Structural Design Assessment

Primary Hull and Cargo Tank Supporting Structure of Type C Tank Liquefied Gas Carriers

January 2017
<table>
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<tr>
<th>Document Date</th>
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<tr>
<td>September 2015</td>
<td>Preliminary release</td>
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<tr>
<td>March 2016</td>
<td>Final release</td>
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</table>
| January 2017    | Consolidated version to revise buckling acceptance criteria in Table 2.6.1 and Corrigenda for March 2016 version’.

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Section 1
Application

1.1 General

1.1.1 In accordance with Lloyd’s Register’s (hereinafter referred to as LR’s) Rules and Regulations for the Construction and Classification of Ships for the Carriage of Liquefied Gases in Bulk (commonly referred to as Rules for Ships for Liquefied Gases), Chapter LR III.5, the ShipRight Structural Design Assessment (SDA) procedures are mandatory for Liquefied Gas Carriers fitted with Independent Type C cargo tanks.

1.1.2 The IMO International Code for the Construction and Equipment of Ships Carrying Liquefied Gases in Bulk (IGC Code) is incorporated in LR’s Rules for Ships for Liquefied Gases. References to the IGC Code in this procedure are equivalent to references to the Rules for Ships for Liquefied Gases.

1.1.3 For compliance with the ShipRight SDA procedure, direct calculations are to be adopted for the determination and verification of the stress level and buckling capability of the ship’s primary structural members, and the structure supporting the cargo tanks as specified in LR’s Rules for Ships for Liquefied Gases.

1.1.4 The minimum requirements specified in this procedure, in addition to the requirements in LR’s Rules and Regulations for the Classification of Ships (commonly referred to as the Rules for Ships) and LR’s Rules for Ships for Liquefied Gases are to be complied with.

1.1.5 The SDA procedure requires the following:

- A detailed analysis of the cargo tank support systems, primary hull structures, as specified in LR’s Rules for Ships for Liquefied Gases. This involves a detailed Finite Element Analysis (FEA) to assess the suitability of the support systems and hull structures in way under applied static and dynamic loads.
- A detailed analysis of the structural response of the ship under applied static and dynamic loads using FEA.
- Other direct calculations as applicable.

1.1.6 In general, the direct calculation is to be based on a three-dimensional (3-D) FEA carried out in accordance with the procedures contained in these guidance notes. Where alternative procedures are proposed, these are to be agreed with LR before commencement.

1.1.7 A detailed report of the calculations is to be submitted and must include the information detailed in Ch 1, 3.1 Contents of SDA report 3.1.1. The report must show compliance with the specified structural design criteria given in Ch 2, 5 Permissible stresses and Ch 2, 6 Buckling acceptance criteria.

1.1.8 If the computer programs employed are not recognised by LR, full particulars of the programs will also be required to be submitted, see Pt 3, Ch 1, 3.1 of the Rules for Ships.

1.1.9 LR may require the submission of computer input and output to further verify the adequacy of the calculations carried out.

1.1.10 Ships which have novel features or unusual hull structural or tank configurations may need special consideration.

1.1.11 This procedure may also be applied to the analysis for primary hull and cargo tank supporting structure of Type C tank LNG carriers.

1.1.12 It is recommended that the designer consults with LR on the SDA analysis requirements early on in the design cycle.

Note The objective of this procedure is to assess the strength of the ship primary hull structure and tank supporting structure to withstand the design loads. The ability of the containment system to accommodate the global or local deformations of the ship structure is not considered in this procedure. Therefore, it is necessary for the designers to
consider and demonstrate separately that the containment system design in terms of strength and fatigue capability can withstand intended loads and comply with the Rules for Ships for Liquefied Gases and IGC Code, see Ch 1, 1.1 General 1.1.2. Additional design requirements with respect to the ship structure specified by the containment system supplier are to be complied with. It is recommended that the designer consult the containment system supplier early on in the design cycle.

Section 2
Symbols

2.1 Definition

2.1.1 The symbols used in these guidance notes are defined as follows:

- \( L \) = Rule length, as defined in Pt 3, Ch 1,6 of the Rules for Ships
- \( B \) = moulded breadth, as defined in Pt 3, Ch 1,6 of the Rules for Ships
- \( D \) = depth of ship, as defined in Pt 3, Ch 1,6 of the Rules for Ships
- \( k \) = higher tensile steel factor, see Pt 3, Ch 2,1.2 of the Rules for Ships
- \( \text{SWBM} \) = Still Water Bending Moment
- \( \text{VWB} \) = Vertical Bending Moment
- \( M_w \) = design vertical bending moment, including hog and sag factor, \( f_2 \), and ship service factor, \( f_1 \), see Pt 3, Ch 4,5 of the Rules for Ships
- \( M_{w0} \) = vertical wave bending moment, excluding hog and sag factor and ship service factor, see Pt 3, Ch 4,5 of the Rules for Ships
- \( f_1 \) = the ship service factor, see Pt 3, Ch 4,5 of the Rules for Ships
- \( f_2 \) = the hogging/sagging factor, see Pt 3, Ch 4,5 of the Rules for Ships
- \( M_s \) = Rule permissible still water bending moment, see Pt 3, Ch 4,5 of the Rules for Ships
- \( M_{sw} \) = design still water bending moment, see Pt 3, Ch 4,5 of the Rules for Ships
- \( M_{sw} \) = still water bending moment distribution envelope to be applied to the FE models for stress and buckling assessments. The values of \( M_{sw} \) are to be greater than \( M_s \) and less than or equal to \( M_s \). These values are to be incorporated into the ship’s Loading Manual and loading instrument as the assigned permissible still water bending moment values. \( M_{sw} \) hereinafter referred to as the permissible still water bending moment. See also Ch 2, 4.1 Introduction 4.1.5

- \( \theta \) = scantling draught
- \( T \) = condition draught
- \( GM \) = transverse metacentric height, including free surface correction, for the loading condition under consideration
- \( \theta \) = heel angle
- \( V \) = service speed (knots)
- \( g \) = acceleration due to gravity
- \( \rho \) = density of sea-water (specific gravity to be taken as 1,025 tonnes/m\(^3\))
- \( h \) = local head for pressure evaluation
- \( \rho_c \) = maximum cargo density at the design temperature
- \( P_0 \) = design vapour pressure, see Ch 4, 4 Cargo Containment 4.1.2 of the Rules for Ships for Liquefied Gases. \( P_0 \) is not to be taken as less than 0,25 bar.
\[ A_x, A_y, A_z = \text{maximum dimensionless acceleration factors (i.e., relative to the acceleration of gravity) in the longitudinal, transverse and vertical directions respectively.} \ A_x \text{ is positive forwards, } \ A_y \text{ is positive to port and } \ A_z \text{ is positive downwards.} \]

\[ h_x, h_y, h_z = \text{local head for pressure evaluation measured from the tank reference point in the longitudinal, transverse and vertical directions respectively.} \ h_x \text{ is positive forwards, } \ h_y \text{ is positive port and } \ h_z \text{ is positive up.} \]

\[ L_t = \text{length of cargo tank} \]
\[ t = \text{thickness of plating} \]
\[ t_c = \text{thickness deduction for corrosion} \]
\[ \sigma_c = \text{elastic critical buckling stress} \]
\[ \sigma_o = \text{specified minimum yield stress of material (special consideration will be given to steel where } \sigma_o \geq 355 \text{ N/mm}^2, \text{ see Pt 3, Ch 2.1 of the Rules for Ships)} \]
\[ \sigma_L = \frac{235}{k_L} \text{ N/mm}^2 \]
\[ \lambda = \text{factor against buckling} \]
\[ \mu_S = \text{static friction coefficient} \]
\[ \mu_d = \text{dynamic friction coefficient} \]
\[ \tau = \text{shear stress} \]
\[ \sigma_e = \text{von Mises equivalent stress given by} \]
\[ \sigma_e = \sqrt{\sigma_x^2 + \sigma_y^2 - \sigma_x \sigma_y + 3\tau_{xy}^2} \]

where
\[ \sigma_x = \text{direct stress in element } x \text{ direction} \]
\[ \sigma_y = \text{direct stress in element } y \text{ direction} \]
\[ \tau_{xy} = \text{shear stress in element } x-y \text{ plane.} \]

2.1.2 Consistent units are to be used throughout all parts of the analysis. Results presentation in N and mm preferred.

2.1.3 All Rule equations are to use units as defined in the Rules for Ships.

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**Section 3**

**Direct calculation procedure report**

3.1 **Contents of SDA report**

A report is to be submitted to LR for approval of the primary structure of the ship and is to contain:

- list of plans used, including dates and versions;
- detailed description of structural model, including all modelling assumptions;
- plots to demonstrate correct structural modelling and assigned properties;
- details of material properties used for all components including all support chocks;
- details of displacement boundary conditions;
- details of all still water and dynamic loading conditions reviewed with calculated shear force (SF) and bending moment (BM) distributions;
- details of the calculations for the waterlines used for the dynamic loading conditions;
- details of the acceleration factors for each loading condition;
- details of applied loadings and confirmation that individual and total applied loads are correct;
- details of boundary support forces and moments;
• details of the derived tank support chock loadings and loads on cradle, anti-flotation, etc., chocks, including gap element forces (if used) and spring forces;
• plots and results that demonstrate the correct behaviour of the structural model to the applied loads;
• summaries and plots of global and local deflections;
• summaries and sufficient plots of von Mises, directional and shear stresses to demonstrate that the design criteria are not exceeded in any member;
• plate buckling analysis and results;
• tabulated results showing compliance, or otherwise, with the design criteria; and
• proposed amendments to structure, where necessary, including revised assessment of stresses and buckling properties.
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<td>ALTERNATIVE PROCEDURES FOR TRANSVERSE LOAD CASES</td>
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Section 1

Objectives

1.1 General

1.1.1 The objectives of the structural analysis are to verify that the stress level and buckling capability of primary hull structure and the cargo support systems under the applied static and dynamic loads are within the acceptable limits for gas carriers with Independent Type C tanks.

1.1.2 The analysis and applied loading are to be sufficient to evaluate the responses of the following primary structural items within the cargo hold region including structural members in way of the foremost bulkhead of No.1 cargo tank, the aftmost bulkhead of the aftmost cargo tank:

(a) Ship’s primary structure includes:
   • upper deck and topside plating,
   • side shell plating,
   • inner bottom and bottom shell plating,
   • double bottom floors and girders,
   • bottom transverse structures where single bottom is applied,
   • hopper tank floors and plating,
   • transverse bulkhead,
   • topside web plating,
   • ordinary web sections,
   • all other structures in way of tank supports and chocks.

(b) Supporting structure for cradle includes:
   • Web sections for cradle,
   • Supporting brackets for cradle.

(c) Structure on upper deck includes:
   • Supporting structures of deck tank, if fitted,
   • Intersection areas between motor/compressor room and upper deck, if arranged.

1.1.3 Adequacy of hull structure supporting the cargo containment system is to be investigated either by incorporation of fine mesh areas or by use of separate fine mesh models.

1.1.4 The calculated load on each tank support assumes perfect fit and alignment of all supports and structural members. Additionally, no account is taken of misalignment caused by contraction of the tank’s structure during cool down, construction tolerances or any other factors in the preparation of the finite element (FE) models. Recognition of these limitations is to be made in the assessment of the chock and support arrangements and guidance for this is given in Ch 2, 4.14 Support and chock design loads.
Section 2

Structural modelling

2.1 FE modelling

2.1.1 A 3-D FE model of the complete ship length is to be used to assess the primary structure of the ship.

2.1.2 Unless there is asymmetry of the ship or cargo tank primary structure about the ship’s centreline, only one side of the ship needs to be modelled with appropriate boundary conditions imposed at the centreline. However, it is recommended that both sides of the ship are modelled as this will simplify the loading and analysis of the asymmetric transverse loading condition.

2.1.3 The FE model is to be represented using a right-handed Cartesian co-ordinate system with:

- $X$ measured in the longitudinal direction, positive forward,
- $Y$ measured in the transverse direction, positive to port from the centreline,
- $Z$ measured in the vertical direction, positive upwards from the baseline.

2.1.4 Typical arrangements representing Type C tank liquefied gas ships are shown in Figure 2.2.1 3-D FE model of a gas carriers with independent Type C tanks to Figure 2.2.5 3-D FE model of web section for cradle. In general, the tank shape of Type C tank can be either bi-lobe or cylindrical shape. The proposed scantlings, excluding Owner’s extras and any additional thicknesses to comply with the optional ShipRight ES Procedure, are to be used throughout the model. The selected size and type of elements are to provide a satisfactory representation of the deflection and stress distributions within the structure.

2.1.5 In general, the plate element mesh is to follow the primary stiffening arrangement for both the structure of the ship and cargo tanks. The minimum mesh size requirements are:

- transversely, one element between every longitudinal stiffener;
- longitudinally, three or more elements between web frames;
- vertically, one element between every stiffener; and
- three or more elements over the depth of double bottom girders, floors, side transverses, the vertical webs and horizontal stringers of transverse cofferdam bulkheads.

The mesh density of the side shell platings in way of the side frames is to be similar to those adjacent to the side shell platings.

2.1.6 The bow and the stern of the ship are to be modelled, but it is not necessary to include all the structure within the bow region forward of the collision bulkhead and stern region aft of the aft peak bulkhead. It is sufficient to model most longitudinal plating, stringers and continuous stiffeners together with sufficient transverse structure to support the modelled longitudinal material.

2.1.7 Secondary stiffening members are to be modelled using line elements positioned in the plane of the plating having axial and bending properties (bars). The bar elements are to have:

- a cross-sectional area representing the stiffener area, excluding the area of attached plating; and
- bending properties representing the combined plating and stiffener inertia.

2.1.8 Face plates and plate panel stiffeners of primary members are to be represented by line elements (rods or bars) with the cross-sectional area modified, where appropriate, in accordance with Table 2.2.1 Line element effective cross-section area and Figure 2.2.6 Effective area of face bars.

2.1.9 In general, the use of triangular plate elements is to be kept to a minimum. Where possible, they are to be avoided in areas where there are likely to be high stresses or a high stress gradient. These include areas:

- in way of tank support items;
- in way of lightening/access holes; and
- adjacent to brackets, knuckles or structural discontinuities.

2.1.10 Dome openings, access openings, lightening holes, etc., in primary structures are to be represented in areas of interest, e.g. in floor plates adjacent to the hopper knuckle, topside tanks, girders at their ends, etc. Additional mesh refinement may be necessary to model these openings, but it may be sufficient to represent the effects of the opening by deleting the appropriate elements.
2.1.11 Lightening holes, access openings, etc., away from the locations referred to in Ch 2, 2.1 FE modelling may be modelled by deleting the appropriate elements or by applying a correction factor to the resulting shear stresses, see Ch 2, 5.1 Permissible stresses.

2.1.12 The modelling of cargo tank supports and chocks is described in Ch 4 Modelling of Supports and Chocks.

2.1.13 The light mass of the ship is to be represented in the model. The weight and inertia forces of the ship engine and other heavy items should be correctly transferred to the supporting structure.

Figure 2.2.1 3-D FE model of a gas carriers with independent Type C tanks
SDA of Primary Hull and Cargo Tank Supporting Structure of Type C Liquefied Gas Carriers

Analysis of Primary Structures of Type C Liquefied Gas Carriers

Chapter 2

Section 2

Figure 2.2.2 3-D FE model of Type C cargo tanks

Figure 2.2.3 3-D FE model showing internal structures of a watertight bulkhead
Figure 2.2.4 3-D FE model showing internal structures in way of a tank swash bulkhead (other configurations may be possible)

Figure 2.2.5 3-D FE model of web section for cradle
Figure 2.2.6 Effective area of face bars
### Table 2.2.1 Line element effective cross-section area

<table>
<thead>
<tr>
<th>Structure represented by element</th>
<th>Symmetrical</th>
<th>Asymmetrical</th>
</tr>
</thead>
<tbody>
<tr>
<td>Primary member face bars</td>
<td></td>
<td></td>
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<tr>
<td>Symmetrical</td>
<td></td>
<td></td>
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<tr>
<td>Asymmetrical</td>
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<tr>
<td>A&lt;sub&gt;e&lt;/sub&gt; = 100% A&lt;sub&gt;n&lt;/sub&gt;</td>
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<tr>
<td>A&lt;sub&gt;e&lt;/sub&gt; = 100% A&lt;sub&gt;n&lt;/sub&gt;</td>
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<tr>
<td>Curved bracket face bars (continuous)</td>
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<tr>
<td>Symmetrical</td>
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<tr>
<td>Asymmetrical</td>
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<tr>
<td>From Figure 2.2.6 Effective area of face bars</td>
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<tr>
<td>Straight bracket face bars (discontinuous)</td>
<td></td>
<td></td>
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<tr>
<td>Symmetrical</td>
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<tr>
<td>Asymmetrical</td>
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<tr>
<td>A&lt;sub&gt;e&lt;/sub&gt; = 100% A&lt;sub&gt;n&lt;/sub&gt;</td>
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<tr>
<td>A&lt;sub&gt;e&lt;/sub&gt; = 60% A&lt;sub&gt;n&lt;/sub&gt;</td>
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<tr>
<td>Straight bracket face bars (continuous around toe curvature)</td>
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<tr>
<td>Symmetrical</td>
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<td>Asymmetrical</td>
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<td>A&lt;sub&gt;e&lt;/sub&gt; = 100% A&lt;sub&gt;n&lt;/sub&gt;</td>
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<td>A&lt;sub&gt;e&lt;/sub&gt; = 60% A&lt;sub&gt;n&lt;/sub&gt;</td>
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<tr>
<td>Curved portion</td>
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<td>Symmetrical</td>
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<td>Asymmetrical</td>
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<tr>
<td>From Figure 2.2.6 Effective area of face bars</td>
<td></td>
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</tr>
<tr>
<td>Web stiffeners – sniped both ends</td>
<td>Flat bars</td>
<td></td>
</tr>
<tr>
<td>A&lt;sub&gt;e&lt;/sub&gt; = 25% stiffener area</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Other sections</td>
<td></td>
<td></td>
</tr>
<tr>
<td>A&lt;sub&gt;e&lt;/sub&gt; = \frac{A}{1 + \left(\frac{Y_0}{r}\right)^2} - A_p</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Web stiffeners – sniped one end, connected other end</td>
<td>Flat bars</td>
<td></td>
</tr>
<tr>
<td>A&lt;sub&gt;e&lt;/sub&gt; = 75% stiffener area</td>
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<td>Other sections</td>
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<tr>
<td>A&lt;sub&gt;e&lt;/sub&gt; = \frac{A}{1 + \left(\frac{Y_0}{r}\right)^2} - A_p</td>
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</tbody>
</table>

**Symbols**

- A = cross-section area of stiffener and associated plating
- A<sub>n</sub> = average face bar area over length of line element
- A<sub>p</sub> = cross-section area of associated plating
- I = moment of inertia of stiffener and associated plating
- Y<sub>0</sub> = distance of neutral axis of stiffener and associated plating from median plane of plate
- r = radius of gyration = \sqrt{\frac{I}{A}}

### Section 3

#### Boundary conditions

#### 3.1 Introduction

3.1.1 The boundary conditions to be applied to the FE model are dependent on the load case to be analysed. Different boundary conditions need to be applied for symmetric, asymmetric and anti-symmetric load cases.

3.1.2 The boundary conditions described in this Section include the different requirements for full-breadth and half-breadth FE ship models for the upright load cases and the requirements for a full-breadth model for the transverse load cases. It is recommended that a full-breadth model is used for the transverse static and dynamic load cases as this simplifies the loading.
boundary conditions and results post-processing. However, if a half-breadth model is used for these transverse load cases, guidance on suitable boundary conditions is given in Ch 5, 1.1 General.

3.1.3 The boundary conditions suitable for each load case are shown in Table 2.3.1 Boundary conditions for full ship model.

3.1.4 The boundary conditions described in this Section are preferred. However, alternative equivalent boundary conditions may be used.

Table 2.3.1 Boundary conditions for full ship model

<table>
<thead>
<tr>
<th>Load case</th>
<th>Boundary conditions</th>
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<tbody>
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<td>See Ch 2, 3.2 Symmetrical boundary conditions for global loads and Figure 2.3.1 Boundary conditions for the application of symmetric global loads</td>
</tr>
<tr>
<td>• Bending moment sub-load cases</td>
<td></td>
</tr>
<tr>
<td>• Wave crest/trough sub-load cases</td>
<td>See Ch 2, 3.3 Symmetrical boundary conditions for local loads and Figure 2.3.2 Boundary conditions for the application of symmetric local loads</td>
</tr>
<tr>
<td>Vertical dynamic load cases, see Ch 2, 4.3 Vertical dynamic load cases</td>
<td>See Ch 2, 3.2 Symmetrical boundary conditions for global loads and Figure 2.3.1 Boundary conditions for the application of symmetric global loads</td>
</tr>
<tr>
<td>Static heel load cases, see Ch 2, 4.4 Static heel load cases</td>
<td>See Ch 2, 3.4 Asymmetric boundary conditions for transverse loads, Inertia Relief solution is recommended or alternative boundary conditions given in Figure 2.3.4 Alternative boundary conditions for transverse cases for a full-breadth model if an Inertia Relief solution cannot be applied</td>
</tr>
<tr>
<td>Transverse dynamic load cases, see Ch 2, 4.5 Transverse dynamic load cases</td>
<td>See Ch 2, 3.4 Asymmetric boundary conditions for transverse loads, Inertia Relief solution is recommended or alternative boundary conditions given in Figure 2.3.4 Alternative boundary conditions for transverse cases for a full-breadth model if an Inertia Relief solution cannot be applied</td>
</tr>
<tr>
<td>Collision load cases, see Ch 2, 4.6 Collision load cases</td>
<td>See Figure 2.3.1 Boundary conditions for the application of symmetric global loads</td>
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<tr>
<td>Tank test condition, see Ch 2, 4.7 Tank test condition</td>
<td>See Ch 2, 3.2 Symmetrical boundary conditions for global loads and Figure 2.3.1 Boundary conditions for the application of symmetric global loads</td>
</tr>
<tr>
<td>Flotation load cases, see Ch 2, 4.8 Flotation load cases</td>
<td>See Ch 2, 3.3 Symmetrical boundary conditions for local loads and Figure 2.3.2 Boundary conditions for the application of symmetric local loads and Figure 2.3.3 Alternative boundary conditions for the application of symmetric local loads</td>
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3.2 Symmetrical boundary conditions for global loads

3.2.1 Symmetric boundary conditions suitable for the analysis of global loads are shown in Figure 2.3.1 Boundary conditions for the application of symmetric global loads. These boundary conditions allow the FE model to deflect globally under the action of hull girder vertical shear forces and bending moments.

3.3 Symmetrical boundary conditions for local loads

3.3.1 Symmetric boundary conditions suitable for the analysis of local loads are shown in Figure 2.3.2 Boundary conditions for the application of symmetric local loads. Vertical grounded springs are to be distributed to nodes of the elements in the side shell in way of the transverse bulkheads. The distribution may be in proportion to the projected vertical cross-sectional area of the elements. The spring stiffness, \( k_s \), may be obtained as:
\[ k_s = G \cdot \frac{A}{T} \]

where

\[ G = \text{modulus of rigidity} \]

\[ l = \text{distance between transverse bulkheads} \]

\[ A = \text{projected vertical cross-sectional area of the element in the side shell} \]

3.3.2 Alternatively, instead of the application of grounded springs as described in Ch 2, 3.3 Symmetrical boundary conditions for local loads 3.3.1, vertical balance forces, which may be obtained as the reaction force at the deck constraints, may be distributed to nodes of the elements in the side shell in way of the transverse bulkheads as shown in Figure 2.3.3 Alternative boundary conditions for the application of symmetric local loads. The distribution may be in proportion to the projected vertical area of the elements.

3.3.3 These boundary conditions remove the effects of hull girder bending from the FE model and are therefore only suitable for calculating stresses resulting from local loads.

3.4 Asymmetric boundary conditions for transverse loads

3.4.1 For a full-breadth model, it is recommended that an Inertia Relief solution is used for removing rigid body motion for these cases. The position of the reference points is not critical providing that the point selected has stiffness in the required degrees of freedom and the complete set describes the rigid body motion. The centre of gravity of the ship may be selected as a reference point. The feature of this solution sequence is that any out-of-balance loads are reacted by inertial forces acting on the mass elements of the model, the reference points are only reference values and hence no inappropriate stress and deflections are generated at these positions. However, if the FE package being used does not provide this facility, then alternative boundary conditions given in Figure 2.3.4 Alternative boundary conditions for transverse cases for a full–breadth model if an Inertia Relief solution cannot be applied may be used.

3.4.2 For a half-breadth model, two load cases need to be considered where the symmetric and anti-symmetric load components are applied separately. These separated load components are then applied to the FE model with symmetric and anti-symmetric boundary conditions respectively. See Ch 5 Alternative Procedures for Transverse Load Cases.
Figure 2.3.2 Boundary conditions for the application of symmetric local loads

- Vertical grounded springs are to be distributed to nodes of the elements in the side shell in way of the transverse bulkheads. Where a double cofferdam bulkhead is fitted, the ground springs may be distributed only to the watertight bulkhead.

- **Centreline Plane**
  - Symmetry constraints: \( \delta_z = 0 \) apply to all grid points on the ship’s structure and cargo tank’s structure (only applicable for a half-breadth model)

- **Points**
  - A: \( \delta_x = \delta_y = \delta_z = 0 \) at the A.P. of the keel or transom, where appropriate, on the centreline
  - B: \( \delta_z = 0 \) at the deck on the centreline at the aft end
  - C: \( \delta_z = 0 \) at the F.P. of the keel on the centreline

---

Figure 2.3.3 Alternative boundary conditions for the application of symmetric local loads

- Balance forces to be applied to both port and starboard sides in a full breadth model. Alternatively, instead of balance forces, grounded springs may be applied.

- **Centreline Plane**
  - Symmetry constraints: \( \delta_x = \delta_y = \delta_z = 0 \) apply to all grid points on the ship’s structure and cargo tank’s structure (only applicable for a half-breadth model)

- **Points**
  - A: \( \delta_x = \delta_y = \delta_z = 0 \) at the A.P. of the keel or transom, where appropriate, on the centreline
  - B: \( \delta_z = 0 \) at the deck on the centreline at the aft end
  - C: \( \delta_z = 0 \) at the F.P. of the keel on the centreline
Section 4
Loading conditions

4.1 Introduction
4.1.1 This Section specifies the standard load cases which are to be considered in the stress and buckling assessments. These include load components arising from static and dynamic effects.

4.1.2 Some of the standard load cases in this Section may not need to be examined if the ship is not to operate in such loading conditions. In this case, a note is to be included in the Loading Manual stating that these loading conditions are not permitted. The load cases to be analysed should be discussed and agreed with LR at the earliest opportunity.

4.1.3 Additional load cases may need to be examined if the ship is in an unusual configuration or is to be operated in conditions which would give rise to higher stresses. If the ship’s Loading Manual contains conditions in which ballast tanks in way of empty cargo tanks are empty or have reduced filling level, then these conditions are to be analysed.

4.1.4 The fully loaded conditions are defined on the basis that the ship’s scantling draught is not significantly different from the operating draught. If this is not so, then special consideration will be given.

4.1.5 The assigned permissible still water bending moment, $M^\text{s,w}$, to be used in the analysis may be less than the Rule permissible still water bending moments, $M^\text{s}$, $M^\text{s,w}$ is not to be taken as less than 0.25 times $M^\text{s}$. The values of $M^\text{s,w}$ used in the analysis are to be incorporated into the ship’s Loading Manual and loading instrument as the assigned permissible still water bending moment values, see Figure 2.4.1 Permissible still water bending moment envelopes $M^\text{s,w}$.

4.2 Wave load cases
4.2.1 The wave load cases to be analysed are specified in Table 2.4.1 Wave load cases. Each wave load case consists of two sub-load cases; bending moment sub-load case and local wave crest/trough sub-load case.

4.2.2 The bending moment sub-load cases are based on the ship’s operating loading conditions specified in Table 2.4.1 Wave load cases and includes all static load components. The Rule design vertical wave bending moment and the permissible
vertical still water bending moment envelope, \( M_{SW} \) (see Ch 2, 4.1 Introduction 4.1.5 and Ch 1, 2.1 Definition), are to be applied. Explanatory notes for the application of the vertical bending moments are given in Ch 2, 4.12 Application of assigned permissible still water and design vertical wave bending moment envelope. All deadweight and lightweight items are to be applied. The ship is balanced on a trimmed waterline.

4.2.3 For the local wave crest/trough sub-load cases, external pressure due to a local wave crest or wave trough is applied to the model, see Ch 2, 4.13 Procedure to apply local wave crest or trough. The model is balanced by vertical grounded springs or vertical forces distributed to the grid points on the side shell in way of the transverse bulkheads; the procedure is described in Ch 2, 3.3 Symmetrical boundary conditions for local loads. No other loads are to be applied.

4.3 Vertical dynamic load cases

4.3.1 The following loads are to be applied to the FE model:

- External hydrostatic pressures due to the quasi-static trimmed waterline.
- Cargo pressure loads acting on the cargo tank structure. These loads are to include design cargo vapour pressure, static pressure and dynamic pressure due to vertical and longitudinal accelerations.
- Inertia forces of lightship mass and other major deadweight items (see Ch 2, 4.3 Vertical dynamic load cases 4.3.7) are to include vertical and longitudinal acceleration factors.
- Rule design vertical wave bending moment distribution.
- Permissible vertical still water bending moment envelope, \( M_{SW} \), see Ch 2, 4.1 Introduction 4.1.5 and Ch 1, 2.1 Definition.

Explanatory notes for the application of these load components are given in Ch 2, 4.9 Application of loads to Ch 2, 4.12 Application of assigned permissible still water and design vertical wave bending moment envelope.

4.3.2 The vertical dynamic load cases to be analysed are specified in Table 2.4.2 Vertical dynamic load cases.

4.3.3 For the vertical dynamic load cases, two quasi-dynamic conditions are to be considered.

(a) Vertical dynamic condition 1: Bow pitched down, which loading condition reflects the ship at:
- maximum downward heave;
- maximum bow down pitch and hence deep draught forward; and
- maximum downwards inertial acceleration over the forward end.

(b) Vertical dynamic condition 2: Bow pitched up, which loading condition reflects the ship at:
- maximum downward heave;
- maximum bow up pitch and hence deep draught aft; and
- maximum downwards inertial acceleration over the aft end.

4.3.4 The longitudinal and vertical acceleration factors are to be calculated as described in Ch 2, 4.10 Calculation of acceleration factors.

4.3.5 The longitudinal and vertical acceleration factors to be applied to a particular cargo tank for a given loading condition in Table 2.4.2 Vertical dynamic load cases are to be taken as the maximum acceleration factors for that cargo tank from the loading conditions in the ship’s Loading Manual with the same loading pattern. Attention is to be given to ensure the correct maximum vertical accelerations are used for bow pitched up and bow pitched down cases, see Ch 2, 4.10 Calculation of acceleration factors 4.10.3.

4.3.6 The acceleration factors to be applied to other large volume tanks, such as ballast tanks and fuel oil tanks, are to be determined in a similar manner as given in Ch 2, 4.3 Vertical dynamic load cases 4.3.4 and Ch 2, 4.3 Vertical dynamic load cases 4.3.5. However, the loading conditions used to determine the acceleration values can be those used to determine the cargo tank acceleration factors. In calculating the acceleration factors the centre of gravity of the particular tank is to be used. It is not necessary to apply acceleration factors to small tanks such as fresh water tanks, dally-use tanks, lubrication oil tanks etc.

4.3.7 Longitudinal and vertical acceleration factors are also to be applied to the ship’s lightweight. To simplify the application of these acceleration factors, large equipment items should be represented as mass elements. If the FE package being used does not support the application of varying acceleration values, alternative methods should be discussed with LR.

4.3.8 These load cases are to be evaluated using quasi-static techniques. The derived values of vertical acceleration are applied to all deadweight and lightweight items and the resulting dynamic condition is balanced on a trimmed waterline. The effect of longitudinal acceleration may be ignored in the determination of the balanced trimmed waterline.
4.4 Static heel load cases

4.4.1 The static heel load cases to be analysed are specified in Table 2.4.3 Static heel load cases. These load cases are for the compliance with the IGC Code requirement that tank and tank support structures are to be able to sustain a 30° static heel condition. The analysis is to be carried out based on full load condition.

4.4.2 Static load due to all deadweight and lightweight items is to be applied. The ship is to be heeled at an angle of 30 degrees and balanced vertically on a trimmed waterline.

4.4.3 No additional vertical bending moment is required to be applied to the FE model, i.e., the vertical bending moment in the model is equal to the still water bending moment generated by the loads when the model is heeled. However, if the actual hull girder bending moment achieved in the model exceeds $M_{sw} + 0.6M_{sw}$, then correcting the bending moment in the model to $M_{sw} + 0.6M_{sw}$ is permitted.

4.5 Transverse dynamic load cases

4.5.1 The following loads are to be applied to the FE model:

- External hydrostatic pressures due to the trimmed and heeled waterline. The pressure head distribution is given in Figure 2.4.4 Hydrostatic pressure distribution for asymmetric load cases using a full-breadth model.
- Cargo pressure loads acting on the cargo tank structure. These loads are to include design cargo vapour pressure, static pressure and dynamic pressure due to vertical and transverse accelerations.
- Inertia forces of lightship mass and other deadweight items are to include transverse and vertical acceleration factors, see Ch 2, 4.5 Transverse dynamic load cases 4.5.11.
- Bending moment correction, if required, see Ch 2, 4.5 Transverse dynamic load cases 4.5.13. Explanatory notes regarding the derivation and application of these load components are given in Ch 2, 4.12 Application of assigned permissible still water and design vertical wave bending moment envelope.

4.5.2 The transverse dynamic load cases to be analysed are specified in Table 2.4.4 Transverse dynamic load cases. The analysis is to be carried out based on two ship loading configurations, i.e. alternate load 1 (odd numbered cargo tanks loaded; even numbered cargo tanks empty) and alternate load 2 (even numbered cargo tanks loaded; odd numbered cargo tanks empty). If it is clearly stated in the ship’s Loading Manual that partly loaded conditions, such as alternate loading conditions, single tank loading conditions or one tank only empty loading conditions, are prohibited in a seagoing condition, then a full load condition can be used for the analysis.

4.5.3 The transverse acceleration (i.e. $A_y$ at point I of the acceleration ellipsoid in Figure 2.4.2 Acceleration ellipsoid) to be used to derive the resultant acceleration of a given tank is to be the maximum transverse acceleration for that tank from all the loading conditions in the ship’s Loading Manual, including any single tank loading condition. When calculating the transverse acceleration factors, $A_y$, for a given tank, the maximum GM from the ship’s loading conditions, which will result in the maximum transverse acceleration component, is to be used. Therefore the GM of a given tank used for the calculation of $A_y$ for the application to the FE analysis may be related to different loading conditions in the Loading Manual. This GM is to be selected by reviewing all alternate tank and single tank loading conditions in the Loading Manual. For example, the GM for the alternate 1 loading condition may be less than the GM for the No.1 or No.3 tank single loading condition, in which case the higher value should be used for the Alternate 1 load case analysis. Similarly, the GM for the alternate 2 loading condition may be less than the GM for the No.2 or No.4 tank single loading condition, in which case the higher value should be used in the alternate 2 load case analysis. Single tank loading conditions are to be considered unless it is specifically stated in the ship’s Loading Manual that single tank loading conditions are prohibited.

4.5.4 For each loading configuration, the vertical and longitudinal accelerations (i.e. $A_z$ at point II and $A_x$ at point III of the acceleration ellipsoid in Figure 2.4.2 Acceleration ellipsoid) to be used to derive the resultant acceleration of a given tank are to be the maximum vertical downward acceleration and maximum longitudinal acceleration (i.e. the maximum of the bow pitched up and bow pitched down load cases) of each tank for the same loading configuration, as determined in accordance with Ch 2, 4.3 Vertical dynamic load cases.

4.5.5 The combination of longitudinal, transverse and vertical acceleration components is to be in accordance with the acceleration ellipsoid concept as described in Ch 4, Sec 3, 4.28.1 of the Rules for Ships for Liquefied Gases. This acceleration ellipsoid is shown in Figure 2.4.2 Acceleration ellipsoid.

4.5.6 Depending on the phase relationship between longitudinal, transverse and vertical acceleration components, the resultant acceleration vector, $a_R$, will have varying magnitude and direction $\beta$. These relationships are given by the acceleration ellipsoid, see Figure 2.4.2 Acceleration ellipsoid. The acceleration ellipsoid can be represented by the following equation:
\[
\frac{a_x^2}{A_x^2} + \frac{a_y^2}{A_y^2} + \frac{(a_z - 1)^2}{A_z^2} = 1
\]

where \(a_x, a_y, a_z\) are instantaneous longitudinal, transverse and vertical acceleration factors on the surface of the acceleration ellipsoid.

- when \(a_x = 0\) and \(a_z = 1\),
  
  Point I represents the maximum transverse acceleration factor \(a_y = A_y\)

- when \(a_y = 0\) and \(a_x = 0\),
  
  Point II represents the maximum vertical acceleration factor \(a_z = 1 + A_x\)

- when \(a_y = 0\) and \(a_x = 1\),
  
  Point III represents the maximum longitudinal acceleration factor \(a_x = A_x\)

4.5.7 The pressures calculated from Ch 2, 4.9 Application of loads 4.9.3 gives the instantaneous pressures at the tank boundary. The maximum pressure acting at individual points on the tank boundary in general does not occur simultaneously, thus, the combination of \(a_x, a_y, a_z\) that generates the maximum pressure at individual points on the tank boundary may vary. While these pressures should be used for the design of the local structures, the primary structures; e.g. transverse web, transverse bulkheads, stringers girders, inner hull plating, deck and shell plating; requires to be designed to also meet the most onerous condition in which the pressure act simultaneously. A number of load cases need to be investigated to show compliance with relevant stress and buckling criteria when the tanks are subjected to combination of \(a_x, a_y, a_z\); the derivation of these cases is described in Ch 2, 4.5 Transverse dynamic load cases 4.5.8 to Ch 2, 4.5 Transverse dynamic load cases 4.5.12.

4.5.8 All design conditions in which internal pressure distribution is calculated with an angle which produces the maximum pressure at the supporting location of each cradle located between the intersection of inner bottom and hopper plate and the side shell plating, are to be considered. As example, the following design cases with the structural arrangement on the web section of the cradle are to be examined, see Figure 2.4.3 Definition of concerned points of tank A through H.

- P1: condition in which an internal pressure distribution is calculated using a combination of \(a_x, a_y, a_z\) to produce the maximum pressure at point A or point E, whichever is the highest.
- P2: condition in which an internal pressure distribution is calculated using a combination of \(a_x, a_y, a_z\) to produce the maximum pressure at point B or point F, whichever is the highest.
- P3: condition in which an internal pressure distribution is calculated using a combination of \(a_x, a_y, a_z\) to produce the maximum pressure at point C or point G, whichever is the highest.
- P4: condition in which an internal pressure distribution is calculated using a combination of \(a_x, a_y, a_z\) to produce the maximum pressure at point D or point H, whichever is the highest.

For a tank with uniform cross section, where the maximum pressure at forward and aft sections of the tank is equal, the points at the forward section are to be considered. For each of these conditions, there is a specific set of values of \(a_y\) and angle \(\beta\) to be applied to generate the maximum pressure at the point considered on the boundary of each tank, where

\[a_\beta\] is the resultant acceleration vector of \(a_x, a_y\) and \(a_z\).

\(\beta\) is the angle of the resultant acceleration vector as shown in Figure 2.4.2 Acceleration ellipsoid. When the cradle arrangement of a ship is different, the points checked for the maximum pressure are to be agreed with LR before commencement.

4.5.9 The design conditions described in Ch 2, 4.5 Transverse dynamic load cases 4.5.8 are to be considered for each loading configuration specified in Ch 2, 4.5 Transverse dynamic load cases 4.5.2.

4.5.10 External sea pressure applied to the model is to be taken as hydrostatic pressure calculated based on the ship heeled at an angle equal to the lesser of the following angles and vertically balanced on a trimmed waterline:

- \(\beta_{average}\)’
- 30 degrees,
- Angle of heel equivalent to upper deck edge immersion at zero trim.
4.5.11 Inertia loads due to lightship mass and other deadweight items, other than contents in cargo tanks, are to be included by applying vertical and transverse acceleration factors,

\[ a_z = a_{\text{average}} \cos(\beta_{\text{average}}) \]

and

\[ a_y = a_{\text{average}} \sin(\beta_{\text{average}}) \]

These acceleration factors include the effect of gravity.

4.5.12 The value of \( a_{\text{average}} \) and \( \beta_{\text{average}} \) is to be calculated as follows:

\[
a_{\text{average}} = \frac{1}{n} \left( \sum_{i=1}^{n} a_{\beta 0_i} \cos \beta_{0_i} \right)^2 + \left( \sum_{i=1}^{n} a_{\beta 0_i} \sin \beta_{0_i} \right)^2
\]

\[
\tan(\beta_{\text{average}}) = \frac{\sum_{i=1}^{n} a_{\beta 0_i} \sin \beta_{0_i}}{\sum_{i=1}^{n} a_{\beta 0_i} \cos \beta_{0_i}}
\]

where

- \( n \) = the number of loaded cargo tanks
- \( a_{\beta 0_i} \) = the resultant vertical and transverse acceleration factor that generates the maximum pressure at the point considered on the boundary of loaded cargo tank \( i \), see Figure 2.4.2 Acceleration ellipsoid
- \( \beta_{0_i} \) = the angle of the resultant acceleration factor \( a_{\beta 0_i} \), see Figure 2.4.2 Acceleration ellipsoid

4.5.13 The analysis of the transverse dynamic load cases may be based on the actual bending moment resulting from the application of loads to the FE model. If, however, the actual hull girder bending moment exceeds \( M_{\text{sw}} + 0.6 M_{\text{w}} \), then correcting the bending moment in the model to \( M_{\text{sw}} + 0.6 M_{\text{w}} \) is permitted.

4.6 Collision load cases

4.6.1 This load case is to consider the capability of the tank supporting structures to withstand the collision load on the model surface corresponding to one half the weight of the tank and cargo in the forward direction and one quarter the weight of the tank and cargo in the aft direction according to LR’s Rules for Ships for Liquefied Gases.

4.6.2 The transverse swash bulkheads, where fitted, should be able to withstand half the collision load as described in Ch 2, 4.6 Collision load cases 4.6.1.

4.6.3 The collision load cases to be analysed are specified in Table 2.4.5 Collision load cases. These load cases are to be analysed based on full load condition.

4.6.4 All static load components and external hydrostatic pressures due to the static waterline for these conditions are to be applied. The cargo design vapour pressure is to be applied.

4.6.5 No additional vertical bending moment is required to be applied to the FE model, i.e. the vertical bending moment in the model is equal to the still water bending moment generated by the applied loads.

4.6.6 The pressure at the tank boundary is to be calculated according to Ch 2, 4.9 Application of loads 4.9.3.

4.6.7 The following two scenarios are to be considered:

- Friction forces based on dynamic friction coefficient.
- Friction forces based on static friction coefficient.

4.7 Tank test condition

4.7.1 These case(s) are to consider the ship in the actual loading conditions when the tank test procedures are undertaken. It may be necessary to analyse load cases for the testing of each cargo tank separately. All still water loads and external pressure due to the actual test conditions are to be applied. The pressures in the tanks are to correspond to the test values. The tank test case(s) to be analysed are specified in Table 2.4.6 Special load cases.
4.7.2 The actual ship conditions proposed for the tests of cargo tanks 1 and 2 must be used together with the lightest
draughts chosen for each condition.
4.7.3 The cargo pressures in each tank are to represent the test pressure and all still water load items are to be applied.
External hydrostatic pressures due to the still waterline are to be applied.

4.8 Flotation load cases
4.8.1 The objective of the load case is to examine the yield and buckling capability for the topside tank and transverse in way
of anti-floating chock to withstand the flotation load in the cargo holds.
4.8.2 The flotation load cases to be analysed are specified in Table 2.4.6 Special load cases. The load case has been greatly
simplified to include only the forces acting on the anti-flotation chocks when they are seated on top of the cargo tanks. A load
case to study each tank individually may be required.
4.8.3 Only the void space between the inside of the hold and the outside of the cargo tank boundaries is assumed to be
flooded up to the scantling draught. The cargo tanks are to be empty. External buoyancy, deadweight and lightweight items and all
other loads are not to be applied.
4.8.4 This load case is not based on an actual loading condition. If damage stability calculations are available, then the depth
of flooding from the damage stability calculations may be used in lieu of the scantling draught.

4.9 Application of loads
4.9.1 All components of a loading condition are to be included in the analysis. The lightship is to be included by adjusting the
self-weight of the model to equal the required lightweight and LCG position. Acceleration factors are to be applied to lightship
mass items, including the ship’s engine, in the case of dynamic load cases. See also Ch 2, 2.1 FE modelling 2.1.13.
4.9.2 Buoyancy loads are to be applied as pressures, \( \rho gh \), to wetted shell elements, where \( h \) is the vertical distance from the
waterline to the centre of the element. See Figure 2.4.9 Pressure head distributions for local wave crest or trough.
4.9.3 Cargo loads, including the design vapour pressure, static and dynamic loads due to additional acceleration factors, are
to be applied as pressures directly to the elements representing the tank plating. The following equations are to be used to
determine the pressure values:

For still water cases:
\[
P = \rho_c A_p g h z + P_0
\]

For vertical dynamic cases:
\[
P = \rho_c A_p g h z \left( 1 + A_x \right) + \rho_c A_p g l x A_x + P_0
\]

For transverse dynamic cases:
\[
P = \rho_c A_p g Z \beta + P_0
\]

For forward collision case:
\[
P = \rho_c A_p g h z + \rho_c A_p 0.5 g l x + P_0
\]

For aft collision case:
\[
P = \rho_c A_p g h z + \rho_c A_p 0.25 g l x + P_0
\]

where

\[
P_0 = \text{is the designed vapour pressure which is generally to be taken as given in the specification and to be not less then:}
\]
\[
= 2 + AC \left( \rho \right)^{1.5} \text{ (bar)}
\]

where

\[
A = 0.0185 \left( \frac{\sigma_m}{A \sigma} \right)^2
\]

with:

Lloyd’s Register
\(\sigma_m = \) design primary membrane stress

\(\Delta\sigma_A = \) allowable dynamic membrane stress (double amplitude at probability level \(Q = 10^{-8}\))

- \(55 \text{ N/mm}^2\) for ferritic-perlitic, martensitic and austenitic steels,
- \(25 \text{ N/mm}^2\) for aluminium alloy (5083-O).

\(C = \) a characteristic tank dimension to be taken as the greatest of the following:

- \(\text{Max}(h; 0.75b; 0.45l)\)

with

- \(h = \) height of tank (dimension in ship’s vertical direction), in metres,
- \(b = \) width of tank (dimension in ship’s transverse direction), in metres,
- \(l = \) length of tank (dimension in ship’s longitudinal direction), in metres.

\(\rho_r = \) the relative density of the cargo (\(\rho_r = 1\) for fresh water) at the design temperature

\(l_x = \) is the following reference distance:

- for bow pitched down vertical dynamic case and the forward collision case, the longitudinal distance from the aft end of tank to centre of element,
- for bow pitched up vertical dynamic case and the aft collision case, the longitudinal distance from the forward end of tank to centre of element.

\(h_z = \) is the vertical distance from the highest point of a tank to centre of element (see Figure 2.4.6 Static pressure load distribution \(P\) for cargo tanks).

\(a_{\beta} = \) is the resultant acceleration factor vector (at angle \(\beta\)) that generates the maximum pressure at a required point on the tank boundary, see Ch 2, 4.5 Transverse dynamic load cases 4.5.7 and Ch 2, 4.5 Transverse dynamic load cases 4.5.8. The acceleration vectors required to maximise the pressure at a given point on the tank boundary depend on tank geometry, hence they are in general different for each cargo tank. These maximum pressures may be determined using LR’s RulesCalc software.

\(Z_{\beta} = \) is the largest liquid height above the point where the pressure is to be determined measured from the tank shell in the \(\beta\) direction, see Figure 2.4.5 Internal tank pressure for transverse load cases.

\(\Lambda_P = \leq 1.0\), a factor to account for the difference in cargo tank volume measured from the primary barrier and that measured to the ship structure. For tanks with insulation fitted to their external surface such as Type A tanks, \(\Lambda_P\) is to be taken as 1.0.

4.9.4 Tank domes considered to be part of the accepted total tank volume shall be taken into account when determining \(Z_{\beta}\) and \(h_z\), unless the total volume of tank domes \(V_d\) does not exceed the following value:

\[V_d = V_t \left(1 - \frac{FL}{V_t}ight)\]

where

- \(V_t = \) is the tank volume without any domes.
- \(FL = \) is the filling limit in accordance with Ch 15 of LR’s Rules for Liquefied Gases.

4.10 Calculation of acceleration factors

4.10.1 The maximum longitudinal and transverse acceleration factors may be obtained using the following guidance formulae.
where

\[ z = \text{is the vertical distance, in metres, from the ship's actual waterline to the centre of gravity of tank with contents; } z \text{ is positive above and negative below the waterline} \]

\[ x = \text{is the longitudinal distance, in metres, from amidships to the centre of gravity of the tank with contents; } x \text{ is positive forward of amidships, negative aft of amidships} \]

\[ B = \text{is the greatest moulded breadth of the ship, in metres} \]

\[ K = 13GM/B, \text{ where } K \geq 1 \text{ and } GM \text{ is the metacentric height, in metres} \]

\[ A_0 = \text{See Ch 2, 4.10 Calculation of acceleration factors 4.10.3} \]

\[ C_b = \text{is the block coefficient, see Ch 2, 4.10 Calculation of acceleration factors 4.10.5} \]

4.10.2 LR's RulesCalc software may be used to calculate the longitudinal and transverse accelerations at the centre of each cargo tank and also at the selected positions along the ship length.

4.10.3 The guidance vertical acceleration formula given in the Rules for Ships for Liquefied Gases is modified, as below, to maintain a consistent acceleration curve over the model length. This modification takes account of the fact that the pitch motion in the aft end of the ship results in a vertical acceleration component acting in the opposite direction to that at the forward end of the ship. The following formulae may be used to obtain the vertical acceleration factors.

For heave downward and bow pitch down case:

- For \( x \geq -0.05L \)
  \[ A_z = A_0 \left[ 1 + \left( \frac{5.3 - 45L}{L} \right)^2 \left( \frac{x}{L} - 0.05 \right)^2 \left( \frac{0.6}{C_b} \right)^{1.5} + \left( \frac{0.6yK^{1.5}L}{B} \right)^2 \right] \]

- For \( x < -0.05L \)
  \[ A_z = A_0 \left[ 2 \left( \frac{0.6yK^{1.5}L}{B} \right)^2 - 1 + \left( \frac{5.3 - 45L}{L} \right)^2 \left( \frac{x}{L} + 0.05 \right)^2 \left( \frac{0.6}{C_b} \right)^{1.5} + \left( \frac{0.6yK^{1.5}L}{B} \right)^2 \right] \]

For heave downward and bow pitch up case:

- For \( x \geq -0.05L \)
  \[ A_z = A_0 \left[ 2 \left( \frac{0.6yK^{1.5}L}{B} \right)^2 - 1 + \left( \frac{5.3 - 45L}{L} \right)^2 \left( \frac{x}{L} + 0.05 \right)^2 \left( \frac{0.6}{C_b} \right)^{1.5} + \left( \frac{0.6yK^{1.5}L}{B} \right)^2 \right] \]

- For \( x < -0.05L \)
  \[ A_z = A_0 \left[ 1 + \left( \frac{5.3 - 45L}{L} \right)^2 \left( \frac{x}{L} - 0.05 \right)^2 \left( \frac{0.6}{C_b} \right)^{1.5} + \left( \frac{0.6yK^{1.5}L}{B} \right)^2 \right] \]

where

\[ A_0 = \frac{0.2V}{\sqrt{L}} + \frac{34 - 600L}{L} \]

\[ x = \text{as defined in Ch 2, 4.10 Calculation of acceleration factors 4.10.1} \]

\[ y = \text{is the transverse distance, in meters, from the ship's centreline to the centre of gravity of the tank with contents} \]

\[ C_b = \text{is the block coefficient, see Ch 2, 4.10 Calculation of acceleration factors 4.10.5} \]

\[ A_z = \text{is the vertical acceleration factor (positive downward)} \]

4.10.4 The distribution of vertical acceleration factor for the case of ship motion of heave downwards/bow pitched down and heave downwards/bow pitched up is illustrated in Figure 2.4.7 Distribution of vertical acceleration factor, \( A_z \).
4.10.5 The block coefficient based on the summer draught or the scantling draught, whichever is greater, may be used for the calculation of the acceleration factors.

4.10.6 Alternatively, direct calculation procedures using an appropriate ship motion program may be used to derive the acceleration factors after consultation with LR.

4.11 Procedure to derive the quasi-static waterline for dynamic load cases

4.11.1 This procedure may be used to calculate the dynamic loads acting on all deadweight and lightweight items to determine the resulting quasi-static external pressure distribution acting on the shell plating.

4.11.2 The longitudinal weight distribution is to be broken down into convenient longitudinal sections for all lightweight and deadweight items in a similar way to that required for a still water loads analysis.

4.11.3 A vertical acceleration factor (relative to $g$) at the longitudinal centre of gravity of each section is to be calculated and added to the static gravity of $g$.

4.11.4 Each section of lightweight and deadweight is to be multiplied by its corresponding vertical acceleration factor to give the combined static and dynamic weight distribution, and this is to be balanced on a suitable waterline using a still water loads program. This waterline should not include any added wave profile.

4.11.5 The resulting quasi-static trimmed waterline is to be used to apply the external hydrostatic pressures to shell plating elements.

4.11.6 For static heel and dynamic transverse cases, the external hydrostatic pressure is to be based on the ship heeled at the required angle and balanced vertically on a trimmed waterline. This position may be obtained by first achieving the vertical balance of the upright ship and then heeling the ship at the required heel angle.

4.12 Application of assigned permissible still water and design vertical wave bending moment envelope

4.12.1 Where required, the vertical wave bending moment and permissible vertical still water bending moment envelope are to be applied to the FE model.

4.12.2 The additional bending moment distribution that is required to be applied to the FE model to generate the permissible still water and Rule design vertical wave bending moments is illustrated in Figure 2.4.8 Procedure to derive required VBM distribution for applying to FE model. This bending moment distribution takes account of the bending moment generated by the loading condition of the FE load case. The total bending moment, i.e. the sum of the applied additional bending moment and the still water bending moment from the FE load case, need not exceed the required value. Care is to be taken in the sign convention of sagging and hogging in deriving the required bending moment distribution.

4.12.3 The vertical load distribution that is required to produce the bending moment distribution can be obtained by numerical differentiation method. The load distribution calculated is to be approximated by a series of vertical forces acting along the length of the FE model. These vertical forces are to be applied as a series of nodal forces at the side shell and inner skin in proportion to the projected vertical cross-sectional area. The distribution of the vertical forces is to be such that the required bending moment distribution can be closely reproduced. It is recommended that the nodal forces be applied to every web frame position.

4.12.4 Other proposed methods of applying the Rule vertical wave bending moment distribution and permissible vertical still water bending moment envelope will be specially considered.

4.13 Procedure to apply local wave crest or trough

4.13.1 For the wave load cases, an additional wave head is to be applied over the full length of the FE model using the pressure distribution shown in Figure 2.4.9 Pressure head distributions for local wave crest or trough.

4.13.2 The ship’s scantling draught may be used for deriving the pressure head distribution.

4.14 Support and chock design loads

4.14.1 For the assessment of tank support chocks, an additional load factor is to be applied to the calculated loads to allow unequal chock support distribution due to construction tolerance and the relative flexibility of the cargo tank and the web section of the cradle. A 12 per cent increase in the calculated loads is generally recommended when a 1mm tolerance for the fitting gap is specified on supports made of resin impregnated laminated wood construction. For the application, see Ch 2, 5.1 Permissible stresses 5.1.8.

4.14.2 Allowance for structural misalignment between cargo tank and cradle members can be made using a separate fine mesh model.
4.14.3 A forward longitudinal acceleration will increase loads on fore end supports and decrease them at aft end supports, conversely for an aft acceleration. Therefore, for a parallel sided tank, the scantlings, its support and supporting hull structure should be symmetrical about the mid-tank position; hence, in general, the scantlings over the aft part of the tank, including its support and hull structure in way are not to be less than that calculated for the forward part of the tank.

4.14.4 Proposals for additional load factors, other than the recommended additional load factors, are to be submitted for verification. The additional load factors for supports and chocks made from alternative materials are to be agreed with LR.

4.14.5 A preliminary investigation is to be made on the effect of misalignment between the tank doubler plates and their seatings on the cradle due to a combination of thermal contraction, construction tolerances, and design gap at the chocks. If it is found that this misalignment could cause a significant increase to, or redistribution of, the stress levels in the support chocks or the reinforced structures, then the method of examination is to be discussed and agreed with LR.

4.14.6 Coefficients of chock friction are sensitive to atmospheric dust and humidity, surface finish, velocity of sliding, temperature, vibration and extent of contamination. The friction coefficient in Table 2.4.7 Friction coefficient (reference values) may be used, unless otherwise provided by the makers. For details to modelling the support and chocks, see Ch 4 Modelling of Supports and Chocks.
### Table 2.4.1 Wave load cases

<table>
<thead>
<tr>
<th>Load case</th>
<th>Rule vertical wave bending moment</th>
<th>External pressure</th>
<th>Internal pressure</th>
<th>Additional sub-load cases</th>
<th>Tank loading pattern</th>
<th>Boundary conditions</th>
</tr>
</thead>
<tbody>
<tr>
<td>Full Load</td>
<td>S1.1</td>
<td>$M_{sw}$ Hog</td>
<td>$M_{wv}$ Hog</td>
<td>Balanced waterline</td>
<td>-</td>
<td>(a)</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>See Ch 2, 3.2 Symmetrical boundary conditions for global loads and Figure 2.3.1 Boundary conditions for the application of symmetric global loads</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>(b)</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>See Ch 2, 3.3 Symmetrical boundary conditions for local loads and Figure 2.3.2 Boundary conditions for the application of symmetric local loads and Figure 2.3.3 Alternative boundary conditions for the application of symmetric local loads</td>
</tr>
</tbody>
</table>

#### Tank loading pattern images

- **(a)**
- **(b)**
<table>
<thead>
<tr>
<th>Full Load</th>
<th>S1.2</th>
</tr>
</thead>
<tbody>
<tr>
<td>$M_{lw}$ Sag</td>
<td>$M_{nv}$ Sag</td>
</tr>
<tr>
<td>-</td>
<td>-</td>
</tr>
</tbody>
</table>

Symmetrical boundary conditions for global loads and Figure 2.3.1
Boundary conditions for the application of symmetric global loads

Symmetrical boundary conditions for local loads and Figure 2.3.2
Boundary conditions for the application of symmetric local loads and Figure 2.3.3 Alternative boundary conditions for the application of symmetric local loads

Tank loading pattern images

See Ch 2. 3.2

See Ch 2. 3.3

(c)

(d)
### Analysis of Primary Structures of Type C Liquified Gas Carriers

#### Chapter 2

**Section 4**

<table>
<thead>
<tr>
<th>Alternate 1</th>
<th>( M_{lw} ) Hog</th>
<th>( M_{sv} ) Hog</th>
<th>Balanced waterline</th>
<th>( g, P_0 )</th>
<th>-</th>
<th>(e)</th>
</tr>
</thead>
<tbody>
<tr>
<td>S2.1, see Notes 5 and 7</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>Wave crest</td>
<td>(f)</td>
</tr>
</tbody>
</table>

**Notes:**
- See Ch 2, 3.2 Symmetrical boundary conditions for global loads and Figure 2.3.1 Boundary conditions for the application of symmetric global loads.
- See Ch 2, 3.3 Symmetrical boundary conditions for local loads and Figure 2.3.2 Boundary conditions for the application of symmetric local loads and Figure 2.3.3 Alternative boundary conditions for the application of symmetric local loads.

**Tank loading pattern images**

(e) [Image]

(f) [Image]
Alternate 2

<table>
<thead>
<tr>
<th>$M_{sw}$ Hog</th>
<th>$M_{lv}$ Hog</th>
<th>Balanced</th>
<th>$g$, $P_0$</th>
<th>-</th>
<th>(g)</th>
</tr>
</thead>
<tbody>
<tr>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>Wave crest</td>
<td>(h)</td>
</tr>
</tbody>
</table>

**Notes:**
1. See Ch 2, 3.2 Symmetrical boundary conditions for global loads and Figure 2.3.1 Boundary conditions for the application of symmetric global loads.
2. See Ch 2, 3.3 Symmetrical boundary conditions for local loads and Figure 2.3.2 Boundary conditions for the application of symmetric local loads.
3. See Ch 2, 3.3 Alternative boundary conditions for the application of symmetric local loads.

**Tank loading pattern images**

(g) 

(h)
### Heavy Ballast

<table>
<thead>
<tr>
<th></th>
<th>$M_{sw}$ Hog</th>
<th>$M_{sw}$ Hog</th>
<th>Balanced waterline</th>
<th>$g, P_0$</th>
<th>-</th>
<th>(i)</th>
</tr>
</thead>
</table>

- - - - Wave crest \(i\)

See Ch 2, 3.2 Symmetrical boundary conditions for global loads and Figure 2.3.1 Boundary conditions for the application of symmetric global loads

- - - - Wave crest \(i\)

See Ch 2, 3.3 Symmetrical boundary conditions for local loads and Figure 2.3.2 Boundary conditions for the application of symmetric local loads and Figure 2.3.3 Alternative boundary conditions for the application of symmetric local loads

#### Tank loading pattern images

**Note 1.** The Rule design vertical wave bending moment and the permissible vertical still water bending moment envelope, $M_{sw}$, is to be applied to bending moment sub-load cases, see Ch 1, 2.1 Definition 2.1.1 and Ch 2, 4.1 Introduction 4.1.5 for definition of $M_{sw}$.

**Note 2.** All lightweight and deadweight items are to be applied in bending moment sub-load cases.

**Note 3.** External pressure is applied to the wave crest and wave trough sub-load cases. No lightweight, deadweight and other load items are to be applied. See Ch 2, 4.2 Wave load cases 4.2.3 and Ch 2, 4.13 Procedure to apply local wave crest or trough for the application of wave crest and wave trough. See Ch 2, 3.3 Symmetrical boundary conditions for local loads for the application of vertical ground springs and vertical balance forces.

**Note 4.** Bending moment sub-load case and wave crest/trough sub-load case are run separately. The resultant stresses are to be combined by superposition.

**Note 5.** Alternately loaded condition with odd number tanks full and even number tanks empty. For illustration, figure shown is a ship with four cargo tanks.

**Note 6.** Alternately loaded condition with even number tanks full and odd number tanks empty. For illustration, figure shown is a ship with four cargo tanks.

**Note 7.** Full ballast tanks are shown in way of empty cargo tanks. If the ship’s Loading Manual contains conditions in which ballast tanks in way of empty cargo tanks are empty or have reduced filling level, then these conditions are to be analysed.

**Note 8.** A deep draught loading condition with most water ballast tanks filled.
### Table 2.4.2 Vertical dynamic load cases

<table>
<thead>
<tr>
<th>Load case</th>
<th>Still water bending moment</th>
<th>Rule vertical wave bending moment</th>
<th>External pressure</th>
<th>Internal pressure</th>
<th>Additional sub-load cases to apply</th>
<th>Tank loading pattern</th>
<th>Boundary conditions</th>
</tr>
</thead>
<tbody>
<tr>
<td>Full load</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>See Ch 2, 3.2 Symmetrical boundary conditions for global loads and Figure 2.3.1</td>
</tr>
<tr>
<td>V1.1 Bow pitch down, see Note 3</td>
<td>$M_{sw}$ Hog</td>
<td>$M_{w}$ Hog</td>
<td>Hydrostatic pressure due to quasi-static trimmed waterline</td>
<td>$(1+A_3)g, A_5g,$ $P_0$</td>
<td>-</td>
<td></td>
<td>Boundary conditions for the application of symmetric global loads</td>
</tr>
<tr>
<td>V1.2 Bow pitch up, see Note 3</td>
<td>$M_{sw}$ Hog</td>
<td>$M_{w}$ Hog</td>
<td>Hydrostatic pressure due to quasi-static trimmed waterline</td>
<td>$(1+A_3)g, A_5g,$ $P_0$</td>
<td>-</td>
<td></td>
<td>See Ch 2, 3.2 Symmetrical boundary conditions for global loads and Figure 2.3.1</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Boundary conditions for the application of symmetric global loads</td>
</tr>
</tbody>
</table>

#### Tank loading pattern images

![Tank loading pattern images](image_url)
<table>
<thead>
<tr>
<th>Alternate 1</th>
<th>V2.1 Bow pitch down, see Note 4</th>
<th>$M_{sw}$ Hog</th>
<th>$M_{nv}$ Hog</th>
<th>Hydrostatic pressure due to quasi-static trimmed waterline</th>
<th>(1+$A_{z}$)g, $A_{x}$g, $P_0$</th>
<th>-</th>
<th>See Ch 2, 3.2 Symmetrical boundary conditions for global loads and Figure 2.3.1 Boundary conditions for the application of symmetric global loads</th>
</tr>
</thead>
<tbody>
<tr>
<td>V2.2 Bow pitch up, see Note 4</td>
<td>$M_{sw}$ Hog</td>
<td>$M_{nv}$ Hog</td>
<td>Hydrostatic pressure due to quasi-static trimmed waterline</td>
<td>(1+$A_{z}$)g, $A_{x}$g, $P_0$</td>
<td>-</td>
<td>See Ch 2, 3.2 Symmetrical boundary conditions for global loads and Figure 2.3.1 Boundary conditions for the application of symmetric global loads</td>
<td></td>
</tr>
</tbody>
</table>

**Tank loading pattern image**

(b)
### Alternate 2

| V3.1 Bow pitch down, see Note 5 | $M_{sw}$ Hog | $M_w$ Hog | Hydrostatic pressure due to quasi-static trimmed waterline | $(1+A_z)g$,$A_xg$,$P_0$ | - | See Ch 2, 3.2
Symmetrical boundary conditions for global loads and Figure 2.3.1
Boundary conditions for the application of symmetric global loads |
| V3.2 Bow pitch up, see Note 5 | $M_{sw}$ Hog | $M_w$ Hog | Hydrostatic pressure due to quasi-static trimmed waterline | $(1+A_z)g$,$A_xg$,$P_0$ | - | See Ch 2, 3.2
Symmetrical boundary conditions for global loads and Figure 2.3.1
Boundary conditions for the application of symmetric global loads |

### Tank loading pattern image

(c)

**Note 1.** The Rule design vertical wave bending moment and the permissible vertical still water bending moment envelope, $M_{sw}$, are to be applied. See Ch 1, 2.1 Definition 2.1.1 and Ch 2, 4.1 Introduction 4.1.5 for definition of $M_{sw}$.

**Note 2.** All liquefied cargo is to include vertical and longitudinal accelerations. All lightweight and deadweight items, other than liquefied cargo, are to include vertical acceleration factor.

**Note 3.** Vertical and longitudinal acceleration factors of a tank are to be taken as the maximum acceleration factors for that tank from all loading conditions in the ship’s Loading Manual with all cargo tanks filled.

**Note 4.** Vertical and longitudinal acceleration factors of a tank are to be taken as the maximum acceleration factors for that tank from all loaded conditions in the ship’s Loading Manual with odd number tank(s) full and even number tanks empty. For illustration, figure shown is a ship with three cargo tanks.

**Note 5.** Vertical and longitudinal acceleration factors of a tank are to be taken as the maximum acceleration factors for that tank from all loaded conditions in the ship’s Loading Manual with even number tank(s) full and odd number tanks empty. For illustration, figure shown is a ship with three cargo tanks.

**Note 6.** Full ballast tanks are shown in way of empty cargo tanks. If the ship’s Loading Manual contains conditions in which ballast tanks in way of empty cargo tanks are empty or have reduced filling level, then these conditions are to be analysed.
### Table 2.4.3 Static heel load cases

<table>
<thead>
<tr>
<th>Load case</th>
<th>Still water bending moment</th>
<th>Rule vertical wave bending moment</th>
<th>External pressure</th>
<th>Internal pressure</th>
<th>Additional sub-load cases to apply</th>
<th>Tank loading pattern</th>
<th>Boundary conditions</th>
</tr>
</thead>
<tbody>
<tr>
<td>Full Load</td>
<td>SH-1</td>
<td>See Note 2</td>
<td>Hydrostatic pressure due to heel angle of 30 degrees, see Note 1</td>
<td>g, ( P_0 ) See Note 3</td>
<td>-</td>
<td>(a)</td>
<td>An Inertia Relief solution is recommended. If the FE package being used does not provide this facility, then alternative boundary conditions given in Figure 2.3.1 Boundary conditions for the application of asymmetric global loads may be used. See Ch 2, 3.4 Asymmetric boundary conditions for transverse loads.</td>
</tr>
</tbody>
</table>

**Note 1.** The ship is to be heeled at an angle of 30 degrees and balanced vertically on a trimmed waterline. Internal and external loads are to be calculated based on the heeled/trimmed condition. All lightweight items are to be included.

**Note 2.** No additional vertical bending moment is required to be applied to the FE model, i.e. the vertical bending moment in the model is equal to the still water bending moment generated by the loads when the model is heeled. However, if the actual hull girder bending moment achieved in the model exceeds \( M_{sw} + 0.6 M_w \), then correcting the bending moment in the model to \( M_{sw} + 0.6 M_w \) is permitted.

**Note 3.** Static tank pressure due to 30° heel condition is to be applied for the cargo tanks.
Table 2.4.4 Transverse dynamic load cases

<table>
<thead>
<tr>
<th>Load case</th>
<th>Still water bending moment</th>
<th>Rule vertical wave bending moment</th>
<th>External pressure</th>
<th>Internal pressure</th>
<th>Additional sub-load cases to apply</th>
<th>Tank loading pattern</th>
<th>Boundary conditions</th>
</tr>
</thead>
<tbody>
<tr>
<td>T1-P1, see Note 1</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>An Inertia Relief solution is recommended. If the FE package being used does not provide this facility, then alternative boundary conditions given in Figure 2.3.1. Boundary conditions for the application of symmetric global loads may be used. See Ch. 2, 3.4 Asymmetric boundary conditions for transverse loads.</td>
</tr>
<tr>
<td>T1-P2, see Note 2</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>T1-P3, see Note 3</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>T1-P4, see Note 4</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Alternate 1, see Notes 9 and 12

Tank loading pattern image

(a)}
Note 1. Dynamic condition P1: maximum pressure at the corners A or E of cargo tank, see Ch 2, 4.5 Transverse dynamic load cases 4.5.8 and Figure 2.4.3 Definition of concerned points of tank A through H. See also Note 7.

Note 2. Dynamic condition P2: maximum pressure at the corners B or F of cargo tank, see Ch 2, 4.5 Transverse dynamic load cases 4.5.8 and Figure 2.4.3 Definition of concerned points of tank A through H. See also Note 7.

Note 3. Dynamic condition P3: maximum pressure at the corners C or G of cargo tank, see Ch 2, 4.5 Transverse dynamic load cases 4.5.8 and Figure 2.4.3 Definition of concerned points of tank A through H. See also Note 7.

Note 4. Dynamic condition P4: maximum pressure at the corners D or H of cargo tank, see Ch 2, 4.5 Transverse dynamic load cases 4.5.8 and Figure 2.4.3 Definition of concerned points of tank A through H. See also Note 7.

Note 5. The combination of longitudinal, vertical and transverse acceleration components is to be in accordance with the acceleration ellipsoid concept, see Ch 2, 4.5 Transverse dynamic load cases.

Note 6. The ship is to be heeled at the required angle specified in Ch 2, 4.5 Transverse dynamic load cases 4.5.10 and balanced vertically on a trimmed waterline. External pressure applied to the FE model is to be based on the hydrostatic pressure due to the heeled and trimmed waterline.

Note 7. All lightweight and deadweight items are to be applied and to include vertical and transverse acceleration factors. In general, the acceleration vectors $a_x$, $a_y$ and $a_z$ applied to each cargo tank are different to maximise the tank pressure at the tank boundary. An average acceleration factor $a_{\text{average}}$ may be applied to lightship mass and deadweight items other than contents in cargo tanks. See Ch 2, 4.5 Transverse dynamic load cases.

Note 8. The analysis may be based on the actual bending moment generated by the applied loads. No additional hull girder bending moment is to be applied. If however, the actual bending moment exceeds $M_{\text{sw}} + 0.6M_{\text{sw}}$, then correcting the bending moment in the model to $M_{\text{sw}} + 0.6M_{\text{sw}}$ is permitted.

Note 9. Alternate loaded condition with odd number tanks full and even number tanks empty. For illustration, figure shown is a ship with three cargo tanks.

Note 10. Alternate loaded condition with even number tanks full and odd number tanks empty. For illustration, figure shown is a ship with three cargo tanks.

Note 11. Single cargo tank conditions are to be considered to determine the maximum transverse acceleration for each tank unless it is specifically stated in the ship’s Loading Manual that single tank loading conditions are prohibited.

Note 12. Full ballast tanks are shown in way of empty cargo tanks. If the ship’s Loading Manual contains conditions in which ballast tanks in way of empty cargo tanks are empty or have reduced filling level, then these conditions are to be analysed.

Note 13. If it is clearly stated in the ship’s Loading Manual that partly loaded conditions, such as alternate loading conditions, single tank loading conditions or one tank only empty loading conditions, are prohibited in a seagoing condition, then a full load condition can be used for the analysis.
### Table 2.4.5 Collision load cases

<table>
<thead>
<tr>
<th>Load case</th>
<th>Still water bending moment</th>
<th>Rule vertical wave bending moment</th>
<th>External pressure</th>
<th>Internal pressure</th>
<th>Additional sub-load cases to apply</th>
<th>Tank loading pattern</th>
<th>Boundary conditions</th>
</tr>
</thead>
<tbody>
<tr>
<td>Forward Collision O1, see Notes 1, 2 and 4</td>
<td>Actual, see Note 5</td>
<td>-</td>
<td>Hydrostatic pressure due to balanced waterline</td>
<td>$a_x = 0.5g$</td>
<td>$a_z = g, P_0$</td>
<td>-</td>
<td>(a)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Tank loading pattern image</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>See Ch 2, 3.2 Symmetrical boundary conditions for global loads and Figure 2.3.1 Boundary conditions for the application of symmetric global loads</td>
</tr>
<tr>
<td></td>
<td></td>
<td><img src="image" alt="Tank loading pattern image" /></td>
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<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>$a_x = 0.5g$</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Aft Collision O2, see Notes 1, 3 and 4</td>
<td>Actual, see Note 5</td>
<td>-</td>
<td>Hydrostatic pressure due to balanced waterline</td>
<td>$a_x = 0.25g$</td>
<td>$a_z = g, P_0$</td>
<td>-</td>
<td>(b)</td>
</tr>
<tr>
<td></td>
<td></td>
<td><img src="image" alt="Tank loading pattern image" /></td>
<td></td>
<td></td>
<td></td>
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</tr>
<tr>
<td></td>
<td></td>
<td>$a_x = 0.25g$</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>See Ch 2, 3.2 Symmetrical boundary conditions for global loads and Figure 2.3.1 Boundary conditions for the application of symmetric global loads</td>
</tr>
<tr>
<td></td>
<td></td>
<td><img src="image" alt="Tank loading pattern image" /></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

**Notes:****

1. For forward collision, the hydrostatic pressure is due to the balanced waterline.
2. For aft collision, the hydrostatic pressure is also due to the balanced waterline.
3. $P_0$ represents the atmospheric pressure.
4. $a_x$ and $a_z$ represent accelerations in the $x$ and $z$ directions, respectively.
5. See Chapter 2, Section 3.2 for symmetrical boundary conditions for global loads and Figure 2.3.1 for boundary conditions for the application of symmetric global loads.
Forward Collision Assessment of the swash bulkheads O3, see Notes 2, 4, 6 and 7

- Actual, see Note 5
- Hydrostatic pressure due to balanced waterline
- $a_x = 0.5g$
- $a_z = g, F_0$
- (c) See Ch 2, 3.2 Symmetrical boundary conditions for global loads and Figure 2.3.1 Boundary conditions for the application of symmetric global loads

Note 1. A fully loaded condition with all cargo tanks filled and at full load draught. For illustration, figure shown is a ship with three cargo tanks.

Note 2. For forward collision case, a forward acceleration of 0.5g is to be applied in the longitudinal direction to the lightship mass and cargo in tanks.

Note 3. For aft collision case, an aft acceleration of 0.25g is to be applied in the longitudinal direction to the lightship mass and cargo in tanks.

Note 4. Vertical downward acceleration of 1.0g is to be applied to all lightweight and deadweight items.

Note 5. The analysis is based on the actual bending moment generated by the applied loads. No additional hull girder bending moment is to be applied.

Note 6. A loading condition with the cargo tank aft of the swash bulkhead filled and at full load draught.

Note 7. The transverse swash bulkheads, where fitted, should be able to withstand half the collision load as described in Ch 2, 4.6 Collision load cases 4.6.1. However, the transverse swash bulkheads need not be considered in the analysis if the scantlings are greater than half of the end transverse bulkheads, see Ch 2, 4.6 Collision load cases 4.6.2.

Note 8. Two scenarios are to be investigated; friction forces based on dynamic friction coefficient and friction forces based on static friction coefficient, see Ch 2, 4.6 Collision load cases 4.6.7.

Table 2.4.6 Special load cases

<table>
<thead>
<tr>
<th>Load case</th>
<th>Still water bending moment</th>
<th>Rule vertical wave bending moment</th>
<th>External pressure</th>
<th>Internal pressure</th>
<th>Additional sub-load cases to apply</th>
<th>Tank loading pattern</th>
<th>Boundary conditions</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tank test condition</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Tank test condition</td>
<td>Actual</td>
<td>-</td>
<td>Hydrostatic pressure due to balanced waterline</td>
<td>Actual loadings</td>
<td>-</td>
<td>(a)</td>
<td></td>
</tr>
<tr>
<td>---------------------</td>
<td>--------</td>
<td>---</td>
<td>-----------------------------------------------</td>
<td>----------------</td>
<td>---</td>
<td>----</td>
<td></td>
</tr>
<tr>
<td>04</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>see Notes 1, 2 and 3</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

See Ch 2, 3.2 Symmetrical boundary conditions for global loads and Figure 2.3.1 Boundary conditions for the application of symmetric global loads

Tank loading pattern image

![Reference plot (sample) of No.1 tank test loading](image)

Flotation case
Analysis of Primary Structures of Type C Liquified Gas Carriers

Chapter 2

Section 4

Table 2.4.7 Friction coefficient (reference values)

<table>
<thead>
<tr>
<th>Material 1</th>
<th>Material 2</th>
<th>Static friction coefficient, $\mu_s$</th>
<th>Dynamic friction coefficient, $\mu_d$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Steel</td>
<td>Wood</td>
<td>0.5</td>
<td>0.2</td>
</tr>
<tr>
<td>Steel</td>
<td>Synthetic resin</td>
<td>0.5</td>
<td>0.2</td>
</tr>
<tr>
<td>Steel</td>
<td>Steel</td>
<td>0.15</td>
<td>0.15</td>
</tr>
</tbody>
</table>

Note 1. The tank test case(s) are to consider the ship in the actual loading conditions when the tank test procedures are undertaken. It may be necessary to analyse load cases for the testing of each cargo tank separately. The actual ship conditions proposed for the tests of cargo tanks 1 and 2 must be used together with the lightest draughts chosen for each condition as described in Ch 2, 4.7 Tank test condition 4.7.1.

Note 2. All still water load items are to be applied.

Note 3. External hydrostatic pressures due to the still waterline are to be applied.

Note 4. Cargo tanks are to be empty. External buoyancy, deadweight and lightweight items and all other loads are not to be applied.

Note 5. This load case is not based on an actual loading condition.

Note 6. Only the void space between the inside of the hold and the outside of the cargo tank boundaries is assumed to be flooded up to the scantling draught.

Note 7. If damage stability calculations are available, then the depth of flooding from the damage stability calculations may be used in lieu of the scantling draught as described in Ch 2, 4.8 Flotation load cases 4.8.4.

Note 8. For illustration, figure shown is ship with three cargo tanks.
Note 1. $M_{sw}$ is not to be taken as less than $0.25M_{sw}$, permissible still water bending moment determined in accordance with Pt 3, Ch 4, 5.5 of the Rules for Ships.

Note 2. The ship’s seagoing, sheltered water and short voyage permissible still water bending moment limits are to be derived based on $M_{sw}$ and these limits are to be clearly stated in the ship’s Loading Manual and implemented in the ship’s loading computer software.

Figure 2.4.1 Permissible still water bending moment envelopes $M_{sw}$
Components shown are acceleration factors based on acceleration $1g$.

Figure 2.4.2 Acceleration ellipsoid
Figure 2.4.3 Definition of concerned points of tank A through H

(other configurations may be possible)
The pressure head, $h$, for any point around the hull is as follows:

$$ h = z \cos \theta + y \sin \theta $$

where

- $z$ = vertical distance below the mean waterline, measured at the centreline
- $y$ = transverse distance from centreline, positive to port
- $\theta$ = heel angle
- $\rho$ = pressure = $\rho g h$
- $T$ = local draught

**Figure 2.4.4** Hydrostatic pressure distribution for asymmetric load cases using a full-breadth model
Figure 2.4.5 Internal tank pressure for transverse load cases
Figure 2.4.6 Static pressure load distribution $P$ for cargo tanks
Figure 2.4.7 Distribution of vertical acceleration factor, $A_z$
Figure 2.4.8 Procedure to derive required VBM distribution for applying to FE model
5.1 Permissible stresses

5.1.1 The stresses resulting from the application of all load cases, with the exception of load cases O1, O2, O3 of collision cases and O5 of flotation cases, are not to exceed the maximum permissible values given in Table 2.5.1 Maximum permissible membrane stresses for wave load case, vertical dynamic load cases, static heeled cases, transverse dynamic cases and tank test conditions. The structural items indicated in Table 2.5.1 Maximum permissible membrane stresses for wave load case, vertical dynamic load cases, static heeled cases, transverse dynamic cases and tank test conditions are provided for guidance as to the most likely critical areas. All stresses for all parts of the model, however, are to be examined.

5.1.2 The maximum permissible stresses applicable for load cases O1, O2 and O3 of the collision cases and O5 of flotation cases are given in Table 2.5.2 Maximum permissible membrane stresses for collision load (O1, O2 and O3) and flotation load cases (O5).

5.1.3 In addition, it should be noted that the longitudinal hull girder elements should comply, as a minimum, with the requirements in Pt 3, Ch 4 of the Rules for Ships, see Ch 7, 1.1 General 1.1.4.
5.1.4 The permissible stress criteria in Table 2.5.1 Maximum permissible membrane stresses for wave load case, vertical dynamic load cases, static heeled cases, transverse dynamic cases and tank test conditions and Table 2.5.2 Maximum permissible membrane stresses for collision load (O1, O2 and O3) and flotation load cases (O5) are based on the recommended mesh size indicated in Ch 2, 2 Structural modelling.

5.1.5 The structural items indicated in Table 2.5.1 Maximum permissible membrane stresses for wave load case, vertical dynamic load cases, static heeled cases, transverse dynamic cases and tank test conditions and Table 2.5.2 Maximum permissible membrane stresses for collision load (O1, O2 and O3) and flotation load cases (O5) are provided for guidance as to the most likely critical areas. All stresses for all parts of the model, however, are to be examined for high values.

5.1.6 Where openings are not represented in the structural model the element shear stress, \( \tau_{xy} \), is to be increased in direct proportion to the modelled web shear area divided by the actual web area. The revised \( \tau_{xy} \) is to be used to calculate the combined equivalent stress, \( \sigma_e \). Where the resulting stresses are greater than 90 per cent of the maximum permitted, a more detailed analysis using a fine mesh representing the opening may be required or the scantlings increased accordingly.

5.1.7 The stresses in the synthetic materials usually incorporated between the steel faces of the supports and chocks are not to exceed the values given in Table 2.5.3 Maximum permissible stresses for synthetic materials usually incorporated between the steel faces of the supports and chocks.

5.1.8 To allow for a tank support set-up as referred to in Ch 2, 4.14 Support and chock design loads, all components of the computed stresses in the model of the support are to be increased by 12 per cent before comparison with the assessment criteria in Table 2.5.1 Maximum permissible membrane stresses for wave load case, vertical dynamic load cases, static heeled cases, transverse dynamic cases and tank test conditions. In addition, vertical direct stresses in the double bottom floor and girder elements in way, and in the tank primary structure in way, are to be increased by 12 per cent, and the combined stress recalculated for comparison with Table 2.5.1 Maximum permissible membrane stresses for wave load case, vertical dynamic load cases, static heeled cases, transverse dynamic cases and tank test conditions.

5.1.9 To allow for constructional tolerances in chocks as referred to in Ch 2, 4.14 Support and chock design loads, all components of the computed stresses in these supports are to be increased by 10 per cent before comparison with Table 2.5.1 Maximum permissible membrane stresses for wave load case, vertical dynamic load cases, static heeled cases, transverse dynamic cases and tank test conditions. In addition, vertical direct stresses in the deck and cargo tank top primary structures in way are to be increased by 10 per cent, and the combined stress recalculated for comparison with the assessment criteria in Table 2.5.1 Maximum permissible membrane stresses for wave load case, vertical dynamic load cases, static heeled cases, transverse dynamic cases and tank test conditions.

<table>
<thead>
<tr>
<th>Structural item</th>
<th>Combined stress, ( \sigma_e )</th>
<th>Direct stress, ( \sigma )</th>
<th>Shear stress, ( \tau ) (see Note 3)</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Double bottom structure</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Bottom shell plating</td>
<td>0,92 ( \sigma_L )</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Inner bottom plating</td>
<td>0,92 ( \sigma_L )</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Double bottom girders</td>
<td>0,92 ( \sigma_L )</td>
<td>-</td>
<td>0,46( \sigma_L )</td>
</tr>
<tr>
<td>Double bottom floors</td>
<td>0,75 ( \sigma_D )</td>
<td>-</td>
<td>0,35 ( \sigma_D )</td>
</tr>
<tr>
<td>Hopper tank web plating</td>
<td>0,75 ( \sigma_D )</td>
<td>-</td>
<td>0,35 ( \sigma_D )</td>
</tr>
<tr>
<td><strong>Side structure</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Side shell plating</td>
<td>0,92 ( \sigma_L )</td>
<td>-</td>
<td>0,46 ( \sigma_L )</td>
</tr>
<tr>
<td>Hopper plating</td>
<td>0,92 ( \sigma_L )</td>
<td>-</td>
<td>0,46 ( \sigma_L )</td>
</tr>
<tr>
<td>Topside slanted plating</td>
<td>0,92 ( \sigma_L ), 0,75 ( \sigma_L )</td>
<td>0,46 ( \sigma_L )</td>
<td></td>
</tr>
<tr>
<td>Side web frame</td>
<td>0,75 ( \sigma_D )</td>
<td>-</td>
<td>0,35 ( \sigma_D )</td>
</tr>
<tr>
<td>Structural item</td>
<td>Load cases</td>
<td>Combined stress, $\sigma_e$</td>
<td>Direct stress, $\sigma$</td>
</tr>
<tr>
<td>-----------------</td>
<td>------------</td>
<td>-----------------------------</td>
<td>------------------------</td>
</tr>
<tr>
<td><strong>Tank support systems and structure in way of</strong></td>
<td></td>
<td>0.75 $\sigma_0$</td>
<td></td>
</tr>
<tr>
<td>Cradle support (bracket)</td>
<td>O1, O2 and O3</td>
<td>$\sigma_0$</td>
<td>-</td>
</tr>
<tr>
<td>Anti-flotation chock seating</td>
<td>O5</td>
<td>$\sigma_0$</td>
<td>-</td>
</tr>
</tbody>
</table>

**Face plate of web structure**

<table>
<thead>
<tr>
<th>Structural item</th>
<th>Load cases</th>
<th>Combined stress, $\sigma_e$</th>
<th>Direct stress, $\sigma$</th>
<th>Shear stress, $\tau$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Transverse structure face plate</td>
<td></td>
<td>-</td>
<td>0.75 $\sigma_0$</td>
<td>-</td>
</tr>
</tbody>
</table>

**Note 1.** Stress criteria relate to the coarse mesh described in Ch 2, 2.1 FE modelling 2.1.5.

**Note 2.** If a finer mesh size is used, then stresses may be averaged over an area equal to the size of the coarse mesh element in way of the structure being considered. The averaging is to be based only on elements with their boundary located within the desired area. Stress average is not to be carried out across structural discontinuity or abutting structure.

**Note 3.** For girders, stringers, vertical webs and floors the specified values relate to the mean shear stress over the depth of the member. For bulkhead, shell and deck plating, in general mean shear stress relate to the average shear stress over three elements.

---

### Table 2.5.2 Maximum permissible membrane stresses for collision load (O1, O2 and O3) and flotation load cases (O5)

<table>
<thead>
<tr>
<th>Structural item</th>
<th>Load cases</th>
<th>Permissible stresses</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Tank support systems and structure in way of</strong></td>
<td></td>
<td>Combined stress, $\sigma_e$</td>
</tr>
<tr>
<td>Cradle support (bracket)</td>
<td>O1, O2 and O3</td>
<td>$\sigma_0$</td>
</tr>
<tr>
<td>Anti-flotation chock seating</td>
<td>O5</td>
<td>$\sigma_0$</td>
</tr>
</tbody>
</table>

**Note 1.** Stress criteria are based on the coarse mesh described in Ch 2, 2.1 FE modelling 2.1.5.

**Note 2.** If a finer mesh size is used, then stresses may be averaged over an area equal to the size of the coarse mesh element in way of the structure being considered. The averaging is to be based only on elements with their boundary located within the desired area. Stress average is not to be carried out across structural discontinuity or abutting structure.

**Note 3.** Criteria relate to single FE element.

---

### Table 2.5.3 Maximum permissible stresses for synthetic materials usually incorporated between the steel faces of the supports and chocks

<table>
<thead>
<tr>
<th>Condition</th>
<th>Not to exceed the lesser of</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cargo conditions</td>
<td>0.5 $\sigma_u$, 0.9 $\sigma_y$</td>
</tr>
</tbody>
</table>
### Section 6

#### Buckling acceptance criteria

6.1 **Buckling criteria**

6.1.1 For all load cases, with the exception of the collision load cases O1, O2 and O3 and flotation load cases O5, the buckling criteria are given in Table 2.6.1 Local plate panel required factor against buckling (see Note 1). Plate buckling is to be investigated for all areas of primary structure, but particular attention is to be paid to the areas specified in Table 2.6.1 Local plate panel required factor against buckling (see Note 1).

6.1.2 The buckling criteria applicable to collision load cases O1, O2 and O3 and the flotation load cases O5, are given in Table 2.6.2 Local plate panel required factor against buckling (collision and flotation load cases).

6.1.3 In addition, it should be noted that the longitudinal hull girder elements should comply, as a minimum, with the requirements in Pt 3, Ch 4, 7 of the Rules for Ships, see Ch 1, 1.1 General 1.1.4.

6.1.4 The combined effects of bi-axial compressive stress, shear stress and ‘in plane’ bending stress are to be included in the buckling calculation. In general, the average stresses acting within the plate panel are to be used for the buckling calculation.

6.1.5 Panel buckling calculations are to be based on the proposed thickness of the plating reduced by a thickness deduction for corrosion. A corrosion deduction of 1 mm is to be made for all structural items for each surface in contact with ballast water. No deduction is to be made for other surfaces.

6.1.6 In general, the applied stresses for buckling assessment are to be increased by a factor equal to the original thickness divided by the thickness after corrosion.

6.1.7 For the direct stress component which includes hull girder bending stress, it is permissible to adjust only the local stress component by the corrosion deduction. All other stress components are to be applied in accordance with Ch 2, 6.1 Buckling criteria 6.1.6.

6.1.8 For the buckling assessment of a tank support free edge, the elastic critical buckling stress can be calculated as follows. This elastic critical buckling stress is to be compared to the mean stress over the central half of the length and a plate width of 4t.

\[
\sigma_{FE} = \frac{\pi^2}{8} \left( \frac{t}{l} \right)^2 E
\]

where

\[
l = \text{free edge length}
\]

6.1.9 In calculating the factors against buckling, the edge restraint factor ‘c’ defined in Pt 3, Ch 4, 7 of the Rules for Ships may be taken into account in calculating the critical buckling stress of wide panels subjected to compressive loading on the long edge of the panel. The edge restraint factor ‘c’ is not to be used in the calculation of the critical buckling stress for compression applied on the short edges.

6.1.10 When the calculated elastic critical buckling stress, \( \sigma_c \), exceeds 50 per cent of the specified minimum yield stress, then the buckling stress is to be adjusted for the effects of plasticity using the Johnson-Ostenfeld correction formula, given below:

\[
\sigma_{cr} = \text{critical buckling stress corrected for plasticity effects;}
\]

\[
\sigma_{cr}, \sigma_0 = \text{see Ch 1, 2.1 Definition.}
\]

- when \( \sigma_c \leq 0,5 \sigma_0 \)

\[
\sigma_{cr} = \sigma_c
\]
when $\sigma_c > 0.5 \sigma_0$

$$\sigma_{cr} = \sigma_0 \left(1 - \frac{\sigma_0}{4\sigma_c}\right)$$

6.1.11 The buckling factors of safety of plate panels are to be derived, taking into account all relevant stress components specified in Ch 2, 6.1 Buckling criteria 6.1.4, using LR Buckle method in ShipRight Software. An alternative buckling assessment method, such as first principle direct calculation method, may be specially considered.

Table 2.6.1 Local plate panel required factor against buckling (see Note 1)

<table>
<thead>
<tr>
<th>Structural item</th>
<th>Factor against buckling $\lambda$</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Double bottom structure</strong></td>
<td></td>
</tr>
<tr>
<td>Bottom shell plating</td>
<td></td>
</tr>
<tr>
<td>Inner bottom plating</td>
<td>1.0</td>
</tr>
<tr>
<td>Hopper tank plating</td>
<td></td>
</tr>
<tr>
<td>Double bottom girders</td>
<td>1.0</td>
</tr>
<tr>
<td>Double bottom floors</td>
<td></td>
</tr>
<tr>
<td>Hopper tank web plating</td>
<td>1.1</td>
</tr>
<tr>
<td><strong>Side structure</strong></td>
<td></td>
</tr>
<tr>
<td>Side shell</td>
<td>1.0</td>
</tr>
<tr>
<td><strong>Cargo tank structure</strong></td>
<td></td>
</tr>
<tr>
<td>Cargo tank transverse webs</td>
<td></td>
</tr>
<tr>
<td>Cargo tank end bulkhead webs</td>
<td>1.1</td>
</tr>
<tr>
<td>Cargo tank top plating</td>
<td></td>
</tr>
<tr>
<td><strong>Deck structure</strong></td>
<td></td>
</tr>
<tr>
<td>Upper deck plating</td>
<td>1.0</td>
</tr>
<tr>
<td>Upper deck transverse webs</td>
<td>1.1</td>
</tr>
<tr>
<td><strong>Bulkhead structure</strong></td>
<td></td>
</tr>
<tr>
<td>Bulkhead plating</td>
<td></td>
</tr>
<tr>
<td>Bulkhead vertical webs</td>
<td>1.1</td>
</tr>
<tr>
<td>Bulkhead stringers</td>
<td></td>
</tr>
<tr>
<td><strong>Tank support structure</strong></td>
<td></td>
</tr>
<tr>
<td>Tank support seatings</td>
<td>1.0</td>
</tr>
<tr>
<td>Ship and tank structure in way of flotation chocks</td>
<td>1.0</td>
</tr>
</tbody>
</table>

**Note 1.** Applicable to wave load cases, vertical dynamic load cases, static heel load cases, transverse dynamic load cases and tank test load cases.
<table>
<thead>
<tr>
<th>Structural item</th>
<th>Factor against buckling $\lambda$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bulkhead structure and immediate areas of integration</td>
<td></td>
</tr>
<tr>
<td>Plating, vertical webs, stringers and girders</td>
<td>1.0</td>
</tr>
</tbody>
</table>

**Note 1.** Immediate areas of integration are to include side, bottom and deck structure from one web frame aft of the bulkhead to one web frame forward of the bulkhead.
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</tr>
</thead>
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<tr>
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<td>CHAPTER</td>
<td>5</td>
<td>ALTERNATIVE PROCEDURES FOR TRANSVERSE LOAD CASES</td>
</tr>
</tbody>
</table>
Section 1

Objectives

1.1 General

1.1.1 The objective of the structural analysis is to verify that the stress level in structural components and details under the applied static and dynamic loads are within acceptable limits.

1.1.2 Structural components and details in the following locations are to be investigated:

- Dome opening connections to upper deck,
- Forward and aft web sections for cradle,
- Other locations where the mesh size of the 3-D full ship FE model is insufficiently detailed to represent areas of high stress concentrations.

Section 2

Structural modelling

2.1 Applications

2.1.1 A screening analysis is recommended selecting the critical fine mesh areas specified in Ch 3, 2.1 Applications 2.1.3. The areas to be sub-modelled or subject to finer meshing are to be discussed with LR at the earliest opportunity.

2.1.2 Areas of high stress concentration in way of structural components and details are to be investigated by incorporating local fine meshed zones into the main model. Alternatively, separate local fine mesh models with boundary conditions derived from the main model may be used. Clear of areas to be analysed in detail, a coarse mesh arrangement may be adopted.

2.1.3 Areas where a fine mesh is needed include:

(a) Cradle and its supporting members for FWD and AFT web sections;
(b) Anti-floatation chocks;
(c) Anti-rolling chocks (for cylindrical Type C);
(d) Significant dome openings including coamings and girders.

2.1.4 The mesh size adopted should be such that the structural geometry can be adequately represented and the stress concentrations can be adequately determined. In general, the minimum required mesh size in fine mesh areas is not to be greater than 1/10 of the depth of the member (smallest dimension), 15t x 15t or 150 x 150 mm, whichever is the lesser, where t is the main plating thickness. In some locations a finer mesh may be necessary to represent the structural geometry. The mesh size need not be less than t x t unless adequate representation of the structural geometry requires a finer mesh. Triangular plate elements are to be avoided.

2.1.5 The element mesh size in way of dome openings is to comply with Ch 3, 2.1 Applications 2.1.4. Additionally, at the radius corner the element size should be smaller than R/10, where R is the radius of the corner. However, the mesh size need not be less than the thickness of the plating. For analysis codes in which reliable nodal stresses are not given, a line element of nominal (small) area should be arranged at the radius free edge. The stresses from this line element should be compared to the peak stress criteria given in Table 3.4.1 Maximum permissible stresses in Fine Mesh regions in way of stress concentrations.
2.1.6 Typical examples of fine mesh areas are shown in Figure 3.2.1 Typical fine mesh FE model for dome opening area to Figure 3.2.2 Typical fine mesh area in way of a cradle.

![Figure 3.2.1 Typical fine mesh FE model for dome opening area](image1)

![Figure 3.2.2 Typical fine mesh area in way of a cradle](image2)

---

**Section 3**

**Loading and boundary conditions**

3.1 General

3.1.1 Load cases specified in Ch 2, 4 Loading conditions are to be analysed.
3.1.2 Where a separate local fine mesh model is used, enforced displacements obtained from the full ship FE model in Ch 2 Analysis of Primary Structures of Type C Liquified Gas Carriers are to be applied to the boundaries of the fine mesh model. All local loadings are to be applied to the fine mesh model.

Section 4
Permissible stresses

4.1 Application

4.1.1 The stresses resulting from the application of the load cases referenced in Table 2.4.1 Wave load cases to Table 2.4.4 Transverse dynamic load cases in Ch 2 Analysis of Primary Structures of Type C Liquified Gas Carriers and the tank test load cases are not to exceed the maximum permissible values given in Table 3.4.1 Maximum permissible stresses in Fine Mesh regions in way of stress concentrations.

4.1.2 The stresses resulting from the application of the collision and flotation load cases are not to exceed the criteria given in Table 3.4.2 Maximum permissible stresses in Fine Mesh regions in way of stress concentrations (Collision and Flotation load cases).

Table 3.4.1 Maximum permissible stresses in Fine Mesh regions in way of stress concentrations

<table>
<thead>
<tr>
<th>Load cases</th>
<th>Combined stress $\sigma$</th>
<th>Direct stress $\sigma$</th>
<th>Shear stress $\tau_{xy}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\sigma_{\text{coarse}}$, see Note 1</td>
<td>See Table 2.5.1 Maximum permissible membrane stresses for wave load case, vertical dynamic load cases, static heeled cases, transverse dynamic cases and tank test conditions in Ch 2 Analysis of Primary Structures of Type C Liquified Gas Carriers</td>
<td>$\sigma_{0}$</td>
<td>See Table 2.5.1 Maximum permissible membrane stresses for wave load case, vertical dynamic load cases, static heeled cases, transverse dynamic cases and tank test conditions in Ch 2 Analysis of Primary Structures of Type C Liquified Gas Carriers</td>
</tr>
<tr>
<td>Average combined stress, $\sigma_{\text{average}}$, see Note 2</td>
<td>$\sigma_{0}$</td>
<td>$\sigma_{0}$</td>
<td>$0.55 \sigma_{0}$</td>
</tr>
<tr>
<td>Individual element</td>
<td>$1.2 \sigma_{0}$</td>
<td>$\sigma_{0}$</td>
<td>$0.55 \sigma_{0}$</td>
</tr>
</tbody>
</table>
In way of dome openings, see Ch 2, 2.1 FE modelling 2.1.5

| Peak stress in radius: clear of welds | Cases as in Table 2.4.1 Wave load cases to Table 2.4.4 Transverse dynamic load cases in Ch 2 Analysis of Primary Structures of Type C Liquefied Gas Carriers and the tank test load cases | — | 1.5 \( \sigma_0 \) | — |
| Peak stress in radius: in way of welds | Cases as in Table 2.4.3 Wave load cases in Ch 2 Analysis of Primary Structures of Type C Liquefied Gas Carriers and the tank test load cases | — | 1.2 \( \sigma_0 \) | — |

**Note 1.** \( \sigma_{\text{coarse}} \) are the values of combined stress, direct stress and shear stress, as required, averaged over an area equal to the size of the coarse mesh element in way of the structure being considered, see Figure 3.4.1 Mesh area for the calculation of \( \sigma_{\text{coarse}} \). The averaging is to be based only on elements with their boundary located within the desired area. Stress average is not to be carried out across structural discontinuity or abutting structure.

**Note 2.** \( \sigma_{\text{average}} \) is the average combined stress from element being assessed and the elements connected to its boundary nodes. However, averaging is not to be carried across structural discontinuity or abutting structure.

Table 3.4.2 Maximum permissible stresses in Fine Mesh regions in way of stress concentrations (Collision and Flotation load cases)

<table>
<thead>
<tr>
<th>Load cases</th>
<th>Permissible stresses</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Combined stress ( \sigma_{\text{coarse}} )</td>
</tr>
<tr>
<td>Collision and Flotation load cases</td>
<td>See Table 2.5.2 Maximum permissible membrane stresses for collision load (O1, O2 and O3) and flotation load cases (O5) in Ch 2 Analysis of Primary Structures of Type C Liquefied Gas Carriers</td>
</tr>
<tr>
<td>Average combined stress, ( \sigma_{\text{average}} )</td>
<td>1,3 ( \sigma_0 )</td>
</tr>
</tbody>
</table>

**Note 1.** \( \sigma_{\text{coarse}} \) are the values of combined stress, direct stress and shear stress, as required, averaged over an area equal to the size of the coarse mesh element in way of the structure being considered, see Figure 3.4.1 Mesh area for the calculation of \( \sigma_{\text{coarse}} \). The averaging is to be based only on elements with their boundary located within the desired area. Stress average is not to be carried out across structural discontinuity or abutting structure.

**Note 2.** \( \sigma_{\text{average}} \) is the average combined stress from element being assessed and the elements connected to its boundary nodes. However, averaging is not to be carried across structural discontinuity or abutting structure.
Figure 3.4.1 Mesh area for the calculation of $\sigma_{\text{coarse}}$
CHAPTER 1 INTRODUCTION
CHAPTER 2 ANALYSIS OF PRIMARY STRUCTURES OF TYPE C LIQUEFIED GAS CARRIERS
CHAPTER 3 ANALYSIS OF STRUCTURAL DETAILS
CHAPTER 4 MODELLING OF SUPPORTS AND CHOcks
    SECTION 1 MODELLING OF SUPPORT AND CHOcks
CHAPTER 5 ALTERNATIVE PROCEDURES FOR TRANSVERSE LOAD CASES
Section 1
Modelling of support and chocks

1.1 Modelling of supports and chocks

1.1.1 Cargo tank supports (steel seatings) of cradle chocks and anti-flotation chocks, see Figure 4.1.1 Location of cargo tank chocks (example), may be idealised by plate elements as shown in Figure 4.1.2 Idealisation of fixed (a) and sliding (b) supporting chocks.

1.1.2 For the chock, the analysis requires solving of a contact problem as there are no constraints to restrict the chocks and their seatings from separating. A chock can only withstand compressive loads when the chock and its support seating are in contact. Any tensile load will cause the chock to separate.

1.1.3 The contact problem can be modelled using non-linear contact type elements (such as GAP elements in NASTRAN) or using linear spring or rod elements with a suitable iteration technique.

1.1.4 For coarse mesh analysis, each chock connection can normally be modelled with a single element. However, for long supporting chocks, more elements are required to correctly represent the structural behaviour.

1.1.5 Multiple elements are required to represent the chock for fine mesh analysis to determine the local deformation behaviour of the chock.

1.1.6 Axial compressive and tensile stiffness properties can be defined based on the chock characteristics for non-linear GAP elements. The analysis will be non-linear and automatically iterate until the solution converges. Initial gap and preload can be taken as zero unless specified by the designer. The stiffness, $k_{\text{chock}}$, can be obtained as:

$$ k_{\text{chock}} = \frac{A_{\text{chock}} E_{\text{chock}}}{L_{\text{chock}}} $$

where $A_{\text{chock}}$ is the area of the chock represented by the element, $E_{\text{chock}}$ is the Young’s modulus of the chock and $L_{\text{chock}}$ is the height of the chock.

1.1.7 When linear spring or rod elements are used to model the connection, a manual iterative process is required. In each step of the iteration, elements which are in tension (i.e. no longer in contact) are removed or their stiffness value is set to zero. The process is repeated until the solution converges (i.e. no elements are in tension). The stiffness or area of elements can be defined based on the chock characteristics. For rod elements, the rod cross sectional area, $A_{\text{rod}}$, may be modelled as:

$$ A_{\text{rod}} = \frac{k_{\text{chock}} L_{\text{rod}}}{E_{\text{rod}}} $$

where $k_{\text{chock}}$ is the chock stiffness, see Ch 4, 1.1 Modelling of supports and chocks 1.1.6, $L_{\text{rod}}$ is the length of the rod element and $E_{\text{rod}}$ is the Young’s modulus of the rod element.

1.1.8 Frictional forces between the chocks and seatings are to be considered based on static and dynamic friction coefficients ($\mu_s$ and $\mu_d$). The magnitude of the frictional force is related to the reaction (compressive axial force) between the chocks and seating. Friction coefficients depend on the materials in contact and typical values are specified in Table 2.4.7 Friction coefficient (reference values).

1.1.9 For non-linear GAP elements, static and dynamic friction coefficients can be defined in the element property.

1.1.10 Spring and rod elements are unable to incorporate frictional forces automatically and require manual application. The total frictional force applied to the chocks can be calculated as follows:

$$ F_{\text{total}} = M_{\text{tank}} a_{\text{tank}} $$

if $M_{\text{tank}} a_{\text{tank}} \leq \mu_s \Sigma R_{\text{Element Reaction}}$

$$ F_{\text{total}} = \mu_d \sum R_{\text{Element Reaction}} $$

if $M_{\text{tank}} a_{\text{tank}} \leq \mu_s \Sigma R_{\text{Element Reaction}}$
where $M_{\text{tank}}$ is the mass of the tank and cargo, $a_{\text{tank}}$ is the acceleration experienced by the tank and $R_{\text{Element Reaction}}$ is the axial compressive force of the chock.

The frictional force to be applied to an individual compressed chock may be obtained as follows:

$$F_{\text{chock}} = \frac{F_{\text{total}} \times R_{\text{Element Reaction}}}{\sum R_{\text{Element Reaction}}}$$

where the total frictional force is less than the tank inertial force the difference of these forces is to be distributed to the stoppers fitted.

1.1.11 For collision load case, see Table 2.4.5 Collision load cases.

![Figure 4.1.1 Location of cargo tank chocks (example)](image)
Figure 4.1.2 Idealisation of fixed (a) and sliding (b) supporting chocks
<table>
<thead>
<tr>
<th>CHAPTER</th>
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<th>CONTENTS</th>
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</thead>
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<tr>
<td>1</td>
<td>1</td>
<td>INTRODUCTION</td>
</tr>
<tr>
<td>2</td>
<td>2</td>
<td>ANALYSIS OF PRIMARY STRUCTURES OF TYPE C LIQUEFIED GAS CARRIERS</td>
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<tr>
<td>3</td>
<td>3</td>
<td>ANALYSIS OF STRUCTURAL DETAILS</td>
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<tr>
<td>4</td>
<td>4</td>
<td>MODELLING OF SUPPORTS AND CHOCS</td>
</tr>
<tr>
<td>5</td>
<td>1</td>
<td>ALTERNATIVE PROCEDURES FOR TRANSVERSE LOAD CASES</td>
</tr>
</tbody>
</table>

**SECTION 1**

PROCEDURE TO APPLY TRANSVERSE ASYMMETRIC LOADS TO A HALF-BREADTH FE MODEL
Section 1
Procedure to apply transverse asymmetric loads to a half-breadth FE model

1.1 General

1.1.1 The preferred method for analysing the transverse load cases is to use a full-breadth model as described in Ch 2 Analysis of Primary Structures of Type C Liquefied Gas Carriers. If however, the analyst chooses to use a half-breadth model for the transverse cases, the procedure described in this Section is to be used.

1.1.2 In order to generate a transverse asymmetric load case for a half-breadth model, it is necessary to apply the transverse loads by combining two separate load cases. These two load cases consist of:

(a) The symmetric load case. This case applies symmetric loading components and boundary conditions to the FE model, see Figure 5.1.2 Boundary conditions for the application of symmetric loads to a half-breadth model.

(b) The anti-symmetric load case. This case applies anti-symmetric loading components and boundary conditions to the FE model, see Figure 5.1.3 Boundary conditions for the application of anti-symmetric loads to a half-breadth model.

1.1.3 If any of the loads do not conform to the above description, or if the structure is not symmetric about the centreline, then this technique is not strictly valid and a full-breadth FE model is required. In this case, it may also be necessary to consider an additional transverse dynamic load case which is heeled in starboard direction with negative transverse acceleration factors.

1.1.4 Using the above two load cases, the different structural response of both sides of the ship to the transverse loads can be derived as follows:

- port asymmetric = symmetric + anti-symmetric
- starboard asymmetric = symmetric - anti-symmetric

This is illustrated in Figure 5.1.1 Derivation of the asymmetric load cases for a half-breadth model from the symmetric and anti-symmetric load cases.

1.1.5 Application of the external hydrostatic pressure corresponding to the heeled waterline for symmetric and anti-symmetric load cases is illustrated in Figure 5.1.1 Derivation of the asymmetric load cases for a half-breadth model from the symmetric and anti-symmetric load cases and described as follows:

- The symmetric load component for the hydrostatic pressure is applied as half of the sum of the pressures on the port and starboard sides. Note it is necessary to modify the side of the shell pressure distribution as shown in Figure 5.1.1 Derivation of the asymmetric load cases for a half-breadth model from the symmetric and anti-symmetric load cases to satisfy the symmetric load definition stated above.

- The anti-symmetric load component for the external hydrostatic pressure is applied as half the difference of pressure on the port and starboard sides.

1.1.6 The cargo tank pressure loadings are specified in Ch 2, 4.9 Application of loads 4.9.3. If a half-breadth model is used to analyse the transverse cases, then, in a similar method as indicated in Ch 5, 1.1 General 1.1.4, it is necessary to express this pressure distribution as symmetric and anti-symmetric pressure cases.

1.1.7 The total pressure at a point in the tank is given by:

\[ P = \rho_c \cdot \Lambda_p \cdot a_{\beta} \cdot g \cdot Z_{\beta} + P_0 \]

1.1.8 The pressures to be applied to the symmetric load case, \( P_{\text{symmetric}} \), and the anti-symmetric load case, \( P_{\text{anti-symmetric}} \), are given by:

\[ P_{\text{symmetric}} = P_0 + 0.5 \cdot \rho_c \cdot \Lambda_p \cdot a_{\beta} \cdot g \cdot \left( Z_{\beta, \text{port}} + Z_{\beta, \text{stbd}} \right) \]

\[ P_{\text{anti-symmetric}} = 0.5 \cdot \rho_c \cdot \Lambda_p \cdot a_{\beta} \cdot g \cdot \left( Z_{\beta, \text{port}} - Z_{\beta, \text{stbd}} \right) \]

where
\( a_\beta = \) is the resultant acceleration vector at angle \( \beta \) see Ch 2, 4.9 Application of loads 4.9.3 and Figure 2.4.5 Internal tank pressure for transverse load cases

\( Z_\beta = \) is the largest liquid height above the point where the pressure is to be determined, see Ch 2, 4.9 Application of loads 4.9.3 and Figure 2.4.5 Internal tank pressure for transverse load cases

\( Z_{\beta, \text{port}} = \) is the height, \( Z_\beta \), measured to a point on the tank boundary at the port side of the ship.

\( Z_{\beta, \text{stbd}} = \) is the height, \( Z_\beta \), measured to the point on the tank boundary at the starboard side of the ship corresponding to the point used for \( Z_{\beta, \text{port}} \).

See Ch 2, 4.9 Application of loads 4.9.3 for definition of other symbols.

1.1.9 The boundary conditions for the symmetric load case and the anti-symmetric load case are as follows:

- Symmetric load case: See Figure 5.1.2 Boundary conditions for the application of symmetric loads to a half-breadth model.
- Anti-symmetric load case: See Figure 5.1.3 Boundary conditions for the application of anti-symmetric loads to a half-breadth model.

1.1.10 The FE model is to be analysed as follows:

- Sub-load case 1: Symmetric condition comprising symmetric loads and symmetric boundary conditions.
- Sub-load case 2: Anti-symmetric condition comprising anti-symmetric loads and anti-symmetric boundary conditions.

1.1.11 The load cases to be compared with the stress and buckling criteria given in Ch 2 Analysis of Primary Structures of Type C Liquefied Gas Carriers are obtained as follows:

- For the port side of the ship: Sub-load case 1 + Sub-load case 2.
- For the starboard side of the ship: Sub-load case 1 - Sub-load case 2.

In carrying out these combinations, direct and shear stresses should be combined as indicated above and von Mises stress is to be recalculated from the values of direct and shear stress resulting from the required combination. Buckling factor of safety is to be calculated based on the combined stresses.
Figure 5.1.1 Derivation of the asymmetric load cases for a half-breadth model from the symmetric and anti-symmetric load cases
Figure 5.1.2 Boundary conditions for the application of symmetric loads to a half-breadth model

Symmetry constraints: \( \delta_y = \delta_z = 0 \) apply to all grid points on the ship's structure and cargo tank's structure (only applicable for a half-breadth model)

A: \( \delta_x = 0 \) At the A.P. of the keel or transom, where appropriate, on the centreline
B: \( \delta_y = 0 \) At the deck on the centreline at the aft end
C: \( \delta_z = 0 \) At the F.P. of the keel on the centreline

Figure 5.1.3 Boundary conditions for the application of anti-symmetric loads to a half-breadth model

Anti-symmetry constraints: \( \delta_y = \delta_z = 0 \) apply to all grid points on the ship’s structure and cargo tank’s structure

A: \( \delta_x = \delta_y = \delta_z = 0 \) At the A.P. of the keel or transom, where appropriate, on the centreline
B: \( \delta_y = 0 \) At the deck on the centreline at the aft end
C: \( \delta_z = 0 \) At the F.P. of the keel on the centreline