Strength Analysis of Liquefied Gas Carriers with Independent Type B Prismatic Tanks

JULY 2013

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Classification Notes

Classification Notes are publications that give practical information on classification of ships and other objects. Examples of design solutions, calculation methods, specifications of test procedures, as well as acceptable repair methods for some components are given as interpretations of the more general rule requirements.

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CHANGES – CURRENT

General
This is a new document.
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1 General

1.1 Introduction

Classification Notes shall be considered in connection with DNV Rules for Classification of Ships, Pt.3 Ch.1, Hull Structural Design, Ships with Length 100 metres and above, /1/, and Pt.5 Ch.5, Liquefied Gas Carriers, /2/. The aim of this Classification Note is to describe the scope and applicable procedures for strength analysis of the hull in the cargo area, the cargo tanks and the supporting hull structures of LNG carriers with IMO Type B cargo tanks constructed mainly of plane surfaces.

Structural analysis carried out in accordance with the procedures outlined in this Classification Note will normally be basis for plan approval.

In general, LNG carriers should satisfy the strength criteria to main class *1A1 as given in the Rules Pt.3 Ch.1. The criteria for classification notation “Tanker for Liquefied Gas” as given in the Rules Pt.5 Ch.5 should be complied with for tank structures, tank supports*, and the hull structures supporting the cargo tank. The requirements of DNV Rules Pt.5 Ch.5 are considered to meet the requirements of the International Code for the Construction and Equipment of Ships Carrying Liquefied Gases in Bulk, IGC Code.

*) See Pt.5 Ch.5 A1100: Prismatic B-tank design have normally supports blocks distributed on the inner bottom and the whole double bottom, depending on the load case considered, shall then be regarded as supporting structure.

Class notation NAUTICUS(Newbuilding) is normally mandatory for LNG carriers with independent tanks of type B. Additional class notations PLUS and CSA as defined in DNV rules Pt.3 Ch.1, may impose additional requirements for the hull structure.

Attention should be given to the additional class requirements by flag or port authorities. Please note that USCG may have additional requirements for vessels trading to US ports.

1.2 Safety principles for Independent tanks Type B

Where the cargo temperature at atmospheric pressure is below –10°C, a secondary barrier is to be provided to act as a temporary containment for any envisaged leakage of liquid cargo through the primary barrier.

Where the cargo temperature at atmospheric pressure is not below –55°C, the hull structure may act as a secondary barrier. For lower cargo temperatures a separate secondary barrier is needed.

The main purpose of the secondary barrier is to protect the hull structure materials from exposure to low or cryogenic temperature liquids.

For independent tanks Type B the governing codes allow the secondary barrier to be replaced by a small leak protection system typically consisting of a gas detection system, containers to collect liquid spills, spray shields to deflect liquids into the containers, and equipment to dispose of the liquid as required. This, however, requires significant engineering efforts to practically eliminate the likelihood of massive leakages from the primary containment, and to document the type and extent of potential leakages to design and dimension the small leak protection system. The safety principles can be summarized as follows:

1) Failure developments that can be reliably detected before reaching a critical state (e.g. by gas detection or inspection) shall have a sufficiently long development time for remedial actions to be taken

2) Failure developments that cannot be safely detected before reaching a critical state shall have a predicted development time that is much longer than the expected lifetime of the tank.

For Moss type containment systems normal practice has been to assume that failures can originate from any dynamically loaded weld connection in the tank shell. As detection by inspection is not considered sufficiently reliable, it is generally required to document that such failure developments can be safely detected by leakage detection. In practice this means that a crack in the tank shell must be shown by fracture mechanics to exhibit stable growth through the shell thickness until the through thickness crack is sufficiently large to cause a leakage likely to be detected by the gas detection system, and continue to grow in a stable manner for a period of at least 15 days in specified environmental conditions. The small leak protection system must be designed to safely contain and dispose of the expected leakage rate at the end of the 15 day period. This is often referred to as leak before failure (LBF).

In weld connections in particular areas of the tank shell experiencing mainly dynamic bending action, cracks are likely to grow in the plane of the shell instead of through the thickness. Such failure developments will not necessarily satisfy leak before failure criteria. As an alternative, enhanced fatigue strength criteria have been introduced to ensure that the failure development time is sufficiently long compared to the expected lifetime of the tank.

In a similar way, leakage detection shall be the primary means of detection of failure developments originating in the tank shell of prismatic independent Type B tanks. This includes the weld connections in the tank shell plates and attached/adjacent structure from which failures developments may propagate undetected into the

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shell plate and cause leakages. This can typically be failure developments starting in stiffener to/through web connections or other typical fatigue critical details of which there are hundreds or thousands in a tank, and for which detection by inspection is not a sufficiently reliable means of detection. In such cases the fracture mechanics analyses for the tank shell shall be carried out considering the existence of a failure development in the attached/adjacent structure.

Detection by inspection will in many cases be an acceptable and reliable alternative for failure development in parts of girders, stringers and web frames easily accessible for inspection. In such cases it shall be documented by fracture mechanics that the time for the failure to reach a critical state is sufficiently long compared to the inspection interval.

Weld connections in other areas of girders, stringers and frames where failure developments cannot be reliably detected by inspection will be subject to enhanced fatigue strength requirements, and the time for the failure to reach a critical state shall be shown by fracture mechanics to be sufficiently long compared to the expected lifetime of the tank. Similar requirements may be relevant for particular weld connection in shell plates and adjacent structure where effective failure detection by leakage cannot be assured.

More specific requirements to fatigue and fracture mechanics analyses can be found in Sec.8 and Sec.9.

### 1.3 Scope of Work

Documentation required for approval of Independent Tank Type B, constructed mainly of plane surfaces as well as and hull structures is summarized in Table 1-1.

<table>
<thead>
<tr>
<th>Table 1-1</th>
<th>Overview of scope of work</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Classification requirements – cargo tanks and supporting structure in way of cargo tank</strong></td>
<td><strong>Scope</strong></td>
</tr>
<tr>
<td><strong>Design Loads</strong></td>
<td>— A complete wave load analysis is required for cargo tanks of type B. Loads defined at 10^-8 probability level and based on North Atlantic wave scatter diagram.</td>
</tr>
<tr>
<td></td>
<td>— Design loads for the tanks are to be determined based on the principles in Pt.5 Ch.5 Sec.5 A.</td>
</tr>
<tr>
<td></td>
<td>— Before direct wave load analyses results are available, design loads for the tanks can be determined according to Pt.3 Ch.1 Sec.4 and with guidance formulas for accelerations in Pt.5 Ch.5 Sec.5 A.</td>
</tr>
<tr>
<td><strong>Stress analysis of cargo tanks</strong></td>
<td>— A three dimensional FE cargo hold and tank analysis shall be carried out for midship region and fore and/or aft cargo hold region depending on the actual tank/ship design if fore and aft region deviates significantly from the midship region.</td>
</tr>
<tr>
<td></td>
<td>— The calculations including modelling, loading conditions, strength assessment and allowable stress checks to be carried out according to the procedure described in Chapter Pt.5 Ch.5 Sec.5 H.</td>
</tr>
<tr>
<td></td>
<td>— Mechanical properties for the material of a cargo tank to be documented</td>
</tr>
<tr>
<td></td>
<td>— For buckling analysis fabrication tolerances shall be considered.</td>
</tr>
<tr>
<td><strong>Local structural fine mesh analysis</strong></td>
<td>— Tank support analyses to be carried out for yield and buckling control;</td>
</tr>
<tr>
<td></td>
<td>— Vertical supports</td>
</tr>
<tr>
<td></td>
<td>— Upper/lower transverse supports</td>
</tr>
<tr>
<td></td>
<td>— Longitudinal support</td>
</tr>
<tr>
<td></td>
<td>— Anti-floating supports</td>
</tr>
<tr>
<td></td>
<td>— Other local details where high stress occurs</td>
</tr>
<tr>
<td></td>
<td>— Secondary stiffener analysis to be carried out for the following locations;</td>
</tr>
<tr>
<td></td>
<td>— Forward/aft stiffener in way of end bulkhead of cargo tanks and cargo hold double bottom.</td>
</tr>
<tr>
<td></td>
<td>— Other local details, as found necessary</td>
</tr>
<tr>
<td></td>
<td>— The calculations to be carried out according to the procedure described in Sec.5</td>
</tr>
<tr>
<td>Classification requirements – cargo tanks and supporting structure in way of cargo tank</td>
<td></td>
</tr>
<tr>
<td>---------------------------------</td>
<td>-------------------------------------------------</td>
</tr>
<tr>
<td><strong>Fatigue analysis</strong></td>
<td>For cargo tanks, fatigue analysis shall be carried out and shall cover areas of stress concentration, e.g. tank structures close to the supports, tank supports, tower supports, and other local details where high stresses occur.</td>
</tr>
<tr>
<td></td>
<td>The calculated fatigue life time in North Atlantic environmental conditions during $10^8$ wave encounters shall not have a fatigue damage factor larger than $C_W = 0.5$. In any case the fatigue life shall not be less than 20 years (the minimum design life for ships in DNV rules).</td>
</tr>
<tr>
<td></td>
<td>Calculations shall include maximum fabrication tolerances</td>
</tr>
<tr>
<td></td>
<td>S-N curves shall be relevant for the actual design detail.</td>
</tr>
<tr>
<td></td>
<td>Cumulative effect of fatigue, Miner’s sum, shall be based on mean value minus two standard deviations of the S-N test data according to current industry standard, see DNV Classification Notes 30.7.</td>
</tr>
<tr>
<td><strong>Crack propagation analysis</strong></td>
<td>A fatigue crack propagation analysis shall be carried out for areas with high dynamic stresses. The analysis shall consider propagation rates in parent material, weld metal and heat-affected zone.</td>
</tr>
<tr>
<td></td>
<td>The analysis shall establish the size and shape of possible fatigue cracks at penetration of the tank wall, taking into account the stress distribution through the tank wall.</td>
</tr>
<tr>
<td></td>
<td>The largest crack dimension at penetration is to be defined. The propagation of the through-thickness crack during a 15 day North Atlantic storm shall be determined as basis for estimation of leakage rates, and assessment of possible unstable crack behaviour.</td>
</tr>
<tr>
<td></td>
<td>Material data used in the calculation shall be documented</td>
</tr>
<tr>
<td></td>
<td>Crack propagation data to be based on mean value plus two standard deviations of the test data</td>
</tr>
<tr>
<td></td>
<td>Crack propagation analysis shall be carried out for selected areas of stress concentration taking into account maximum fabrication tolerances</td>
</tr>
<tr>
<td></td>
<td>The initial cracks used in the calculations shall be larger than those found by NDT methods</td>
</tr>
<tr>
<td></td>
<td>If leak before failure cannot be proven, i.e. the crack becomes unstable and/or the small leak protection system is not capable of containing and disposing of the leakage in a safe way, enhanced requirements to S-N fatigue data analysis and fracture mechanics analyses applies as defined in Table 8.3.</td>
</tr>
<tr>
<td><strong>Sloshing analysis</strong></td>
<td>Natural periods of liquid motions in the tank for each anticipated filling level to be documented</td>
</tr>
<tr>
<td></td>
<td>A simplified sloshing calculation as per Pt.3 Ch.1 Sec.4 C303 to C310 to be satisfied, as minimum requirement</td>
</tr>
<tr>
<td></td>
<td>In addition, numerical sloshing analyses and/or model testing may be required by the Society as found necessary.</td>
</tr>
<tr>
<td>Table 1-1  Overview of scope of work (Continued)</td>
<td></td>
</tr>
<tr>
<td>------------------------------------------------</td>
<td></td>
</tr>
<tr>
<td><strong>Classification requirements – cargo tanks and supporting structure in way of cargo tank</strong></td>
<td></td>
</tr>
<tr>
<td><strong>Vibration analysis</strong></td>
<td>— A vibration analysis to be carried out to show that no harmful vibrations will be excited by the propulsion system or other machinery. Added mass of LNG to be considered as relevant. — Local vibration analysis of panels to be documented.</td>
</tr>
<tr>
<td><strong>Thermal analysis</strong></td>
<td>Thermal analysis to be documented for the following cases: — Steady state temperature distribution of cargo hold to determine temperature in tank supporting structure and inner hull as basis for material grade selection — Transient thermal loads during cooling down periods. — For full load condition, combined stresses (thermal stress + local stress from lateral pressure) to be assessed.</td>
</tr>
<tr>
<td><strong>Insulation/reduced secondary barrier</strong></td>
<td>— Insulation system for cargo containment system to be documented with respect to material and design — Leak rates shall be determined for the purpose of sizing the partial secondary barrier in order to keep the temperature of the hull structure at a safe level. — Documentation of the suitability of the insulation system acting as a spray shield to deflect any liquid cargo down into the space between the primary and secondary barrier at low temperature. — A risk analysis to be performed to verify that any leaks are contained by the drip tray for at least 15 days and that the leaked gas can be disposed of in a safe way. Due account shall be taken of liquid evaporation, rate of leakage, pumping capacity etc.</td>
</tr>
<tr>
<td><strong>Classification requirements – hull structures</strong></td>
<td></td>
</tr>
<tr>
<td><strong>Damage Stability and Separation of Cargo Hold Spaces</strong></td>
<td>— The ship shall comply with the requirements for ship type 2G — The calculation shall include all relevant loading conditions with partially filled tanks</td>
</tr>
<tr>
<td><strong>Cargo hold analysis</strong></td>
<td>— A three dimensional integrated ship hull cargo hold and cargo tank model FE analysis shall be carried out for the midship region, see Sec.4. — Additional analyses for the fore and/or aft cargo hold regions may have to be carried out depending on the actual tank/ship design configuration if fore and aft region deviates significantly from the midship region.</td>
</tr>
</tbody>
</table>
1.4 Stresses and strength members

The following definitions are used in the report:

\[ \sigma_B = \text{the specified minimum tensile strength at room temperature (N/mm}^2\text{). For welded connections in aluminium alloys the tensile strength in annealed condition shall be used} \]

\[ \sigma_F = \text{the specified minimum upper yield stress at room temperature (N/mm}^2\text{). If the stress-strain curve does not show a defined yield stress, the 0.2\% proof stress applies} \]

\[ \sigma_{0.2} = \text{the specified minimum 0.2\% proof stress at room temperature (N/mm}^2\text{). For welded connections in aluminium alloys the 0.2\% proof stress in annealed condition shall be used} \]

Definition of strength member types is as follows:

— Primary members: supporting members such as webs, girders and stringers consisting of web plates, face plate and effective plating.
— Secondary members: stiffeners and beams, consisting of web plate, face plate (if any) and effective plating.
— Tertiary members: plate panels between stiffeners.

1.5 Units and Notations

The following SI-units (International System of units) are used in this Classification Notes:

- Mass: tonnes (t)
- Length: millimetres (mm) or metres (m), stated in each case
- Time: seconds (s)
- Force: kilo-newtons (kN)
- Acceleration: metres per second square (m/s²)

The following notations have been applied:

- \( L = \text{length of the vessel in m as defined in the Rules Pt.3 Ch.1 Sec.1 B101} \)
- \( B = \text{greatest moulded breadth in m, measured at the summer waterline} \)
- \( D = \text{moulded depth defined as the vertical distance in m from the top of the keel to the moulded deck line} \)
- \( T = \text{mean moulded summer draught in m, may be replaced by scantling draught} \ T_s \text{ in m (greater than summer draught)} \)
- \( T_M = \text{minimum design draught in m amidships, normally taken as} \ 2 + 0.02 \ L \)
- \( T_{\text{MIN}} = \text{Min. relevant seagoing draught in m, may be taken as} \ 0.35 \ D \text{ if not known} \)
- \( F.P. = \text{forward perpendicular, see the Rules Pt.3 Ch.1 Sec.1 B101} \)
- \( C_B = \text{block coefficient as defined in the Rules Pt.3 Ch.1 Sec.1 B101} \)
- \( g_0 = \text{standard acceleration of gravity} = 9.81 \text{ m/s}^2 \)
- \( f_1 = \text{material factor depending on material strength group, see the Rules Pt.3 Ch.1 Sec.2} \)
- \( x = \text{axis in the ship’s longitudinal direction} \)

<table>
<thead>
<tr>
<th>Table 1-1 Overview of scope of work (Continued)</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Classification requirements – cargo tanks and supporting structure in way of cargo tank</strong></td>
</tr>
<tr>
<td>Local structural analysis</td>
</tr>
</tbody>
</table>
| Fatigue strength analysis | — Fatigue strength assessment to be carried out for the following locations in the cargo area:  
— Supports and associated structures  
— Other details where high stresses occur.  
— Rule load to be applied, unless CSA notation is given.  
— Design target life of minimum 20 years based on 80\% North Atlantic operation (equivalent to world-wide scatter diagram). | Pt.3 Ch.1 Sec.15 |
| Stern slamming analysis | — A calculation of stern slamming to be documented, if found necessary. | |
| Temperature calculation | — A temperature calculation to be documented for material selection of hull structures  
— If a partial secondary barrier is required, the partial secondary barrier is to be designed for the same design temperature as the primary barrier.  
— For definition of ambient temperatures see Sec.2. | Pt.5 Ch.5 Sec.2 B |
y = axis in the ship’s athwart ships direction (to port)
z = axis in the ship’s vertical direction (upwards)
E = modulus of elasticity of the steel material $2.06 \cdot 10^5$ N/mm$^2$
\( \sigma \) = Normal Stress
\( \tau \) = Shear Stress
LT = material grade intended for low temperature service.
\( \sigma_e \) = Equivalent stress as defined in Pt.3.Ch.1
\( \tau_m \) = Mean shear stress over a net cross section
\( \eta \) = usage factor
\( \eta_s \) = usage factor related to static loads
\( \eta_{S+D} \) = usage factor related to static plus dynamic loads (ULS condition)
ULS = Ultimate Limit State; design condition related to static (S) plus dynamic (S+D), $10^{-8}$ loads
FLS = Fatigue Limit State; design condition related to repeated dynamic fatigue loads, $10^{-4}$ loads
ALS = Accidental Limit State; accidental design condition.
2 Material Selection

Presence of the cold cargo will cause a low temperature for parts of the hull steel structure. The temperature for hull structures shall be calculated in accordance with Pt.5 Ch.5 Sec.2. The calculation is normally to be based on empty ballast tanks if this assumption will cause the lowest steel temperature. Further guidance is given, Sec.6.

2.1 Material of Cargo Tank and Hull Structures

Specification of materials in cargo tank shall be submitted for approval, ref. Pt.5 Ch.5 Sec.1 C202.

For certain materials, subject to special consideration by the Society, enhanced yield strength and tensile strength at design temperatures below -105°C. ref. Pt.5 Ch.5 Sec.5 A401 may be considered.

Materials for cargo tanks and hull structures shall comply with the minimum requirements given in Pt.5 Ch.5 Sec.2 C and D.

2.2 Ambient Temperature

Design ambient temperatures for the analysis may be taken in accordance with Table 2-1.

<table>
<thead>
<tr>
<th>Regulations</th>
<th>Still sea water temperature, °C</th>
<th>Air temperature, °C</th>
<th>Speed, knots</th>
<th>Applicable areas</th>
</tr>
</thead>
<tbody>
<tr>
<td>IGC code</td>
<td>0.0</td>
<td>+5.0</td>
<td>0.0</td>
<td>All hull structure in cargo area</td>
</tr>
<tr>
<td>USCG requirements, except Alaskan water</td>
<td>0.0</td>
<td>-18.0</td>
<td>5.0</td>
<td>Inner hull and members connected to inner hull in cargo area</td>
</tr>
<tr>
<td>USCG requirements, Alaskan water</td>
<td>-2.0</td>
<td>-29.0</td>
<td>5.0</td>
<td>Inner hull and members connected to inner hull in cargo area</td>
</tr>
</tbody>
</table>

1) If DAT or Winterized notation has been specified, the specified design material temperature should be used as design ambient temperature.

2.3 Outer Hull Structures

The outer hull structure includes the shell and deck plating of the ship and all stiffeners attached thereto. The material of the outer hull structure is to be in accordance with Pt.3 Ch.1 Sec.2, unless the calculated temperature of the material in the design condition is below -5°C due to the effect of low temperature cargo. In this case the material is to be in accordance with the rules Pt.5 Ch.5 Sec.2.

If the DAT notation has been specified, the exposed members above the ballast waterline of the vessel shall comply with Pt.5 Ch.1 Sec.7

Note: Additional USCG requirements apply to hull plating along the length of the cargo area as follows:

— Deck stringer and sheer strake must be at least Grade E steel
— Bilge strake at the turn of the bilge must be of Grade D or Grade E

2.4 Inner Hull Structures

The inner hull structure includes inner bottom plating, longitudinal bulkhead plating, transverse bulkhead plating, floors, webs, stringers and all stiffeners attached thereto. For ships intended for trading in areas where the ambient temperatures differ from those in Table 2-1, the lower ambient temperatures shall be used for the temperature calculation.

The load condition giving the lowest draft among the load conditions with two tanks empty and the other tanks full may be used for the temperature calculations.

Steel grade of load carrying stiffeners (e.g. deck longitudinal or bulkhead stiffeners) shall be as for the plating to which the stiffener is attached. This also applies to structural members where direct loads are not applied, e.g. brackets, top stiffeners, ribs, lugs attached to web frames, floors and girders.

For structural members connecting inner and outer hull, possibly containing small openings, the mean temperature may be taken for selection of steel grade as given in Pt.5 Ch.5 Sec.2 B501.

Engine room temperature of 5°C is normally assumed. It is assumed that heating coil in fuel oil tank is inactive.
3 Local Strength of Cargo Tanks

3.1 Design Density of Cargo
For design the specific cargo density is to be taken as minimum 0.5 t/m³ for LNG cargo irrespective if a cargo with lower density is planned for the actual trade.

3.2 Cargo Tank Pressure based on IGC Code
Cargo tank pressure at $10^{-8}$ probability level is calculated using the acceleration ellipsoid with component accelerations at $10^{-8}$ probability level as determined by direct wave load analysis, DNV rules Pt.5 Ch.5 Sec.5 A700. However, the guidance formulas in A704 may be used for initial design assessment in lack of direct wave load analysis. These formulas will normally give more conservative estimates than found by wave load analysis which in any case has to be carried out for final documentation of a B-tank design.

The cargo pressure for a full tank is given by:

$$p_{eq} = p_0 + (p_{gd})_{\text{max}} \quad \text{(bar)}$$

$$(p_{gd})_{\text{max}} = a_\beta Z_\beta \rho / (1.02 \cdot 10^4) \quad \text{(bar)}$$

Where:

- $p_0$ = design vapour pressure is the maximum gauge pressure at the top of the tank, not to be taken less than 0.25 bar. To be conservatively set to zero, $p_0 = 0$ bar, for buckling control.
- $(p_{gd})_{\text{max}}$ = maximum combined internal liquid pressure, resulting from combined effects of gravity and dynamic acceleration
- $a_\beta$ = the dimensionless acceleration (relative to the acceleration of gravity) resulting from gravitational and dynamic loads, in an arbitrary direction $\beta$ (a more detailed description is given below)
- $\rho$ = the maximum density of the cargo in tonnes/m³ at the design temperature
- $Z_\beta$ = largest liquid height (m) above the point where the pressure is to be determined measured from the tank shell in the $a_\beta$ direction (see Figure 3-1).

Influence of the dome on the pressure height shall be taken into account as described in the Rules Pt.5 Ch.5 Sec.5 A706.

The acceleration $a_\beta$ is calculated by combining the three component accelerations $a_x$, $a_y$ and $a_z$ values according to an ellipsoid surface, as given in the Rules Pt.5 Ch.5 Sec.5 A704. The acceleration shall be based on wave load analysis as outlined in Classification Notes 34.1 CSA Direct analyses of ship structures, Ref. /9/.

For different directions of $a_\beta$ in the ellipsoid, the pressure at different corner locations in the cargo tank, are calculated according to the formula above. Based on the calculated pressures, the maximum pressures at corner points are found. Between corner points the pressure may be found by linear interpolation. The design vapour pressure in tanks should normally be set to 0.25 bars in seagoing conditions.

The cargo tank is normally divided by a liquid tight centre line bulkhead with opening in way of liquid dome area forming a common gas phase. The same filling height at both sides of centreline longitudinal bulkhead is assumed for sea going conditions.

If divided by a liquid tight centreline bulkhead the centreline longitudinal bulkhead is normally to be designed for one side filling in harbour. The fatigue strength of stiffeners in the upper part of the centreline bulkhead shall be checked in sea going condition.
Depending on the actual design and the actual operational conditions additional analyses and/or tests may be required by the Class Society and/or by the Port State. Shelf State requirements will apply if the vessel is to operate under an offshore regulatory regime.

3.3 Corrosion Addition
Cargo tanks and inner side of hold space do generally not require corrosion thickness on cargo hold side.

3.4 Requirements for Local Scantlings
The scantlings of the tank's strength members shall be based on a complete structural analysis of the tank and are generally not to be less than those for Independent Tanks Type A, ref. DNV Rules Pt.5 Ch.5 Sec.5 H103. Scantlings of plates and stiffeners shall as a minimum satisfy the A-tank requirements of DNV Rules Pt.5 Ch.5 Sec.5 E200.
In addition, stress in plates and secondary stiffeners of tank boundaries (the primary tank barrier) should also comply with the requirements in DNV Rules Pt.5 Ch.5 Sec.5 H300 and H400.

Figure 3-1
Resulting acceleration (static + dynamic) in arbitrary direction
Figure 3-2
Calculation of plate stresses

A conservative assessment method to fulfil the requirements in H300 and H400 is outlined below:
The plate stress to be controlled is composed of the most conservative combination of the stresses from three
main structural responses classified as follows:

1) local plate bending ($\sigma_y$ stress) at the middle of the plate panel
2) local stiffener bending ($\sigma_x$ stress) at the middle of the plate panel
3) girder bending stresses ($\sigma_y$ stress) taken from global finite element model at the middle of the plate panel.

For the slanted or vertical stiffened panels, in addition to above, as minimum requirement the plate stress is
composed as follows, see Figure 3-2.

1) local plate bending ($\sigma_y$ stress) at the half of the stiffener spacing above the lower support of the plate panel
2) local stiffener bending ($\sigma_x$ stress) at the half of the stiffener spacing above the lower support of the plate panel
3) girder bending stresses ($\sigma_y$ stress) taken from global finite element model at the half of the stiffener spacing
   above the lower support of the plate panel.

Local plate bending:
$$\sigma_y = \frac{0.25 P_{eq} s^2 y}{1000} \quad (\text{N/mm}^2)$$
At midpoint of plate

Local stiffener bending:
$$\sigma_x = \frac{1000 P_{eq} s^2 I_z}{24 Z_{act}} \quad (\text{N/mm}^2)$$
at a midpoint of span for plate scantling (conservative estimate)
$$\sigma_x = \frac{1000 P_{eq} s^2 I_z}{24 Z_{act}} \left[ 2 - 6 \left( \frac{S}{I} \right) - 3 \left( \frac{S}{I} \right)^2 \right] \quad (\text{N/mm}^2)$$
at the half of the stiffener spacing above the lower support of the plate panel
where
\[ s = \text{stiffener spacing in m} \]
\[ t = \text{plate thickness in mm} \]
\[ l = \text{stiffener span in m} \]
\[ Z_{act} = \text{actual section modulus of the stiffener at base plate in cm}^3 \]
\[ P_{eq} = \text{lateral pressure in kN/m}^2. \]

The effective von Mises stresses may conservatively be defined as follows:
\[
\sigma_{eq} = \sqrt{\sigma_x^2 + \sigma_y^2 - \sigma_x \sigma_y}
\]
\[
\sigma_x = \sigma_x(\text{stiffener}) + \sigma_x(\text{global})
\]
\[
\sigma_y = \sigma_y(\text{local plate bending})
\]

The stress components defined above are combined using the von Mises equivalent stress given in Pt.5 Ch.5 Sec.5 F201.

3.5 Allowable Stresses
Allowable stresses of plates (tertiary) and stiffeners (secondary) shall be as defined by DNV Rules Pt.5 Ch.5 Sec.5 H400. In addition, allowable stresses for static conditions shall be taken as 70% of the values given in Table H1-3 (also listed as Figure 4-5 below).

Allowable stress of 0.7\( \sigma_f \) against sloshing pressures defined in DNV Rules Pt.3 Ch.1 Sec.4 C300 is to be used. This is the same utilization as used in the ship rules, i.e. (160 \( f_1 \)). For check against sloshing impacts, the impact pressures given in Pt.3 Ch.1 Sec.4 C307, C308, C309 and C310 shall be used together with the impact strength formulae in Sec.9 E400. (See also Sec.7: Sloshing Assessment).
4 Cargo Tank and Hull Finite Element Analysis

For LNG carriers of Independent Tank Type B, constructed mainly of plane surfaces, the structural analysis shall be carried out for the evaluation of a cargo tank, tank supports and hull structures in accordance with IGC code and applicable rules. Modelling of hull structures shall unless defined below follow guidance in DNV Classification Notes 31.3.

An integrated cargo hold and cargo tank finite element model is established to determine reaction forces in supports and to assess the structural adequacy of primary members of the cargo tank, tanks supports and associated hull structure under hull girder bending, external and internal loads.

This section describes a procedure for modelling and strength assessment of the cargo tank and hull structures with respect to yield and buckling (ULS condition) based on finite element analysis.

A flow diagram showing the minimum requirement of finite element analysis is shown in Figure 4-1.

![Figure 4-1](Minimum requirement on finite element analysis)

4.1 Structural Idealization

4.1.1 Coordinate system

A right-handed orthogonal coordinate system is normally employed to describe the model. The origin is at the intersection between the ship’s baseline and the aft perpendicular. The global X-axis runs in the longitudinal direction of the ship, positive forward. The global Y-axis runs in the ship’s transverse direction, positive to port side. The global Z-axis points in the vertical direction, positive upwards.

4.1.2 Required information for the analysis

The following information is necessary for the structural analyses:

- General arrangement
- Trim & stability booklet including lightweight distribution
- Key plans, e.g. midship section, construction profiles and decks
- Cargo tanks construction drawings
- Cargo supports arrangement and scantlings.

4.1.3 Model Extent

The necessary longitudinal extent of the model will depend on the structural arrangement and the loading conditions. The analysis model is normally extended over two hold lengths (½+ 1 + ½), where the middle tank/hold of the model is used to assess the yield and buckling strength.

The model shall cover the full breadth of the ship in order to account for asymmetric structural layout of the cargo tank/supporting hull structure and design load conditions (heeled or unsymmetrical loading conditions).

Additional cargo hold models of the foremost or/and aft-most cargo tank may be required if the design of these tanks is significantly deviating from amid-ship.
4.1.4 Elements and Mesh Size

The structural assessment is to be based on linear finite element analysis of three dimensional structural models of the vessel. The general types of finite elements to be used in the finite element analysis are outlined below:

- Rod (or truss) elements are line elements with axial stiffness only and constant cross sectional area along the length of the element.
- Beam elements are line elements with axial, torsional and bi-directional shear and bending stiffness and with constant properties along the length of the element.
- Shell elements are elements with in-plane stiffness and out-of-plane bending stiffness with constant thickness.

Two node line elements and four node shell elements are, in general, considered sufficient for the representation of both the tank structure and the hull structure. The mesh requirements given in this chapter are based on the assumption that these elements are used in the finite element models. However, higher order elements may also be used. In general 8 node curved rectangular and 6 node curved triangular elements will be more stable and less sensitive to non-uniform mesh configurations.

The use of 3 node (constant stress) shell element shall be kept to a minimum. 2 node beam elements are usually used.

In general, hull and cargo tank structures may be meshed with one element between stiffeners (e.g. longitudinals) and a sufficient number of elements between stiffener supports (e.g. girders, web frames and stringers) to maintain an aspect ratio less than 3.0. Where possible, the aspect ratio of plate elements in areas where there are likely to be high stresses or a high stress gradient is to be kept close to one. The element mesh should represent the actual stiffening system of the structure as far as practicable so that the stresses for the control of yield and buckling strength can be read and averaged from the results without interpolation or extrapolation.

In special cases it may not be possible to idealize the geometry and stress distribution into suitable parts in order to use simplified buckling check and/or the PULS code. Typical cases may be frames and girders with free flanges and structural parts with irregular geometry. In such cases a FE buckling calculation may need to be carried out of those areas. This can be done as sub-model with a mesh designed to capture the dominating buckling modes.

When using non-linear FE programmes like ABAQUS special considerations with respect to modelling (mesh fineness), imperfection levels, imperfection modes and acceptance levels is required and will be considered by the Society.

Elastic buckling of plates and stiffeners in the cargo tanks shall not be allowed as frequently occurring elastic buckling of plates will increase the probability of crack initiation along plate boundaries.

Hence, with PULS use analysis option BS, Buckling Strength, for tank and associated support structures. Use analysis option UC, Ultimate Capacity, for the hull structures in general.

The UC option allows for elastic buckling for slender structures which shall not be allowed for the tank structure.

For definition of average stress and selection of suitable buckling panels (equivalent plate panels-EPP) see Ch.8 Sec.3 in the harmonized CSR rules, ref. /10/.

4.1.5 Modelling of Geometry

All the relevant structural members of the tank and hull structure shall be modelled. Plating members such as deck, bottom, inner bottom, side shell, transverse webs, watertight bulkheads, cargo tank bulkheads and shell, stringers, etc. shall be modelled by shell elements.

Stiffeners under action of local lateral loads, e.g. longitudinal stiffeners and vertical stiffeners attached to cargo tank bulkheads shall be represented by beam elements or shell elements. Face plates of primary supporting elements may be represented by beam or truss elements in order to represent bending properties properly. Secondary stiffeners, such as buckling stiffeners on transverse webs, girders/stringers, etc. may be modelled by truss or beam elements.

Small openings in double bottom floors/girders and web frames and bulkheads of cargo tank may be not modelled. Openings in way of critical areas shall be specially evaluated after opening area reduction with regard to shear strength. Modelling is to be according to the procedure in Classification Notes 31.3.

The modelled scantlings for all hull structures shall be obtained by deducing corrosion addition $t_k$ from the as built scantlings as required by the Rules.
4.1.6 Modelling of supports

Vertical supports, anti-rolling/pitching supports and anti-floating supports may be idealized by shell elements. The supports on hull and cargo tank may be interconnected with solid elements or beam/truss elements representing the support blocks. If linear elements are employed, the connection elements shall be disconnected when they are in tension (i.e. no contact). An iterative procedure may be required; supports in tension shall be disconnected and the FE model rerun until all active supports are in compression.

The cargo tanks are supported by the following supports.

— Vertical supports in global Z direction (vertical, positive upwards)
— Anti-rolling keys in global Y direction (transverse, positive to port side)
— Anti-pitch/anti-collision keys in global X direction (longitudinal, positive forward)
— Anti-flotation keys in global Z direction.

Unless otherwise documented by the designer, friction coefficients to use with the analyses of the supports are shown in Table 4-1.

<table>
<thead>
<tr>
<th>Surface Material 1</th>
<th>Surface 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>0.17</td>
<td>0.15</td>
</tr>
</tbody>
</table>

A common lay-out of cargo tank supports are shown in Figure 4-2 below.

![Figure 4-2](image)

**Figure 4-2**
Location of cargo tank supports (example)

4.1.7 Analysis strategy for ULS assessment

4.1.7.1 Vertical Supports

Various types of vertical supports are used. The following figure shows an example of a vertical supports. Normally Wood mounted on resin will contribute to levelling of vertical supports in a cargo hold. Dam plates are fitted to avoid movement of wood in case of damages in resin or bond between resin and top plate of a vertical support.
The support chocks can be modelled in several ways; either by solid elements combined with contact elements or modelled as beams with representative axial and shear stiffness. It is important that the model correctly remove the vertical load on supports in tension. This means that beams/contact elements in tension are removed (the vertical stiffness set to zero or a very small value) and the analysis repeated until all active vertical supports are in compression.

If it is seen that the combined longitudinal and transverse force exceeds the static friction force limit, the shear stiffness has to be removed for the connection elements and the calculation rerun with no shear coupling. The friction force acting on the interfacing surfaces must be applied to both parts of the supports with a magnitude of the static friction coefficient ($\mu_S$) times the vertical force acting on each individual support. The static friction coefficient can, if not otherwise specified by the designer, be conservatively set to 0.5, Table 4-1.

4.1.7.2 Transverse anti-roll supports
The anti-roll supports are designed based on the transverse acceleration load cases (LC 6 and 7) and the heeled load cases (LC8 and 9). In these load cases some of the transverse force is carried by friction in the vertical supports and the rest is taken by the upper and lower anti roll supports.

In order to be able to predict the distribution of forces between the upper and lower anti-roll supports a refined cargo hold analysis procedure should be used. This is based on an iterative approach.

1) All vertical supports are initially modelled with actual shear and bending stiffness.
2) The transverse forces in each of the supports are calculated.
3) If the dynamic friction force of a vertical support is exceeded, the shear and bending stiffness of the support is set to zero and the dynamic friction force is applied as a force couple.

4) The analysis shall be repeated to determine the new distribution of horizontal support forces.

This procedure shall be repeated until all the transverse support forces are less or equal to the dynamic friction force.

The dynamic friction force is calculated as the dynamic friction coefficient ($\mu_d$) times the vertical force acting on each individual support. If not otherwise documented by the maker of the wood blocks, a dynamic friction coefficient of 0.2 can be applied, Table 4-1.

Effects of intentional clearances between support surfaces should be include if this is expected to significantly affect the distribution of forces between the upper and the lower roll supports.

![Figure 4-5](image)

**Figure 4-5**
Example of anti-pitch support

### 4.1.7.3 Longitudinal anti-pitch supports

The ULS assessment of the anti-pitch supports is based on the collision load cases (LC10 and 11). Normally LC 10 will be governing. As for the anti-roll supports, some of the load in longitudinal direction will be taken by the friction in vertical supports and the rest will be taken by the anti-pitch supports.

The same iterative approach as described in the previous section should be utilized. If sliding occurs (the longitudinal force exceeds the static friction force), the analysis should be rerun with applied dynamic friction forces. Then it is possible to obtain the distribution of forces between outer and inner anti-pitch supports.

![Figure 4-6](image)

**Figure 4-6**
Example of an anti-floatation support

### 4.1.7.4 Anti-floatation supports

These supports are analysed similarly as the vertical supports. Deformations of the local models are taken from the cargo hold model analysed with the flooding condition (LC 12). The friction forces are to be calculated as described in [4.1.7.1], but the direction of these forces should be found from the cargo hold analysis.
4.2 Boundary Conditions

Boundary conditions for the application of hull girder load, local loads and asymmetric load are given in Figure 4-7 to Figure 4-9.

---

**Figure 4-7**
Boundary conditions for application of hull girder loads

**Figure 4-8**
Boundary conditions for the application of local load

**Figure 4-9**
Boundary conditions for the application of asymmetric loads
Weight of cargo tanks, support blocks and hull structures shall be taken into account. Reaction forces or springs to be applied to the transverse bulkheads to counteract for the imbalance of vertical forces. In addition, transverse horizontal constraint should be added to a node at the intersection between the transverse bulkhead and bottom/deck. Transverse directional spring elements are recommended for both ends of model instead of fixed condition \((dy = 0)\).

For application of bending moment at both ends of model, nodal points of all longitudinal elements should be rigidly linked \((dx, Ry \text{ and } Rz)\) to independent node (Master node) at neutral axis on centre line. This ensures the intersection plane to be planar. The longitudinal translation of a master node of one end of the model needs to be fixed.

Bending moment and shear force adjustment should be carried out to get target global bending moment and shear force. Reaction forces or springs to be applied to the transverse bulkheads to counteract for the imbalance of vertical and/or horizontal forces. Spring constants may be estimated as shown in Classification Notes 31.3 ignoring the effect of bending deflection:

\[
K = 8 \cdot A_s \cdot E / (7.8 \cdot 3 \cdot l_h)
\]

where:

- \(A_s\) = shear area for side (side and inner side for hull structures)
- \(E\) = modulus of elasticity, \(2.06 \cdot 10^{5}\) N/mm\(^2\) for steel
- \(l_h\) = length of one cargo hold.

### 4.3 Loading Conditions and Design Load Cases

The design load cases are selected based on actual loading conditions from vessel’s loading manual.

The design load conditions should include fully loaded condition, alternate conditions (realistic combinations of full and empty cargo tanks) with static (S) and static and dynamic (S+D) sea pressure/tank pressure, giving maximum net loads on double bottom structures.

The basis for the selection of load conditions is to maximize the cargo tank and hull stress response by combining internal and external loads with hull girder bending.

Design loads including actual bending moments and maximum cargo accelerations and sea pressure should be applied to the global cargo hold finite element model. The loads shall be the expected extreme for 10\(^8\) wave encounters in North Atlantic conditions and serve as basis for design against yield and buckling of the cargo tank, the supports and the supporting double bottom structures. Note that the worst combination of loads shall be considered. In some cases this may result in removing loads (shall be realistic) to archive the largest stresses for particular elements.

**Table 4-2** and **Table 4-3** list applicable design load cases for analysis of amid-ship cargo areas given in the Rules Pt.5 Ch.5 Sec.5. The tables include an indication of the structural components governed by the various load cases. These design load cases are normally covering all relevant loading conditions for independent tank type B constructed mainly of plane surfaces.

Some conditions given in **Table 4-2** and **Table 4-3** may be omitted in the analysis considering actual hull girder bending moment and its effect on the vertical supports.

Sea pressure shall be calculated from one of the following rules, depending on the probability level employed.

- **Pt.3 Ch.1 Sec.4 C200**, referring to probability level of \(10^{-4}\)
- **Pt.5 Ch.5 Sec.5 E303**, referring to probability level of \(10^{-8}\).

Cargo loads, static (S) and dynamic (D) loads shall be calculated, depending on the probability level employed.

#### Static pressure (S)

\[
P_{sta.} = \rho_c g h_z + P_0
\]

#### Vertical (S+D) pressure

\[
P_{dyn.} = \rho_c g (1 + a_z) h_z + P_0
\]

at probability level of \(10^{-8}\)

\[
P_{dyn.} = \rho_c g (1 + 0.5 a_z) h_z + P_0
\]

at probability level of \(10^{-4}\)

#### Transverse (S+D) pressure

\[
P_{dyn.} = (\rho_c g h_z + P_0) + \rho_c g h_y a_y
\]

at probability level of \(10^{-8}\)

\[
P_{dyn.} = (\rho_c g h_z + P_0) + 0.5 \rho_c g h_y a_y
\]

at probability level of \(10^{-4}\)

### Pressure for ballast tanks:

#### Static pressure (S)

\[
P_{sta.} = \rho_w g h_z
\]

#### Vertical (S+D) pressure

\[
P_{dyn.} = \rho_w g (1 + a_z) h_z
\]

at probability level \(10^{-8}\)

\[
P_{dyn.} = \rho_w g (1 + 0.5 a_z) h_z
\]

at probability level \(10^{-4}\)
where:

\[\rho_c = \text{design density of cargo, } 0.5 \text{ t/m}^3\]
\[\rho_{bw} = \text{design density of ballast water, } 1.025 \text{ t/m}^3\]
\[g = \text{gravity, } 9.81 \text{ m/s}^2\]
\[P_0 = \text{design vapour pressure at seagoing condition, subject to special considerations.}\]

A vapour pressure higher than \(P_0\) may be accepted in harbour condition where dynamic loads are reduced.

\[h_y, h_z = \text{local head for pressure measured from the tank reference point in the transverse and vertical direction, respectively}\]
\[a_y, a_z = \text{maximum dimensionless acceleration (relative to the acceleration of gravity) at the centre of gravity of the tank in the transverse and vertical direction, respectively}\]

The weight of a cargo tank and hull structures is to be included in the FE analysis. For buckling control the cargo vapour pressure shall be taken equal to zero, \(P_0 = 0 \text{ bar}\).

### 4.4 Design Application of Load Cases

Primary members shall be designed considering the loading conditions summarized in Table 4-2 and Table 4-3. Table 4-3 gives indications on which Load Cases will influence on the design of the various areas in the tank and hull structure. However, other areas (members) not mentioned may need to be reviewed with additional relevant load cases.

The transverse acceleration load cases (LC 6 and 7) and the heeled load cases (LC 8 and 9) are used to verify the keying arrangement (the transverse roll supports), cargo tank web frame structure and hopper frames. A heeling angle of 30° shall be used as specified in Pt.5 Ch.5 Sec.5 A1104 and the IGC code. The internal pressure in the cargo tank shall be based on the combined effect of gravity \(g_o\) and a transverse acceleration component of gravity amounting to \(a_y = g_o \sin(30) = 0.5g_o\).

For Load Case 16, the double side ballast tank and the cargo hold is assumed punctured with water ingress into the hold space between the hull and the cargo tank. The maximum static pressure from the inclined damaged waterline is to be applied to the transverse bulkhead. It is to be ensured that the cargo tank is intact and no cargo leakage into the cargo hold (void space) takes place, see Table 4-2 and Table 4-3.

The Load Cases and Loading Conditions shown in Table 4-2 and Table 4-3 shall be applied for evaluation of Tank 2. Similar load cases need to be applied for other tanks.

It should be noted that the loading conditions given in Table 4-2 and Table 4-3 are minimum loading conditions. If more severe loading conditions, e.g. two adjacent cargo tanks empty or full, etc. are given in the loading manual, these conditions shall also be taken into account.
Table 4-2  Load Cases for Tank, Hull and Tank Supporting Structures

<table>
<thead>
<tr>
<th>Load Case</th>
<th>Loading condition</th>
<th>Tank load</th>
<th>Sea press.</th>
<th>Draught</th>
<th>Illustration</th>
<th>Comments</th>
</tr>
</thead>
<tbody>
<tr>
<td>LC 1</td>
<td>Full load</td>
<td>S</td>
<td>S</td>
<td>TS</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Hogging Static (S)</td>
<td></td>
<td></td>
<td></td>
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<td></td>
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<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>— Still water bending moment $M_S$: max. hogging bending moment from Trim and Stability (T&amp;S) booklet: All tanks full or any one tank empty others full. $M_S \geq 0.5 M_S_{rule}$ hogging. — Wave bending moment: $M_W = 0$</td>
</tr>
<tr>
<td>LC 2</td>
<td>Alternate Loading</td>
<td>S</td>
<td>S</td>
<td>$T_{MIN}$</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Sagging Static (S)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
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<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>— Still water bending moment $M_S$: max. sagging bending moment from T&amp;D booklet. Alternate tank filling or any one tank full others empty. $M_S \geq 0.5 M_S_{rule}$ sagging — Wave bending moment: $M_W = 0$</td>
</tr>
<tr>
<td>LC 3</td>
<td>Head Sea (ULS)</td>
<td>S+D</td>
<td>S</td>
<td>TS</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Full load</td>
<td></td>
<td></td>
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<td></td>
</tr>
<tr>
<td></td>
<td>Seagoing Hogging</td>
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<tr>
<td></td>
<td>$S+D (10^{-8})$</td>
<td></td>
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<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>— Still water bending moment $M_S = LC 1$ (hogging): — Wave bending moment: $M_W = 1.0 M_W_{hogging}$ calculated — Vertical acceleration ($a_x$) of a cargo tank combined with gravity, ($g_o$).</td>
</tr>
<tr>
<td>LC 4</td>
<td>Head Sea (ULS)</td>
<td>S+D</td>
<td>S</td>
<td>$T_{MIN}$</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Alternate Loading</td>
<td></td>
<td></td>
<td></td>
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<tr>
<td></td>
<td>Seagoing Sagging</td>
<td></td>
<td></td>
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</tr>
<tr>
<td></td>
<td>$S+D (10^{-8})$</td>
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<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>— Still water bending moment $M_S = LC 2$ (sagging): — Wave bending moment: $M_W = 1.0 M_W_{sagging}$ calculated — Vertical acceleration ($a_y$) of cargo tank combined with gravity, ($g_o$).</td>
</tr>
<tr>
<td>LC 5</td>
<td>Head Sea (ULS)</td>
<td>S</td>
<td>S+D</td>
<td>TS</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Full load</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Seagoing Hogging</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>$S+D (10^{-8})$</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>— Still water bending moment $M_S = LC 1$ (hogging) — Wave bending moment: $M_W = 1.0 M_W_{hogging}$ calculated — Max. dynamic sea pressure Pt.5 Ch.5 Sec. 5.</td>
</tr>
<tr>
<td>LC 6</td>
<td>Beam Sea (ULS)</td>
<td>S</td>
<td>S</td>
<td>TS</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Full load</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Seagoing Max a_y Dynamic condition</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>$S+D (10^{-8})$</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>— Still water bending moment $M_S = LC 1$ (hogging): — Wave bending moment: $M_W = 0$ — Transverse acceleration ($a_y$) of cargo tank combined with gravity, ($g_o$).</td>
</tr>
<tr>
<td>Load Case</td>
<td>Loading condition</td>
<td>Tank load</td>
<td>Sea press.</td>
<td>Draught</td>
<td>Illustration</td>
<td>Comments</td>
</tr>
<tr>
<td>-----------</td>
<td>-------------------</td>
<td>-----------</td>
<td>------------</td>
<td>----------</td>
<td>-------------</td>
<td>----------</td>
</tr>
<tr>
<td>LC 7 Beam Sea (ULS)</td>
<td>Alternate load Seagoing Max ay, Dynamic (S+D)</td>
<td>S+D (10⁻⁵)</td>
<td>S</td>
<td>T_MIN</td>
<td>Still water bending moment ( M_S = LC 2 ) (sagging):</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Wave bending moment: ( M_W = 0 )</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Transverse rule acceleration ( a_y ) of cargo tank combined with gravity ( g_o )</td>
<td></td>
</tr>
<tr>
<td>LC 8 (ULS)</td>
<td>Full load Heeled condition Static (S)</td>
<td>S</td>
<td>S</td>
<td>T_S</td>
<td>Still water bending moment ( M_S = LC 1 ) (hogging):</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Wave bending moment: ( M_W = 0 )</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Inclination 30° with static sea pressure from Pt.5 Ch.5 Sec.5 (Pt.3 Ch.1)</td>
<td></td>
</tr>
<tr>
<td>LC 9 (ULS)</td>
<td>Alternate load Heeled condition Static (S)</td>
<td>S</td>
<td>S</td>
<td>T_MIN</td>
<td>Still water bending moment ( M_S = LC 2 ) (sagging):</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Wave bending moment: ( M_W = 0 )</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Inclination 30° with static sea pressure from Pt.5 Ch.5 Sec.5 (Pt.3 Ch.1)</td>
<td></td>
</tr>
<tr>
<td>LC10 (ALS)</td>
<td>Full load Collision ( a_x = 0.5 g_o ), forward</td>
<td>S+D</td>
<td>S</td>
<td>T_S</td>
<td>Still water bending moment ( M_S = LC 1 ) (hogging):</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Wave bending moment: ( M_W = 0 )</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Acceleration ( a_x = 0.5 g_o ), forward combined with gravity ( g_o )</td>
<td></td>
</tr>
<tr>
<td>LC11 (ALS)</td>
<td>Full load Collision ( a_x = 0.25 g_o ), aftwards</td>
<td>S+D</td>
<td>S</td>
<td>T_S</td>
<td>Still water bending moment ( M_S = LC 1 ) (hogging):</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Wave bending moment: ( M_W = 0 )</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Acceleration ( a_x = 0.25 g_o ), aftwards combined with gravity ( g_o )</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>LC 10 will normally be governing</td>
<td></td>
</tr>
<tr>
<td>LC12 Head Sea (ALS)</td>
<td>Flooded one hold empty</td>
<td>S</td>
<td>S</td>
<td>T_S</td>
<td>Maximum still water hogging bending moment ( M_S ) from T&amp;S booklet (one tank empty): ( M_S \geq 0.5 M_S ) rule hogging.</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Wave bending moment: ( M_W = 0.67 M_{W_hogging} ) in World-Wide environment, ( M_{W_WW} = 0.8 M_{W_NA} )</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Use full draught (Ts) to maximise upward force (anti flotation keys)</td>
<td></td>
</tr>
</tbody>
</table>
### Table 4-2 Load Cases for Tank, Hull and Tank Supporting Structures (Continued)

<table>
<thead>
<tr>
<th>Load Case</th>
<th>Loading condition</th>
<th>Tank load</th>
<th>Sea press.</th>
<th>Draught</th>
<th>Illustration</th>
<th>Comments</th>
</tr>
</thead>
</table>
| LC13 (ULS) | Cargo tank centreline bulkhead Harbour Static (S) | S | S | 0.5TS | ![Still water bending moment: M_S = 0](image) | — Still water bending moment: M_S = 0  
— Wave bending moment: M_W = 0  
— Vapour pressure (P_o) in harbour condition to be added in loaded and empty holds. |
| LC14 (ULS) | Full load, hull design Seagoing Hogging Dynamic (S+D) | S | S+D (10^-4) | T_S | ![Still water bending moment M_S = LC 1 (hogging):](image) | — Still water bending moment M_S = LC 1 (hogging):  
— Wave bending moment: M_W = 0.59 M_W_rule hogging  
This condition will in most cases be overruled by LC 5 at 10^-8 level. |
| LC15 (ULS) | Alternate load, hull design seagoing Hogging Dynamic (S+D) | S | S+D (10^-4) | T_MAX | ![Maximum still water hogging bending moment M_S from T&S booklet, alternate condition or one tank empty others full:](image) | — Maximum still water hogging bending moment M_S from T&S booklet, alternate condition or one tank empty others full:  
— M_S \geq 0.5 M_S_rule hogging.  
— Wave bending moment: M_W = 0.59 M_W_rule hogging |
| LC16 (ALS) | Damaged condition | S | | T_DAM | ![Still water bending moment: M_S = 0](image) | — Still water bending moment: M_S = 0  
— Wave bending moment: M_W = 0  
— Heeled damage waterline to be applied to the transverse bulkhead. The vertical distance shall not be less than up to the bulkhead deck.  
— Primary tank structure to be intact and no leakage from tank. |

**Note:**

1) T_S: scantling draught  
T_MIN: actual minimum draught at any hold loaded condition  
T_MAX: actual maximum draught at any hold empty condition  
T_DAM: damaged draught from damage stability calculation

2) The design conditions given in Table 4.2 assume that the ballast tank under a loaded cargo tank is empty, and the ballast tank under an empty cargo tank is full. If the actual conditions in the loading manual include more severe assumption than above, the actual condition of the ballast tank shall be applied.

3) The design loading conditions given in Table 4.2 and 4.3 are valid for homogeneous and alternate loading of the vessel. However special loading conditions as shown below should be considered for the evaluation of transverse hull bulkhead, if applicable, based on vessel’s loading manual.

- Tank nos. 1 and 4 full, tank 2 and 3 empty  
- Tank nos. 1 and 4 empty, tank 2 and 3 full

When these loading conditions are applied, all of the cargo tanks in the analysis should be empty and full with maximum actual draft and minimum actual draft, respectively.
4.5 Acceptance criteria for ultimate strength

Table 4-4 give usage factors for the load cases in Table 4-2 and Table 4-3 above.

The tanks with supports directly attached to the tanks and the tank support areas shall be designed according to the principles in the International Gas code (IGC) and the DNV gas carrier rules, Pt.5 Ch.5 Sec.5. This means that the dynamic loads in the Ultimate Limit State Condition (ULS) are referred to 10⁸ load cycles in the North Atlantic. This goes for both Ultimate Strength and Fatigue (FLS). For the rest of the ship hull the DNV ship rules (Pt.3 Ch.1) refer to dynamic loads at a 10⁻⁴ probability level both for ULS and FLS.

Allowable stresses and buckling usage factors for each design loading condition are summarized in Table 4-4. Scantlings of the transverse and longitudinal primary cargo tank structures shall be determined according to the yielding and buckling criteria given in Pt.5 Ch.5 Sec.5 H. For convenience Tables H1 to H3 for allowable stress control and Table H4 for stability control in the rules are given below as Table 4-5.
The recommended buckling check procedure is to use the PULS /17/ buckling and ultimate capacity code for both the hull and the cargo tank structure. Buckling usage factors for PULS prescriptive rule criteria are listed in Table 4-4.

In case of highly irregular geometries and/or boundary conditions, nonlinear FE analyses may have to be carried out in order to determine the buckling strength of specific areas. In such cases (e.g. using programmes like ABAQUS) special considerations with respect to modelling (mesh fineness), imperfection levels/modes and acceptance levels is required and will be considered by the Society.

For load conditions designed for evaluation of the hull structure supporting a full cargo tank, e.g. applied probability level $Q = 10^{-8}$, the corresponding strength criteria described in the Rule Pt.5 Ch.5 Sec.5 E300 shall be used.

It should be noted that the stiffener bending stress between transverse girders is not a part of the girder bending stresses. The magnitude of the stiffener bending stress included in the stress results depends on the mesh division and the element type used.

The usage factors for static load cases are set to 70% of dynamic load cases in line with Pt.5 Ch.5 Sec.5 H for both equivalent stress and buckling.

LC1 and LC2 are static load cases. LC3-LC9 and LC13-LC15 are extreme dynamic ULS load cases (dynamic loads referring to North Atlantic conditions with $10^8$ load cycles). The 30° heel cases, LC8 and LC9, are static cases, but will set up stress levels in the tank similar to what will be the case with a transverse acceleration of 0.5 g and is therefore assigned the same acceptance levels as the ULS load cases. LC13 is a static harbour condition for check of the tank centreline bulkhead.

LC14-15 are pure hull design conditions with dynamic loads at $10^{-4}$ probability level, and the usage factors have been scaled down to the lower dynamic load level (see footnotes to Table 4-4).

LC10-11 (collision), LC12 (flooded condition) and LC16 (damaged condition) are accidental conditions (ALS) where the primary tank structures can be allowed to have a higher utilization than an ULS condition, typically full yield capacity for equivalent stress.

LC16 is a flooded condition for check of the integrity of the transverse hull bulkheads. Usage factors for check of the tank are also included.

The acceptance criteria for PULS are set based on a static usage factor of 0.6 for both the hull and the tank, whereas for the ULS conditions the common CSA rule value of 0.9 for the hull is used but with the stricter value of 0.85 for the tank structure.

*In cases where the acceptance criteria are referring to equivalent stress no separate requirement to shear stress is given as shear is already included in the equivalent stress concept.*
Table 4-4  Acceptance criteria for Cargo Tank and Cargo Hold coarse mesh FE Analysis

<table>
<thead>
<tr>
<th>Load Case</th>
<th>Equivalent stress</th>
<th>Directional stress</th>
<th>Mean shear stress</th>
<th>Buckling Criteria</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Hull structures</td>
<td>Cargo tanks and</td>
<td>Hull structures</td>
<td>Cargo tanks and</td>
</tr>
<tr>
<td></td>
<td>Supporting cargo</td>
<td>supporting structure</td>
<td></td>
<td>supporting structure</td>
</tr>
<tr>
<td></td>
<td>tanks</td>
<td>attached to cargo</td>
<td></td>
<td>attached to cargo</td>
</tr>
<tr>
<td></td>
<td>Supporting</td>
<td>tanks</td>
<td></td>
<td>tanks</td>
</tr>
<tr>
<td></td>
<td>structure</td>
<td></td>
<td></td>
<td>(biaxial + shear)</td>
</tr>
<tr>
<td>LC1-LC2</td>
<td>hog. &amp; sag.</td>
<td>S</td>
<td></td>
<td></td>
</tr>
<tr>
<td>LC3-LC5</td>
<td>ULS (hog. &amp; sag.)</td>
<td>S+D</td>
<td></td>
<td></td>
</tr>
<tr>
<td>LC6-LC7</td>
<td>ULS (a_y &amp; g_0)</td>
<td>S+D</td>
<td></td>
<td></td>
</tr>
<tr>
<td>LC8-LC9</td>
<td>ULS (Heel 30°)</td>
<td>S</td>
<td></td>
<td></td>
</tr>
<tr>
<td>LC10-11</td>
<td>ALS (Collision a_y=0.5g_0/0.25g_0)</td>
<td>S+D</td>
<td></td>
<td></td>
</tr>
<tr>
<td>LC12</td>
<td>ALS (Anti-</td>
<td>S+D</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>flotation)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>LC13</td>
<td>ULS (Internal</td>
<td>S</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>struct.)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>LC14-15</td>
<td>ULS (hog.)</td>
<td>S+D</td>
<td></td>
<td></td>
</tr>
<tr>
<td>LC16</td>
<td>ALS (Dam. Flooding)</td>
<td>S</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

| 1) | A/B : A is without hull girder loads / B is with hull girder loads included |
| 2) | When USCG requirements are applied, allowable stress shall be specially considered. |
| 3) | For buckling control the vapour pressure shall be taken as zero, P_o = 0 MPA, and lateral pressure shall be included as relevant. |
| 4) | If two plate flanges, ref. Pt.3 Ch.1 Sec.12 Table B1. |
| 5) | For transverse cargo hold bulkheads |

General comments:

a) Due to the seriousness of tank and tank support system failure the usage factors for static (S) and static plus dynamic (S+D) conditions has been reduced as compared to the values of 0.8 and 1.0 commonly applied for hull structures in general.

b) When using PULS use:
   — analysis option BS, Buckling Strength, for buckling control of primary support members and associated tank support structure.
   — the UC option, Ultimate Capacity, for hull structures in general.
**Table 4-5 General Acceptance Levels for Cargo Hold coarse mesh analyses, Pt.5 Ch.5 Sec.5 H, \( \eta_{S+D} \)**

<table>
<thead>
<tr>
<th>Type of strength member</th>
<th>Design load condition</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Item A708 and A600</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th></th>
<th>( \sigma_e / \sigma_F )</th>
<th>( \sigma_e / \sigma_B )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Primary</td>
<td>0.70</td>
<td>0.50</td>
</tr>
<tr>
<td>Secondary</td>
<td>0.75</td>
<td>0.525</td>
</tr>
<tr>
<td>Tertiary</td>
<td>0.80</td>
<td>0.56</td>
</tr>
</tbody>
</table>

**Table H2 Austenitic steels**

<table>
<thead>
<tr>
<th>Type of strength member</th>
<th>Design load condition</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Item A708 and A600</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th></th>
<th>( \sigma_e / \sigma_{0.2} )</th>
<th>( \sigma_e / \sigma_B )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Primary</td>
<td>0.80</td>
<td>0.375</td>
</tr>
<tr>
<td>Secondary</td>
<td>0.85</td>
<td>0.40</td>
</tr>
<tr>
<td>Tertiary</td>
<td>0.90</td>
<td>0.425</td>
</tr>
</tbody>
</table>

**Table H3 Aluminium alloys**

<table>
<thead>
<tr>
<th>Type of strength member</th>
<th>Design load condition</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Item A708 and A600</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th></th>
<th>( \sigma_e / \sigma_{0.2} )</th>
<th>( \sigma_e / \sigma_B )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Primary</td>
<td>0.75</td>
<td>0.35</td>
</tr>
<tr>
<td>Secondary</td>
<td>0.80</td>
<td>0.375</td>
</tr>
<tr>
<td>Tertiary</td>
<td>0.85</td>
<td>0.40</td>
</tr>
</tbody>
</table>

**Table H4 Allowable stability factor \( \eta \)**

<table>
<thead>
<tr>
<th></th>
<th>Design load condition</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Item A600</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th></th>
<th>( \eta )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Local buckling failure</td>
<td>0.7</td>
</tr>
<tr>
<td>Overall or torsional buckling failure</td>
<td>0.5</td>
</tr>
</tbody>
</table>

**Note:**

1) Allowable stability factor for static (S) load cases has been changed to 70% of the values for (S+D) ULS load cases at 10^{-8} probability level. This brings the static acceptance level for buckling in line with the static value for allowable stresses (H1-3).
5 Local Structural Fine Mesh Analysis (ULS)

5.1 General
Local structural analyses are to be carried out to analyse stresses in high loaded areas of cargo tanks, the tanks supports and supporting hull structures. Stresses in laterally loaded local plate and stiffeners may need to be investigated. Further, stiffeners subjected to large relative deformations between girders or frames and bulkhead shall be investigated along with stress increase in critical areas such as brackets with continuous flanges.

5.2 Locations to be checked
The following areas shown in Table 5-1 in the midship cargo region is to be investigated with fine mesh analysis. The need for fine mesh analysis of these areas may be determined based on a screening of the actual geometry and the results from the cargo hold analysis. Additional locations may also be required to be analysed based on the outcome of the screening.

<table>
<thead>
<tr>
<th>Locations to check</th>
<th>Applied loads</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cargo tanks and Tank supports</td>
<td>— Maximum reaction force from cargo hold analysis (LC 1, 2, 3, 4, 5) in Sec.4 to be combined with horizontal friction force.</td>
</tr>
<tr>
<td></td>
<td>— Vertical reaction force + horizontal transverse friction force</td>
</tr>
<tr>
<td></td>
<td>— Vertical reaction force + horizontal longitudinal friction force</td>
</tr>
<tr>
<td></td>
<td>— Static friction coefficient (before sliding) of min. 0.5 to be used if not otherwise documented by the designer</td>
</tr>
<tr>
<td></td>
<td>— Each representative design to be assessed</td>
</tr>
<tr>
<td>Upper &amp; lower transverse support</td>
<td>— Maximum reaction force from cargo hold analysis (LC 6, 7, 8, 9) in Sec.4 to be applied.</td>
</tr>
<tr>
<td></td>
<td>— For lower transverse supports, total transverse dynamic (when sliding) friction force due to vertical supports to be deducted and not to be taken greater than 0.2 times total weight of the cargo and cargo tank</td>
</tr>
<tr>
<td></td>
<td>— Each representative design to be assessed</td>
</tr>
<tr>
<td>Upper and lower longitudinal support</td>
<td>— Maximum reaction force from cargo hold analysis (LC10 and 11) in Sec.4 to be applied.</td>
</tr>
<tr>
<td></td>
<td>— For lower longitudinal supports, total longitudinal friction force due to vertical supports to be deducted and not to be taken greater than 0.2 times total weight of the cargo and cargo tank</td>
</tr>
<tr>
<td></td>
<td>— Each representative design to be assessed</td>
</tr>
<tr>
<td>Anti-floatation supports</td>
<td>— Reaction force from cargo hold analysis (LC 12) in Sec.4 to be applied.</td>
</tr>
<tr>
<td></td>
<td>— Each representative design to be assessed.</td>
</tr>
<tr>
<td>Fwd &amp; aft end secondary stiffener structures</td>
<td>— Maximum reaction force from cargo hold analysis in Sec.4 to be combined with horizontal friction force, see Figure 5-1.</td>
</tr>
<tr>
<td></td>
<td>— Internal inertia pressure due to</td>
</tr>
<tr>
<td></td>
<td>— Vertical acceleration</td>
</tr>
<tr>
<td></td>
<td>— Transverse acceleration</td>
</tr>
<tr>
<td></td>
<td>— Longitudinal acceleration.</td>
</tr>
<tr>
<td></td>
<td>— Sloshing forces</td>
</tr>
<tr>
<td></td>
<td>— General information on pump tower loads are given in Classification Notes 30.9 Ch.4.</td>
</tr>
<tr>
<td>Hull structures</td>
<td>— Cargo tank full + min. Draft, LC2 in Table 4-2 and Table 4-3</td>
</tr>
<tr>
<td></td>
<td>— Cargo tank full + min. Draft, LC4 in Table 4-2 and Table 4-3</td>
</tr>
<tr>
<td></td>
<td>— Cargo tank empty + max. draft, LC15 in Table 4-3 and Table 4-3</td>
</tr>
<tr>
<td></td>
<td>— Ref. also Nauticus (Newbuilding) requirements</td>
</tr>
<tr>
<td>Transverse Bulkheads</td>
<td>— Vertical stiffeners to inner bottom</td>
</tr>
<tr>
<td></td>
<td>— Relative deflection due to cargo loads</td>
</tr>
</tbody>
</table>
5.3 Structural Modelling

The fine mesh analysis shall be carried out by means of a separate local finite element model with fine mesh zones, in conjunction with the boundary conditions obtained from the cargo tank model, or by incorporating fine mesh zones into the cargo tank model.

The extent of the local finite element models is to be such that the calculated stresses at the areas of interest are not significantly affected by the imposed boundary conditions and application of loads. The boundary of the fine mesh model is to coincide with primary support members, such as girders, stringers and floors, in the cargo tank model.

The fine mesh zone shall represent the geometry of the localised area with high stress. The finite element mesh size within the fine mesh zones is not to be greater than 50 mm $\times$ 50 mm. In general, the extent of the fine mesh zone is not to be less than 10 elements in all directions from the area under investigation.

All plating within the fine mesh zone is to be represented by shell elements. A smooth transition of mesh density is to be maintained. The aspect ratio of elements within the fine mesh zone is to be kept as close to 1:1 as possible. Variation of mesh density within the fine mesh zone and the use of triangular elements are to be avoided. In all cases, the elements are to have an aspect ratio not exceeding 3:1. Distorted elements, with element corner angle less than 45° or greater than 135°, are to be avoided. Stiffeners inside the fine mesh zone are to be modelled using shell elements. Stiffeners outside the fine mesh zones may be modelled using beam elements.

Where fine mesh analysis is required for an opening, the first two layers of elements around the opening are to be modelled with mesh size not greater than 50 mm $\times$ 50 mm. A smooth transition from the fine mesh to the coarser mesh is to be maintained. Edge stiffeners which are welded directly to the edge of an opening are to be modelled with shell elements. Web stiffeners close to an opening may be modelled using rod or beam elements located at a distance of at least 50 mm from the edge of the opening.

Where fine mesh analysis is required for main bracket end connections, the fine mesh zone is to be extended at least 10 elements in all directions from the area subject to assessment.

Face plates of openings, primary support members and associated brackets are to be modelled with at least two elements across their width on either side.

The fine mesh models are to be based on gross scantlings reduced by $t_k$.

The extensions of the local FE support models described below refers to the borders where tapering from coarse global model to the 50 mm $\times$ 50 mm start. The support models shall not only cover the support itself with the associated parts of the hull structure, but also the associated area of the cargo tank in way of the supports.

The most critically loaded:

- vertical
- transverse
- longitudinal
- anti-floatation

supports should generally follow the requirements below:
5.3.1 Modelling of vertical supports
— Two web frame spaces is to be modelled in way of aft or forward end bulkhead in general.
— The local model should extend one web frame spacing forward and aft of the vertical support in longitudinal direction.
— In the transverse direction, the model should in general include the neighbouring primary supporting structures.
— Hull and cargo tank structures in way of the above supports should normally be modelled in full breadth.

5.3.2 Modelling of transverse supports
For modelling of transverse supports, the transverse extension of the local model should in general be as for the vertical supports. In longitudinal direction, the extension is required to be two web frame spaces.
— One web frame space on either side of the support.
— If different types are employed, make each model.
— Upper and lower supports shall be modelled.
— Hull and cargo tank structures in way of the above supports shall be modelled.

5.3.3 Modelling of longitudinal supports
— Longitudinal extension of the model may be two web frame spaces, i.e. forward and aft of the support.
— In the transverse direction, symmetry may be considered and the extension of the model should normally be one longitudinal space from the edge of the support.
— One anti-pitch support to be modelled.
— Hull and cargo tank structures in way of the above supports shall be modelled.

5.3.4 Modelling of anti-floatation supports
— One typical support should be modelled. The model should include necessary surrounding structure and using boundary conditions from the global model.
— Two frame space model, port only in way of end bulkhead.
— Hull and cargo tank structures in way of the above supports shall be modelled (port only).

5.3.5 Modelling of stiffeners subject to large lateral deformation
— Forward & aft end secondary stiffener in cargo tank, double bottom longitudinal with brackets subjected to large deformations should be modelled.
— The stiffener model is to be extended longitudinally at least two web frame spaces from the areas under investigation.
— The model width is to be at least 1+1 longitudinal spaces.
— The web of the longitudinal stiffener should be represented by at least 3 shell element across its depth
— The face plate of the longitudinal stiffener and bracket should be modelled with at least two elements across its width on either side.
— The prescribed displacements obtained from the cargo tank FE model should be applied to all boundary nodes which coincide with the cargo tank model.

5.4 Load Cases
The fine mesh analysis in way of cargo tanks and tank supports is to be carried out for the load cases specified in Table 4-2 and Table 4-3 for the locations outlined in Table 5-1. However, not all the load cases listed in Table 4-2 and Table 4-3 may be governing. The actual tank design and support configuration may vary and the applicable load cases will have to be selected accordingly.

The fine mesh analysis of double bottom longitudinal with brackets subject to large relative deformation is to be carried out for the load cases LC2, LC4 and LC16 in the areas specified in Table 5-1.

5.5 Application of Loads and Boundary Conditions
Where a separate local finite element model is used for the fine mesh detailed stress analysis, the nodal displacements from the cargo tank model are to be applied to the corresponding boundary nodes on the local model as prescribed displacements. Alternatively, equivalent nodal forces from the cargo tank model may be applied to the boundary nodes. The fine mesh model can also be an integral part of the cargo hold model.

Where there are nodes on the local model boundaries which are not coincident with the nodal points on the cargo tank model, it is acceptable to impose prescribed displacements on these nodes using multi-point constraints. The use of linear multi-point constraint equations connecting two neighbouring coincident nodes is considered sufficient.

All local loads are to be applied to the local finite element model.
5.6 Acceptance Criteria

The von Mises stress is to be calculated based on the membrane axial and shear stresses of the plate element evaluated at the element centroid. Where shell elements are used, the stresses are to be evaluated at the midplane of the element (membrane stress).

It is required that the resulting von Mises stresses are not exceeding the allowable membrane values specified in Table 5-2. These criteria apply to regions where stress concentrations occur due to irregular geometries. Nominal stresses shall remain within the limits given in in Table 4-4.

When mesh sizes smaller than 50 mm $\times$ 50 mm is used, the average stress is to be calculated based on stresses at the element centroid. Stress averaging is not to be carried across structural discontinuities and abutting structure.

### Table 5-2 Maximum allowable membrane stresses for local fine mesh analysis

<table>
<thead>
<tr>
<th>Element stress</th>
<th>Allowable stresses</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cargo tank and hull structures supporting cargo tank</td>
<td></td>
</tr>
<tr>
<td>Element not adjacent to weld (base material)</td>
<td>$1.36\sigma_F$</td>
</tr>
<tr>
<td>Element adjacent to weld</td>
<td>$1.2\sigma_F$</td>
</tr>
<tr>
<td>Other hull Structures</td>
<td></td>
</tr>
<tr>
<td>Element not adjacent to weld (base material)</td>
<td>$1.7\sigma_F$</td>
</tr>
<tr>
<td>Element adjacent to weld</td>
<td>$1.5\sigma_F$</td>
</tr>
</tbody>
</table>

**Note:**

1) The maximum allowable stresses are based on the mesh size of 50 mm $\times$ 50 mm. Where a smaller mesh size is used, an average von Mises stress calculated over an area equal to the specified mesh size may be used to compare with the permissible stresses.

2) Average von Mises stress is to be calculated based on weighted average against element areas:

$$\sigma_{von\_av} = \frac{\sum_{i=1}^{n} A_i \sigma_{von\_i}}{\sum_{i=1}^{n} A_i}$$

where

- $\sigma_{von\_av}$: the average von Mises stress
- $\sigma_{von\_i}$: the von Mises stress of the $i$th plate element within the area considered
- $A_i$: the area of the $i$th plate element within the area considered
- $n$: the number of elements within the area considered

3) Stress averaging is not to be carried across structural discontinuities and abutting structure and stress values obtained by interpolation and/or extrapolation are not to be used.

5.7 Structural verification for wood and dam plate

The detailed configuration of the wood and the dam plate should preferably be included in the local FE model of the supports. A simplified approach is outlined in [5.7.1] and [5.7.2].

5.7.1 Strength of wood

Strength of wood should carefully be checked in view of compressive strength and shear strength. Figure 5-2 shows an assumed of force transmission to wood if applied from the support. It is assumed that the reaction force from a support will be transmitted through the top plate of the supports with angle of 90 degrees. The compressive strength at wood can be checked as follows;

$$\sigma_r = \frac{f_z}{A_{loaded}} \leq \frac{\sigma_{wood}}{\gamma_{SF}}$$

where

- $f_z$ = load on the support normal to the support surface.
- $\gamma_{SF}$ = safety factor, 3.0 for wood
Figure 5-2
Assumption of force transmitted

The shear strength of wood shall be checked using the transverse friction force due to the maximum vertical force or maximum longitudinal force applied the support to be considered. Thus, the shear strength at wood shall be satisfied as follows;

\[ \tau_r = \frac{f_h}{A_w} < \frac{\tau_{wood}}{\gamma_{SF}} \]

where,

- \( f_h = \mu f_z \) Friction force at support
- \( \gamma_{SF} = \) safety factor, 3.0 for wood
- \( A_w = \) shear area of wood, mm\(^2\)
- \( \tau_{wood} = \) minimum shear strength of wood, N/mm\(^2\).

5.7.2 Assessment of dam plate

Dam plate shall be fitted against the friction force and be designed with 10% of the maximum force applied to the support to be considered, when adhesive and resin strength may be damaged. The required shear area of dam plate is given as follows;

\[ A_d \geq \frac{f_h}{\tau_{allow}} \text{, mm} \]

where

- \( \tau_{allow} = \) allowable shear stress of dam plate, N/mm\(^2\)
  \[ = 0.95 \sigma_f \]

A small size bracket may be fitted to prevent yielding of the dam plate, if dam plate area is not sufficient.

Figure 5-3
Force applied to dam plate

Bending strength of dam plate shall also be checked with designed with 10% of the maximum force applied to the support to be considered. Allowable stress is 0.95\( \sigma_f \).

5.7.3 Material data

Material strength data shall supplied by the designer based on certification of the relevant materials.
6 Thermal Analysis

6.1 General
To determine the grade of plate and sections used in the hull structure, a temperature calculation shall be performed for all tank types when the cargo temperature is below -10°C, Pt.5 Ch.5 Sec.2 B500.

Steady state thermal analysis of hold area and the cargo tank shall be performed for all tank types when the cargo temperature is below -10°C (Rules Pt.5 Ch.5 Sec.2 B500) to:

— determine steel temperature as basis for material quality selection of the surrounding hull structure, and
— as input to thermal stress analysis to confirm the structural integrity of the cargo tank and support system with respect to yield and buckling in partial and full load conditions.

Transient thermally induced loads during cooling down periods shall be considered for tanks intended for cargo temperatures below -55°C as required by the rules Pt.5 Ch.5 Sec.5 A901.

6.2 Thermal stress analysis
Load cases should at least be considered as follows;

— LC1: full load condition (98% filling) to determine maximum cool-down of surrounding hull structure.
— LC2: partial load condition, filling to each stringer level as relevant to determine stress ranges for low cycle fatigue analysis for the full thermal cycle due to loading and unloading of cargo.

Thermal loads (temperature distribution loads) should be specified along the tank height for each design load case. For partial load conditions and full load condition, thermal load, static cargo pressure and minimum design vapour pressure should be applied. Deflection of double bottom structure shall be taken into account for all load conditions.

6.3 Acceptance Criteria
Allowable stress for the design load cases including thermal stress shall not exceed two times the relevant values given in Table 4-4.

Local buckling of plates between stiffeners under thermal stress shall be checked against structural stability. Allowable stability factors are given in the rules, Pt.5 Ch.5 Sec.5 E308.
7 Sloshing Assessment
For partial tank fillings the risk of significant loads due to sloshing induced by ship motions shall be considered.

7.1 Sloshing strength analysis
The tank boundary structure shall be designed to withstand loads caused by liquid sloshing. The design sloshing pressures are to be explicitly considered in the scantling requirements of plates and stiffeners.

As a minimum the tank shall be designed for the sloshing inertia and impact pressure loads given in DNV Rules Pt.3 Ch.1 Sec.4. Based on experience, this will normally be considered sufficient if swash bulkheads are arranged to reduce liquid sloshing resonances in the tanks.

The acceptance criteria for sloshing strength analysis shall be according to the rules, Pt.3 Ch.1 Sec.4. Please note that the rule sloshing pressures are referred to $10^{-4}$ probability level.

For tanks built without swash bulkheads and/or longitudinal bulkhead, or where the liquid motion resonance period is found to be close to the natural motion periods of the ship (see [7.2]) the need for documentation by more comprehensive sloshing assessments (e.g. CFD and/or model testing) will be considered by the Society in each case.

7.2 Liquid resonance interaction
Interaction of liquid sloshing motion with the natural ship motion periods may cause violent sloshing motion of liquids inside the tanks. Normally, the lowest natural liquid periods should be 20% away from the natural ship motion periods to limit this effect. The fitting of swash bulkheads can move the liquid resonance periods away from the motion periods of the ship and significantly reduce sloshing loads inside the tanks.

The natural periods for liquid motion for a prismatic tank can be approximated by, Ref. /8/;

$$T_i = \frac{2\pi}{\sqrt{g\kappa_i}} : \text{Natural sloshing period for mode } i=1, 2, \ldots$$

where

$$f_i = \sqrt{g\kappa_i} : \text{Natural sloshing frequency for mode } i=1, 2, \ldots$$

$$\kappa_i = \frac{\pi i}{l} \tanh \left( \frac{\pi i}{l} h \right)$$

$l = \text{length or breath of free liquid surface at filling height } h$

$h = \text{filling height (distance from tank bottom to free surface)}$
8 Fatigue Analysis

The fatigue assessment shall be carried out for details in the hull, cargo area, the cargo tanks and tank supports. Hull and cargo tank structures shall be designed to satisfy a design fatigue life of at least 20 years in worldwide operation for the hull structure and $10^8$ wave encounters in the North Atlantic for cargo tank and tank support structure.

This section describes the procedure to perform fatigue analysis of cargo tank and hull structures. Additional requirements may apply for the hull structure depending on class notations (e.g. PLUS, CSA).

Fatigue analysis of a cargo tank and its supports shall be carried out in accordance with DNV Rules Pt.5 Ch.5 Sec.5 A1400.

Fatigue analysis of hull structures shall be carried out in accordance with Pt.3 Ch.1. For details of fatigue strength assessment, it is referred to Classification Notes 30.7, *Fatigue Assessment of Ship Structures*.

The fatigue requirements for the tank specified in this section are linked to the fracture mechanics analysis, Pt.5 Ch.5 Sec.5 A1300.

8.1 Fatigue damage accumulation

The fatigue analysis type B-carriers is normally based on rule loads for the hull, but wave load analyses shall be carried out for analysis of tanks and the tank support system, Pt.5 Ch.5 Sec.5 F102. The stresses shall be determined by the use of finite element models.

A fatigue analysis shall be carried out in accordance with DNV Rules Pt.5 Ch.5 Sec.5 A1400. The total fatigue damage may be obtained for cargo tanks as follows:

$$D = \sum_{i=1}^{k} \left( \frac{n_i}{N_i} \right) + \frac{n_l}{N_j} \leq C_w$$

where

- $D$ = accumulated fatigue damage ratio
- $n_i$ = number of cycles in stress block “i”
- $N_i$ = number of cycles to failure at constant stress range $\Delta \sigma$ as determined by an appropriate S-N curve
- $n_l$ = number of loading and unloading cycles covering the complete pressure and temperature range during the lifetime of the vessel. To be taken as $10^3$ for a trading carriers (IGC). Can be considerably larger for vessels engaged in offshore operations as SRV - LNG Shuttle and Regasification Vessel System, FSRU - Floating Storage and Regasification Unit, FLNG - Floating Liquefied Natural Gas etc. that frequently will operate with part tank fillings.
- $N_j$ = number of load cycles to failure for fatigue loads due to variable fillings, loading and unloading
- $k$ = number of stress blocks, $\geq 8$
- $\bar{a}, m$ = parameters defining the fatigue S-N curve
- $\Delta \sigma$ = stress range in stress block “i”
- $C_w$ = acceptable accumulated fatigue damage levels as given in Table 8-3.

**Guidance note:**

The first term in the damage equations above can most conveniently be determined by the alternative formulation in Classification Notes 30.7 (Sec. 2) using a Gamma function and a long term Weibull load distribution with a shape parameter $h = 1.0$ as per IGC requirement.

---end---of---Guidance---note---

8.2 Fatigue damage evaluations

The long term distributions of stresses at the critical welds shall be determined for the loaded, part load and ballast conditions. The combined effect for the fatigue analysis can be determined as outlined in bullets a. and b. below considering the operational profile of the vessel. The operational profile is defining the fraction of the total lifetime spent in the actual loading conditions - full load, ballast, part loads and at various heading angles, Table 8-2.
The fatigue life can be determined in basically two different ways.

a) By adding up damage contributions

— Fatigue damage contributions (Miner sums) calculated for each loading condition can be added according to the operational profile of the vessel to give the total fatigue damage contribution over the reference lifetime of the vessel.
— For the part load condition the fatigue damage contributions from the considered filling levels can be added according to the operating time at each filling level. A minimum of three part filling levels are to be used.

b) By establishing a resulting long term Weibull stress distribution representative for the expected operation of the vessel over its life time. This can be done by combining the long term stress distribution for all the load cases as a weighted sum according to the operational profile for the vessel. Fatigue analysis on this basis is to be compared to the total design lifetime.

### 8.3 Locations to be checked for fatigue

The fatigue strength assessment is to be carried out for cargo tank, tank supports and hull structures in the cargo area as specified in Table 8-1. Additional areas may have to be analysed based on specific structural configurations.

<table>
<thead>
<tr>
<th>Structure member</th>
<th>Structural detail</th>
<th>Load type</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hull structures</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Side, bottom, inner side, hopper slope and deck longitudinals.</td>
<td>End connections of longitudinal stiffeners to web frames and transverse bulkheads</td>
<td>Hull girder wave bending</td>
</tr>
<tr>
<td>Hopper knuckles</td>
<td>Lower hopper</td>
<td>Dynamic sea pressure load</td>
</tr>
<tr>
<td>Deck openings. Cargo tank dome</td>
<td>Cargo tank dome and Deck openings and coamings</td>
<td>Dynamic ballast pressure load</td>
</tr>
<tr>
<td>Double bottom longitudinal girder</td>
<td>Longitudinal girder connection to transverse bulkhead</td>
<td>Dynamic cargo loads</td>
</tr>
<tr>
<td>Watertight transverse bulkheads</td>
<td>Vertical stiffeners to inner bottom</td>
<td>Relative deflection due to cargo loads</td>
</tr>
<tr>
<td>Cargo tank, tank supports,* and cargo tank in way of supports</td>
<td>Vertical supports</td>
<td>Internal pressure due to</td>
</tr>
<tr>
<td></td>
<td>Upper &amp; lower transverse supports</td>
<td>— Vertical acceleration</td>
</tr>
<tr>
<td></td>
<td>Fwd &amp; aft end secondary stiffeners</td>
<td>— Transverse acceleration</td>
</tr>
<tr>
<td></td>
<td>High stressed tank structure in way of supports</td>
<td>— Longitudinal acceleration.</td>
</tr>
<tr>
<td></td>
<td>Bracket ends</td>
<td>Dynamic sea pressure</td>
</tr>
<tr>
<td></td>
<td>End connection of stiffeners</td>
<td>Hull girder wave bending</td>
</tr>
<tr>
<td></td>
<td>Primary barrier shell plate to stiffeners and frames / girders</td>
<td>Sloshing (if relevant)</td>
</tr>
<tr>
<td>Cargo tank pump tower (if used)</td>
<td>Tower and tower supports</td>
<td>Internal inertia pressure due to</td>
</tr>
<tr>
<td></td>
<td>Tank dome connection</td>
<td>— Vertical acceleration</td>
</tr>
</tbody>
</table>

* Tank supports includes the part of the tank structure in way of supports and the support structure welded to the hull, and adjacent hull structure where the stress mainly originates from the presence of the tank.

**Note:** Several methods for fatigue analyses are available; simplified beam approach, component spectral (stochastic) analysis and full spectral (stochastic) analysis. See Classification Notes 30.7 for details.

### 8.4 Finite Element Models

The determination of nominal stresses ranges for use with S-N curves and stress concentration factors can be based on cargo hold model meshes or fine mesh analysis models dependent on the suitability for use with the...
actual detail to be analysed. Alternatively, stresses can be extrapolated to the hot spot from a very fine mess $(t \times t)$ mesh FE analysis. The extrapolation procedure is described in Classification Notes 30.7/5/.

8.5 Calculation of dynamic stress range spectra
For each loading condition, local stress components due to simultaneous internal and external pressure loads are to be combined with stress induced by dynamic hull deflections. Detailed description of the combination of the stress components are given in Classification Notes 30.7/5/.

The long term distribution of stress ranges at local details may be described by a Weibull distribution. For the hull structure Weibull slope parameters defined by analytical expression may be used as given in Classification Notes 30.7/5/. Unless otherwise agreed a Weibull shape parameter of 1.0 as given in the IGC code may normally be used for the tank and tank support structure.

As the main contribution to the cumulative fatigue damage comes from the smaller waves, the long term reference stress range should be referred to the $10^{-4}$ probability level.

8.6 Stress processing for S-N curve fatigue analysis
Fatigue and fracture mechanics analyses shall be carried out based on the largest principal stress at the considered location. Geometrical stress concentration factors not accounted for in the FE-model, e.g. shell thickness changes, can be calculated according to Classification Notes 30.7/5/ and applied to the analysed stress ranges.

The principal stresses to be used in the fatigue evaluation shall be calculated as follows:

a) Determine the static and dynamic combined stress for each (all) surface stress component, x, y and shear. This shall be done at the both surfaces.

b) Calculate principal dynamic stress ranges separately at both surfaces. The largest principal dynamic surface stress range within $\pm 45^\circ$ off the perpendicular to the weld (crack) is to be used in the S-N curve fatigue analysis.

8.7 Fatigue Strength Analysis Procedures
For independent B-tanks wave load calculations are to be carried out in compliance with Classification Notes 30.7/5//Classification Notes 34.1. Rule loads shall be used for the hull unless otherwise specified by the owner, or if the voluntary notation CSA is specified.

8.7.1 Loading conditions
Fatigue analyses are to be carried out for representative loading conditions according to the ship’s intended operation as given in the loading manual (the Trim and Stability Booklet). The following loading conditions shall be represented as applicable:

Hull:

— Homogeneous full load condition at design draught (departure).
— Ballast condition at normal ballast draught (departure). If a normal ballast condition is not defined in the loading manual, minimum ballast draught should be used.

Tank and supports:

— Homogeneous full load condition at design draught (departure).
— Partial tank fillings as relevant. The following effects should be considered; sloshing inertia effects and local and global impacts due to sloshing.
— Ballast condition at normal ballast draught (departure). If a normal ballast condition is not defined in the loading manual, minimum ballast draught should be used (10% filling if applicable).

Pump tower or cargo pipe attachment to aft tank transverse bulkhead if pump tower not used:

— Partial fillings: If relevant a minimum of 3 part filling levels are to be used.
— Ballast condition at normal ballast draught (departure). If a normal ballast condition is not defined in the loading manual, minimum ballast draught should be used (10% filling if applicable).

8.7.2 Operating profile
The ship loading conditions to be used in the fatigue and fracture mechanics analyses are given in Table 8-2 below for a normal trading carrier (trading with full load on entire laden voyage and in ballast for return transit).

Other distributions of time fractions (exposure times) may be applied for trades that require the ship to operate more of the time in part load conditions.
The design loading conditions and exposure times giving basis for the fatigue calculations shall be stated in the Appendix to the Class Certificate.

| Table 8-2 Design loading conditions and exposure times, \( p_n \) |
|---------------------------------|-------------------------------------------------|-------------------------------------------------|
| Loading Condition               | Fraction of life time, tank and supports | Fraction of life time, tower and tower supports |
| Full Load, all tanks full       | 45%                                            | -                                              |
| Ballast, all tanks empty        | 40%                                            | *                                              |
| Part load conditions            | -                                              | 5%                                             |

* If the vessel is sailing with heel in tanks (e.g. up to 10% filling height) in ballast voyage, this should be taken into account for the duration of the ballast voyage.

8.7.3 Operation route factor
A correction factor \( f_e \) for different trading routes may be applied for the ship hull as described in Classification Notes 30.7 accounting for the long-term sailing routes of the ship. For worldwide operation the factor may be taken as 0.8. This means that for world-wide operation the stress ranges are taken as 80% of the stress ranges obtained from North Atlantic operation. The North Atlantic scatter diagram is given in Classification Notes 30.7 /5/ and in DNV-RP-C205 Environmental Conditions and Environmental Loads /7/.

The fatigue analyses of the cargo tanks and the supporting structure shall be referred to \( 10^8 \) wave cycles in North Atlantic environmental as specified by the IGC code.

8.7.4 Corrosion
Corrosion addition should be taken into account according to the DNV Rules Pt.3 Ch.1 /1/. The basic S-N curve for welded regions in air is to be applied for joints situated in dry spaces or joints effectively protected against corrosion.

For inner hull joints facing to cargo tanks, the S-N curve in air may normally be used.

For joints efficiently protected only a part of the design life and exposed to corrosive environment the remaining part, the fatigue damage may be calculated as a sum of partial damages according to Classification Notes 30.7.

8.7.5 Fatigue S-N curves
The design S-N curves shall be based on mean-minus-two-standard-deviation curves (m-2s) for relevant experimental data corresponding to 97.6% probability of survival of the S-N test data.

For the cargo tank, fatigue properties of welded and base material used should be documented from experiments relevant for both room temperature and cryogenic temperature.

If relevant S-N curves for the considered material(s) are not available, the designer may need to develop such data to the satisfaction of DNV. General procedures for development and documentation of S-N curves can be found in Classification Notes 30.7 /5/, but it is recommended that this is discussed with DNV before extensive studies are launched.

The hull structure weld joints within the scope of the assessment can be designed based on the S-N curves in Classification Notes 30.7 /5/ provided normal or high strength steel is used. The basic S-N curve for welded regions in air is to be applied for joints situated in dry spaces or joints effectively protected against corrosion. Void cargo hold spaces between hull and cargo tanks are normally considered as dry space.

8.7.6 Stress concentration factors
The fatigue life of a detail is governed by the hot spot stress range. The hot spot stress range is obtained by multiplication of the nominal stress by stress concentration factors (K-factors). K-factors for the most common details are tabulated in Classification Notes 30.7 /5/. For special details (geometries) not covered by the “standard” tabulated details in Classification Notes 30.7 /5/, local very fine mesh analyses with FE meshes of size \( t \times t \), where \( t \) is the plate thickness, can be made to determine stress concentration factors for the actual detail.

8.7.7 Mean stress effect
The stress range may be reduced dependent on whether mean cycling stress is tension or compression. This reduction may be carried out for the base material and weld joints. The calculated stress range obtained may be multiplied by the reduction factor \( f_{m} \). Details of the mean stress factor is given in Classification Notes 30.7 /5/.

8.7.8 Effect of weld toe grinding
According to DNV-RP-C203 Fatigue Design of Offshore Steel Structures /19/, the fatigue life can be improved by grinding with a factor of 0.01 \( \sigma_{y} \) (max. 3.5), where \( \sigma_{y} \) = the characteristic yield strength of the material. As corrosion of ground metal surfaces virtually eliminates the benefit of burr grinding the ground surface must be adequately protected against corrosion.

However, toe grinding at design stage is normally not to be used if the damage factor \( C_w \) is larger than 1.47Cw-allowable according to Table 8-3.
8.8 Fatigue Assessment of Cargo Tanks

8.8.1 Fatigue load cases
The load cases, i.e. LC3, LC5, LC6 and LC 7, from Table 4-2 and Table 4-3 are to be applied for fatigue analysis of the cargo tanks and cargo tank supports.

8.8.2 Fatigue damage ratio
The fatigue damage ratio may be calculated based on the S-N fatigue approach under the assumption of linear cumulative damage (Miner-Palmgren’s rule) as described above. Acceptable fatigue damage ratios are specified in Table 8-3. The table refers to the principle of Leak-Before-Failure (LBF) as defined in [1.2].

In Table 8-4 North Atlantic environmental conditions corresponding to 10^8 wave encounters is used as reference for the cargo tank and its supports in line with IGC (4.3.4) requirements.

### Table 8-3 Required fatigue damage ratios, \( C_W \)

<table>
<thead>
<tr>
<th>Area</th>
<th>Requirement</th>
<th>Comment</th>
</tr>
</thead>
</table>
| Primary barrier (outer tank shell plates, attached stiffeners and adjacent structure) | For failures that can be reliably detected by means of leakage detection;  
— \( C_W \leq 0.5 \)  
— Predicted remaining failure development time, from the point of detection of leakage till reaching a critical state, shall not be less than 15 days unless different requirements apply for ships engaged in particular voyages. | Leak-Before-Failure (LBF) proven. |
| Primary structures (girders, stringers, web frames) | For failures that cannot be detected by leakage but can be reliably be detected at the time of in-service inspections  
— \( C_W \leq 0.5 \)  
— Predicted remaining failure development time, from the largest crack not detectable by in-service inspection methods until reaching a critical state, shall not be less than three (3) times the inspection interval. | No leak, not LBF |
| Primary barrier, secondary and tertiary structures where relevant. | In particular locations of the tank where effective defect or crack development detection cannot be assured, the following, more stringent, fatigue acceptance criteria should be applied as a minimum;  
— \( C_W \leq 0.1 \)  
— Predicted failure development time from the assumed initial defect until reaching a critical state, shall not be less than three (3) times the lifetime of the tank. | No leak, not LBF |
| Tank supports and associated hull structure | — \( C_W \leq 0.5 \)  
— Hull structure in general | — \( C_W \leq 1.0 \)  
Follow procedures in Classification Notes 30.7/5/ |

8.8.3 Global structural analysis
Global structural analysis is performed to identify fatigue prone areas by fatigue screening and serve as basis for the local finite element analyses.

Global fatigue screening analyses are performed based on 20 years operation in North Atlantic operation with exposure times \( p_e \) defined in Table 8-2. The calculations are based on the DNV S-N curve I for welded joints in air. The fatigue screening is based on nominal membrane stresses.

8.8.4 Local structural analysis
The solution (displacements) of the global analysis is transferred to the local models using sub-modelling technique. The idea of sub-modelling is in general that a particular portion of a global model is separated from the rest of the structure, re-meshed and analysed in greater detail. The calculated deformations from the global analysis are applied as boundary conditions on the borders of the sub-models. The appropriate boundary solutions are determined and applied to the sub-models. Loads corresponding to the global model results must be applied to the local model. Local fine mesh models can also be set up as integrated refined parts of coarser models.

8.9 Fatigue assessment of cargo tank supports
The largest dynamic forces acting on each keys and supports will be induced under the full load condition. However, in case of the ballast condition, the dynamic forces from self-weight of cargo tanks may be negligible compared to the full load condition.
8.9.1 Locations to be checked
The following locations may be critical in view of fatigue.

— Vertical support connection to cargo tanks and inner bottom
— Lower roll support connection to cargo tanks and inner bottom in transverse direction
— Upper roll support connection to deck transverse web in transverse direction
— Upper roll support connection to cargo tank top in transverse direction
— Pitch support to cargo tanks and inner bottom in longitudinal direction.

8.9.2 Vertical supports
Dynamic stresses in the vertical supports are caused by the following dynamic loads:

— Horizontal acceleration,
— Vertical acceleration,
— Sea pressure
— Double bottom bending
— Hull girder bending.

The evaluation has to consider each support type considering maximum lateral support force. The static friction coefficient can be taken as 0.5 and the sliding friction coefficient is set to 0.2, unless otherwise documented by the designer, Table 4-1.

The analysis procedure is:

1) Vertical loads are applied.
2) Transverse loads are applied assuming conservative static friction $\mu = 0.5$. The tank will not be sliding for the dominating fatigue loads. See Figure 8-2.
3) Hull girder loads are also to be applied.
4) Correlations to be applied according to Classification Notes 30.7/5/.

8.9.2.1 Structures loaded in the transverse direction
A. Dynamic stress caused by vertical acceleration

The distribution of vertical forces may be found from the load cases in Table 4-2 and Table 4-3.

1) LC 3: fully loaded hogging condition; hull girder loads with dynamic internal (S+D) pressure and static (S) external pressure. and
2) LC 5: fully loaded hogging condition; hull girder loads with internal static (S) pressure and external (S+D) pressure.

Based on the calculated support forces from the mentioned load cases, the relevant stress may be calculated. The stress range in the vertical supports shall be calculated as the difference between the stress values in LC 3 and in LC 5 after deduction of static stress from LC 1. The long term distribution of the stress is shown in Figure 8-1.

This will produce conservative results due to taking maximum down loads and up loads to define the stress range.

This analysis will include the effect of hull girder bending as well as double bottom bending. This will influence on the distribution of vertical forces in the supports.
B. Dynamic stress caused by transverse acceleration

At a certain magnitude of the transverse acceleration the cargo tanks will slide on the vertical supports. The probability level, \( Q = 10^{-N} \), where the change from static to sliding friction may be calculated as

\[
N = \left( \frac{\mu_s}{a_t} \right) \times 8
\]

where,
- \( \mu_s = 0.5 \) the static friction coefficient
- \( a_t \) is the calculated transverse acceleration as fraction of gravity (g) at probability level \( Q = 10^{-8} \)

The position where the change from static to sliding friction will occur is indicated in Figure 8-2.

The FEM calculation shall include 2 load cases specified in Table 8-4.

<table>
<thead>
<tr>
<th>Table 8-4 Load cases for transverse stress calculation</th>
</tr>
</thead>
<tbody>
<tr>
<td>( LC1 )</td>
</tr>
<tr>
<td>Transverse acceleration, ( a_t )</td>
</tr>
<tr>
<td>Point A</td>
</tr>
<tr>
<td>Point B</td>
</tr>
<tr>
<td>Point C</td>
</tr>
</tbody>
</table>

The stress range in the supports shall be calculated considering the transverse forces to port and starboard, Figure 8-3.

Sliding will occur if the transverse acceleration \( \left( a_t / g_o \right) \) is larger than the static friction coefficient, \( \mu_s \).

Normally, the transverse acceleration \( a_t / g_o \) will at \( 10^{-8} \) probability level be of the order 0.4 to 0.6. According to the formula above sliding will then begin (point A in Figure 8-2) at \( 10^{-10} \) to \( 10^{-6.7} \) probability level. This is well beyond the probability level of \( 10^{-4} \) to \( 10^{-2} \) where most of the fatigue damage will be accumulated. Hence, for practical purposes, calculations can in most cases be carried out without taking sliding into account.
The assumption that the smaller loads are only carried by friction implies that there is no initial contact at the transverse supports. In addition, it is assumed that the (shear) deformation of the vertical support and tank will not cause contact.

However, based on the actual configuration, the long term distribution of loads on the transverse support should consider the possibility of interaction (friction force and contact) also for smaller loads, taking into account actual production tolerances and gaps.

**C. Total combined stress**

Combined stress from vertical acceleration and transverse acceleration is calculated as below:

$$\sigma_{comb} = \sqrt{\sigma_v^2 + \sigma_{tv}^2}$$

The calculation is to be carried out at characteristic fatigue sensitive points. The principal stress normal to, or within ± 45° of the normal, to the weld in question shall be used as basis for the fatigue calculation, Ref. Classification Notes 30.7 Sec. 2.3.

**8.9.2.2 Structure loaded in the longitudinal direction**

When considering the longitudinal direction longitudinal acceleration, hull girder bending and double bottom bending need to be considered in order to determine the actual number of vertical supports in contact with the tank.

**A. Stresses caused by vertical acceleration**

To be calculated as described [8.9.2.1]A. This analysis will include the effects of vertical acceleration, double bottom bending and hull girder bending.

**B. Dynamic stress caused by longitudinal acceleration**

Sliding will only occur if the longitudinal acceleration becomes larger than the static friction coefficient. The typical longitudinal acceleration is in the range of 0.15 ~ 0.2g. Hence, with a static friction coefficient of 0.5 no sliding is expected for this load condition. The stress amplitude may be calculated based on the load application shown in Figure 8-4.

$$P_{long} = \left( P_w \cdot a_x \right) / n$$

where,  

- $P_w$ = static cargo weight  
- $a_x$ = longitudinal acceleration  
- $n$ = number of active vertical supports allowing for the deformation of the double bottom.

**Figure 8-4**

Load application to vertical supports (longitudinal direction)

**Figure 8-5**

Long term distribution of stress for longitudinal load

The long term distribution of the stress is shown in Figure 8-5.

**C. Hull girder bending**

The hull girder bending causes shortening/elongation of the inner bottom. The cargo tank is in most cases rather stiff and will resist the imposed deformation from the inner bottom. Thus, it will create longitudinal forces in the vertical supports. The flexibility at/in the vertical supports will have effect on the longitudinal forces.
occurring in the vertical supports. In order to take into account the flexibility of the supports, calculations may have to be carried out. The flexibility will have effect on at which level of the hull girder moment sliding will occur.

D. Total combined stress

The combined stress from vertical acceleration and double bottom bending ($\sigma_v$), longitudinal acceleration ($\sigma_{long}$) and hull girder bending may be calculated by using the formula below:

$$\sigma_{comb} = \sqrt{\sigma_v^2 + (\sigma_{long} + \sigma_{hull\_bending})^2}$$

The calculation is to be carried out at characteristic fatigue sensitive points. The principal stress normal to, or within $\pm 45^\circ$ of the normal, to the weld in question shall be used as basis for the fatigue calculation, Ref. Classification Notes 30.7 Sec. 2.3.

8.9.3 Transverse supports

Dynamic stresses in the transverse supports are caused by transverse acceleration, LC 6 and LC 7, Table 4-2 and Table 4-3.

The total transverse forces acting on the cargo tanks will be supported by upper and lower transverse supports and friction forces in the vertical supports. The distribution of the supporting forces between friction force in the vertical supports and forces in the transverse supports will vary depending on the magnitude of the transverse dynamic force (transverse acceleration) and the actual gaps at the transverse supports. The analysis can be carried out using an iterative procedure as described in [4.1.7.2].

The following assumptions are made as below:

— With small transverse accelerations, the transverse load from the cargo tank will be carried by the friction in the vertical supports only. If the deformation of the tank and the vertical support cause activation of the transverse support, this load will have to be considered.

— With larger transverse acceleration, the transverse cargo tank load will be carried partly by friction in vertical supports and partly by forces in the transverse supports.

— The friction coefficient used should be considered based on a low value for the dynamic friction coefficient that shall not exceed $\mu=0.2$, unless otherwise documented by the designer.

The sum of the friction force in the vertical supports which may be taken into account shall not exceed:

$$P_{friction} = P_w \times \mu$$

where,

$P_w$: static cargo and tank weight.

$\mu$: 0.2, lower bound for sliding friction coefficient

This friction force will reduce the active transverse force acting on the lower supports.

The long term distribution of stress is shown in Figure 8-6. “A” is drawn based on calculated stress, $\sigma_1$, from a transverse load where no friction force is included. The effect of the friction in the vertical supports is that the transverse supports will not be loaded for small values of the transverse acceleration. This calculated friction force, $P_{friction}$, will cause a stress which is equivalent to $\sigma_2$. Thus, transverse accelerations causing a stress less than $\sigma_2$ will not cause stress in the transverse supports and the area marked, B, shall be removed from the fatigue load diagram.

Thus, the effective fatigue load diagram to be used in the fatigue calculations is shown in Figure 8-7. The stress range shall be used in the fatigue analysis.

---

**Figure 8-6**
Long term distribution of stress

**Figure 8-7**
Effective fatigue load diagram
The details of the procedure used in the fatigue analysis are shown in Table 8-5.

### Table 8-5 Procedure used in the fatigue calculation

<table>
<thead>
<tr>
<th>Procedure</th>
</tr>
</thead>
<tbody>
<tr>
<td>Long term stress distribution</td>
</tr>
<tr>
<td>— Long term stress range distribution determined from wave load analyses and structural stress analysis with FE models using a Weibull slope parameter of $h = 1$ as per IGC requirement.</td>
</tr>
<tr>
<td>— Total number of cycles to be calculated considering the effect of friction in the vertical supports</td>
</tr>
<tr>
<td>— Stress range based on $10^{-4}$ probability level as illustrated in Figure 8-7.</td>
</tr>
<tr>
<td>— $\sigma_1$ are $\sigma_2$ are extracted directly from a fine mesh analysis and linear extrapolation is done to get hot spot stresses. See Classification Notes 30.7/5/ for extrapolation procedure</td>
</tr>
<tr>
<td>Mean stress effect</td>
</tr>
<tr>
<td>— See Classification Notes 30.7 Sec 2.3.4 for mean stress effect.</td>
</tr>
<tr>
<td>Fraction time</td>
</tr>
<tr>
<td>— Fraction factor of 0.45 is used for the life time operating under full load condition</td>
</tr>
<tr>
<td>— Stresses from ballast condition is assumed to be negligible</td>
</tr>
<tr>
<td>S-N curves</td>
</tr>
<tr>
<td>— S-N curves defined by parameters given in Classification Note 30.7 Table 2-1 or 2-2 can be used as appropriate.</td>
</tr>
</tbody>
</table>

8.9.4 Longitudinal supports
The longitudinal support is not included in the fatigue calculations as the longitudinal acceleration is relatively small in normal ship operation and the longitudinal load is absorbed by the friction in the vertical supports.

8.9.5 Anti-floatation supports
The anti-floatation supports are supports designed to prevent the tank in an accidental cargo hold flooding situation to float up and need therefore not be subject to fatigue calculations.

8.10 Fatigue Assessment of Hull Structures
8.10.1 Loading conditions
Loading conditions shall be selected from the loading manual that represents the trading pattern of the vessel over the design life time, see [8.7.1] Loading Conditions.

1) Full homogeneous departure.
2) Ballast departure.

Loading conditions suitable for the derivation of the component stress ranges for determining combined stress range and/or as basis for component stochastic analyses as described in Classification Notes 30.7/5/ are given in Table 8-6. Hull girder wave bending stress shall be added, as applicable.

### Table 8-6 Load cases for fatigue strength assessment

<table>
<thead>
<tr>
<th>LC</th>
<th>Loading Condition</th>
<th>Pressure</th>
<th>Ref.</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Fully loaded</td>
<td>Internal dynamic cargo pressure</td>
<td>Dynamic</td>
</tr>
<tr>
<td>2</td>
<td>Fully Loaded</td>
<td>External dynamic sea pressure</td>
<td>Dynamic</td>
</tr>
<tr>
<td>3</td>
<td>Ballast</td>
<td>External dynamic sea pressure</td>
<td>Dynamic</td>
</tr>
<tr>
<td>4</td>
<td>Ballast</td>
<td>Internal dynamic ballast load</td>
<td>Dynamic</td>
</tr>
<tr>
<td>5</td>
<td>Fully loaded</td>
<td>External static pressure plus internal cargo static pressure</td>
<td>Static</td>
</tr>
<tr>
<td>6</td>
<td>Ballast</td>
<td>External static pressure plus internal ballast tank pressure</td>
<td>Static</td>
</tr>
</tbody>
</table>

Hull girder loads and loads calculated from LC1 to LC4 above shall be used for fatigue strength evaluation. LC5 and LC6 may be used for determination of mean stress effects.

8.10.2 Hopper knuckle connections
The angle of hopper knuckle is one of primary design issues. A steep hopper knuckle angle is usually preferred to minimize ballast volume and maximize the cargo volume. On the other hand, a less steep angle is preferred in view of structure design. Thus, it is usually found that the hopper knuckle angle is between 40 and 55 degrees. 8.8 shows an example of a fine element model with thickness size meshes.
Figure 8-8
A finite element model showing meshes

When a hopper knuckle angle is quite steeper and geometry of the hopper knuckle connection outside the midship is different from those amidships, an additional finite element model may be made. An example of this case is shown in Figure 8-9.

If the required fatigue life is not satisfied, weld shape improvement is commonly used. Figure 9-2 shows details of weld profiling. The weld bead should be ground and undercut at the weld toe removed. It should be noted that the final grinding direction should be transverse direction in order to avoid additional notches due to the grinding.

Figure 8-9
A design example of weld profiling

For weld toe grinding, it is required that a toe grinding depth of maximum 5% of thickness and the minimum 0.5 mm is applied according to Classification Notes 30.7 /5/. For thicker plates, the maximum grinding depth should not exceed 2.0 mm. It should be noted that the final grinding direction should be transverse direction in order to avoid additional notches due to the grinding. A burr grinder is normally used for toe grinding. Before performing the burr grinding, the weld should be de-slagged and cleaned by a wire brush.

8.10.3 Acceptance Criteria
The ship hull is to be designed for a minimum fatigue life of 20 years of operation in world-wide conditions according to the procedure in Classification Notes 30.7 /5/. Rule loads are to be applied unless otherwise specified; by owners requirement and/or by voluntary class notation CSA. Accumulated fatigue damage ratio $C_w$ to be less or equal to 1.0 (Table 8-3).
9 Crack Propagation Analysis

Fracture mechanics analyses shall be carried out in accordance with DNV Rules Pt.5 Ch.5 Sec.5 A1300 for dynamically loaded weld connections in the tank shell structure and internal girder/frame structure as indicated in [1.2]. For details located in the tank shell structure including shell stiffeners and parts of webs/girders adjacent to the shell the analyses shall be used to determine if

a) a crack penetrates through the primary barrier (plate) thickness and remains stable for at least 15 days from time of detection in the worst storm conditions, or

b) the crack does not go through the thickness, but grows in length due to dominating bending stress over the plate thickness.

Typical weld connections to be considered are:

a) Plate connections to stiffeners, frames and girders in outer tank shell

b) High stress areas at stiffener transitions through web frames and girders and stiffeners subjected to large relative deformations, where failure developments has potential to propagate into the outer shell before being detected

c) Tank structure in way of supports and tower foundation.

In case of a), the leakage rates shall be determined according to [9.8], and the small leak protection system shall be dimensioned accordingly. This demonstrates compliance with the Leak-Before-Failure (LBF) requirements as outlined in [1.2], and can be considered detectable by leakage detection. On the other hand, if a crack cannot be shown to penetrate the shell thickness before reaching a critical state (condition “b” above) and LBF does not apply, the stricter requirements in Table 8-3 have to be applied.

For details of the internal girder/frame structure away from the shell, and where failure detection by leakage is not an option, the analyses shall be used to determine crack propagation paths and the time until the failure development reaches a critical state where it can compromise the integrity of the tank. Requirements are given in Table 8-3, and shall be selected considering whether or not detection by inspection is applicable.

Dynamic stresses are driving fatigue crack growth, whereas the rupture of a fatigue crack of a given size is governed by a maximum ULS load situation. The primary load effect governing final rupture of a fatigue crack of a certain size is the most probable largest one time stress amplitude (static plus dynamic amplitude) during $10^8$ cycles in the North Atlantic.

Hence, the most convenient approach is to establish the total fatigue load stress range spectrum for fracture mechanics (crack propagation) analyses from the most probable largest load spectrum the ship will experience during $10^8$ wave encounters in the North Atlantic applying a Weibull slope parameter of $h=1$ as per IGC.

9.1 Fracture Mechanics Analysis

An unstable crack means either spontaneous crack growth with no additional input of driving strain energy (brittle fracture) or as plastic tearing needing only marginal input of strain energy for the crack to propagate. If leak-before-failure cannot be proven, enhanced fatigue and fracture mechanics requirements apply depending on possibilities to inspect (access) and the inspection period, Table 8-3.

The start of the analyses is from an initial semi-elliptical crack of length “2c” and depth “a” that will at least be of the maximum defect size that will not be discovered with NDT inspections. The following analyses and initial cracks sizes would normally be anticipated:

a) The estimated number of cycles/years before a leakage, i.e. the number of cycles to propagate through the thickness enabling gas detection of the leakage, or to reach a critical size shall be calculated. The stress spectrum to be used for this case is illustrated in a normalized form in Figure 9-1. Here, 30 stress blocks are used in the spectrum.

b) Further propagation of the penetrated crack shall then be calculated as a “through-thickness” crack for the worst 15 day North Atlantic storm, Figure 9-2.

c) If the crack is stable up to 15 days, and beyond, leakage calculations to be carried out to estimate liquid leakage rates, [9.8] below.

d) If failure monitoring based on gas detection cannot be safely applied the predicted failure development time shall, if reliable in-service inspection is possible, be 3 times the inspection interval, otherwise 3 times the design lifetime of the tank, Table 8-3.

Both the crack propagation and the critical crack size calculations can be performed using the software program CrackWise, ref. /12/. The critical flaw calculations can be carried out based on the level 2B calculation defined in BS 7910:2005, ref. /13/.

---

DET NORSKE VERITAS AS
The crack growth can be calculated by stepwise integration of the Paris’ equation:

\[ \frac{da}{dN} = C(\Delta K)^n \cdot \Delta K = Y \cdot \Delta \sigma \sqrt{a} \]

where

- \( \frac{da}{dN} \) = the crack growth per load cycle
- \( m \) and \( C \) = the crack growth constants, determined from experiment
- \( \Delta K \) = the stress intensity factor range
- \( \Delta \sigma \) = the stress range
- \( Y \) = is a correction factor depending on geometry
- \( a \) = the crack depth

The additional bending stress from eccentricities can be calculated directly from the FE- analyses or from suitable stress concentration formulas as given in Classification Notes 30.7/5/. If the welds are ground the local stress concentration factor at e.g. weld toes, Mk, can be set equal to 1.0, otherwise the Mk definition in the CrackWise program can be applied if relevant for the actual geometry.

9.2 Initial defects to be used

The size of initial defects to be used in the analysis shall be decided considering the production quality of the builder. For failures originating in the primary barrier plate the following initial crack sizes in way of Heat Affected Zone (HAZ) through thickness may be used for builders with a high production quality standard.

- Butt welds: 1.0 mm depth and 5 mm in length
- Fillets: 0.5 mm depth and 5 mm in length

For failures originating elsewhere, e.g. from end connections of primary barrier stiffeners, the initial crack size in the shell plating shall be determined considering the development of the crack through the stiffener.

9.3 Crack propagation data for fracture mechanics analysis

The design crack propagation data are to be based on the mean-plus-two-standard-deviation of the test data. For the cargo tank, crack propagation data (C and m in Paris’ equation) and fracture toughness data, J-values or CTOD values, need to be determined for the tank material and its welded joins for relevant service conditions. Documented test data for both room temperature and cryogenic temperature should be available.

9.4 Welding residual stresses

In the critical flaw size calculations, the residual stress from welding is assumed equal to the yield strength.

9.5 Stress range spectra

9.5.1 Long term wave-induced stress range spectrum

For design against leakage the load spectrum is taken as the most probable largest load spectrum the ship will experience during 10^8 wave encounters in the North Atlantic. The normalized stress range on probability level 10^-8 can set equal to 1 to facilitate easy scaling to other stress levels. The long term wave-induced stress range spectrum (10^8 cycles) is given in Figure 9-1 using 30 stress blocks and assuming a Weibull shape parameter of h=1.0. If no better estimate for “h” is available this is the normal assumption according to the International Gas Code (IGC) /3/.
However, in the direct wave load analysis procedure which is required for independent tanks of type B the Weibull shape parameter can be determined. Hence, the shape of the stress distributions in Figure 9-1 and Figure 9-2 can be modified as follows:

$$\frac{\Delta \sigma}{\Delta \sigma_0} = \left( \frac{1 - \log n}{\log n_0} \right)^{1/h}$$

where

- $\Delta \sigma_0$ = reference stress value at the local detail exceeded once in $n_0$ cycles (shown as normalized to 1.0)
- $n_0$ = total number of cycles associated with the stress range level $\Delta \sigma_0$ (here $10^8$)
- $h$ = the Weibull slope parameter

### 9.5.2 Stress range spectrum for 15 days of storm

For analysing crack propagation of a through-thickness crack the stress range spectrum representing 15 days in the worst $10^8$ storm condition in the North Atlantic is used.

The stress range spectrum at 15 days ($2 \cdot 10^5$ cycles) of storm is shown in Figure 9-2 using 30 stress blocks.

### 9.5.3 Failure criterion for ductile material

The dynamic stress range is the driving force for crack propagation. For through thickness analysis the 20 year stress spectrum is to be used, Figure 9-1. When a through thickness crack has developed, the 15 day spectrum in the most severe storm (same extreme stress value as for the 20 year spectrum) shall be used and the crack length at day 15 determined. The materials normally used for tank type B construction, aluminium 5083-0 and 9% Ni-steel, has been shown to be very ductile and no critical crack length can be precisely defined, ref/14/.

The value of the CTOD parameter can therefore during the through thickness crack propagation phase be set to a fairly large value (e.g. =1000). The analysis should then be carried out well beyond the 15 day duration (up to about 3x15 days) in order to
trace the through thickness crack growth curve (crack length vs. cycles). If the curve shows an accelerated crack growth this indicates an imminent plate failure (rupture).

The driving stress range can based on the results of the investigations reported in ref. /14/ be set to

$$\Delta \sigma = \Delta \sigma_{dm} + \Delta \sigma_{db}/m$$

where

- $\Delta \sigma_{dm}$ = dynamic membrane stress range
- $\Delta \sigma_{db}$ = dynamic bending stress range
- $m$ = dynamic bending reduction factor

The value of the dynamic bending factor “m” can be taken as 3. The strain energy associated with the compressive side of the bending stress distribution will not contribute to open the crack, but the tension side might. Hence, as a conservative measure 1/3 of the surface bending stress range can be included. This corresponds to the centre of gravity of the triangular tension part of the bending stress distribution.

9.6 Acceptance Criteria

The acceptance criteria described in Table 8-3 shall be satisfied.

Leak rates through cracks in the outer shell plates (the primary barrier) shall be determined for the actual size of cargo tank. This is an important input for the evaluation of the capability of the drip tray and gas venting arrangement to dispose of leakages in a safe manner, and hence if the conditions for Leak-Before-Failure is satisfied or not.

9.7 Stress combination for fracture mechanics analysis

In order to evaluate residual fracture of fatigue cracks over the lifetime of the vessel fracture mechanics analysis has most conveniently to be referred to the ULS stress range to be compatible with the total ULS stress amplitude that governs potential fatigue crack rupture. The fatigue stress range can preferably be determined at a $Q=10^{-4}$ probability level as for the S-N fatigue approach and extrapolated to the ULS stress range level using the long term Weibull stress distribution. As the IGC specifies the Weibull shape parameter $h = 1$ this means multiplying the $10^{-4}$ stress range with 2 to arrive at the $10^{-8}$ stress range.
For use in the fracture mechanics analysis the principal stresses determined for S-N curve fatigue analysis shall be further processed as given below:

a) In order to correctly evaluate crack propagation, the static value plus the dynamic design life ULS amplitude of the principal surface stresses shall be calculated in addition to the dynamic stress ranges.

b) Based on the inside and outside values of the principal surface stresses, the stresses are to be split into membrane and bending parts separately for dynamic stress ranges and for static plus ULS amplitude values. This is essential for the fracture mechanics analyses but is not necessary for the S-N fatigue analyses.

c) Select the largest membrane stress for the analysis. This will give the fastest crack growth through the thickness and probably the shortest fatigue life. However, in some cases it might be necessary also to check the maximum bending combination in which the crack will grow faster in length than in depth.

9.8 Leakage calculation

9.8.1 Purpose of calculation

It is important to estimate the leakage rate through a crack of given size in order to establish whether the concentration of leaked gas is sufficient to be detected before the crack becomes of unstable, e.g. by experiencing an accelerating crack propagation rate.

In practice it must be established that the crack between the time of detection and the end of the voyage does not reach the critical crack length. For the possibility of detecting a leak the worst case would be one in which the leakage is detected soon after the start of the voyage.

In this context a conservative leakage estimate is one that underestimates the leakage for a given crack size. A calculation that overestimates the leakage will indicate that the leakage will be detected earlier than in reality.

On the other hand, a larger LNG liquid leakage will be conservative for the dimensioning of the capacity of the small leak protection system (the reduced secondary barrier drip tray).

9.8.2 Cases to consider

The leakage may be of vapour from the region above the liquid surface or of liquid from the lower region of the tank. In practice it is considered sufficient to consider two cases:

1) Through-thickness cracks in the outer primary shell plate in the lower parts of the tank, e.g. in way of supports and tower foundation. The largest liquid leak rate will be when the tank is full, i.e. has the largest liquid pressure head at the position of the crack.

2) A through-thickness crack in the primary shell plate in the upper part of the tank, which may be above or below the liquid surface.

9.8.3 Form and dimensions of crack

The development of a fatigue crack in the tank shell may be considered to consist of three stages as shown in Figure 9-3 and described below:

i) The crack starts to grow from a defect at one surface (the initiation side). It grows in both the in-plane direction and the thickness direction until it reaches the opposite face of the shell (the penetration side).

ii) The length of the crack at the initiation side is given by the fracture mechanics analysis when the crack has propagated through the thickness. The crack shape will be semi-elliptic with axes equal to the crack length at the initiation side and the smaller half axis equal to the plate thickness.

iii) The crack will grow as a trough thickness crack. The length of the crack at the penetration side can be calculated assuming the same elliptic shape (the same ratio between the half axes) as in Stage II throughout the through thickness crack growth. This approach gives similar crack shapes as those found during the fracture and leakage testing reported in refs. /15/ and /16/.

![Initiation Side](image1)

![Penetration Side](image2)

**Figure 9-3**

Three stages of crack growth
The crack length at the penetration side can be derived as follows:

1) Calculate the crack propagation through the plate thickness starting from the defects given in [9.2] above by fracture mechanics analysis (Crackwise). The crack length at the initiation side is then obtained from the analysis as the length when the crack penetrates the thickness.

2) Starting from the crack length on the initiation side carry out a crack growth calculation of the through-thickness crack and determine the crack length on the initiation side after 15 days in the most severe storm using a load spectrum as shown in Figure 9-2.

3) The corresponding crack length at the penetration side can then be estimated as:

\[ a_p^2 = a_i^2 - \left( \frac{a_{i0}}{t_0} \right)^2 t \]

where

- \( a_p \) = half crack length at penetrations side after 15 days storm
- \( a_i \) = half crack length at initiation side after 15 days storm
- \( a_{i0} \) = half crack length at initiation side at penetration of the shell plate thickness
- \( t \) = shell plate thickness
- \( t_0 \) = depth of crack corresponding to \( a_{i0} \) as returned by Crackwise

Due to the numerical solution procedure used in Crackwise \( t_0 \) will normally be slightly less than \( t \). It will, however, be conservative to put \( t_0 = t \).

Effective Crack Opening Stress:

Compression will close the crack and no leakage will occur. The integrated mean tension stress over a dynamic cycle will be the opening crack stress in terms of leakage. The effective crack opening stress can be taken as

\[ \sigma_{eqt} = \sigma_{sm} + \sigma_{db}/m \quad : \text{for } \sigma_{sm} > \sigma_{dm} \quad (a) \]
\[ \sigma_{eqt} = \sigma_{sm} + (\sigma_{dm} - \sigma_{sm})/2 + \sigma_{db}/m \quad : \text{for } \sigma_{sm} \leq \sigma_{dm} \quad (b) \]

where

- \( \sigma_{sm} \) = static membrane stress
- \( \sigma_{dm} \) = dynamic membrane stress amplitude
- \( \sigma_{db} \) = dynamic bending stress
- \( m \) = dynamic bending reduction factor

For the sake of conservatism a \((1/m)\) fraction of the dynamic bending stress has been included with a value of “\(m\)” equal to 3, [9.5.3].

For the purpose of average crack opening for leakage calculations the average effective tension stress will contribute to open the crack.

- a) If the membrane tension stress is larger than the dynamic membrane stress amplitude the net effect of the dynamic stress will be zero, eq. (a).
- b) On the other hand, if the membrane tension stress is less than the dynamic membrane stress amplitude the average effect of the difference should be included, eq. (b).

Effective Crack Opening Area:

For a case of a pure tension stress \( \sigma \) it may be assumed that the shape of the opening in stage III is an ellipse whose major axis length is the crack length \( 2a_p \) and whose minor axis length is equal to the maximum crack opening displacement \( 2\delta \). The area of the opening is then given by

\[ A = \pi a_p \delta \]

where

\[ \delta = \frac{2\sigma_{eqt} a_p}{E} \]

Thus

\[ A = \frac{2\pi a_p^3 \sigma_{eqt}}{E} \]

For cases with combined bending and tension, the crack opening and also the crack length will vary through the thickness.
9.8.4 Leakage rate

The flow of LNG (or LPG) through a fatigue crack is assumed to be orifice flow of the liquid. The volume of liquid of leaked LNG can be estimated by the following equation:

\[ Q = C_{orifice} \cdot A \cdot \sqrt{2g\left(h + \frac{p_1 - p_2}{\gamma}\right)} \]

where

- \( C_{orifice} \) = an orifice coefficient (= 0.1),
- \( h \) = the head of liquid inside the tank at the crack location,
- \( \gamma \) = the specific weight of the leaking fluid and
- \( p_1, p_2 \) = the pressures inside and outside the tank, respectively.

Here \( A \) is the crack opening area on the penetration side as outlined above. The orifice coefficient has been derived from the tests reported in /15/ and /16/. A value of 0.1 has been found to give a reasonable comparison with the test data.

9.8.5 The size of the Secondary Drip Tray

The inner bottom in way of cargo tanks shall be protected against liquid cargo. Away from (clear of) the partial secondary barrier, provisions need to be made to deflect any liquid cargo down into the space between the primary and secondary barriers and to keep the temperature of the hull structure at a safe level (spray-shield).

The secondary drip tray has to be dimensioned with a reasonable margin to be able to contain the leakage at day 15 in the worst expected storm. The total leakage over a given period of time is obtained by integrating the leakage rate with respect to time, taking account of the crack growth.

For determining the necessary size of the partial secondary barrier (drip tray) due account has to be taken of liquid evaporation, rate of leakage, pumping capacity and other relevant factors.
10 Vibration Analysis

According to Pt.5 Ch.5 Sec.5 A 1000 Design of hull and cargo tanks, choice of machinery and propellers shall be aimed at keeping vibration exciting forces and vibratory stresses low. Calculations or other appropriate information pertaining to the excitation forces from machinery and propellers, may be required for membrane tanks, semi-membrane tanks and independent tanks type B, and in special cases, for independent tanks type A and C. Full-scale measurements of vibratory stresses and or frequencies may be required.

Vibration levels on LNG ships will depend on the design of the vessel with regard to structure and excitation sources. Thus it is difficult to establish general guidelines for this type of ship. The risk of unwanted vibration levels in the cargo containment system has therefore to be investigated case by case either by experience from similar vessels and/or by vibration studies. Normally unwanted vibration levels of structure at some distance from the excitation sources are associated with resonance of the structure. The risk for resonances is larger for a full or partially filled tank than for an empty tank due to the lower natural frequencies caused by added mass of liquid.

The risk of unwanted vibration levels will also depend on the type of excitation sources on the vessel. Turbine driven ships and ships with diesel-electrical propulsion system comprising medium speed resiliently mounted diesel engines will have smaller risk of unwanted vibrations than ships propelled by large bore slow speed engine(s). However, similar ships with different excitation frequencies (number of propeller blades/number of cylinders) may perform differently.

The main aim should be to avoid that natural frequencies of the tank system to coincide with the excitation frequencies from relevant sources. The risk of unwanted vibration is in that case low. However, even at resonance the vibration levels may be acceptable. A forced vibration analysis may prove whether acceptable vibration levels are expected.

10.1 Excitation sources and frequencies

An LNG vessel may be equipped with different types of propulsion units giving rise to forces which may excite vibration in the tank system. The following excitation sources shall be considered for different types of propulsion units:

- **Turbine and reduction gear:**
  - Blade passing frequency from main propeller(s).

- **Diesel-electric propulsion with resiliently mounted diesel engines (speed above 500 RPM):**
  - Blade passing frequency from main propeller(s).

- **Geared propulsion units with resiliently mounted diesel engines (speed above 500 RPM):**
  - Blade passing frequency from main propeller(s).

- **Slow speed engines (speed below some 200 RPM):**
  - Blade passing frequency from main propeller(s).
  - Main excitation frequencies from the diesel engine(s) (free moments/guide force moments).

10.2 Natural frequency of Tank System

Global natural frequencies of the tank system can be determined by a FE analysis using the global stress model of the tank. Lateral vibration of panels can be determined separately paying due consideration to the effect of added mass due to liquid filling.

10.3 Forced response analyses

The safest way to achieve acceptable vibration levels of the tank system is to keep the natural frequencies away from the relevant excitation frequencies. However, the actual vibration levels of a structure will depend on the transfer function from the sources to the structure. Thus it may be acceptable to allow natural frequencies in actual speed ranges of the vessel if it can be demonstrated that acceptable vibration levels are achieved. This may be shown by means of a forced vibration analysis of the vessel. The vibration analysis should be performed according to following guidelines:

1) The finite element model shall comprise a full 3-dimensional model of the whole ship.
2) Full load, ballast and possible part load condition should be analysed.
3) The supporting structure of the tank shall be modelled in such a detail that the dynamic behaviour is represented.
4) The tank and tower should be included.
5) The excitation forces from the propeller(s) shall be applied as:
- Pressure forces acting on the hull
- Shaft forces acting in relevant bearings

6) The excitation forces from slow speed main engine(s) shall be applied to a rough model of the main engine including its top staying.

7) The excitation forces shall correspond to 100% Maximum Continuous Rating (MCR).

8) The frequency range to be applied for the relevant excitation forces shall correspond to the excitation frequency at 100% MCR +20% /- 30%.

9) The excitation forces are to be kept constant corresponding to 100% MCR for the above frequency range.

10) For twin screw vessels the forced response shall be calculated for two excitation modes:
- Propellers/main engines acting in phase
- Propellers/ main engines acting in opposite phase.

11) The highest response of the two excitation modes shall be considered.

12) The damping applied may be proportional to frequency, but not exceeding 2% of critical damping.

13) The maximum calculated vibration level in a frequency range corresponding to 100% MCR + 15% and 90% MCR – 15% shall be considered to be excited at full speed.

14) The calculated vibration levels for each of the applied excitation sources may be evaluated separately.

Based on experience from full scale measurements a vibration level above some 20 mm/s may result in risk of fatigue cracks in aluminium structures. In order to account for uncertainties in the analyses and a safety margin, it is considered that calculated vibrations below some 10 mm/s are acceptable for the tower/tank structure.
11 References

/1/ DNV: Rules for Classification of Ships, Pt. 3 Ch.1 *Hull Structural Design, Ships with Length 100 metres and above*

/2/ DNV: Rules for Classification of Ships Pt.5 Ch.5, *Liquefied Gas Carriers*


/5/ DNV: Classification Note 30.7, *Fatigue Assessment of Ship Structures*


/7/ DNV: Recommended Practice DNV-RP-C205, *Environmental Conditions and Environmental Loads*

/8/ A note on *Leakage Calculation*, DNV 2010-03-24/Vals (internal)

/9/ DNV: Classification Note 34.1, *CSA - Direct Analysis of Ship Structures*

/10/ IACS: Common Structural Rule for Oil Tankers and Bulk Carriers, External release, April 2013


/12/ *Crackwise 4*, The Welding Institute, Cambridge, UK 2005

/13/ Guide on the Methods for Assessing the Acceptability of Flaws in Metallic Structures, BS 7910:2005

/14/ Tenge, P. and Sollie, O *Fracture mechanics in the design of large spherical tanks for ship transport of LNG, Norwegian Maritime Research, Vol.1, no. 2, pp.1-18, (1973)*


/16/ Sollie, O. and Tenge, P.: *9 per cent Ni-Steel for the use in the 87 600 m³ LNG Carrier of Moss-Rosenberg Design. Geometry and propagation rate of fatigue cracks initiated from surface defects, fracture toughness, and gas leak rates through penetrating cracks*, Det Norske Veritas, Report no. 84 5015/4, May 9, 1973

/17/ DNV: Recommended Practice, DNV-RP-C201, Part 2, *Buckling Strength of Plated Structures*

/18/ DNV: Classification Notes 30.9 *Sloshing Analysis of LNG Membrane Tanks*

/19/ DNV: Recommended Practice C203 *Fatigue Design of Offshore Steel Structures*