springing responses. The ratio of these two fatigue lives give the springing fatigue reduction factor that is applicable for the structural element being considered.

1.3. The occurrence of springing of a ship can be reliably predicted by hydroelastic ship motion programs. However the exact determination of the magnitude of springing response is more difficult. The current hydroelastic codes are not able to do this very well and hence it is necessary to calibrate the hydroelastic analysis method based on full scale measurements of a reference ship.
1.4. **Figure 1.1.1 Measured and calculated vertical bending moment amidships for a JONSWAP wave spectrum** shows the calculated spectral energy response for vertical bending moment amidships of an 8500 TEU container ship against the measured response from full scale measurements. The responses were calculated for the following sea state which was estimated from the on board WAVEX wave radar instrument: JONSWAP wave spectrum in head seas $H_s = 6.25\, \text{m}$, $T_p = 11.1\, \text{s}$ and $\gamma = 3.3$. As can be seen, the comparison is very good. Some whipping response was observed in the full scale measurements, however it should be noted that the spectral energy of whipping responses will normally be very small, so what is being shown here is mainly the springing response around a wave encounter frequency of 3.5 rad/s.

![Figure 1.1.1](image)

1.5. The following process does not include any allowances for whipping induced fatigue. Whipping responses will result in a decrease in the fatigue life, but it is not thought that whipping is a major contributor to fatigue due to the relatively low number of cycles associated with whipping during a ship’s lifetime.

**Section 2: Outline of assessment procedure**

2.1. Several methods are possible for the determination of the springing response for inclusion in the fatigue assessment. The reason for inclusion of more than one method is due to the fact that this technology is still maturing, there are several valid ways to approach the problem and industry is in the process of establishing the most suitable methods for the calculation of springing and the assessment of fatigue including springing.

2.2. The possible methods are:

1. Assessment of linear stress RAOs based on a global ship 3D FE analysis for all headings and speeds including springing responses. This method is recommended.

2. Assessment of linear bending moment RAOs based on 2D beam FE analysis for all headings and speeds including springing responses. In general it is hard to assess the warping stresses induced by torsional moments as it is not easy to determine exact torsional properties along the ship length. This method is satisfactory for ships with closed cross sections where warping stresses are low; otherwise the linear stress RAO method should be used.
3. Assessment of non-linear springing responses for critical wave frequencies. This approach demands much more computing time; however the method can take into account the non-linear contributions due to the hull form shape and higher harmonics. This method is not recommended for normal use but may be applicable for use for unusual vessels. Please consult with LR before using this method.

2.3. The aim of these assessment procedures is to allow a "springing fatigue reduction factor" to be derived, see Pt D, Ch 1, Section 3 Loading condition. This can be used to adjust the results of the Level 3 Fatigue Design Assessment procedure (FDA3) to allow for springing responses.

2.4. The choice of which springing response method to use is to some extent dependant on the chosen fatigue assessment process and the available hydroelastic and fatigue assessment tools.

2.5. Assessment of linear stress RAOs for all headings and speeds including springing responses

2.5.1. The linear stress RAO approach is the principal method currently used by LR (see Figure 1.2.1 Flowchart of the fatigue assessment using RAOs including springing responses). This approach uses a frequency domain hydroelastic analysis based on a full ship 3D FE model. It allows stress RAOs to be derived on any element of the FE model taking into account all external hydrodynamic and hydroelastic pressures and internal inertial loads; thereby accounting for all local, primary and global hull girder load actions including vertical bending and shear, horizontal bending and shear and torsional and warping effects.
2.5.2. The linear frequency domain method gives the response amplitude operator RAO curves including the springing response. This method is an extension of the normal ship motion theory to include flexible body (hydroelastic) effects. The advantage of a frequency domain hydroelastic method is that it can calculate the motion and load responses including springing actions for all ship speeds and headings in one analysis.

2.5.3. The springing responses can be calculated using a frequency domain hydroelasticity program. This is applicable for the fatigue calculation using the springing RAO method in Pt D, Ch 1, 2.5 and 2.6. The basis of this method is similar to that given for the linear design load calculations described in Pt A. The hydroelastic responses (RAOs) with both rigid and flexible components to unit regular waves are calculated using a frequency domain hydroelasticity program.

2.5.4. Determination of the stress RAOs including springing effects for any element of the full ship FE model allows a full spectral fatigue assessment to be undertaken using spectral techniques suitable for bi-modal spectral response to derive the fatigue damage in each sea state.

2.5.5. The linear stress RAO approach is supported by LR’s software and tools as follows:

- Trident or Nastran FE package to carry out eigenvalue analysis;
• **HydroModeller**  Creates the hydrodynamic mesh from offsets, FE models, STL (Standard Triangle Language data format) models or similar. Applies the eigenvector (modal shapes) to the hydrodynamic mesh;

• **HydroE-FD**  Linear flexible body frequency domain 3D diffraction hydroelastic ship motion code;

• **Springing Spreadsheet**  Spreadsheet to evaluate the fatigue damage including springing effects.

2.5.6. The stress RAOs can be calculated by back substituting the calculated magnitudes of the principle modes from the hydroelastic analysis into the finite element eigenvalue analysis. These stress RAOs are exact in the sense that they include all load actions and structural dynamic responses, so include the responses due to local inertia and dynamic pressure loads.

2.5.7. The stress RAO of any element \( i \) in the FE structure model can be represented by the formula:

\[
\text{Stress RAO}_i = \sum_{k=7}^{n} \text{SE}_{i,k} \times \text{RAO}_k
\]

where,

- \( k \) is the mode number (1 to 6 are rigid body modes (surge, sway, heave, roll, pitch and yaw), 7 to \( n \) are flexible body modes);
- \( \text{SE}_{i,k} \) is the stress eigenvector of flexible mode \( k \) at element \( i \);
- \( \text{RAO}_k \) are the RAOs of each flexible mode \( k \) (complex value with different magnitude and phase angle for each mode).

2.5.8. *Figure 1.2.2 Stress RAOs including springing effects for a very flexible ship* shows the stress RAOs calculated by the process of back substituting the magnitudes of the principle modes into the FE eigenvalue analysis, labelled as "calculated by stress modal shapes". These are compared with the stress RAOs calculated by using the hull girder section modulus, the elastic bending moment response, see "calculated by HydroE-VBM" curve. As can be seen these curves are very similar for this case as it is a head sea case and 2D beam theory is adequate to assess the vertical bending stress for this keel location. Please note that this ship was a very flexible body with a very low natural period of vibration. The hull girder was so flexible that the elastic response to long waves (wave frequency of 0.4 rad/s) was significantly reduced compared to the "rigid" body response, see the "calculated by rigid body-VBM" curve.
2.5.9. In order to derive accurate stress RAOs, sufficient flexible modes have to be included in the hydroelastic analysis. Failure to include sufficient modes will result in inaccurate stress RAOs being calculated. Typically at least 12 flexible modes need to be included in the hydroelasticity calculation, but a check similar to that shown in Figure 1.2.2 Stress RAOs including springing effects for a very flexible ship may be necessary to confirm this. Failure to include a sufficient number of flexible modes could result in the stress RAOs being underestimated. It is also necessary to ensure that the chosen dynamic reduction node set is sufficient to allow the stresses to be correctly predicted for the locations being reviewed, see Pt C, Ch 2, 6.4.

2.5.10. Using this approach, it is possible to derive stress RAOs including springing effects for any element within a coarse mesh FE model.

2.6. Assessment of linear bending moment RAOs for all headings and speeds including springing responses

2.6.1. It is also possible to use a linear bending moment RAO approach. This approach uses a frequency domain hydroelastic analysis applied to a 2D beam element FE model of the ship. It allows the global bending moment RAOs to be derived along the ship taking into account all external pressures and internal inertial loads including those associated with hydroelastic issues.

2.6.2. Determination of the bending moment RAOs including springing effects at any longitudinal location of the ship allows the stress to be derived at selected locations using simple beam analysis techniques. A full spectral fatigue assessment is then undertaken using suitable bi-modal spectral techniques to derive the fatigue damage in each sea state or other suitable techniques. This method is limited by the simple beam element calculation of the stress at a location, for example warping stresses and primary structural responses might not be correctly included.

2.6.3. The linear bending moment RAO approach is fully supported by LR developed software tools.
Section 3: Loading condition

3.1. The loading conditions to apply are specified in PART A, Ch 1, 2.2.

Section 4: Hydrodynamic modelling

4.1. The mesh size, model verification, ship operational conditions for the hydrodynamic model are specified in PART A, Ch 1, Sec 3 to 6.

4.2. The hydroelastic analysis is to cover the frequency range appropriate for the speed of the ship and the required hull girder natural vibration frequencies that need to be considered, see PART A, Ch 1, Sec 3 to 7.

Section 5: Structural model of the ship

5.1. This is taken as specified for the whipping assessment; see PART C, Ch 2, Sec 5.

Section 6: Structural damping

6.1. This is taken as specified for the whipping assessment; see PART C, Ch 2, 5.6.

6.2. Currently exact prediction of springing (and whipping) responses of ship is not possible due to lack of accurate information regarding structural damping and a few other issues, but damping is the key item. The current hydroelastic analysis programs give a good indication of springing, but not an accurate value for the magnitude of the springing response primarily due to the uncertainties in the structural damping of each of the key vibrational modes. The proposed values of structural damping in PART C, Ch 2, 5.6 are considered to be conservative.

6.3. Other values of structural damping may be proposed based on calibration of the hydroelastic ship motion program against some known reference ship. A springing analysis for an existing ship where full scale measurements including springing information have been recorded should be undertaken. Ideally the existing ship should be as similar as possible to the new design being assessed.

6.4. The calibration process can consider the following factors:

- Adjustment of the structural damping to achieve the correct magnitude of springing response.

- Adjustment of the hull girder structural properties to refine the natural vibration frequencies, but this is likely to be a function of incomplete structural modelling.

6.5. A hydroelastic analysis has to be performed for the reference ship. Hence knowledge of the loading condition, drafts, mass distribution, speed, heading and environmental conditions (sea state including spectral energy distribution and directional content) together with hull girder stresses is necessary for several time periods when springing and/or whipping was present. The hydroelastic analysis should be performed for several of these time periods and the hydroelastic analysis is to be adjusted to achieve a calibrated springing response. The resultant adjustments can then be applied to the new ship design.

6.6. Various figures in this document illustrate comparisons between full scale measurements and hydroelastic analyses.

Section 7: Springing fatigue reduction factor (SRF)

7.1. Overview
7.1.1. The traditional spectral fatigue assessment process adopts a quasi-static approach to calculate the element stresses by applying hydrodynamic pressures and inertial loads to the static FE model and this allows any location in the structure to be assessed across the whole frequency band, this is the basis of LR’s FDA3 and FDA2 procedures. These procedures calculate the probability of a ship encountering a particular sea state at a particular speed, heading and loading condition. The statistical stress responses are then obtained for each sea state, with corrections for non-linear effects such as intermittent wetting. The fatigue damage value in that sea state/heading/speed/loading condition combination is then evaluated. The final fatigue damage value is given by the summation of the fatigue damage value for all sea states. Currently these procedures only allow rigid body stress RAOs to be applied and hence this means that dynamic effects near structural resonant frequencies, i.e. springing effects, cannot be considered.

7.1.2. For the springing fatigue assessment, the above process needs to be applied to the stress RAOs including springing effects. Ideally the FDA3 or FDA2 procedures and software should be updated to include this. However inclusion of the non-linear effects as well as springing effects is very complex and hence an alternative approach is recommended.

7.1.3. A process similar to the standard spectral fatigue assessment is undertaken based on the stress RAOs including springing, so effectively a simplified FDA3 or FDA2 calculation.

7.1.4. For the calculation of the fatigue damage, the wave environmental data to be used is to be taken from the FDA2 or FDA3 analysis.

7.1.5. A two stage fatigue calculation is undertaken, the first stage determines the fatigue damage including springing effects for each structural element and the second stage repeats the calculation excluding the springing effects. The ratio of these two fatigue damages gives the springing fatigue reduction factor (SRF) that is applicable for the structural elements being considered.

7.1.6. The resulting springing fatigue reduction factor, so FDA3 or FDA2 results, can then be applied to the results of the standard "rigid body" (no elasticity) fatigue assessment results to obtain a fatigue life including springing effects.

7.1.7. The calculation of the springing reduction factor is supported by a LR spreadsheet which may be available on request.

7.2. Calculation of fatigue damage index including and excluding springing for one sea state

7.2.1. The fatigue damage including springing effects for any structural element and for each sea state, ship speed and heading combination is to be calculated using the spectral techniques adapted for wide band spectra in association with the stress RAO including springing. The total fatigue damage is the summation of all these sea state, ship speed and heading combinations.

7.2.2. Typically the fatigue damage is calculated based on several loading conditions, however as this is a comparative approach, then the calculation of the fatigue damage on the basis of the loading condition most likely to experience springing will simplify the analysis.

7.2.3. A typical stress RAO curve is shown in Figure 1.7.1 Typical stress RAO curve and cut-off frequency, head sea case showing the dominant vertical bending mode. As can be seen in Figure 1.7.1 Typical stress RAO curve and cut-off frequency, head sea case showing the dominant vertical bending mode, the stress RAO is a wideband signal with multi peaks.

7.2.4. The stress response energy of structural element $i$ at a given ship speed and wave heading in a single sea state is:
\[
RS_{w}(\omega) = \int_{0}^{\infty} S(\omega) \left( \text{Stress RAO}(\omega) \right)^2 d\omega
\]

where,

\( S(\omega) \) is the wave energy of the sea state being considered.

7.2.5. To calculate the corresponding fatigue damage excluding springing effects in a manner that is consistent with the hydroelastic springing ship motion analysis, it is necessary to calculate the equivalent “rigid body” fatigue damage. This is achieved by calculating the spectral stress response energy up to the frequency where springing effects are observed. This is denoted as the springing cut off frequency. An example of the cut off frequency is the red line shown in Figure 1.7.1 Typical stress RAO curve and cut-off frequency, head sea case showing the dominant vertical bending mode. The same fatigue calculation process is then applied to the "cut-off" stress RAO response, i.e. the rigid body stress response, for each sea state, heading, speed combination and the total fatigue damage is the summation of all these combinations. Above this frequency it is assumed that these are the springing induced stresses.

7.2.6. The cut-off frequency can be taken as follows:

\[
\omega_{\text{cut-off}} = \omega_i - 5\zeta\omega_i
\]

where,

\( \omega_i \) = the first major contributing natural frequency of the hull girder
\( \zeta \) = modal damping coefficient of the first vibration mode of the hull girder.

7.2.7. The first major contributing natural frequency of the hull girder can usually be taken as one node torsion modal value, two node horizontal bending modal value or two node vertical bending modal value depending on location. Figure 1.7.2 Typical stress RAO curve and cut-off frequency, oblique sea case showing the dominant combined torsional and horizontal vibrational modes shows a flexible body stress RAO curve for an oblique sea case, it can be seen that the torsional and horizontal mode responses are important for the stress RAO results.
7.3. **Fine mesh details and stress concentration factors (SCFs)**

7.3.1. Typically a coarse mesh FE model will be used for the hydroelastic analysis. In this case, it’s not possible to assess the effect of localised fine mesh regions on localised stresses. It is necessary to include suitable SCFs into the springing fatigue analysis in order to obtain a stress history spectrum that is similar to the rigid body fatigue calculations from FDA2 and FDA3.

7.4. **Short term fatigue damage calculation**

7.4.1. The total springing response in a sea state is carried out using standard short term statistical methods that are adapted to calculation of multi peak response spectrum.

7.4.2. From this, a fatigue calculation can be undertaken using the rain-flow counting method or by utilising a multi-peak short term spectral calculation, such as specified in the following papers:


7.5. **Calculation of the final fatigue life and damage index**

7.5.1. The total fatigue damage index including springing effects $D_{total}$ is given by:

$$D_{total} = \frac{D_{r-FDA3}}{S_{RF}}$$

where,
\[ D_{r-FDA3} = \text{rigid body fatigue damage index without springing (FDA Level 3 result)} \]

\[ S_{RF} = \text{springing reduction factor} \]

\[ S_{RF} = \frac{D_{r-HydroE}}{D_{r-HydroE}} \]

\[ D_{r-HydroE} = \text{rigid body fatigue damage index without springing from the hydroelastic analysis} \]

\[ D_{t-HydroE} = \text{total fatigue damage index including springing from the hydroelastic analysis} \]

7.5.2. The resultant fatigue life will be:

\[ FL_{\text{total}} = S_{RF} FL_{FDA3} \]

where, the rigid body FDA Level 3 calculated fatigue life is given by:

\[ FL_{FDA3} = \frac{1}{D_{r-FDA3}} \]

7.5.3. It should be noted that if the rigid body FDA Level 3 calculated fatigue life is 40 years, a springing reduction factor of 50 per cent will give a total fatigue life of 20 years including springing.

7.5.4. Figure 1.7.3 Example of calculated springing reduction factors at a hatch coaming top and Figure 1.7.4 Example of calculated springing reduction factors at an upper deck display examples of calculated springing reduction factors taking into account springing effects. Figure 1.7.5 Example of the effect of heading, wave height and period and ship speed on the contribution of springing effects on fatigue illustrates the effect of heading, wave height and period and ship speed on the contribution of springing effects on fatigue damage on the upper deck for a large container ship. Each graph shows the total percentage contribution to the overall fatigue damage value based on summation of the individual fatigue damage values for the specified heading, wave height, wave period or ship speed. For this example, this clearly illustrates that springing fatigue issues are dominant at high ship speed, low wave periods and low wave heights. Also the critical headings that induce significant springing fatigue damage are close to head seas for the selected structural element.
Figure 1.7.3
Example of calculated springing reduction factors at a hatch coaming top

Figure 1.7.4
Example of calculated springing reduction factors at an upper deck
Figure 1.7.5
Example of the effect of heading, wave height and period and ship speed on the contribution of springing effects on fatigue