DNV·GL

CLASS GUIDELINE

DNVGL-CG-0130

Edition January 2018

Wave loads

DNV GL AS

FOREWORD

DNV GL class guidelines contain methods, technical requirements, principles and acceptance criteria related to classed objects as referred to from the rules.

Any comments may be sent by e-mail to *rules@dnvgl.com*

CHANGES - CURRENT

This is a new document.

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SECTION 1 GENERAL

1 Introduction

1.1 Objective

This class guideline gives a transparent and detailed description of the hydrodynamic analysis methods and procedures to support and satisfy requirements given in DNVGL-RU-SHIP Pt.3, DNVGL-RU-SHIP Pt.5 and DNVGL-RU-SHIP Pt.6.

1.2 Scope

This class guidelines covers the procedures for calculation of wave induced loads, ship motions and accelerations of monohull displacement ships. Sec.7 to Sec.11 describe the approaches to calculate loads according to the rules for classification. Sec.1 to Sec.6 contain more detailed descriptions of wave conditions, statistics, concepts and approaches, which may be referred to in Sec.7 to Sec.11. Sec. 12 covers aspects related to modelling and quality assurance. Hydroelastic responses like whipping and springing are covered by DNVGL-CG-0153.

2 Symbols and abbreviations

2.1 Symbols

The symbols in Table 1 are used in this class guideline. For symbols not defined in this class guideline, reference is made to DNVGL-RU-SHIP Pt.3 Ch.1 Sec.4.

Table 1 Symbol list

Symbol	*Meaning*	*Unit*
β	heading angle	deg
β_p	principal heading angle	deg
θ	roll angle	deg
θ_i	phase angle	rad
φ	pitch angle	deg
λ	wave length	m
ρ	density; salt water = 1.025; fresh water = 1.0	t/m^3
ν	zero up-crossing frequency	1/s
η	surface elevation	m
ω	wave frequency	rad/s
ω_p	peak wave frequency	rad/s
ω_e	encounter frequency	rad/s
ω_n	natural frequency	rad/s

Symbol	*Meaning*	*Unit*
Δ	moulded displacement at draught T_{SC}	t
Γ	gamma function	-
a	acceleration	m/s^2
$\boldsymbol{a}$	nodal acceleration vector (FE model)	m/s^2
A	wave amplitude	m
A_C	crest height	m
A_T	trough height	m
$\boldsymbol{A}$	added mass matrix	t
B	moulded breadth of ship	m
$\boldsymbol{B}$	damping matrix	Ns/m
C_B	block coefficient at draught T_{SC}	-
c_g	group velocity	m/s
c	phase velocity	m/s
$\boldsymbol{C}$	stiffness matrix	N/m
D	moulded depth of ship	m
$D(\beta)$	wave energy spreading	-
d	water depth	m
f	wave frequency	Hz = 1/s
$\boldsymbol{f}$	nodal force vector (FE model)	kN
f_e	environmental factor	-
f_p	down scaling factor	-
f_R	operational factor	-
$\boldsymbol{f'}$	force vector (hydrodynamic model) at panel centres	kN
$\boldsymbol{F}$	generalised force for modal DOF (FE model) based on nodal forces	kN
$\boldsymbol{F'}$	generalised force for modal DOF (hydrodynamic model) based on forces at panel centres	kN
F	force	kN
F_r	froude number; ratio of gravity and inertia loads $F_r = U/(g \cdot L)^{1/2}$	-
g	acceleration of gravity = 9.81	m/s^2
h	height; distance from fluid surface to load point	m
H	wave height	m
H_b	maximum wave height	m
H_s	significant wave height	m

Symbol	Meaning	Unit
I_{44}	mass moment of inertia in roll	t·m^2
I_{55}	mass moment of inertia in pitch	t·m^2
I_{66}	mass moment of inertia in yaw	t·m^2
k	wave number	1/m
k_r	radius of gyration in roll	m
L	rule length	m
L_{PP}	length between perpendiculars	m
LCG	longitudinal centre of gravity from aft perpendicular	m
m	point mass	t
m_λ	mass per meter along the hull	t/m
m_T	total mass of ship	t
M	moment	kNm
$\boldsymbol{M}$	mass matrix (FE model)	t
$\boldsymbol{M'}$	modal mass matrix (hydrodynamic model)	t
M_{wh}	horizontal wave bending moment	kNm
M_{wt}	torsional wave moment	kNm
$M_{wv\text{-}j}$	vertical wave bending moment, j=h, s (hog, sag)	kNm
$M_{sw\text{-}j}$	vertical still water bending moment, j=h, s (hog, sag)	kNm
n	number of cycles over a treshold	-
n_0	number of total cycles during the life time	-
$\boldsymbol{q}$	modal displacements (hydrodynamic model)	m
$\boldsymbol{Q}$	probability of exceedance	-
Q_{sw}	vertical still water shear force	kN
Q_{wv}	vertical wave shear force	kN
P_W	hydrodynamic pressure	kN/m^2
$P_{WL\text{-}i}$	hydrodynamic pressure at water line at probability level 10^{-i} where i is 2 or 8	kN/m^2
$\boldsymbol{r}$	nodal displacement vector (FE model)	m
$\boldsymbol{r'}$	displacement vector (hydrodynamic model) for panel centres	m
r_{55}	radius of gyration in pitch	m
r_{66}	radius of gyration in yaw	m
S	wave steepness of wave event	-
S_{max}	maximum wave steepness of wave event	-

Symbol	Meaning	Unit
S_S	wave steep of sea state based on *Tz*	-
S_P	wave steepness of sea state based on *Tp*	-
$S_i(\omega)$	wave spectrum, i = PM for Pierson-Moskowitz and i = J for Jonswap	m^2/s
$\boldsymbol{T}$	conversion matrix from modal displacements in hydrodynamic model to nodal displacements in FE model	-
$\boldsymbol{T'}$	conversion matrix from modal displacements in hydrodynamic model to displacements for hydrodynamic panel centres	-
T	moulded draught	m
T	wave period	s
T_n	natural period	s
T_1	expected (mean) wave period	s
T_{act}	actual draught at position considered	m
T_{full}	full load design draught	m
T_{BAL}	ballast draught (minimum midship)	m
T_{SC}	scantling draught	m
T_{Design}	design draught	m
T_d	time duration of a sea state	s
T_{DF}	design life	years
T_p	peak period	s
T_z	zero up-crossing period	s
TCG	transverse centre of gravity from centre line	m
U	ship forward speed	m/s
V	maximum service speed	knot
VCG	vertical centre of gravity above base line	m
x	horizontal distance from origin of the coordinate system	m
y	horizontal distance from origin of the coordinate system	m
z	vertical distance from origin of the coordinate system	m
z_s	vertical distance from water line to water surface	m
z_{WL-i}	splash zone extent at probability level 10^{-i}, where i is 2 or 8.	m

2.2 Abbreviations

The abbreviations in Table 2 is used in this class guideline.

Table 2 Abbreviation list

Abbreviation	*Meaning*
AF	axial force
BEM	boundary element method
CFD	computational fluid dynamics
CL	centre line
COG	centre of gravity
CSR	common structural rules
DOF	degree of freedom
EDW	equivalent design wave
FE	finite element
FEA	finite element analysis
FEM	finite element method
HBM	horizontal bending moment
HF	high frequency (vibration response)
HSF	horizontal shear force
LCF	load combination factor
LF	low frequency (slow drift motion)
LTR	long term response
NA	north atlantic
pdf	probability density function
RAO	response amplitude operator
REC. No.	IACS recommendations number
RP	recommended practice
STF	short term response
TRF	transfer function
TM	torsional moment
UR S11	IACS unified requirement - longitudinal strength standard
VBM	vertical bending moment
VSF	vertical shear force
WF	wave frequency (wave response)
WW	world wide

2.3 Coordinate system and sign conventions

The origin of the "right hand" Cartesian coordinate system has co-ordinates at LCG, COG and CL, see Figure 1. *x, y* and *z* are longitudinal, transverse and vertical distance from the origin to the motion point considered. This corresponds to the coordinate system given in DNVGL-RU-SHIP Pt.3 Ch.4 to define positive motions, relevant to define inertia loads or internal pressure loads. Positive hull girder loads are also defined in DNVGL-RU-SHIP Pt.3 Ch.4 and refer to the origin located at the neutral axes. Positive hull girder loads are illustrated in Figure 2.

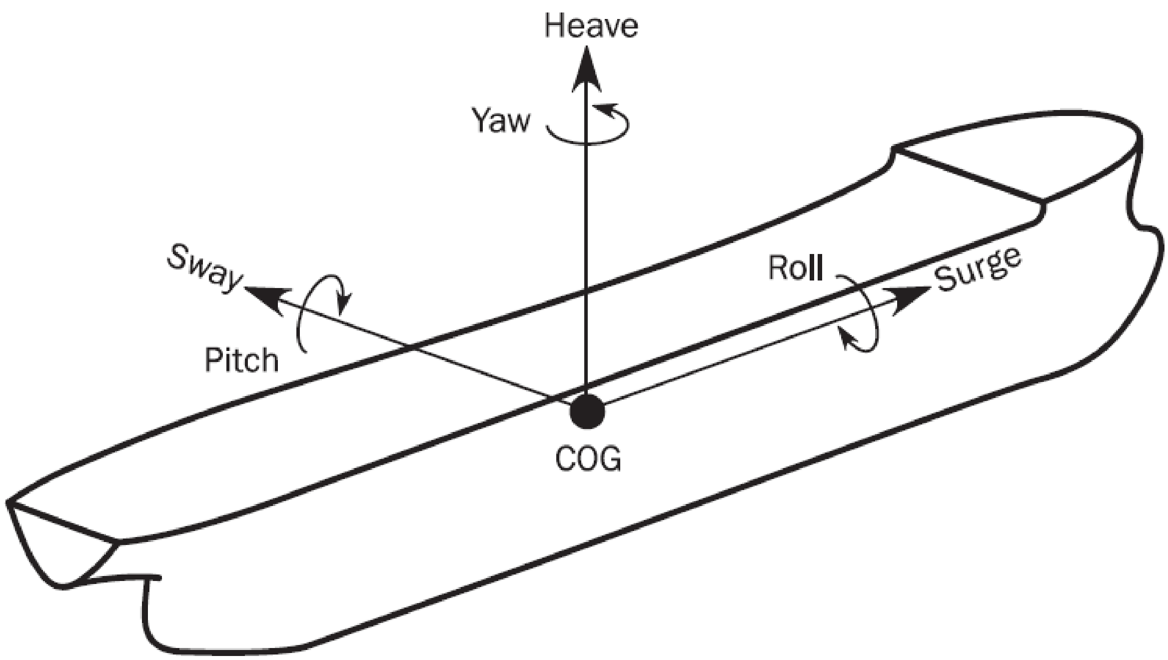

Figure 1 Coordinate system for positive motions

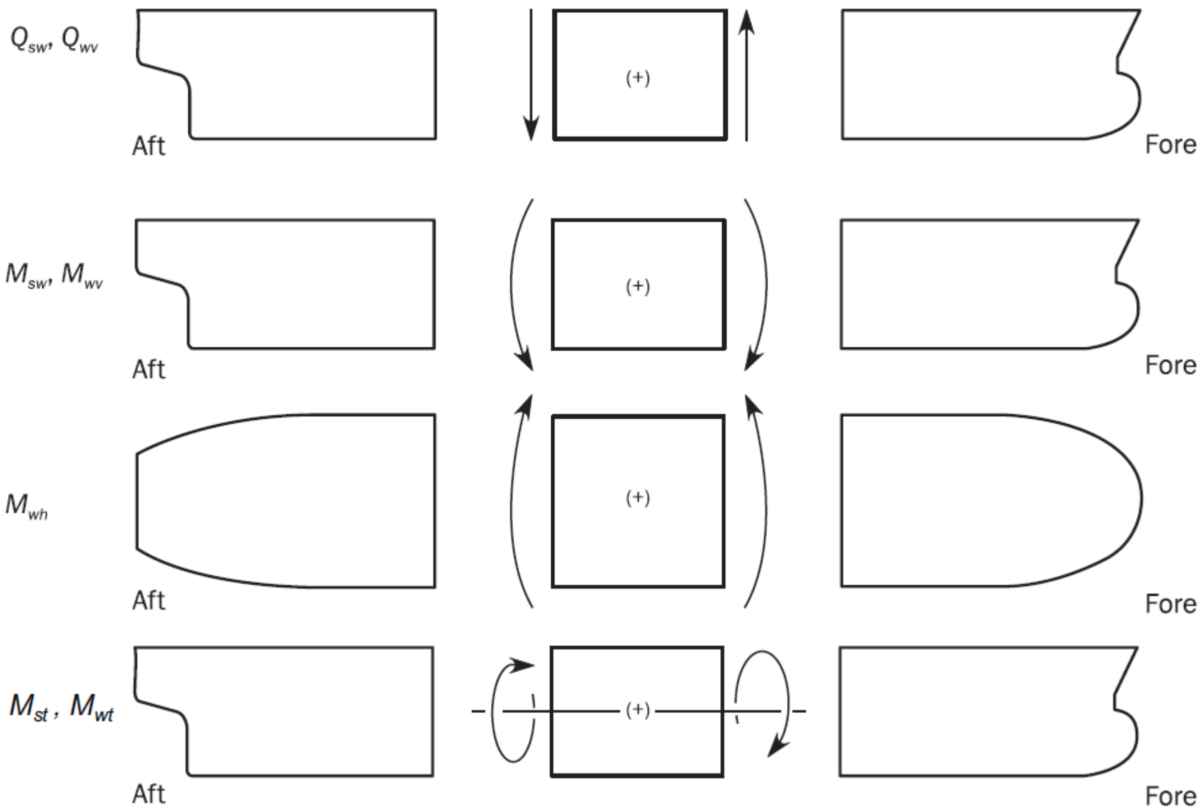

Figure 2 Sign conventions for shear forces Q_{SW}, Q_{WV} and bending moments M_{SW}, M_{WV}, M_{WH}, M_{ST} and M_{WT}

3 Definitions

A list of definitions is given in Table 3.

Table 3 Definitions

Term	*Explanation*
Dispersion relation	The relationship between the wave period T, in s, and the wave length λ, in m. This relation depends on the water depth d, in m.
Expected period	The wave or response period, T_1, in s, in a short term sea state associated with 50% of being exceeded in the probability density function.
Group velocity	c_g is the speed of wave energy transfer, i.e. the speed of the wave front in a wave train, in m/s, i.e. the speed of the moving wave train, being less than the speed of the phase velocity.
Non-linear regular waves	Asymmetric waves, $A_C > A_T$, where the phase velocity depends on wave height, i.e. the dispersion relation is a functional relationship between T, λ and H.

Term	Explanation
Peak period	T_p; the wave or the period of another response like vertical bending moment, in s, with most energy in the wave or response spectrum, i.e. the most probable maximum wave or response in a short term sea state.
Phase velocity	c; the propagation velocity of the wave form, e.g. wave crest, is called phase velocity, wave speed or wave celerity and is denoted by $c = \lambda/T$, in m/s.
Probability density function	pdf; a function pdf(x) showing the probability of occurrence at different levels of the parameter x, which could be any parameter. Integrated it will give the probability within ranges of x.
Significant wave height	H_s; the average of the third highest wave heights in a sea state with a duration of 3 hours.
Short term sea state	A sea state characterised by a wave spectrum, H_s and T_z (or T_p) and with a duration in order of 3 hours.
Surface elevation	The surface elevation $z_s = \eta(x,y,t)$, in m, is the distance between the still water level and the instant wave surface elevation at location (x,y) at time t.
Wave period	The wave period T, in s, is the time interval between successive crests passing a particular point.
Wave amplitude	$A$is the wave amplitude below or above the still water surface, in m.
Wave angular frequency	$\omega = 2\pi/T$, in rad/s.
Wave crest height	A_C is the distance from the still water level to the wave crest, in m.
Wave trough depth	A_T is the distance from the still water level to the trough, in m.
Wave frequency	Wave frequency is the inverse of the wave period, $f = 1/T$, in 1/s.
Wave length	The wave length λ is the distance between successive crests, in m.
Wave height	Wave height, H, between the crest and trough within the wave period in m.
Wave number	$k = 2\pi/\lambda$ in rad/s.
Zero up-crossing period	T_z; The wave or response period, in s, between two up-crossings of the zero level in a specific wave/response event or average based on a short term sea state or long term statistics.

4 Application

4.1 General

This class guideline includes procedures and methods for directly calculated hydrodynamic wave loads and ship motions for the following, but not limited to:

— ships
— linear and weakly non-linear wave induced response
— unlimited service, i.e. North Atlantic or world wide trade
— restricted service with a specific scatter diagram
— hull forms within the limitations given in the DNVGL-RU-SHIP Pt.3 Ch.1 Sec.2 [3.2]
— hull forms outside the limitations given in the DNVGL-RU-SHIP Pt.3 Ch.1 Sec.2 [3.2]

— calculation procedures according to IACS Rec. No. 34 /8/
— calculation procedures for wave loads specified in IACS UR S11A /10/.

4.2 Calculated and observed wave induced loads

Observed wave loads can be measured by hull monitoring systems approved according to the hull monitoring rules DNVGL-RU-SHIP Pt.6 Ch.9 Sec.4 associated with the class notation **HMON**, [8.5]. Calculated design wave loads are based on design assumptions related to e.g. encountered sea states (i.e. design scatter diagram), heading profile, wave spectra, wave energy spreading, loading condition and speed. The design assumptions may deviate from what is encountered in operation.Whipping and freak wave events may increase the wave loads, but weather routing and good seamanship in harsh wave environment, like North Atlantic, may result in lower encountered wave loads than the design wave loads. An illustration of ships avoiding a storm is given in Figure 3.

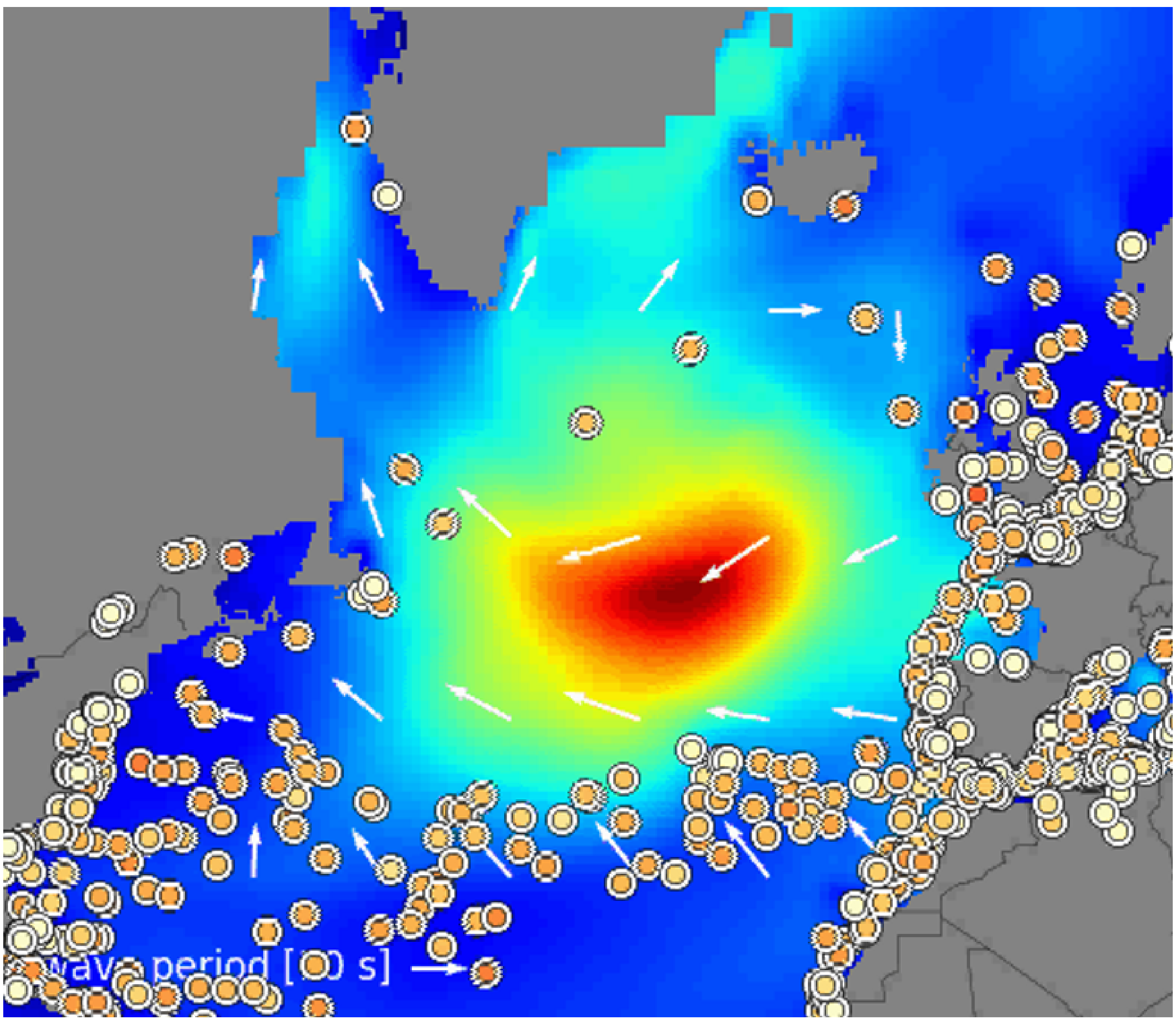

Figure 3 Illustration of ships routing in the North Atlantic to avoid a severe storm

4.3 Wave induced vibrations

Wave induced vibration like whipping is not explicitly considered, but the design wave loads may still account for significant contribution from whipping in combination with good seamanship. For wave induced vibration, reference is made to DNVGL-RU-SHIP Pt.3 Ch.4 Sec.4 [3.1.1], DNVGL-RU-SHIP Pt.5 Ch.2 Sec.4 [2.6], to DNVGL-CG-0153 and to the class notation **WIV** in DNVGL-RU-SHIP Pt.6 Ch.1 Sec.11.

4.4 Displacement ships

DNVGL-RU-SHIP Pt.3 Ch.4 provide loads for displacement ships. Ships with Froude number, F_r, below 0.4 are regarded as displacement ships. Ships with F_r between 0.4 and 1.0 are regarded as high speed vessels, while ships with F_r above 1.0 are regarded as planing ships. Crest and hollow landing situations are relevant for F_r > 0.5, DNVGL-RU-HSLC Pt.3 Ch.1 Sec.4 [1].

5 Rule limitations

5.1 General

The rules for hull structures are associated with the principles given in the DNVGL-RU-SHIP Pt.3 Ch.1 Sec.2 [3.2].

5.2 Seamanship

The wave scatter diagrams used as a basis for classification rules implicitly assume adequate seamanship according to good practice, e.g. avoiding storms, changing course and voluntary speed reduction. Such practice may be assumed on merchant ships.

5.3 Extreme loading

Extreme loading for strength assessment (i.e. ultimate limit state, ULS), is referring to a return period (design life) of 25 years at sea (i.e. largest response of about 10^8 encounter cycles coinciding with a probability of exceedance of about 10^{-8}, DNVGL-RU-SHIP Pt.3 Ch.4 Sec.1 [1.1.3]), and based on ships trading in the North Atlantic wave environment.

For offshore ships:

- without possibility for docking or with limited possibility for inspection and repair a higher return period may be required
- without possibility for docking a site specific wave environment should be considered
- where transit is relevant the transit phase should be regarded as North Atlantic trade.

For ships with service restriction, the correction factor, f_r, should be taken according to DNVGL-RU-SHIP Pt.3 Ch.4 Sec.3 and with service area restrictions in DNVGL-RU-SHIP Pt.1 Ch.2 Sec.5, or a specific trade may be used.

For ships with special operation conditions, specific considerations may be accepted.

5.4 IACS Rec. No.34

IACS Rec. No. 34 /8/ includes procedure and scatter diagram for calculation of rule loads and is the reference used for load calculated in CSR and in the DNV GL rules.

The hull girder loads resulting from calculation according to IACS Rec. No. 34 /8/ are considered to be conservative when taking into account seamanship and associated weather routing. Hence, when using the procedures in IACS Rec. No. 34 /8/, the resulting hull girder loads may be multiplied with a reduction factor, while motions and accelerations may not. Such approach will yield comparable hull girder loads as resulting from the rules.

5.5 Fatigue loading

Fatigue loading is related to a reference level of the probability level of exceedance of 10^{-2} based on ships trading in a world wide wave environment, with reference to the DNVGL-RU-SHIP Pt.3 Ch.9. This wave environment assumes that significant operation time is spent in harsh wave environments such as North Atlantic and North Pacific, but also in other main trades representative for the world's ship fleet. A more severe environment than the world wide wave environment may be specified, e.g. North Atlantic. The target design fatigue life is minimum 25 years.

For offshore ships:

— without possibility of docking, site specific wave environment should be considered
— where transit is relevant, the transit phase should be regarded as world wide trade. The part time in transit and at specific site should be estimated
— at the site the heading profile, wave energy spreading and wave spectra should be specified.

For ships with certain structural details, e.g. on gas carriers, associated with high consequence of failure, limited secondary barriers or limited accessibility during inspection, a target design fatigue life above 25 years and a Miner sum below 1 and North Atlantic rather than world wide may be required by the rules, e.g. DNVGL-RU-SHIP Pt.5 Ch.7 Sec.20 [8.5.1], DNVGL-RU-SHIP Pt.5 Ch.7 Sec.21 [3.3.2] and DNVGL-RU-SHIP Pt.5 Ch.7 Sec.23 [4.2.2].

For ships with special operation conditions, specific considerations may be accepted, e.g. fatigue loading for a specific trade in case of service restrictions.

6 Methods for hydrodynamic analysis

6.1 General

There are different methods and approaches which can be used to estimate wave induced design loads. The following methods are most commonly used:

— analytical calculations when the idealisation is considered valid
— empirical relations when the basis is considered valid
— 3-dimensional potential flow solvers for wave induced loads solving Laplace equation when viscous effects are not regarded important (which would be most cases). This includes boundary element methods (BEM)
— 3-dimensional computational fluid dynamics (CFD) methods for wave induced loads solving Euler or Navier-Stokes equations, e.g. RANS. The latter is regarded necessary if viscous effects are important
— 2-dimensional or 3-dimensional potential flow (e.g. BEM) or CFD for wave impact problems
— model tests based on Froude scaling, but also satisfying other scaling laws when regarded necessary
— full scale measurements by hull monitoring with class notation **HMON**, [8.5].

When the calculations are used in connection with classification, the Society may require documentation of the software including validation of the software, procedures and approaches.

7 Load categories

7.1 General

The loads may be divided into different categories depending on application:

— still water loads (static or steady) and wave induced loads (quasi-static, dynamic or impulsive loads)
— local or global loads

— linear and nonlinear
— load types.

The different categories should be considered when they are relevant.

The subsequent structural assessment should be:

— static
— quasi-static or
— dynamic.

depending on the nature or the loads and the importance of structural dynamic effects. When the wave induced loads are impulsive, coupled hydrodynamic and structural analysis, i.e. hydroelastic analysis, may be regarded necessary. In most cases quasi-static structural analysis is sufficient.

The load types consist of inertia loads (Newton's 2nd law), ballast and cargo loads, sea pressure, impact loads and hull girder loads. Inertia loads, ballast and cargo loads may be derived based on acceleration from the ship motions, but ballast water and fluid cargo loads may also have nonlinear effects in the splash zone and strong nonlinear effects from sloshing.

Still water loads consist of still water hull girder loads caused by the difference between mass and buoyancy distribution. The still water loads includes the hydrostatic pressure of sea water, ballast water and fluid cargo as well as inertia loads due to gravity. Cargo pressure from dry cargo in bulk is specially considered in DNVGL-RU-SHIP Pt.5 Ch.1. In case bulk cargo experience liquefaction, it can be considered as fluid cargo.

The hydrostatic pressure, P_S in kN/m^2, can be formulated as:

$$P_S = \rho g h$$

where:

ρ	=	density of water in t/m^3. Sea water ρ = 1.025
h	=	distance from the still water surface in m as positive value.

The global loads are regarded as hull girder loads, while local loads are regarded as sea pressure, ballast and cargo pressure as well as impact loads like slamming and sloshing. The magnitude of local impact loads needs to be consistent with the load area, which could be plate field, stiffener or girder, DNVGL-RU-SHIP Pt.3 Ch.10.

The still water and wave induced hull girder loads for monohull ships may be separated into:

— axial force, AF, in kN
— horizontal shear force, HSF, in kN
— vertical shear force, VSF, in kN
— torsional moment, TM, in kNm
— vertical bending moment, VBM, in kNm
— horizontal bending moment, HBM, in kNm.

Linear loads are proportional to the wave height, while nonlinear contribution causes a deviation from this linear relation. Linear hydrodynamic theory implies that the wave pressure is acting only up to the still water surface. In non-linear theory, the pressure is acting up to the water surface which may be above or below the still water surface. Non-linear effects should be considered when they may be important, e.g.:

— non-linear effect to sagging and hogging (nonlinear restoring and sea pressure to instantaneous water surface)
— splash zone effect for sea pressure

— splash zone effect for internal fluid pressure
— liquid impact from sloshing in tanks
— slamming
— nonlinear excitation causing whipping and springing response (not only slamming).

8 Additional class notations

8.1 General

The DNVGL-RU-SHIP Pt.6 specifies additional class notations. This subsection contains an overview of the class notations which are related to direct load analysis or measurements of response.

8.2 CSA

Computational ship analysis, **CSA**, as given in the DNVGL-RU-SHIP Pt.6 Ch.1 Sec.7, requires structural strength assessment based on directly calculated loads (3D panel model) and global FE analysis. Scope and extent of structural details are specified for fatigue and extreme loading. Local FE analyses are required.

8.3 RSD

Rational ship design, **RSD**, in the DNVGL-RU-SHIP Pt.6 Ch.1 Sec.8 requires a direct wave load analysis and global FE model for container ships according to DNVGL-CG-0131. Typical details subjected to fatigue assessment are hatch corners, end connections of longitudinals and welded details of upper part of the hull girder. Extreme loading and assessment of yield of hatch coaming and buckling of plate fields are also considered. The wave load analysis is represented by many ship specific equivalent design waves. The phasing between the different load components is accurately handled.

8.4 HMON

HMON in DNVGL-RU-SHIP Pt.6 Ch.9 Sec.4 refers to an approved hull monitoring system. A hull monitoring system, including e.g. strain sensors, GPS, loading computer, wind sensor and accelerometers, is a decision support system (DSS) which gives operational guidance in port and at sea. At sea it acts as a dynamic and static loading computer comparing the total loading against design limitations. Contribution from wave induced vibrations, i.e. springing and whipping, are included for both fatigue and extreme loading. On board trend analysis and the effect of changes to speed and course are displayed, and measurement data can be retrieved to shore for further onshore decision support with focus also on maintenance planning. Loads can be estimated based on the measured stresses.

8.5 NAUT and routing

Weather routing will have a significant impact on the extreme and fatigue loading. Requirements to resolution of data from weather information systems of weather surveillance system are specified within the **NAUT** notation in DNVGL-RU-SHIP Pt.6 Ch.3 Sec.3 [6.7]. A **HMON** system may be a part of a routing system.

8.6 WIV

Wave induced vibrations, typically designated as whipping and springing, cause additional high-frequent dynamic stress in the ship hull structure superimposed to the wave-induced stress. The additional stress has impact on the fatigue and extreme loading. Class notation **WIV**, in DNVGL-RU-SHIP Pt.6 Ch.1 Sec.11, can be assigned if additional load and strength assessment is carried out with explicit consideration of hull girder vibrations.

SECTION 2 WAVE CONDITIONS

1 Introduction

In a sea state, the wave induced response depends on the specific encountered wave event characterized by e.g. a wave height and a wave period. Short term response is a statistical representation of the wave induced response in a sea state, which is defined by a wave spectrum and wave energy spreading. Long term response is based on a statistical representation of all encountered sea states defined by a scatter diagram. These wave conditions are described in the following, and are necessary input to calculation of wave induced response. See also DNVGL-RP-C205.

2 Regular waves

A regular travelling wave is propagating with permanent shape and parameters as illustrated in Figure 1. Relevant parameters are:

— wave length, λ, in m
— wave period, T, in s
— phase velocity, c, in m/s
— wave frequency, f, in 1/s
— wave frequency, ω, in rad/s
— wave number, k, in rad/m
— surface elevation, η, in m
— wave crest height, A_c, in m
— wave trough depth, A_T, in m
— wave height, H, in m
— dispersion relation
— wave crest speed, λ/T, in m/s
— wave group velocity, C_g, in m/s
— wave steepness.

The wave steepness in deep water, S, is defined as:

$$S = 2\pi \frac{H}{gT^2} = \frac{H}{\lambda}$$

The maximum wave height H_{max} to wave length λ ratio (maximum steepness) defining the wave breaking limit depends on the wave length and the water depth, d, in m. The maximum steepness is given by:

$$\frac{H_{\max}}{\lambda} = 0.142 \tanh \frac{2\pi d}{\lambda}$$

In deep water, the breaking wave limit corresponds to a maximum steepness, S_{max}:

$$S_{\max} = H_{\max}/\lambda = 1/7$$

In shallow water the limit of the wave height can be taken as 0.78 times the local water depth.

The dispersion relation in deep water can be written as:

$$\omega^2 = kg$$

Based on this, the wave length in m, may be calculated as:

$$\lambda = \frac{g}{2\pi}T^2 = 1.56T^2$$

The wave speed of the propagating crest, *c* in m/s, can be estimated as:

$$c = \frac{\lambda}{T} = \frac{\frac{g}{2\pi}T^2}{T} = \frac{gT}{2\pi} = 1.56T$$

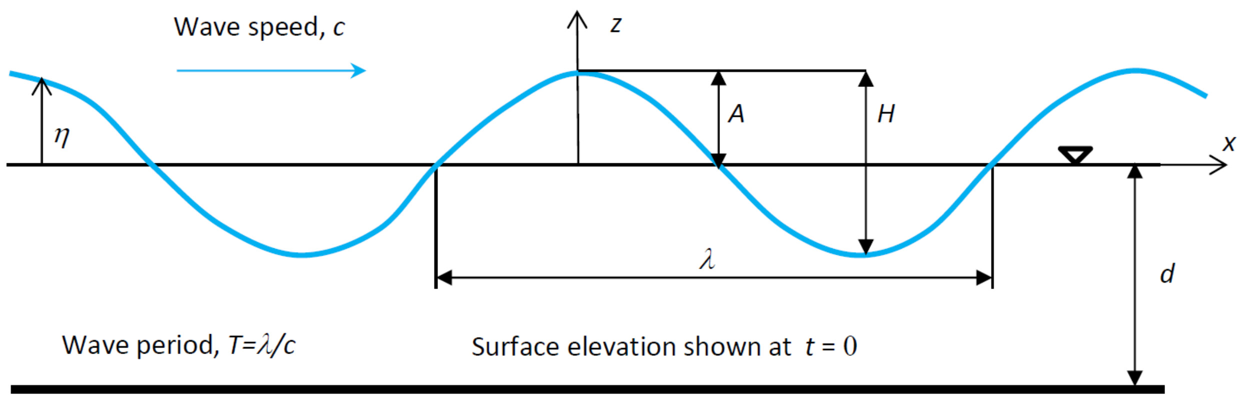

Figure 1 Regular travelling wave properties

3 Wave spectrum

A wave spectrum represents the wave energy (or wave amplitude) distribution of individual wave frequencies in a stationary sea state.

A large set of standardized wave spectra are provided in DNVGL-RP-C205 including recommended parameters. Wave spectra most relevant for ships are the two-parameter Pierson–Moskowitz (PM) (i.e. Bretschneider) and Jonswap wave spectra. These are unidirectional wave spectra referred to as single peak one-dimensional wave spectra, i.e. without wave energy spreading.

The PM wave spectrum for fully developed sea is given by:

$$S_{PM}(\omega) = \frac{5}{16} \cdot H_S^2 \omega_p^4 \cdot \omega^{-5} \exp\left(-\frac{5}{4}\left(\frac{\omega}{\omega_p}\right)^{-4}\right)$$

where:

ω = wave frequency in rad/s

ω_p = spectral peak frequency = $2\pi/T_p$, in rad/s

The Jonswap wave spectrum is formulated as a modification of the PM wave spectrum, and Jonswap represents a developing sea state in a fetch limited situation:

$$S_J(\omega) = A_\gamma S_{PM}(\omega)\ \gamma^{\exp\left(-0.5\left(\frac{\omega-\omega_p}{\sigma\ \omega_p}\right)^2\right)}$$

where:

γ = non-dimensional peak shape parameter

σ = spectral width parameter

σ_a for $\omega \leq \omega_p$

σ_b for $\omega > \omega_p$

A_γ = 1-0.287 $ln(\gamma)$ is a normalizing factor

Average values for the Jonswap experimental data are γ = 3.3, σ_a = 0.07, σ_b = 0.09. For γ = 1, the Jonswap wave spectrum reduces to the PM wave spectrum.

The Jonswap wave spectrum is expected to be a reasonable model for:

$$3.6 < T_p/\sqrt{H_s} < 5$$

and should be used with caution outside this interval. The effect of the peak shape parameter, γ, is shown in Figure 2.

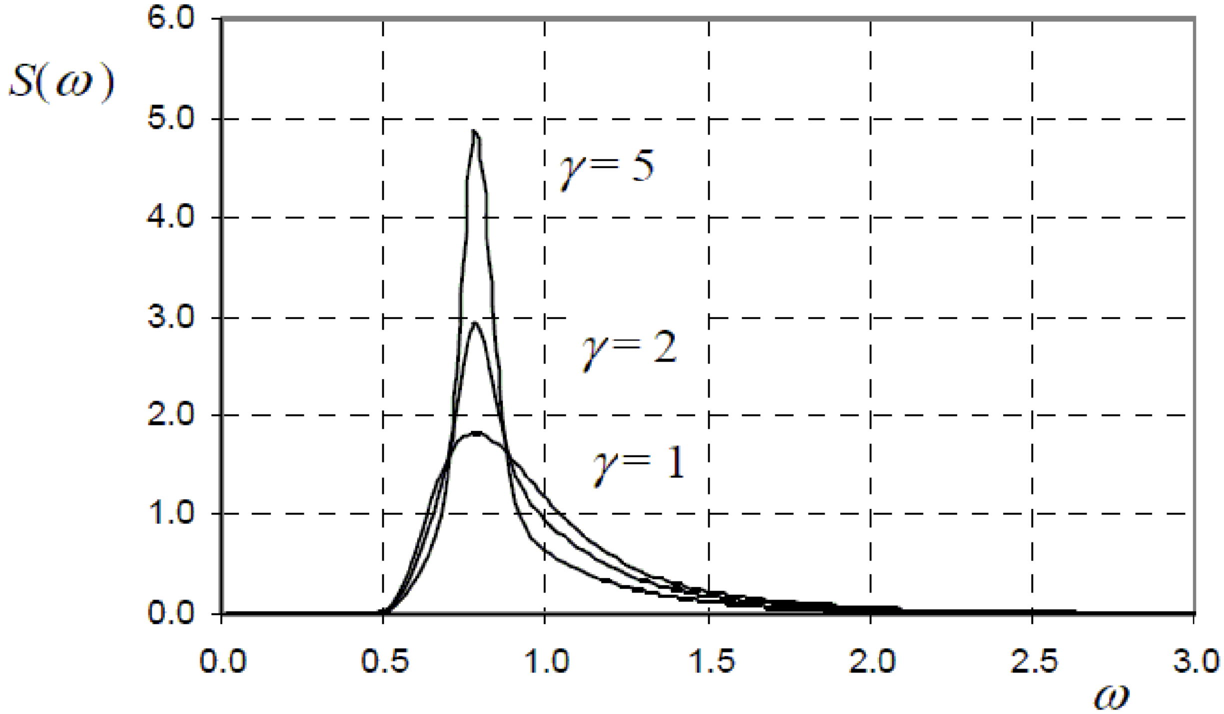

Figure 2 Jonswap spectrum for H_s = 4.0 m, T_p = 8.0 s and for γ = 1, 2 and 5.

If no particular values are given for the peak shape parameter, γ, the following value may be applied:

$$\gamma = 5 \;\text{ for }\; T_p/\sqrt{H_s} \leq 3.6$$

$$\gamma = \exp\left(5.75 - 1.15\frac{T_p}{\sqrt{H_s}}\right) \;\text{ for }\; 3.6 < T_p/\sqrt{H_s} < 5$$

$$\gamma = 1 \;\text{ for }\; 5 \leq T_p/\sqrt{H_s}$$

Both Jonswap and PM wave spectra adopt ω^{-5} as the governing high frequency tail behaviour. There is empirical support for a tail shape closer to the theoretical shape ω^{-4}. The difference between ω^{-4} and ω^{-5} tail behaviour may be of importance for structural dynamic response of ships, e.g. linear and non-linear springing.

3.1 Measured and simulated spectra

Wave spectra may be available from hind cast data or measured by a directional wave radar. Often these data are represented by two-dimensional frequency-heading tables of spectral values, i.e. they include directional and frequency information of the sea state simultaneously. Such data can be applied in sea-keeping assessments similar to usage of standardized parameterized wave spectra. To achieve unidirectional spectrum, the wave energy over the headings needs to be integrated. Information about bi-directional wind and swell sea is then lost. The multi-peak behaviour may however be kept.

4 Wave energy spreading

The wave energy spreading describes how the wave energy is distributed over different headings relative to the main heading. A large spreading implies that the waves are short crested, while a narrow spreading implies that the waves are long crested or unidirectional.

Directional short-crested wave spectra $S(\omega,\beta)$ may be expressed in terms of the unidirectional wave spectra as:

$$S(\omega,\beta) = S(\omega)D(\beta,\omega) = S(\omega)D(\beta)$$

Where the latter equality represents a simplification often used in practice. Here $D(\beta,\omega)$ and $D(\beta)$ are directional functions. β is the angle between the direction of elementary wave trains and the main wave direction of the short crested wave system. The directional function fulfils the requirement:

$$\int_\beta D(\beta,\omega)d\beta = 1$$

For a two-peak spectrum expressed as a sum of a swell component and a wind-sea component, the total directional frequency spectrum $S(\omega,\beta)$ can be expressed as:

$$S(\omega,\beta) = S_{wind\ sea}(\omega)D_{wind\ sea}(\beta) + S_{swell}(\omega)D_{swell}(\beta)$$

A directional function often used for wind sea is:

$$D(\beta) = \frac{\Gamma(1+n/2)}{\sqrt{\pi}\Gamma(1/2+n/2)}\cos^n(\beta-\beta_p)$$

where:

Γ	=	the Gamma function
$\|\beta-\beta_p\|$	=	relative angle ≤ $\pi/2$
β	=	relative spreading around the main direction
β_p	=	main direction, which may be set to the prevailing wind direction if directional wave data are not available.

The wave induced response may be sensitive to the wave energy spreading, i.e. the constant n, which should be chosen to be representative for the actual sea state. Values representative for wind sea are n = 2 to 4, while for swell n > 7 is more appropriate. Higher sea states may also be associated with a higher n.

5 Zero up-crossing period, T_z and peak period T_p

The T_z and the mean wave period T_1, both in s, may be related to the peak period by the following approximate relations ($1 \le \gamma < 7$) based on Jonswap:

$$\frac{T_z}{T_p} = 0.6673 + 0.05037\gamma - 0.006230\gamma^2 + 0.0003341\gamma^3$$

$$\frac{T_1}{T_p} = 0.7303 + 0.04936\gamma - 0.006556\gamma^2 + 0.0003610\gamma^3$$

Combining these two equations give the following relations:

For γ = 3.3; $T_p = 1.286T_z$ and $T_1 = 1.073Tz$

For γ = 1; $T_p = 1.405T_z$ and $T_1 = 1.087T_z$

An alternative is to estimate the zero up-crossing period, T_z, and mean wave period, T_1, directly from the spectral density function and its moments. ω_p corresponds to the highest peak in the spectral density function representing the wave spectrum, e.g. as seen in Figure 2. From this the peak period can be estimated:

$$T_p = \frac{2\pi}{\omega_p}$$

The spectral moments of order *n* of the response process for a given heading may be described as:

$$m_n = \int_\omega \sum_{\beta_p - 90°}^{\beta_p + 90°} D(\beta)\omega^n \cdot S(\omega / H_s, T_z, \beta) d\omega$$

and the zero up-crossing period and mean period are is defined as:

$$T_z = 2\pi\sqrt{\frac{m_0}{m_2}}$$

$$T_1 = 2\pi\sqrt{\frac{m_0}{m_1}}$$

6 Scatter diagram

A scatter diagram defines the probability of occurrence of the different sea states. Each sea state is defined by the significant wave height, H_s, in m, and the zero up-crossing period, T_z (or peak period T_p) in s. The North Atlantic scatter diagram is given in Table 1 and the world wide scatter diagram is given in Table 2. These are based on /1/. For the North Atlantic, the nautical zones 8, 9, 15 and 16 are considered. For world wide many nautical zones are considered, including operation time in North Pacific and North Atlantic. The H_s and T_z combinations are class midpoints, e.g. observed wave heights with significant wave height between 1.0 m and 2.0 m are placed as observations related to the class midpoint of 1.5 m. The nautical zones are illustrated in Figure 3.

DNVGL-RP-C205 provides approximate scatter diagrams for 104 nautical zones based on /1/.

Table 1 Scatter diagram for North Atlantic operation with H_s in m and T_z in s

T_Z (s)	3.5	4.5	5.5	6.5	7.5	8.5	9.5	10.5	11.5	12.5	13.5	14.5	15.5	16.5	17.5	18.5	Sum
H_S (m)																	
0.5	1.3	133.7	865.6	1186.0	634.2	186.3	36.9	5.6	0.7	0.1	0.0	0.0	0.0	0.0	0.0	0.0	3050
1.5	0.0	29.3	986.0	4976.0	7738.0	5569.7	2375.7	703.5	160.7	30.5	5.1	0.8	0.1	0.0	0.0	0.0	22575
2.5	0.0	2.2	197.5	2158.8	6230.0	7449.5	4860.4	2066.0	644.5	160.2	33.7	6.3	1.1	0.2	0.0	0.0	23810
3.5	0.0	0.0	34.9	695.5	3226.5	5675.0	5099.1	2838.0	1114.1	337.7	84.3	18.2	3.5	0.6	0.1	0.0	19128
4.5	0.0	0.0	6.0	196.1	1354.3	3288.5	3857.5	2685.5	1275.2	455.1	130.9	31.9	6.9	1.3	0.2	0.0	13289
5.5	0.0	0.0	1.0	51.0	498.4	1602.9	2372.7	2008.3	1126.0	463.6	150.9	41.0	9.7	2.1	0.4	0.1	8328
6.5	0.0	0.0	0.2	12.6	167.0	690.3	1257.9	1268.6	825.9	386.8	140.8	42.2	10.9	2.5	0.5	0.1	4806
7.5	0.0	0.0	0.0	3.0	52.1	270.1	594.4	703.2	524.9	276.7	111.7	36.7	10.2	2.5	0.6	0.1	2586
8.5	0.0	0.0	0.0	0.7	15.4	97.9	255.9	350.6	296.9	174.6	77.6	27.7	8.4	2.2	0.5	0.1	1309
9.5	0.0	0.0	0.0	0.2	4.3	33.2	101.9	159.9	152.2	99.2	48.3	18.7	6.1	1.7	0.4	0.1	626
10.5	0.0	0.0	0.0	0.0	1.2	10.7	37.9	67.5	71.7	51.5	27.3	11.4	4.0	1.2	0.3	0.1	285
11.5	0.0	0.0	0.0	0.0	0.3	3.3	13.3	26.6	31.4	24.7	14.2	6.4	2.4	0.7	0.2	0.1	124
12.5	0.0	0.0	0.0	0.0	0.1	1.0	4.4	9.9	12.8	11.0	6.8	3.3	1.3	0.4	0.1	0.0	51
13.5	0.0	0.0	0.0	0.0	0.0	0.3	1.4	3.5	5.0	4.6	3.1	1.6	0.7	0.2	0.1	0.0	21
14.5	0.0	0.0	0.0	0.0	0.0	0.1	0.4	1.2	1.8	1.8	1.3	0.7	0.3	0.1	0.0	0.0	8
15.5	0.0	0.0	0.0	0.0	0.0	0.0	0.1	0.4	0.6	0.7	0.5	0.3	0.1	0.1	0.0	0.0	3
16.5	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.1	0.2	0.2	0.2	0.1	0.1	0.0	0.0	0.0	1
Sum	1	165	2091	9280	19922	24879	20870	12898	6245	2479	837	247	66	16	3	1	100000

Table 2 Scatter diagram for world wide operation with H_s in m and T_z in s

T_Z(s)	3.5	4.5	5.5	6.5	7.5	8.5	9.5	10.5	11.5	12.5	13.5	14.5	15.5	16.5	17.5	Sum
H_s (m)																
1.0	311	2734	6402	7132	5071	2711	1202	470	169	57	19	6	2	1	0	26287
2.0	20	764	4453	8841	9045	6020	3000	1225	435	140	42	12	3	1	0	34001
3.0	0	57	902	3474	5549	4973	3004	1377	518	169	50	14	4	1	0	20092
4.0	0	4	150	1007	2401	2881	2156	1154	485	171	53	15	4	1	0	10482
5.0	0	0	25	258	859	1338	1230	776	372	146	49	15	4	1	0	5073
6.0	0	0	4	63	277	540	597	440	240	105	39	13	4	1	0	2323
7.0	0	0	1	15	84	198	258	219	136	66	27	10	3	1	0	1018
8.0	0	0	0	4	25	69	103	99	69	37	17	6	2	1	0	432
9.0	0	0	0	1	7	23	39	42	32	19	9	4	1	1	0	178
10.0	0	0	0	0	2	7	14	16	14	9	5	2	1	0	0	70
11.0	0	0	0	0	1	2	5	6	6	4	2	1	1	0	0	28
12.0	0	0	0	0	0	1	2	2	2	2	1	1	0	0	0	11
13.0	0	0	0	0	0	0	1	1	1	1	0	0	0	0	0	4
14.0	0	0	0	0	0	0	0	0	1	0	0	0	0	0	0	1
Sum	331	3559	11937	20795	23321	18763	11611	5827	2489	926	313	99	29	9	0	100000

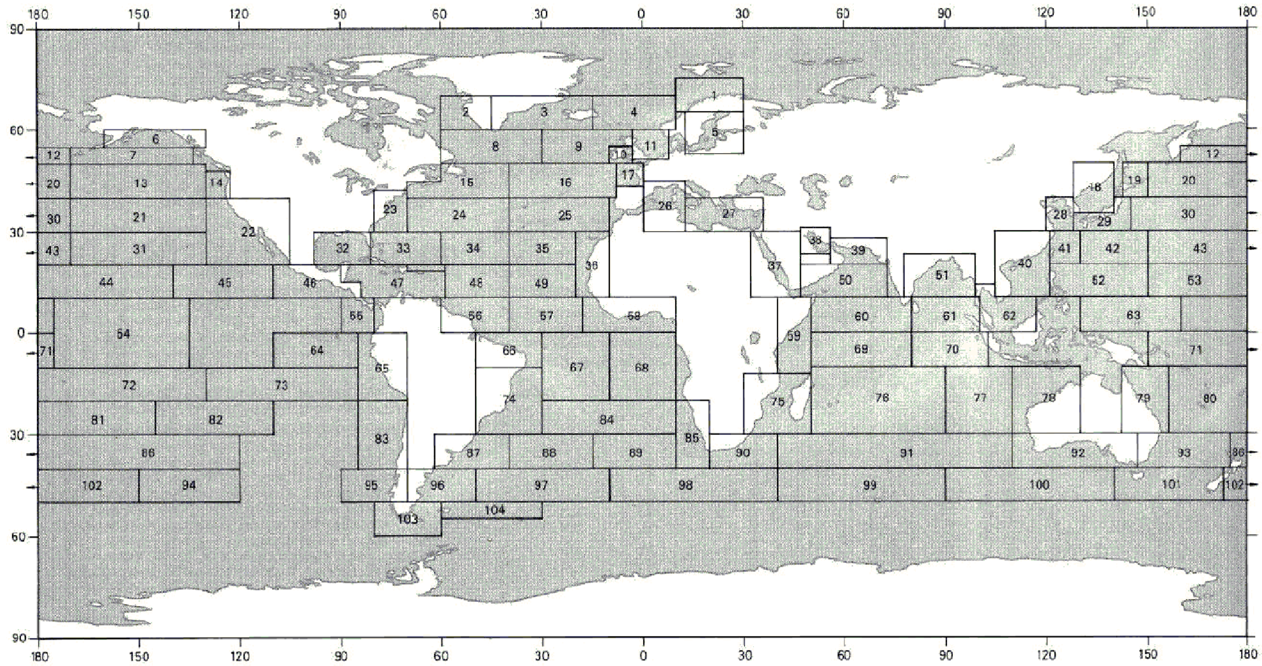

Figure 3 Nautical zones

7 Adjustment for routing and vibration

The scatter diagrams for North Atlantic and world wide in Table 1 and Table 2, respectively, and the scatter diagrams in DNVGL-RP-C205 are based on observations done between 1949 and 1986 and do not fully include the effect of modern weather routing.

7.1 Fatigue loads for world wide or North Atlantic

If hydrodynamic analysis is carried out to calculate rule loads using design scatter diagrams for North Atlantic, the operational factor, f_R, the vibration factor, f_{vib}, and the fatigue coefficient, f_{fa}, should be applied consistently with the prescriptive loads in the rules. The environmental factor, f_e, in DNVGL-RU-SHIP Pt.3 Ch.9 Sec.4 [4.2] is also relevant to make the loads representative for world wide (if not calculated for world wide).

The stress range obtained from direct hydrodynamic analysis (component stochastic or full stochastic analysis) for either North Atlantic or world wide should be multiplied with the operational factor, f_R, in DNVGL-RU-SHIP Pt.3 Ch.9 Sec.4 [4.3], the vibration factor, f_{vib}, in DNVGL-RU-SHIP Pt.3 Ch.4 Sec.4 [3.1.1] and the fatigue coefficient, f_{fa}, in DNVGL-RU-SHIP Pt.3 Ch.4 Sec.3.

7.2 Fatigue loads for site specific operation or specific trade

If hydrodynamic analysis is carried out for a site-specific field, the operational factor, f_R, should be taken as 1.0, and the scatter diagram should be site-specific and not include any routing effect. The vibration factor, f_{vib}, should be taken as 1.05 and the fatigue coefficient, f_{fa}, should be taken as 1.0.

If hydrodynamic analysis is carried out for a specific trade, the operational factor, f_R, should be taken as 1.0 when the scatter diagram is based on matching ship positions and wave data and contain data representative for operation after year 2000, otherwise operational factor as for prescriptive loads should be used. The vibration factor, f_{vib}, in DNVGL-RU-SHIP Pt.3 Ch.4 Sec.4 [3.1.1] will apply, as well as the fatigue coefficient, f_{fa}, in DNVGL-RU-SHIP Pt.3 Ch.4 Sec.3.

7.3 Extreme loads for North Atlantic

If hydrodynamic analysis is carried out to calculate rule loads using design scatter diagrams for North Atlantic, the operational factor, f_R, as for prescriptive loads in DNVGL-RU-SHIP Pt.3 Ch.4 Sec.4 [3.1.1] should be used. The operational factor, f_R, is only applicable for the hull girder loads.

The stress range obtained from direct hydrodynamic analysis (component stochastic or full stochastic analysis) should be multiplied with the operational factor, f_R.

7.4 Extreme loads for site specific operation or specific trade

If hydrodynamic analysis is carried out for a site-specific field, the operational factor, f_R, should be taken as 1.0, and the scatter diagram should be site-specific and not include any routing effect.

If hydrodynamic analysis is carried out for a specific trade, the operational factor,f_R, should be taken as 1.0 when the scatter diagram is based on matching ship positions and wave data and contain data representative for operation after year 2000, otherwise operational factors as for prescriptive loads should be used.

SECTION 3 CONCEPTS

1 Introduction

Several concepts, as explain in the following subsections, may be relevant in the calculation of hydrodynamic loads.

2 Equation of motion

Ship motions follows from Newton's 2nd law:

$$\left(\boldsymbol{M} + \boldsymbol{A}(\boldsymbol{x})\right)\ddot{\boldsymbol{x}} + \boldsymbol{B}(\boldsymbol{x},\dot{\boldsymbol{x}})\dot{\boldsymbol{x}} + \boldsymbol{C}(\boldsymbol{x})\boldsymbol{x} = \boldsymbol{F}(\boldsymbol{x},\dot{\boldsymbol{x}},t)$$

Where the vector ***x*** denotes the ship responses, e.g. representing the 6 DOF rigid body motions. The vector ***F*** represents the excitation forces from the incident undisturbed waves (Froude-Krylov force) and diffracted (reflected) waves. The excitation forces depends on the wave encounter frequencies. The radiation forces, $\boldsymbol{F}_{\boldsymbol{rad}}$, takes the form:

$$\boldsymbol{F}_{\boldsymbol{rad}}(\ddot{\boldsymbol{x}},\dot{\boldsymbol{x}},\boldsymbol{x},t) = -\boldsymbol{A}(\boldsymbol{x})\ddot{\boldsymbol{x}} - \boldsymbol{B}(\boldsymbol{x},\dot{\boldsymbol{x}})\dot{\boldsymbol{x}}$$

Where ***A*** and ***B*** are the added mass and damping coefficients. Radiation represent the situation when the ship generates waves going away from the ship. The mass matrix, ***M***, represents the inertia of the ship in air. In most formulations the restoring matrix, ***C***, is due to hydrostatics only.

The added mass and damping are frequency dependent. For low and high frequencies the linear damping associated with wave generation goes towards zero. The damping can also be proportional to the speed squared. It is then mainly related to viscous effects such as skin friction and vortex shedding. An important example is roll damping due to bilge keels. At high speed also hull lift damping can contribute.

For low frequencies the added mass goes towards zero, but at high frequencies it tends to approach a constant value. Added mass is associated with pressure being proportional to acceleration.

The restoring is independent of frequency and is related to the water plane area. It acts like a spring stiffness in case of changed buoyancy. It can be nonlinear in case of non-vertical ship sides and large motions and roll/pitch angles.

The equation of motion above represents forced motions. The response on board will be felt at the encounter frequency, ω_e, which can be formulated as:

$$\omega_e = \omega - \frac{\omega^2 U}{g}\cos\beta$$

where:

U = ship forward speed in m/s

β = relative heading in degrees between the ship and the wave propagation direction, $\beta = 0°$ in following seas.

In beam seas the encounter period corresponds to the wave period. In following sea the ship speed, which cause zero encounter frequency, can be estimated as:

$$U = \sqrt{\frac{g\lambda}{2\pi}} = 1.25\sqrt{\lambda}$$

For higher speeds the ship is overtaking the waves.

The equation of motion above without the right hand side refers to free motion. In this case the natural frequency can be estimated neglecting the damping. The natural frequency, ω_n, in rad/s, is expressed as:

$$\omega_n = 2\pi\sqrt{\frac{C}{M+A}}$$

The natural period, in s, is the inverse relation of the natural frequency:

$$T_n = \frac{2\pi}{\omega_n}$$

3 Irregular waves

Regular waves are sinusoidal waves as illustrated in Sec.2 Figure 1. The surface propagation, $\eta(x,t)$ in m, can be written as:

$$\eta(x,t) = A\cos(\omega t - kx)$$

Where:

A = is the wave amplitude in m
ω = the wave frequency in rad/s
k = wave number $k = 2\pi/\lambda$, in rad/m.

Irregular waves representing a sea state can be expressed as a linear summation of regular wave components, i:

$$\eta(x,t) = \sum_{i=1}^{n} A_i\cos(\omega_i t - k_i x + \varepsilon_i)$$

where:

ε_i = random phase variable evenly distributed between 0 and 2π.

The amplitude A_i can be estimated from the wave spectrum $S(\omega)$ described in Sec.2 [4]:

$$A_i = \sqrt{2S(\omega_i) \Delta \omega_i}$$

where:

$\Delta\omega_i$ = Frequency interval considered, in rad/s.

The maximum wave steepness of regular waves is defined in Sec.2 [2]. In irregular sea states, the significant wave steepness, S_S, can be estimated as:

$$S_S = \frac{2\pi}{g}\frac{Hs}{Tz^2} = \frac{2}{\pi g}\frac{m_2}{\sqrt{m_0}}$$

where:

— m_0 and m_2 are the zeroth and second spectral moment from Sec.2 [5]
— the significant wave height, H_s, in m, is given by:

$$H_s = 4\sqrt{m_0}$$

— and zero up-crossing period, T_z, in s, from Sec.2 [5]

The limiting steepness values may be taken as:

$$S_S = 1/10 \text{ for } T_z \le 6\text{s}$$

$$S_S = 1/15 \text{ for } T_z \ge 12\text{s}$$

with linear interpolation in between. Based on the peak period the limiting criteria can be written:

$$S_p = 1/15 \text{ for } T_p \le 8\text{s}$$

$$S_p = 1/25 \text{ for } T_p \ge 15\text{s}$$

which resembles the limit which may be used in towing tanks:

$$S_p = \frac{2\pi H_s}{gT_p^2} < 0.03$$

4 Wave induced response

The waves serve as the excitation for the wave induced response. The wave induced response is thereby the consequence of the waves acting on the hull surface. The wave induced response can be amongst others:

— all rigid motions including accelerations and velocities
— flexible motion and deformation of the structure like springing and whipping
— global hull girder loads
— local loads
— internal stress in the structure.

Mathematically it may be expressed as:

$$y = ax$$

Where x is the input (excitation), a is the conversion factor and y is the output (response). The wave induced response is often referred to as only the response.

5 Linear responses

5.1 Introduction

Transfer functions (RAO and phase), short term and long term response are elements in most direct hydrodynamic calculations, and the concepts are briefly explained in the following.

5.2 Response amplitude operator (RAO)

The concept of transfer function, i.e. RAO is essential for sea-keeping assessments. An illustration of a poor and improved transfer function is given in Figure 1.

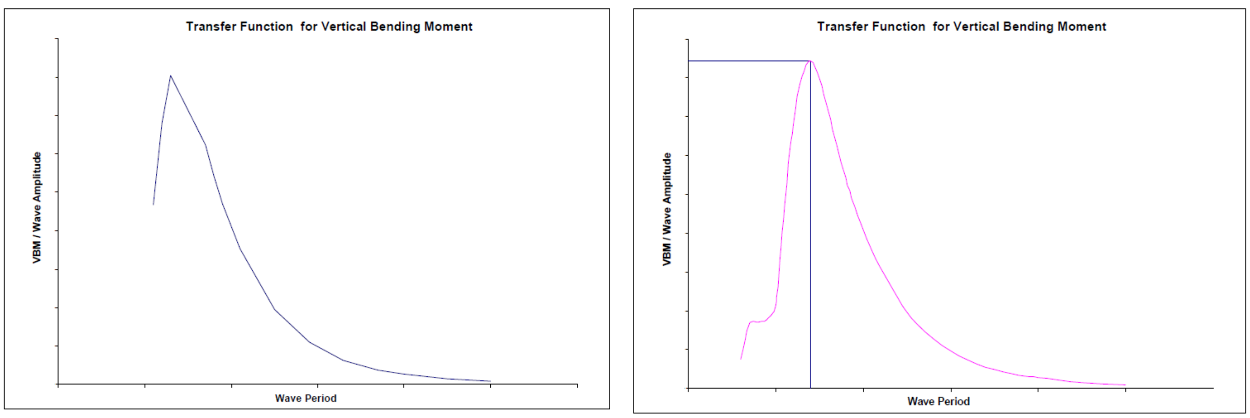

Figure 1 Poor representation of a transfer function on the left, and on the right a transfer function where the peak and shorter wave periods are well represented

In case of linear theory the steady state solution $x(t)$ of the equation of motions can be written as:

$$\boldsymbol{x}(t) = A \cdot \mathrm{Re}\{\boldsymbol{\eta}(\omega, \beta) \cdot \exp(i\omega_e t)\}, \qquad i = \sqrt{-1}$$

Where A is the incident single wave amplitude of a regular wave of frequency, ω, from wave direction, β, relative to the ship. The corresponding encounter frequency is denoted as ω_e. The real part of a complex number is denoted as $Re\{\ldots\}$. The complex function $\boldsymbol{\eta}(\omega,\beta)$ is the actual response-amplitude-operator RAO of response $\boldsymbol{x}$. This function is provided by linear sea-keeping programs. Each component η_i of the vector $\boldsymbol{\eta}$ can always be expressed like:

$$\eta_i(\omega, \beta) = |\eta_i| \cdot \exp(i\theta_i)$$

where $|\eta_i|$ is the amplitude of the corresponding component of $\boldsymbol{x}(t)$ in case A = 1. The actual response component will have a peak when:

$$\omega_e t + \theta_i = 2\pi \cdot n$$

where n is an arbitrary integer. The real parameter θ_i is denoted the phase of the response component.

The phase explains the phase angle relative to the incident wave, e.g. phase equal to zero may be defined as cosine wave with crest amidships in the software. The phase, θ_i, lagging behind with e.g. positive $\pi/2$ implies that the peak response occurs $1/4^{th}$ of the wave period later, i.e. after the wave crest has passed amidships and is located at the aft quarter length, if the wave length corresponds to the ship length in head sea. The phase tends to shift by about π, i.e. 180°, when the encounter frequency moves past the resonance frequency.

The concept of RAOs is applied for all linear responses including motions, accelerations, pressures, hull girder loads, disturbed wave elevation etc. By including a linear structural model, RAOs can be established for stresses and strains at requested locations. However, stress formulations that are non-linear with respect to stresses, like Von-Mises stresses (equivalent stresses), cannot be represented by RAOs.

5.3 Short term response

Short term response implies that the transfer function has been combined with the wave spectrum and the wave spreading function. It defines the response in a sea state and the statistical properties can be estimated based on the response spectrum. The short term response gives properties as significant response, maximum value exceeded once during a sea state, mean response period, zero crossing period and kurtosis and skewness.

Wave spectra are explained in Sec.2 [3] while wave energy spreading is explained in Sec.2 [4]. The response spectra can be interpreted as the energy distribution of the response with respect to frequency content similar to the wave spectra distribution. Derivation of response spectra is explained in Sec.4 [2]. The short term distribution of peaks may be represented by a distribution such as Rayleigh for Gaussian distributed responses (i.e. normal distributed). This is a good assumption in many linear cases. Another relevant distribution could be 2-parameter Weibull. See DNVGL-CG-0129 App.C for Weibull distributions.

5.4 Long term response

Long term response is estimated by combining the short term response from the individual sea states with the scatter diagram in Sec.2 [6]. Long term response is associated with a certain probability of being exceeded, e.g. 10^{-2} or 10^{-8}, or with a certain return period, e.g. 25 years. The return period is associated with a response which is exceeded once within the return period. An exceedance probability of 10^{-8} is the response, which is exceeded once amongst 10^{8} response cycles. A response with a return period of 25 years is close to equivalent with an exceedance probability of 10^{-8}.

The long term distribution is frequently represented by a 2-parameter Weibull distribution. The term straight line spectrum refers to the case where the inverse Weibull slope is 1.0. See DNVGL-CG-0129 App.C for Weibull distributions.

The long term response value may be referred to as the most probable maximum. This concept is related to the extreme value distribution, which is a probability density function (pdf), and the peak value of this pdf is referred to as the most probable maximum. Even though there is a high likelihood of exceeding the most probable maximum, the extreme value distribution is narrow around the most probable maximum. The most probable maximum is also close to the expected maximum value in the extreme value distribution. The expected value is associated with 50% probability of being exceeded.

SECTION 4 STATISTICS

1 Introduction

The wave induced loads are stochastic, and probabilistic methods are needed in order to estimate the extreme (ULS) and fatigue loads (FLS). For such assessments linear sea-keeping (RAOs) can be combined with linear wave statistics in order to provide initial estimates for extreme and fatigue loads. However, if important non-linear effects are relevant for the actual response, statistical corrections may be applied.

2 Short term statistics

Short-term statistics is used to estimate the statistical response induced by a single stationary sea state defined by a wave spectrum. The ship speed and course are fixed. For linear response statistics, analytical results based on RAOs, given wave spectrum and duration can be established.

For non-linear responses the statistics become complicated especially for extreme value assessments. As an alternative, conservative scaling of linear results may be applied based on rule values.

2.1 Short term extreme values from linear responses

Consistent with linear theory short term extreme values can be derived from the standard deviation σ of the response. The response spectrum can be interpreted as the energy distribution of the response with respect to wave (encounter) frequency ω and heading angle β. Given the RAO η(ω,β) for a given response, the response spectrum R(ω,β) is defined as:

$$R(\omega,\beta) = |\eta(\omega,\beta)|^2 S(\omega,\beta)$$

where S(ω,β) is the wave spectrum and |**η**(ω,β)| is the amplitude of the RAO. The moments of the response spectrum are defined as:

$$m_{\eta,n} = \int_0^{2\pi}\int_0^{\infty} \omega^n R(\omega,\beta)\, d\omega d\beta \quad , \quad n = 0,1,2,3,\ldots$$

where n = 0 provides the variance (standard deviation squared), n = 1 the first moment and n = 2 is the moment of inertia of the spectra. Short term extreme values can be derived from the standard deviation σ of the response corresponding to the zero moment

$$\sigma^2 = m_{\eta,0}$$

The significant response amplitude, $\bar{\eta}_{1/3}$, is defined as the mean value of the highest one-third part of the amplitudes:

$$\bar{\eta}_{1/3} = 2\sqrt{m_{\eta,0}}$$

The mean period T_1 and the average zero-crossing period T_z, in s, of the response are defined as:

$$T_1 = 2\pi\sqrt{\frac{m_{\eta,0}}{m_{\eta,1}}}$$

$$T_z = 2\pi\sqrt{\frac{m_{\eta,0}}{m_{\eta,2}}}$$

The expected extreme value $E[\eta_{max}]$ depends on the exposure time T_d in the actual sea state. For narrow banded linear processes a good approximation is given by:

$$E[\eta_{max}] = \sigma \cdot \sqrt{2\ln\left(\frac{T_d}{T_z}\right)}$$

where T_z is the zero-crossing encounter period of the response. It should be noted that the probability of exceeding this value is 63%. However, the extreme value distribution is narrow for large durations and the response will usually not exceed this estimate significantly.

From a time series the expected value can be estimated as:

$$E[y] = \frac{1}{n}\sum_{i=1}^{n} y_i = \sum_{i=1}^{1} y_i p_i$$

where n is the number of samples of the response y. The second relation is more general in case all samples do not have equal probability, while in the first relation equal probability is assumed, i.e. $p_i = 1/n$.

Another often applied metric is the extreme value, $\eta_{\max,\alpha}$, associated with a small prescribed exceedance probability, α. A typical probability level is $\alpha = 0.05$ which means that out of 20 cases, one is expected to exceed the limit. An approximate formula valid for narrow banded linear processes is:

$$\eta_{max,\alpha} = \sigma \cdot \sqrt{2\ln\left(\frac{T_d}{\alpha\, T_z}\right)}$$

The above formulas assume Rayleigh distributed response amplitudes. For broad-banded linear responses the Rayleigh distributions should in principle be replaced by the Rice distribution. However, for practical computations the uncertainty related to the assumption of linear responses exceed the level of statistical effects from broad banded effects.

For long term statistics, the short term cumulative probability distribution, $F_{ST}(y|h, t, \beta, u)$ can be important. The probability of a response value being below y is conditional on the wave height, h, wave period, t, relative wave heading, β, and ship speed, u. In a short term case, these quantities can be considered as fixed, and at least the ship speed is often considered constant in long term statistics. The short term probability of exceeding the value y is then $Q_{ST}(y\,|\,h,t,\beta) = 1 - F_{ST}(y\,|\,h,t,\beta)$.

There are several concepts and theories for extreme value prediction. A thorough introduction is given in /2/.

2.2 Extreme value estimation based on non-linear time series

For non-linear processes, extreme values must often be deduced from finite time series of the responses. The estimated extreme value should not be taken as the maximum observed value in the time series. The associated probability level may be too high or too low.

In general 20-30 repeated experiments are needed for reliable estimates of extreme values. Hence, for prediction of expected max response in a 3-hour sea state, 60-90 hours of available data is needed. The expected value can be estimated as the average of the maximum value in each 3-hour interval of the total signal. Depending on the type of response, the scatter of each maximum observation may be large. For e.g. slamming responses, this scatter is large and long time series are needed for reliable estimates.

In practice, sufficiently long times series are difficult to obtain for many non-linear responses. In such cases various extrapolation techniques may be applied, e.g. 2 or 3 parameter Weibull fit to the observations on higher probability levels, and then subsequently extrapolation to the required probability level. Another recognized extrapolation method is the peak-over-threshold (POT) method as explained in /2/.

It should be noted that extrapolation techniques may provide misleading estimates if other physical effects occur at rare events. As an example, water-on-deck will significantly influence the extreme value distribution of sagging loads. This effect cannot be predicted by extrapolation of results from lower wave amplitudes.

3 Long term statistics

The long term statistics of ship responses take into account the various sea conditions that the ship is expected to encounter during its life time. As a consequence, the long term statistics combine results from various short term assessments as described in the previous section.

The most common method for assessing long term statistics is to apply simple weighting of short term responses based on the probability of encountering various sea conditions. The weighting factor is defined by the probability of occurrence of each sea state as defined by the actual wave scatter diagram, e.g. in Sec.2 [6].

Based on the short term cumulative probability distribution, $F_{ST}(y|h, t, \beta)$ for a constant speed case, the discretized long term distribution, $F_{LT}(y)$, can be given as:

$$F_{LT}\left(y\right) = \sum_{k=1}^{k_{\max}} \sum_{j=1}^{j_{\max}} \sum_{i=1}^{i_{\max}} F_{ST}\left(y|h_k, t_j, \beta\right) f_{H_s, T_z}\left(h_k, t_j\right) f_\beta\left(\beta_i \middle| h_k, t_j\right) \omega_{h_k, t_j, \beta} \Delta h \Delta t \Delta \beta$$

where the deltas are the grid size, $f_{Hs,Tz}$ is the long term joint probability distribution taken from the scatter diagram, f_β is the heading distribution and $\omega_{h,\ t,\ \beta}$ is the weight function. The latter is expressed by:

$$\omega_{h_k, t_j, \beta_i} = \frac{T_{z,avg}}{T_z}$$

where T_z is the zero up-crossing period for the specific sea state and $T_{z,avg}$ is the long term average zero up-crossing period. The long term value y_D for a given return period of T_{DF} years is given by:

$$F_{LT}\left(y_D\right) = 1 - \frac{1}{N_D} = 1 - \frac{T_{z,avg}}{T_{DF} \cdot 365.25 \cdot 24 \cdot 3600}$$

The probability that the long term value will be larger than y_D is given by:

$$Q_{LT}\left(y_D\right) = 1 - F_{LT}\left(y_D\right) = \frac{1}{N_D}$$

where N_D is the number of peaks during the T_{DF} years.

4 Probability of exceedance

The probability of exceedance, Q, of a response, y, depends on the number of cycles, n, exceeding a threshold, y', versus the total number of cycles, n_0, during a specified time period, e.g. design life. It can be written as:

$$Q(y > y') = \frac{n}{n_0}$$

where n_0 can be estimated based on the design life, T_{DF}, in years divided by the long term zero up-crossing period, T_z, in s, of the response:

$$n_0 = \frac{T_{DF} \cdot 3600 \cdot 24 \cdot 365.25}{T_z}$$

The probability of exceedance associated with the extreme loading (ULS) is considered as 10^{-8}, while the probability of exceedance associated with fatigue loading is 10^{-2}. Both refer to the long term distribution during the design life and then the average T_z.

The probability of exceedance can also be associated with short term response in a sea state. Of interest is often the maximum response being exceeded only once during all the encountered response cycles during a sea state, which refer to a duration in the order of 3 hours. This maximum response may correspond to the maximum probable value from the extreme value distribution in case of Rayleigh distributed responses.

5 Return period and up-crossing rate

The return period T_R is always related to an event like exceeding a specific load level, L_L. The average time between each time the load exceed L_L is T_R. The up-crossing frequency or up-crossing rate, ν, in 1/s, is defined as $\nu = 1/T$. For combined loads the terminology out-crossing rate is applied for $\nu = 1/T$.

A common misunderstanding exists that e.g. a 25-year event is likely to occur only once in a 25-year period. The correct interpretation is explained above. In general it should be noted that the probability of load level, L_L, to occur in any given year is $P = 1/T_R$. Hence each (operational) year 4 of 100 sister ships are expected to exceed load level L_L if T_R = 25 years.

6 Coefficient of contribution

Coefficient of contribution (CoC) /2/ is a method to identify the sea state, which contributes most to the long term statistics. For linear response and statistics, this method is very efficient.

The long term statistics of responses are accumulated from the short term statistics of each cell of the scatter diagram. The long term statistics takes into account the expected exposure time and the response level in each cell as explained in [3]. It follows that some cells contribute more to the long term values than other cells. In terms of contour lines CoC reveals the importance of different areas of the scatter diagram. For extreme values of typical responses, only a few sea states are important. The set of important sea states is response dependent. For fatigue more and lower sea states may be important. For extreme loading with speed sensitive whipping response, also lower sea states may be important.

Linear CoC only requires the actual RAO, wave spectrum and the wave scatter diagram. The method is often used to identify critical sea states for which advanced non-linear methods will be applied for more accurate response predictions. For responses with strong non-linear effects , linear CoC may not provide a proper selection of critical sea states and a larger range of sea states may need to be considered. Slamming induced whipping is an example where linear CoC may not provide correct results.

Based on the short term probability distribution in [2.1] and long term value in [3] the CoC may be expressed by:

$$CoC\left(h_k, t_j\right) = \frac{Q_{ST}\left(y_D \middle| h_k, t_j, \beta_i\right) f_{H_s, T_z}\left(h_k, t_j\right) \omega_{h_k, t_j, \beta_i}}{Q_{LT}\left(y_D\right)}$$

Further details can be found in /2/ or /12/.

SECTION 5 LINEAR METHODS

1 Introduction

Linear theory is used to calculate key motion and load characteristics considered sufficient for e.g.FLS assessments. For ULS assessments, non-linear methods may be needed as described in Sec.6. However, non-linear effects can often be accounted for by non-linear scaling of linear results as outlined in Sec.6.

Linear analyses can usually be applied as a screening tool for identification of critical sea states or design waves to be assessed with non-linear methods as long as the nonlinear effects are considered weak.

Wave impact loads like slamming are not handled by linear methods. However, indication of characteristic flow parameters, like expected extreme relative velocities can be obtained from linear simulations combined with statistical methods. Such flow parameters may be applicable as input for local models for assessment of extreme impact loads as discussed in Sec.6.

2 Linearization

Linear sea-keeping software can be classified as either frequency or time domain solvers. All oscillating forces are assumed to be linear with respect to incident wave amplitudes and body motions. Responses can be superimposed. As a consequence the responses caused by individual regular harmonic waves of unit amplitudes (RAOs) are important characteristics of the ship.

The radiation coefficients ***A***, ***B*** and ***C***, as well as the excitation force ***F***, given in Sec.3 [2] are independent of the ship response and can be considered as constants. However, the radiation coefficients depend on actual oscillation frequency, and the excitation forces depend on the wave length and related oscillation period. Linear sea-keeping programs calculates the force coefficients and solve the equation of motions for the response ***x***, expressed as RAO.

Linear theory is based on the assumption of small incident wave amplitudes. The time varying wetted surface is replaced by the average wetted surface. Most formulations simplify this further and consider only the wetted surface below the calm water line. Depending on the actual response, ship geometry, speed, water depth and wave conditions, linear theory can indicate the response level also for severe sea states. In case of large incident waves and ship motions, the effects of time varying wetted surface are considerable both with respect radiation and diffraction forces. In such cases non-linear effects needs to be considered.

3 Frequency domain

Frequency domain programs predict amplitudes and relative phases of the responses induced by sinusoidal incident waves in terms of RAOs. Frequency domain programs typically divide the solution into sub-problems to be combined by linear superposition.

1) The radiation solution is the response induced by harmonic ship oscillations in calm water. The related forces and loads are typical provided in term of added mass, damping and stiffness components.
2) The excitation solution is the response induced by regular waves when the ship is restrained from motion. This solution is often split into:
    a) Froude-Krylov response caused by the undisturbed incident wave pressure field.
    b) Diffraction response derived from the pressure field generated by the ship to counteract the incident wave flux through the hull.

4 Time domain

Time domain programs simulate directly the time history of all responses given actual incident waves. The incident waves could represent regular or irregular waves. Both RAOs and statistical information can be deduced by signal processing of the time series of the simulated responses.

5 Methods

There are various numerical methods for predicting the wave induced motions and loads. All methods require proper modelling and correct execution parameters in order to produce reliable results.

5.1 Strip theory

5.1.1 2D

Strip theory /3/ may be used as a screening tool for sea-keeping assessments of ships moving with $F_r < 0.3$. However, if 3D effects are pronounced other methods should be applied.

A common principal assumption is that the length of the ship is much larger than the draft and beam. As a consequence certain components of the radiation and diffraction wave field vary slowly along the ship length leading to simplifications of the flow. The flow can then be represented as 2-dimensional at each longitudinal section of the ship. These assumptions are not sound near the bow and stern of blunt ships and for high speed ships. All strip theories assumes high frequency of oscillations. Hence, for ships with forward speed results are better for head and bow seas, and poor agreement can be expected for following seas.

5.1.2 High speed theory, 2D+t =2.5D

For high speed displacement ships with $F_r > 0.4$ the 2.5D theories are applicable for response assessments.

The principle assumption for 2.5D theories is that the ship speed is so high that no hydrodynamic effects propagate upstream. E.g. the flow at midship will not affect the flow at the bow. As a consequence a local solution can be established at the bow. This solution will again influence the flow at the next section. By repetition, the load on the complete ship is then obtained by solving 2D problems in sequence from the bow to the stern by taking upstream effects into account.

For very high speeds the 2.5D free surface conditions approximate the 3D free surface conditions.

5.2 Panel methods

For sea-keeping assessments there are two main categories of 3D panel methods:

— Free surface Green's function methods requires mesh of the wetted hull. Only the simple Neuman-Kelvin free surface conditions, see [5.2.2], can be handled. Fake solutions may occur for certain wave frequencies called irregular frequencies corresponding to artificial sloshing resonance modes inside the hull. This is a particular challenge for short waves. Results containing irregular frequencies need to be excluded and confirmed by inspections of RAOs, which should not contain spikes.

— Rankine methods require panels on the free surface in addition to the hull. These methods introduce more flexibility with respect to handling of physical free surface conditions, which is in particular important for resolving the flow for blunt ships in case of forward speed. Rankine methods are applicable for air cushion supported ships.

5.2.1 Zero speed

In case of zero forward speed both free surface Green's function methods and Rankine methods are applicable for response assessments.

Due to the simple free surface conditions for zero speed, the free surface Green's function methods is a particular popular choice. In particular for cases requiring fine mesh discretization.

5.2.2 Forward speed formulations

Response predictions in forward speed are significantly more complex. The main method characteristic is related to selecting the dominant part of the flow, which all other flow components can be considered small and be linearized with respect to. This flow is referred to as the base flow:

— The Neuman-Kelvin formulation assumes a slender ship in the sense that a uniform current can be considered as the base flow.
— The double-body formulation assumes a simple stationary 3D flow as the base flow. This is the steady flow around the ship when replacing the free-surface with a lid.
— The steady-wave formulation applies the physical flow around the ship in absent of incident waves as the base flow. This base flow is typical based on a non-linear solution of the steady field.

For most responses all these formulations may provide acceptable results. However, for responses where the local flow field is important, the double-body and stead-wave formulations should be considered. In case of large incident waves all these formulations are questionable and non-linear effects should be considered.

In case of forward speed, the free surface Green's function methods are too complex to apply consistently. A common by-passing of this problem is to apply the zero-speed formulation with a simple adjustment for actual encounter frequency. Such simplifications should not be applied for assessing responses that are sensitive to the ship generated wave field like e.g. relative motions between ship and waves, non-viscous damping and added-resistance.

The Rankine methods are applicable for all above base flow formulations. The Free Surface Green's function methods can only deal with the Neuman-Kelvin formulation.

For forward speed problems, the parameter τ is often important:

$$\tau = \frac{U\,\omega_e}{g}$$

where U is the ship speed in m/s, ω_e is encounter frequency in rad/s and g is the acceleration of gravity in m/s^2. If τ > 1/4, waves will not propagate upstream of the ship. In the limit case τ = 1/4 numerical problems may occur. This is not a problem for time domain Rankine methods.

In following seas the encounter frequency, ω_e, in rad/s, may approach zero which are associated with very long radiation waves and significant numerical errors may occur.

In case of high speed, the temporal and spatial gradients of the flow are high. As a consequence a fine mesh and small time steps are in general required. Some programs provide special time integration solvers for high speed cases.

5.3 Planing ship

Planing ships are ships moving with very high speed with F_r > 1.0. Buoyancy only has a minor contribution to the hydrodynamic force. The hydrodynamic force model needs to account for flow separation. The load above the zero speed calm surface is significant and must be handled by a spray flow model. As a consequence classical linear sea-keeping programs are not applicable for assessing the response of planing ships.

Special methods for sea-keeping assessments of planing ships have been developed. A comprehensive overview can be found in /4/.

5.4 Damping

Proper handling of damping is important for the study of resonant response levels. Outside resonance the actual damping level is not critical for the response level.

Potential theory is limited to calculation of damping represented by outgoing free surface waves. Other damping components may be needed. In particular, viscous effects are important for assessments of resonant roll motion. Depending on the geometry it may also be important for other responses, e.g. if considerable vortex shedding is expected to occur.

In case of large amplitude motions of flared bodies, the linear model will not capture the additional non-linear wave making damping induced by the flare. This may be relevant for resonant pitch motion where linear theory may over-predict the response.

Ships are often equipped with control devices to provide additional damping of roll motion, e.g. bilge keels, fin stabilizers and anti-roll tanks. As a consequence potential flow programs need user specified coefficients to account for missing damping. The damping levels depend on the actual response level. Hence, for linear models a linearization is required to obtain the realistic damping levels in specific sea conditions. It will vary with e.g. H_S.

In order to provide realistic damping it is necessary to know the actual damping model in the program. Often the model requires input coefficients to be obtained by model tests, CFD or empirical knowledge.

For roll motion, damping data can be based on information from roll decay tests in calm water, or measured roll moment in case of forced roll motion with given motion amplitude.

At roll resonance the potential theory will under predict the total roll damping. The roll motion will, consequently, be over-predicted. The effect from damping mechanisms not related to wave-making, such as vortex-induced damping (eddy-making) near sharp bilges, drag of the hull (skin friction), skegs and bilge keels (normal forces and flow separation), should be included. Example of non-linear roll damping methods for ship hulls includes those published by /5/, /6/ and /7/.

Non-linear roll damping on a ship hull is a function of roll angle, wave frequency and forward speed. As the roll angle is generally unknown and depends on the scatter diagram considered, an iteration process is required. The following 4-step iteration procedure may be used for guidance:

1) Input a roll angle, θ_x^{input}, to compute non-linear roll damping
2) Perform ship motion analysis including damping from 1)
3) Calculate long term roll motion, θ_x^{update}, with probability level 10^{-2} for FLS and 10^{-8} for ULS, using design wave scatter diagram
4) If θ_x^{update} from 3) is close to θ_x^{input} from 1), stop the iteration. Otherwise, set a new θ_x^{input} as the mean value of θ_x^{input} and θ_x^{update}, and go back to 1).

Roll motion can affect responses such as acceleration, pressure and torsion. Viscous damping should be evaluated for beam and quartering seas. The viscous damping has little influences in cases where the natural period of the roll is far away from the exciting frequencies. For fatigue it is regarded sufficient to calibrate the viscous damping for beam sea and use the same damping for all headings.

5.5 Acceleration including gravity

Linear rigid body velocities and acceleration follows directly by scaling of the motion RAOs with $i\omega_e$ and $-\omega_e^2$ respectively. In order to calculate the relevant acceleration for prediction of inertia loads on internal fluid, cargo and lashing systems, the additional dynamic acceleration effects from gravity must be added. See /4/ for further details.

SECTION 6 NON LINEAR METHODS

1 Introduction

Linear theory assumes infinitesimal small wave amplitudes. This assumption becomes more questionable as the wave heights, steepness and response amplitudes increase. For large sea states, results from linear theory need to be corrected. Such corrections can be obtained by results from non-linear methods. Non-linear responses may also require special statistical methods.

2 Non-linear effects

In large waves the wetted part of the hull may vary considerable as the wave trains pass by. This may contribute to additional loads e.g. on the bow, stern, deck and superstructure. Additional loads may include impact loads, which may cause considerable contribution to local and global ULS and FLS loads. Except for hydrostatic pressure below calm water surface, linear simulations do not take into account any time variations of the wetted surface.

The motion characteristics can be non-linear, in particular roll motion. Motion control systems like anti-roll tanks have strong non-linear characteristics. Another means of controlling the roll motion is by active fins, which can be effective at high speed, but with less effect at zero speed. This may affect acceleration levels and design loads.

3 Frequency domain

Non-linear frequency domain programs are commercial available for zero speed assessments of sum and difference frequency problems, especially relevant for offshore problems. Commercial tools for relevant sum frequency problems like springing are not available for forward speed problems.

Lower order drift forces including forward speed effects like added resistance can be assessed by some frequency domain tools.

4 Time domain

Most non-linear sea-keeping tools for ships are based on time domain simulation formulations. Both potential flow and CFD codes are relevant. Such programs simulate directly the time history of all responses. Incident waves may be regular or transient based on linear or non-linear wave theory.

5 Methods

All non-linear methods apply some simplifications of the physics. For practical simulations, it is important to select a method which balances requirements of accuracy and computational time. This selection depends on the type of response and the available computer capacity. As a consequence there is a large variety of non-linear sea-keeping methods. The most common methods are:

— Non-linear hydrostatics and Froude-Krylov based methods. These methods are based on linear solvers with the exception that hydrostatic pressure and the undisturbed incident wave pressure are integrated over the instantaneous wetted hull. The wetted hull is defined by only considering the ship motions and the incident wave profile.
— Non-linear strip theory is similar to the above method, but can include non-linear radiation and diffraction forces by considering the instant submergence of each 2D section. The 2D theory neglects important physical effects.
— Weak scatter non-linear panel methods takes into account the effect of large incident wave in a simplified way. Panels are distributed on the incident wave surface and the instantaneous wetted hull. The flow field is linearized with respect to this control volume. Hence, the method accounts for non-linear radiation and diffraction effect, e.g. of bow flares. This method is considerable more robust than fully non-linear panel

methods. However, it is more computational demanding than non-linear hydrostatics and Froude-Krylov methods.

— Fully non-linear panel methods are in general not robust for sea-keeping assessment of ships. The main reason for this is the handling of overturning waves e.g. close to the hull. Panel methods cannot simulate after the occurrence of wave breaking and simplifications of the wave surface is needed.
— CFD methods are the most complete methods in terms of covering physical effects. For sea-keeping simulations RANS/VOF solvers are the most common CFD methods. For commercial assessments CFD is typically limited to the study of responses in short wave sequences due to severe computational requirements.
— Nonlinear pressure extrapolation is an approximate nonlinear pressure correction method, which can be used in some cases, extrapolating pressure from the linear sea-keeping analysis above the time-average waterline, taking into account hull geometry. For the hull surface below and above the time-average waterline, pressure is interpolated and extrapolated, respectively; the instantaneous water line is then found where the pressure equals atmospheric pressure, and pressure integration is repeated taking into account the actual submerged area. Care should be taken to ensure equilibrium of forces and moments after the pressure is extrapolated and integrated over corrected wet surface. This can also be done in the frequency domain.

Due to limitations in all numerical models, physical model tests are still needed occasionally for assessment of ship responses for novel or unusual designs or operational conditions.

5.1 Water on deck

In case of large relative motion, in particular in the bow region, green sea loads may occur due to water on deck. This may introduce significant local or global loads. Local green sea loads may refer to impact loads. Based on relative motion at the bow and domain decomposition techniques, commercial potential flow solvers need to include a separate flow model in order to approximate the load on deck. For global loads like sagging, it is conservative to exclude water on deck loads. In general, green sea loads reduce the sagging moment.

Water on deck is best handled by CFD methods provided a sufficiently fine model of the bow region and a proper surface capturing method is applied.

5.2 Selection of methods

Only a small number of sea-keeping responses require a viscous flow model. Even roll damping can often be calculated accurately using the Euler formulation. The Euler formulation apply zero viscosity, but account for intrinsically generated numerical vorticity. In particular, in case of bilge keels this is sufficient for simulating proper roll characteristics also for large amplitude motions. CFD programs can be used for establishing relevant roll damping coefficients to be applied in potential flow solvers. Wave damping effects from the CFD calculations should not be included into the roll damping coefficients. Such effects are already handled by the potential flow solver.

CFD is in particular useful when significant variation of the wetted surface occur. When the bow and stern gets in and out of water and when green sea loads are relevant. This is mainly due to robust surface capturing methods like VOF often implemented in CFD solvers. This is not related to Navier-Stokes equations.

At present commercial potential flow solvers are based on surface tracking methods instead of surface capturing methods. As a consequence they are not as robust for dealing with large variations of the wetted surface, especially in cases when large areas completely exit the water and then plunge into the water again. This is often the case when extreme loads occur.

Simplified potential flow solvers taking important non-linear effects into account can provide useful extreme load predictions based on proper calibration with CFD or model test results combined with proper statistics. Such solvers can also be used for identifying critical sea states and wave trains to be analysed with CFD or model test.

6 Approaches

It is not feasibly to apply the most sophisticated numerical methods or model tests to assess the complete set of life time load cycles corresponding to a scatter diagram. A hierarchy of methods should be applied in a cost efficient, sound and reliable manner.

6.1 Screening: Weak non-linear versus strong non-linear responses

Weakly non-linear responses are responses which are only moderately influenced by non-linear effects. For such responses, it is reasonable to assume that there is a positive correlation between the response level based on linear and non-linear calculations. Hence, sea states that induce a large linear response will most likely also induce a large response based on non-linear calculations and vice versa.

For weakly non-linear responses, it makes sense to apply linear tools to identify a set of critical sea states and wave trains according to Sec.4 [6]. Advanced methods like non-linear potential flow solvers or CFD should then be applied to simulate the response in those critical conditions. These simulated results should be combined with proper statistical assessments as described in Sec.4.

For strongly non-linear responses like slamming induced whipping, screening based on linear results may be misleading because important non-linear effects are not considered in the screening phase. In order to apply reliable screening, the screening tool needs to account for important non-linear effects. If the screening method is very approximate a larger set of critical sea states and wave trains needs to be assessed by the sophisticated tool. In case it is not possible to account for the physical effects directly in the screening tool, screening should be based on a set of strongly correlated parameters, e.g. in case of green sea loads the screening may be partly based on large relative motions in the bow region.

6.1.1 Non Linear factor

In case it is not possible to perform direct non-linear simulations, it is possible to account for non-linear effects by sound scaling of linear results. Sound scaling factors can be obtained by DNVGL-RU-SHIP Pt.3 Ch.4, model test results or non-linear analysis of similar cases. Scaling factors can only be applied for corresponding ship type, main dimensions, loading condition, ship speed and wave conditions.

An example is to consider the linear part of the vertical wave bending moment from DNVGL-RU-SHIP Pt.3 Ch.4, and replace this with the linear calculated results, and thereby use the nonlinear factors in DNVGL-RU-SHIP Pt.3 Ch.4 to obtain the sagging and hogging moment. The calculations ensure then that the linear calculated results are ship specific; however the non-linear factors are not.

6.2 Regular design wave

A regular design waves is a regular wave of a particular wave length, wave height and heading. The corresponding wave induced response simulated by a non-linear program is assumed to be the expected worst non-linear response. Each response has its specific design wave parameters and cannot be used for assessing other responses.

Most regular design waves are based on linear theory as explained in [6.8].

6.3 Conditioned wave train

A conditioned wave train is applied similar to a regular design wave. However, the wave profile has a spatial and temporal deformation corresponding to actual critical wave profile encountered at seas. Each response has its specific conditioned wave train and cannot be used for assessing other responses.

Most conditioned wave trains are based on linear theory as explained in [6.8]. Sets of non-linear conditioned wave trains can be established based on sampling of wave profiles from non-linear screening tool simulations.

6.4 Design sea state

A design sea state is a sea state where the life time extreme value of a particular response is likely to occur. For most responses a small set of design sea states is likely. The set will vary from response to response. For moderately non-linear responses, design sea states are typically provided by the linear CoC method as described in Sec.4 [6].

For strongly non-linear responses, the design sea states should be selected based on simplified non-linear simulations taking the essential non-linear physics into account. In case of limited capabilities in the screening tool, a larger set of design sea states should be defined. For each design sea state, the response should be simulated by model tests or a non-linear program taking the relevant physics into account. The expected duration in the actual sea state should be taken into account based on expected life time and the relevant scatter diagram. The expected maximum response should be estimated based on the statistical methods explained in Sec.4 [2.1].

6.5 Contour analysis

Contour analyses are only relevant for ULS assessments. In particular for strongly non-linear responses, it may not be possible to define a proper set of design sea states using the methods described in [6.4]. This could be due to limited physical effects accounted for by the screening tool. In such cases it is possible to apply the contour line approach. Hence, for each T_z in the scatter diagram, add the sea state with the highest probable H_s to the set of design sea states, i.e. the highest H_s that the actual ship is expected to encounter in the life time based on actual scatter diagram and expected life time.

It should be taken into account that the duration of the actual sea state will affect the extreme value. If the duration of the sea state at the contour line is very small it is recommended to also include the sea state just below the contour line in the set of design sea states. Due to longer duration the maximum response may occur in sea states with slightly lower significant wave heights.

The contour line may also be established based on given return period.

6.6 Wave height dependent RAOs

Non-linear responses can often be linearized with respect to a response level. E.g., the equivalent linearized vortex induced roll damping coefficient differs with respect to actual H_s. This can be accounted for by introducing wave height dependent damping coefficients, which implies wave height dependent RAOs. When doing linear short-term and long-term statistics, the actual wave height dependent RAOs should be applied for each sea state.

Similar to wave height dependent RAOs, the RAOs may also vary with respect to zero-crossing periods.

Due to speed-sea state restrictions, the concept of H_s and T_z dependent RAOs is also relevant for linear responses of high speed ships.

6.7 Non-linear CoC

The concept of CoC as described in Sec.4 [6] can also be applied for non-linear responses based on non-linear screening tools. E.g. by taking bow slamming into account in the screening tool, it is possible to establish tables for which sea states it is likely to encounter the maximum whipping induced loads.

6.8 Linear design waves

Design waves are short wave trains representing an event, which will induce the design load. Most design waves are constructed based on information from linear RAOs and wave statistics.

From non-linear simulations of a ship response given a design wave, it is possible to establish correction factors to be applied on linear results.

Design waves are response dependent. Hence a set of design waves are needed for assessing several responses.

The most classical design wave is a regular sinusoidal wave. The wave period correspond to the peak period of the RAO. The wave amplitude is defined as the amplitude, for which a linear simulation will reproduce the long term extreme response as explained in Sec.8 [4.1] .

There are several methods for constructing irregular linear design waves. The objective with such wave trains is to construct wave profiles corresponding to the most critical wave profile in real sea states. Such wave trains are short.

There is not a standard terminology for the large amount of design wave methods. Irregular design waves are often referred to as conditioned waves. Examples of irregular linear design waves are MLER (most likely extreme response), EMLER, MLRW and CRRW.

SECTION 7 PROCEDURE FOR DIRECTLY CALCULATED RULE LOADS

1 Introduction

This section describes how hydrodynamic calculations can be used to derive rule loads such as hull girder loads, sea pressure, accelerations and motions.

2 Scatter diagrams

For ULS and FLS the North Atlantic scatter diagram in Sec.2 [6] is applicable.

For FLS, however, a correction should be made in the fatigue assessment to convert the stress from North Atlantic into world wide trade by using the environmental factor f_e. Alternatively, the world wide scatter diagram in Sec.2 [6] may be used directly, and calculations can be done at a probability level of exceedance of 10^{-2}, thereby also avoiding the down scaling factor f_p in DNVGL-RU-SHIP Pt.3 Ch.4.

3 Wave spectrum

The Pierson-Moskowitz wave spectrum in Sec.2 [3] is applicable.

4 Wave energy spreading

The $\cos^2$ wave energy spreading in Sec.2 [4] is applicable.

5 Heading profile

All wave headings from 0 to 360° should be included in the linear long term calculations with a maximum step (spacing) of 30°. The heading distribution should be evenly distributed, i.e. with the same probability for all headings.

6 Speed

For estimation of hull girder loads, motions, sea pressures and accelerations, for the purpose of ULS, the speed should be taken as:

— 5 knots for CSR ships
— 5 knots for ships with $L \geq 150$m and 0 knots for ships with $L < 100$m
  and with linear interpolation in between.

For local impact pressures on the hull surface a speed of 2/3 of the design speed should be used, but may be replaced with maximum achievable speed in different headings and sea states. In this assumption seamanship in terms of voluntary speed reduction is not included.

For the purpose of FLS, the speed should be taken as:

— 2/3 of the maximum service speed, V.

7 Weather routing

When the long term target rule load for North Atlantic in DNVGL-RU-SHIP Pt.3 Ch.4 is directly calculated to replace corresponding rule values, the long term hull girder loads, motions, accelerations and pressures should be multiplied with the factor such as f_R, f_{vib} and f_{fa} consistently with the rules, see Sec.2 [7]. The factors are however included explicitly in many rule formulas, and correction should not be included twice. The resulting long term load components can then be used in the prescriptive assessment.

8 Return period

The loads should be calculated at a return period of 25 years for ULS. This would correspond to approximately a probability level of exceedance equal to 10^{-8} depending on the response considered.

If f_p is not used for downscaling to estimate fatigue loads, the FLS loads should be calculated at 10^{-2} probability level of exceedance. The loads for FLS can be taken at another speed than for ULS, so f_p may be interpreted to contain also the speed effect.

9 Non-linear corrections

9.1 Hull girder loads

When calculating vertical bending moments (sagging and hogging) and vertical shear forces, corrections for non-linear effects should be included. The non-linear corrections shall be based on DNVGL-RU-SHIP Pt.3 Ch.4 unless particular values are available for actual ship or similar ship.

When calculating other hull girder loads, non-linear corrections can be disregarded.

9.2 Motions and accelerations

In calculations of extreme motions, non-linear damping in roll should be considered, e.g. as with the iterative procedure in Sec.5 [5.4]. Other non-linear effects are to be accounted for in line with DNVGL-RU-SHIP Pt.3 Ch.4.

9.3 Pressures and splash zone

The splash zone is considered slightly different for ULS and FLS. For FLS a splash zone correction is made to represent a combined case. For ULS it is either wave crest or wave trough phase.

When the ULS dynamic sea pressure with a return period of 25 years has been derived, a correction above or below the mean water line (WL) in a wave crest or wave trough situation, respectively, can be made based on linear theory. The correction above and below WL should be based on the pressure at WL, P_{WL-8}, and the distribution is based on a hydrostatic consideration. For a wave crest, the pressure below the water surface down to WL is considered hydrostatic, with the water surface as the reference. For a wave trough, the pressure above the water surface, being a distance in m equal to $z_{WL-8}=P_{WL-8}/\rho g$ below the WL, is set to zero, and the pressure below is simply derived by adding the negative dynamic and hydrostatic pressure (this will lead to a small non-zero pressure at the trough surface). This is illustrated in Figure 1.

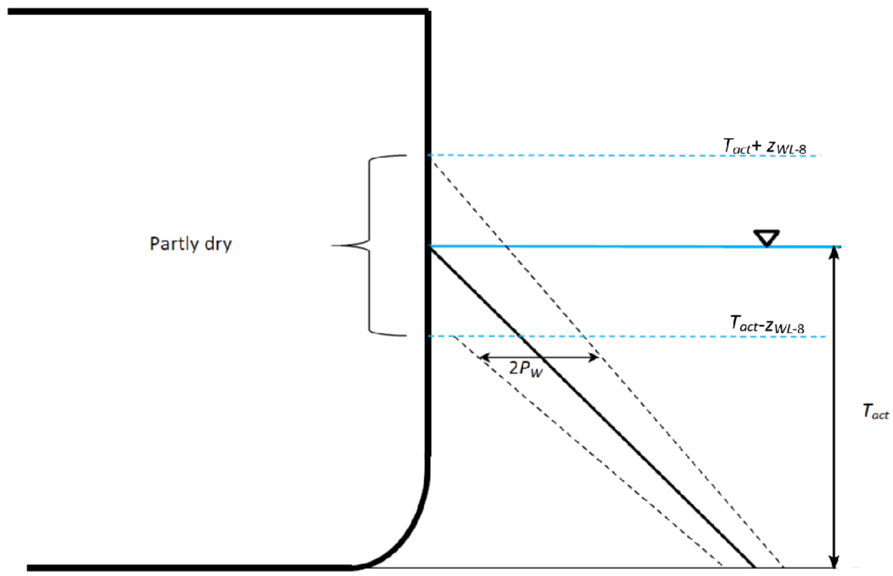

Figure 1 Illustration of splash zone for ULS. Dashed lines are the pressure distribution in the crest and trough phase, respectively

For FLS, the dynamic sea pressure at 10^{-2} level can be derived, and a correction can be made for the partly wet and dry splash zone. The extent of the splash zone, $z_{WL\text{-}2}$ in m, above and below the water line, WL, can be estimated based the pressure at WL, $P_{WL\text{-}2}$, divided by the hydrostatic pressure per meter ρg as:

$$z_{WL-2} = \frac{2P_{WL-2}}{\rho \cdot g}$$

This implies that the extent of the splash zone is defined at a 10^{-4} probability level, which is also used in DNVGL-RU-SHIP Pt.3 Ch.4 Sec.5 [1.4]. The dynamic pressure above the WL can be divided by 2, while a linear interpolation factor from ½ to 1 is combined with the pressure from WL at $z = T_{act}$ down to the pressure at $z = T_{act} - z_{WL\text{-}2}$, respectively, where T_{act} = the draught, in m, at the mean water line considered. An illustration of the splash zone for FLS is given in Figure 2. This is the basis for the component stochastic approach in DNVGL CG DNVGL-CG-0129 Sec.5 [3].

Alternatively for FLS, the pressure can be considered with a return period of 25 years, but downscaled to 10^{-2} probability level of exceedance by f_p where the stress range (rather than the stress amplitude) is based on the difference between wave crest and wave trough load phase, rather than a combined case used in component stochastic approach. This is the basis for the EDW approach in DNVGL-RU-SHIP Pt.3 Ch.4 Sec.5 [1.4].

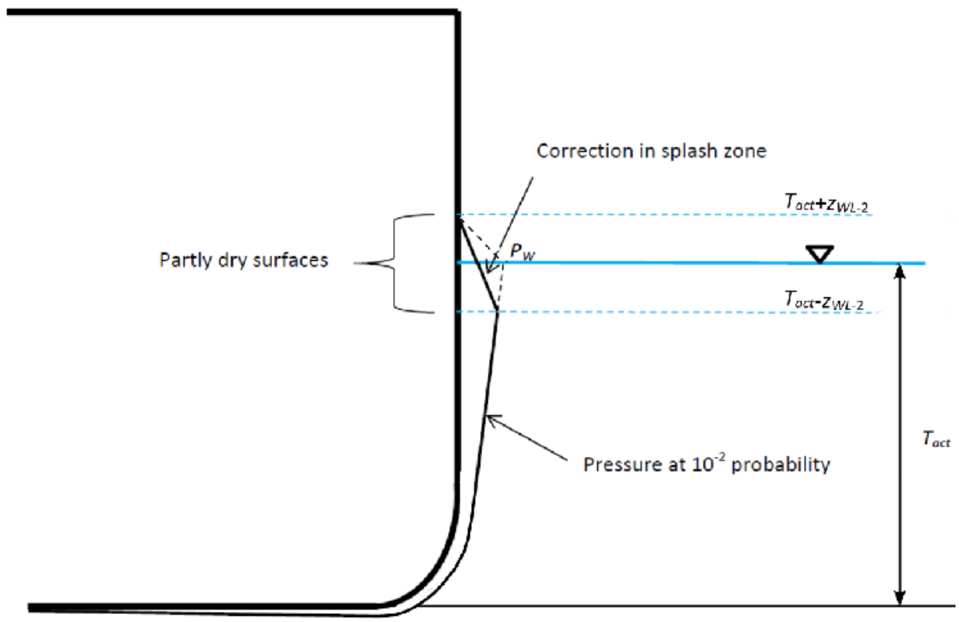

Figure 2 Illustration of splash zone for FLS. Dashed line is the crest pressure without correction for wet and dry zone, while the solid line shows the combined pressure for component stochastic fatigue

For sea pressure towards the ship ends for extreme loading at 25 year return period, due consideration should be made to the non-linear impact loads as slamming on the non-vertical ship sides, stem and bottom.

When calculating the pressure head from green sea on horizontal deck plates and hatch covers, the relative motion between the undisturbed wave surface and ship motion at the centre line for the considered area at a return period of 25 years can be applied as a first approximation. The pressure in kN/m^2 can be taken as the hydrostatic pressure below the water surface.

10 Loading condition

For ULS loading with return period of 25 years, unfavourable loading conditions according to the loading manual should be used. The rules DNVGL-RU SHIP Pt.3 Ch.4 may point to loading conditions that are regarded unfavourable.

For FLS loading at 10^{-2} probability level of exceedance, frequent loading conditions according to the loading manual should be used. The rules DNVGL-RU-SHIP Pt.3 Ch.4 may point to loading conditions which are regarded frequent.

The loading conditions should be included in the loading manual, and may also be defined in DNVGL-RU-SHIP Pt.5 for the different ship types.

11 Combination of loads

RU SHIP Pt.3 Ch.4 Sec.2 defines load combination factors for the different equivalent design waves. These load combination factors can be used if not calculated for a specific ship or available for a similar ship.

SECTION 8 EQUIVALENT DESIGN WAVES

1 Introduction

Equivalent design waves (EDWs) are the basis for the wave loads in DNVGL-RU-SHIP Pt.3 Ch.4 for both FLS and ULS. The EDWs listed in [3] are referred to as rule EDWs.

EDWs are a convenient alternative to statistical based loads, e.g. phase information is better maintained. Secondly, the long term value of the vertical bending moment is dominated by head seas, while the long term value of the side shell pressure is dominated by beam seas. These maximum values should not be used simultaneously, hence, in the statistical analysis correlation factors are needed. For EDWs in head seas, the vertical bending moment should be used with corresponding pressure, and in beam sea from port side, the port side pressure should be used with corresponding vertical bending. The phase information can then be included directly in a consistent way, which is important when many load components are included simultaneously. At the same time, the rule loads must assume a relevant distribution along the length, so that the number of EDW's can be reduced to a number that can be handled in a rule check. For this reason, the rule EDW are not fit for use on global FE analysis. There are then in principle two EDW approaches; one for global FE models and one for local strength assessment.

By applying the EDWs for different responses and headings in the structural analysis, the long term value of the stress at any position can be reproduced with sufficient accuracy.

For each EDW there are two phases corresponding to a crest phase and a trough phase. In DNVGL-RU-SHIP Pt.3 Ch.4 Sec.2 the two load cases are denoted EDW1 and EDW2, and represent two snap shots when the wave is passing the ship.

The intention with this section is to illustrate the basis for the EDW approach, and how the rule EDWs in DNVGL-RU-SHIP Pt.3 Ch.4 can be adjusted based on direct analysis if required.

2 Basic assumptions

The basic assumptions for the long term response of the individual loads are the same as in Sec.7, and include the following:

- scatter diagrams as in Sec.7 [2]
- wave spectrum as in Sec.7 [3]
- wave energy spreading as in Sec.7 [4]
- heading profile as in Sec.7 [5]
- speed as in Sec.7 [6], but speed for local pressure corresponding to speed for hull girder loads, i.e. not for impact loads
- return period as in Sec.7 [7]
- non-linear correction as in Sec.7 [9]
- loading conditions as in Sec.7 [10].

2.1 Weather routing

When the long term target value is calculated for rule corrected EDW, or when the EDW is directly calculated from first principles, the probability level should be consistent. The factors f_R, f_{vib} and f_{fa} should be handled consistently with the rules, see Sec.2 [7].

The EDW for beam sea for ULS loads are associated with a heading factor f_β in DNVGL-RU-SHIP Pt.3 Ch.4, and this factor should be applied. Alternatively, the EDW for beam sea associated with a dead ship with a return period of 1 year can be calculated at a probability level of exceedance of $10^{-6.5}$.

3 EDWs used in the rules

3.1 ULS load cases

The EDWs are denoted with a name where the two first letters represent the heading, while the third letter represents the response parameter and the fourth letter the direction of the incoming wave. The following EDWs with a return period of 25 years are relevant for ULS, see DNVGL-RU-SHIP Pt.3 Ch.4 Sec.2 [2] for further description:

— HSM-1/HSM-2 – Head sea moment
— FSM-1/FSM-2 – Following sea moment
— HSA-1/HSA-2 – Head sea acceleration
— BSP-1P/ BSP-2P –Beam sea pressure from port side
— BSP-1S/ BSP-2S- Beam sea pressure from starboard side
— BSR-1P/ BSR-2P – Beam sea roll from port side
— BSR-1S/ BSR-2S - Beam sea roll from starboard side
— OST-1P/ OST-2P - Oblique sea torsion from stern quartering sea from Port side
— OST-1S/ OST-1S - Oblique sea torsion from stern quartering sea from Starboard side
— OSA-1P/ OSA-2P - Oblique sea acceleration from bow quartering sea from Port side
— OSA-1S/ OSA-2S - Oblique sea acceleration from bow quartering sea from Starboard side

This is eleven EDW pairs. For certain ship types a reduced number of EDWs may be acceptable.

3.2 FLS load cases

The following EDWs with a return period of 25 years (combined with downscaling by f_p) are relevant for FLS:

— HSM-1/ HSM-2 – Head sea moment
— FSM-1/ FSM-2 – Following sea moment
— BSP-1P/ BSP-2P –Beam sea pressure from port side
— BSP-1S/ BSP-2S- Beam sea pressure from starboard side
— BSR-1P/ BSR-2P – Beam sea roll from port side
— BSR-1S/ BSR-2S - Beam sea roll from starboard side
— OST-1P/ OST-2P- Oblique sea torsion from stern quartering sea from port side
— OST-1S/ OST-2S- Oblique sea torsion from stern quartering sea from starboard side.

This is eight EDW pairs covering four response parameters and six headings. For certain ship types a reduced number of EDWs may be acceptable.

The EDWs are basically covered by the ULS EDWs, so when rule EDW loads are corrected there is no need to run specific EDWs for FLS.

In the case in global FE analysis separate EDWs for ULS and FLS are convenient calculating the FLS loads with another scatter diagram and for another probability level.

3.3 Load combination factors

Combination of loads is handled by a consistent set of load components with phase information specific for each EDW load case. The relative value of each load component compare to its long term value for each load case is handled by the load combination factors (LCFs) denoted C_{xx}, where xx refers to the different load components in DNVGL-RU-SHIP Pt.3 Ch.4 Sec.2 [2.2] for ULS and in DNVGL-RU-SHIP Pt.3 Ch.4 Sec.2 [3.2] for FLS. The LCF is 1 or -1 for the load components which correspond to the governing load component (or which is in phase or out of phase compared to governing load component) which the EDW pair is based on. E.g. for HSM-1 the LCF for vertical wave bending moment is -1 corresponding to sagging.

In case of global FE analysis, the load combination factors are not necessary to derive as pressures and accelerations (inertia) are directly transferred.

4 Establishing EDWs from first principles

4.1 ULS

In DNVGL-RU-SHIP Pt.3 Ch.4 the EDWs, listed in [3.1] and [3.2], and their load components and LCF have been expressed by empirical relations, which are derived based on a number of different ships with varying parameters. These load components and LCF may be obtained from directly calculated ship specific EDWs for other EDW responses, where the rule values are not representative. The EDWs can be established in the following way:

— the long term response value LTR_R for the response R should be calculated for a return period of 25 years for ULS according to Sec.4 [3] with basic assumptions in [2]
— the wave heading contributing most to the long term value should be determined based on the CoC method in Sec.4 [6]
— the amplitude, A_R, in m, is estimated as LTR_R/RAO_R where RAO_R is the peak value in the transfer function for the heading determined above
— wave length based on wave period T_R, in s, refers to the wave period giving the corresponding peak value, RAO_R
— the wave steepness, S,should be checked, i.e. the steepness criterion $S<S_{max}$ in Sec.2 [2] should be fulfilled. If not, a slightly longer wave period, T_R, should be chosen with a corresponding RAO_R value, and the check should be repeated until the steepness criterion is fulfilled.

Thereafter the crest and trough should be considered, i.e. when the response is $\pm LTR_R$. The load combination factors (LCFs) can be established for all the load components. The relative phase information in the transfer function for RAO_R and RAOs of the other load components should be determined. This is the basis for defining the LCFs for the other responses, i.e. the LCF accounts for the relative amplitude (compared to its long term value), but also the phase information from the respective RAOs. The non-linear factors in the crest and trough phase should be considered for the specific response after determination of the linear LTR_R. I.e. the nonlinear contribution does not affect the EDW, but the long term values of the individual responses used to find the LCF. This may result in LCF above 1 or less than -1 even for the governing responses.

In case of global FE analysis, the load combination factors are not necessary to derive as pressures and accelerations (inertia) are directly transferred.

4.2 FLS

The EDW for FLS is established in the same way as for ULS in [4.1], but to avoid using f_p and environmental factor f_e, it can be based on a probability level of 10^{-2} and for the relevant scatter diagram and speed (2V/3). The steepness criterion is less relevant, so in certain cases it may imply that the wave length becomes shorter than in the ULS case when the steepness criterion is violated in the ULS case, i.e. longer wave amplitudes may occur for ULS.

5 Establishing corrected rule EDWs

5.1 Wave bending moment

The correction of the wave bending moment consists of calculating a new long term wave bending moment distribution with 25 year return period and replace the wave bending moment in the DNVGL-RU-SHIP Pt.3 Ch.4 Sec.4 [3.1.1] with the maximum wave bending moment amidships. In this context, the operational factor, f_R, should be multiplied with the maximum wave bending moment amidships. This applies to all the hull girder loads, while the distributions of the hull girder loads are maintained from the rules. For shear force and torsion, this is done separately for aft and foreship. The load combination factors are maintained from the rules.

5.2 Pressures

The correction of the pressure consists in calculating the long term pressure at the water line at midship for head sea, following sea and beam sea. For oblique sea the long term pressure is calculated at L/4 from aft end (AE). Nonlinear factors and heading correction factors as given in DNVGL-RU-SHIP Pt.3 Ch.4 should be accounted for when comparing the calculated pressure with the rule value. The pressure distribution along the ship and the load combination factors are maintained from the rules.

5.3 Acceleration

The correction of the acceleration consists in calculating the long term accelerations in the six degrees of freedom at the centre of gravity. The acceleration is replacing the rule values, and they are not scaled. The distribution is maintained from the rules, but the load combination factors for accelerations are recalculated replacing rule values.

6 Load application for different ship types

6.1 General

When required, corrected rule EDWs can be used according to [5] for cargo hold or partial ship models and cross sectional analysis, while load application based on first principles can be applied to global FE models according to approach in [4]. Agreement to which responses parameters (e.g. wave bending moment midship) to include, should be made with the Society.

6.2 Gas carriers

Direct calculations of EDWs are regarded relevant for supports of B-tanks and B-tanks of gas tankers as well as cargo containment system of novel designs, DNVGL-RU-SHIP Pt.5 Ch.7 Ch.20, DNVGL-RU-SHIP Pt.5 Ch.7 Ch.21 and DNVGL-RU-SHIP Pt.5 Ch.7 App.B, respectively. For liquefied gas tankers with independent prismatic tanks of type B, see also DNVGL-CG-0133 and for spherical tank type B, see DNVGL-CG-0134.

For the global FE model, all the response parameters used in the rules, see [3], are relevant based on first principles according to the approach explained in [4].

For the cargo hold model, the rule EDWs are corrected according to [5].

6.3 Ro/Ro and Passenger ships

For category II ships of Ro/Ro ships in DNVGL-RU-SHIP Pt.5 Ch.3 and for passenger ships in DNVGL-RU-SHIP Pt.5 Ch.4 global finite element models are required. Load application procedures in DNVGL-CG-0137 and in DNVGL-CG-0138, respectively, should be followed. The approaches follow the load application from first principles according to [4]. However, for the Ro-Ro vessel, the maximum transverse acceleration at top deck at midship is considered (a simplified approach is also acceptable, but balancing becomes cumbersome), while for passenger ships, the wave bending moment at midship is considered.

SECTION 9 PROCEDURE FOR CSA - COMPONENT STOCHASTIC ANALYSIS

1 Introduction

In component stochastic analysis the phase relations between different load components are included in a more sophisticated way than by the EDW approach or simple combination of loads, and further much more wave headings, wave periods and amplitudes are included. For each load component, the stress per unit load is estimated (stress factor) and combined with the transfer function of the load component, to establish a stress transfer function. Then all the stress transfer functions from all the load components are combined before statistical processing is done. In this way the phase relations are well taken care of at all probability levels.

The method is especially convenient for linear response used for FLS, but it is possible to include non-linear factors as an additional stress factor (or amplification of the transfer function) also for ULS.

The component stochastic approach is applicable when the principal stress directions from different load components are similar. Stress transfer functions for yield stress (nonlinear interaction) cannot be created, but the stress components at a certain probability level may be calculated and finally combined to estimate the yield stress.

The intention is to use the directly calculated wave induced loads in the subsequent strength assessment and maintaining a more accurate combination of the load components. In addition more load components like axial force can be consistently included. A major benefit is also that the importance of individual load components can easily be revealed. A flow diagram of the component stochastic fatigue analysis is included in Figure 1.

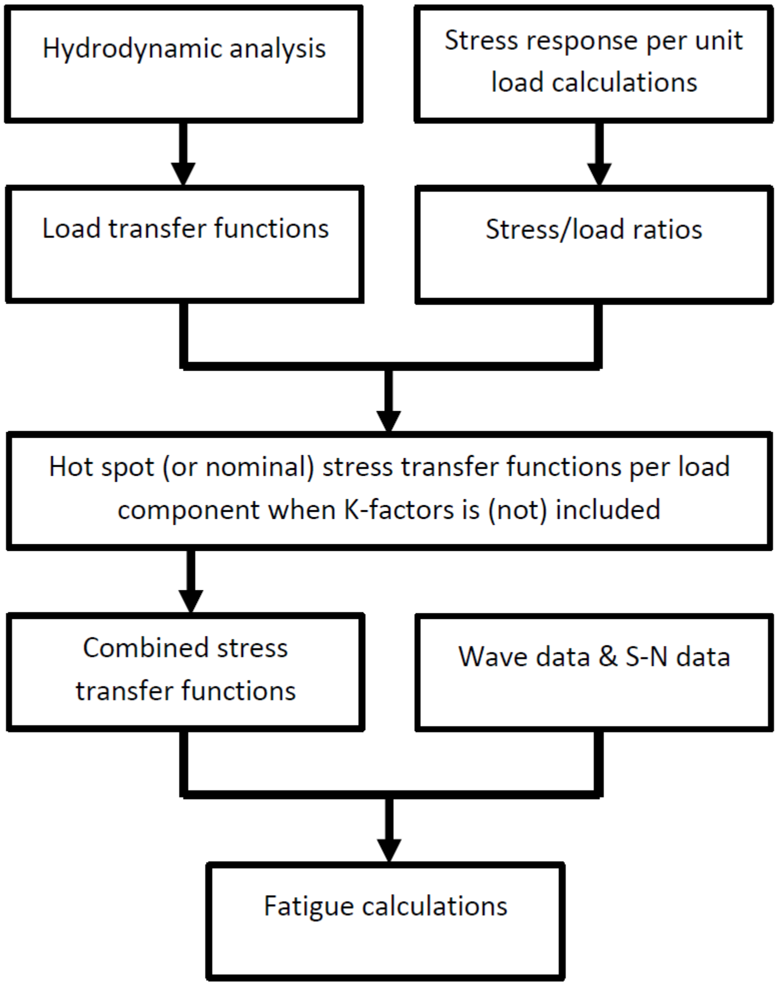

Figure 1 Flow diagram for component stochastic fatigue calculations

2 Basic assumptions

The basic assumptions is as for Sec.8 [2] for the purpose of using this approach as an alternative to the procedure in Sec.7 or Sec.8.

It is convenient with the component stochastic analysis to use a specific scatter diagram or a specific heading profile, i.e. using different input assumptions.

3 Structural model

The structural model can be represented by the same beam models as used for prescriptive FLS analysis as covered by DNVGL-CG-0129 Sec.4. In certain cases also a partial ship (cargo hold) FE model, covered by DNVGL-CG-0127, is needed with applied unit load cases to determine stress factors for relative deflection and double hull bending. These components may also be represented by beam formulations according to DNVGL-CG-0129 App.D.

4 Load components

The load transfer functions may include:

— vertical hull girder bending moment
— horizontal hull girder bending moment
— axial force
— torsional moments
— external (panel) pressures
— internal pressure due to dynamic gravity components from ship motions and due to inertia forces from accelerations.

Load transfer functions for internal cargo and ballast fluid pressure due to accelerations in *x*-, *y*- and *z*-direction can be derived from the ship motions as:

$$H_{p,ax}(\omega|\beta) = \rho \cdot x_s \cdot H_{ax}(\omega|\beta)$$
$$H_{p,ay}(\omega|\beta) = \rho \cdot y_s \cdot H_{ay}(\omega|\beta)$$
$$H_{p,az}(\omega|\beta) = \rho \cdot z_s \cdot H_{az}(\omega|\beta)$$

Where x_s, y_s, z_s is the distance from the centre of free liquid surface to the load point in *x*-, *y*- and *z*-direction defined by the coordinate of the free surface centre minus the coordinate of the load point.

The acceleration transfer functions should be determined in the tank centre of gravity and the accelerations should include the gravity component due to pitch and roll motions. The gravity components are expressed as:

$$H_{p,axg}(\omega|\beta) = -g \cdot \sin(\varphi) \approx -g\varphi$$
$$H_{p,ayg}(\omega|\beta) = g \cdot \sin(\theta) \approx g\theta$$

5 Stress factors per unit load

The following stress component factors may be relevant to determine the combined stress in stiffeners and plating:

A_1 = axial stress per unit vertical hull girder wave bending moment in N/mm^2/kNm
A_2 = axial stress per unit horizontal hull girder wave bending moment in N/mm^2/kNm
A_3 = axial stress per unit global axial wave force in N/mm^2/kN
A_4 = warping stress per unit bi-moment (from torsion) for ships with open deck structure in N/mm^2/kNm^2
A_5 = bending stress per unit local external wave pressure in N/mm^2/kNm^{-2}
A_6 = bending stress per unit local internal wave induced pressure (to be combined with accelerations in *x*-, *y*- and *z*-direction) in N/mm^2/kNm^{-2}
A_7 = axial stress from double hull bending per unit external wave pressure in N/mm^2/kNm^{-2}
A_8 = axial stress from double hull bending per unit internal wave induced pressure (to be combined with accelerations in *x*-, *y*- and *z*-direction) in N/mm^2/kNm^{-2}
A_9 = axial stress from relative deflection per unit external wave pressure in N/mm^2/kNm^{-2}
A_{10} = axial stress from relative deflection per unit internal wave induced pressure (to be combined with accelerations in *x*-, *y*- and *z*-direction) in N/mm^2/kNm^{-2}.

The units given above is based on stresses in N/mm^2 (consistent with S-N curves), hull girder bending moments in kNm, hull girder forces in kN, bi-moment from hull girder torsion in kNm^2 and local pressure in kN/m^2.

The stress factors A_k may be either positive of negative depending on the position in the structure, type of loading and sign convention of sectional loads used in the wave load programme.

The warping stress factor A_4 is related to the bi-moment, which is a result of the torsional moment distribution. To establish the warping stress transfer function, the response from the torsional moment distribution needs to be assessed for each period and wave heading to establish the transfer function of the bi-moment. It is recognised that it is regarded necessary to use a global FE model to derive the warping stress component for component stochastic analysis, and for these cases the component stochastic analysis is regarded cumbersome. This is however a way to reveal the importance of warping stress for longitudinal end connections. Due to this, the warping stress may be neglected which can both produce conservative and non-conservative results depending on location. A combined procedure is therefore most accurate.

Depending on the detail to be investigated, the stress per unit load is either calculated directly by FE analysis or derived from beam formulations.

6 From load to stress transfer function

For each load transfer function, the corresponding stress transfer function is determined as:

$$H_{\sigma,k}(\omega|\beta) = A_k \cdot H_k(\omega|\beta)$$

where:

A_k = stress factor (stress/load ratio) for load component *k*
$H_k(\omega/\beta)$ = load transfer function for load component *k*.

The combined stress response is determined by a linear complex summation of the *n* numbers of stress transfer functions:

$$H_\sigma(\omega|\beta) = \sum_{k=1}^{n} H_{\sigma,k}(\omega|\beta)$$

6.1 Weather routing

The operational factor, f_R, should be applied to the final stress range (or dynamic stress amplitude in case of ULS) while the combination of the vibration factor, f_{vib}, and fatigue coefficient, f_{fa}, should be applied to the wave bending moment transfer function, see also Sec.2 [7].

7 Splash zone correction

7.1 FLS

The splash zone correction for FLS is given in Sec.7 [9.3]. The splash zone extent is based the pressure at the water line and taken at a probability level of exceedance of 10^{-2}. However the extend is multiplied with two to represent an extent at 10^{-4} level. Similar splash zone effect is also relevant for internal fluid tanks.

7.2 ULS

The splash zone correction for ULS is given in Sec.7 [9.3]. The splash zone extent is based the pressure at the water line and taken at a probability level of exceedance of 10^{-8}. Similar splash zone effect is also relevant for internal fluid tanks.

The splash zone effect in the trough phase is a result of combining the static and dynamic pressure and could result in a small negative pressure at the trough surface. In this case the negative pressure may be replaced with zero pressure.

The splash zone effect needs to be accounted for when it significantly affects the stress level of the detail considered. This may apply to other details than those in the side shell, e.g. for hopper tank knuckles.

8 Fatigue damage calculation

When the long term stress range distribution is represented by a Weibull distribution, the fatigue damage for bi-linear S-N curves is given in DNVGL-CG-0129 App.C [2.3]. Alternatively, the fatigue damage can be calculated by a summation of the short term sea states, which are defined by a short term Rayleigh distribution, in DNVGL-CG-0129 App.C [2.5]. This implies that the zeroth moment of the hot spot stress according to Sec.4 [2.1] must be calculated for each sea state. By the short term sea state summation, the contribution from the different sea states can be compared directly in order to identify the most contributing sea states. This can also be done to reveal the most important headings.

9 Extreme loads

The long term distribution of stress can be calculated according to Sec.4 [3] and the extreme values in a short term sea state can be calculated according to Sec.4 [2]. The important sea states can be identified by the CoC method in Sec.4 [6].

SECTION 10 PROCEDURE FOR CSA - FULL STOCHASTIC ANALYSIS

1 Introduction

Full stochastic analysis is the most advanced analysis and can be applied to any kind of structure. Hydrodynamic loads are directly transferred from the wave load analysis program to the FE models, and the directly calculated loads include sea pressures, internal fluid pressures and inertia forces from rigid body accelerations. The loads, phase information and stress directions are preserved through the calculations. The method is suitable for FLS and ULS calculations of details with complex stress pattern and loading. Typical examples are panel knuckles, bracket terminations of the main girder system, top side equipment contributing to the stiffness of the ship, larger openings and hatch corners.

The analysis can be based on a global FE model of the ship. Local FE models can be sub-models, or integrated local FE models in the more course global FE model can be used. Alternatively, a cargo hold or partial ship FE model can be used transferring section loads calculated by the wave load program to the forward and aft end of the cargo hold or partial ship FE model. A flow diagram for full stochastic analysis, as used for fatigue, is illustrated in Figure 1.

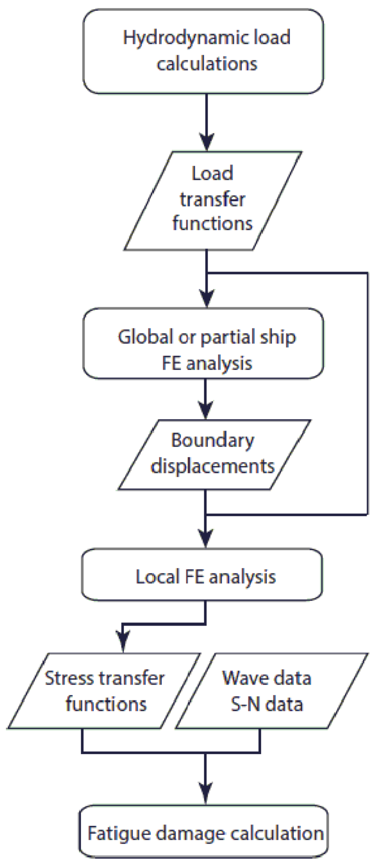

Figure 1 Flow diagram for full stochastic fatigue analysis

The stress transfer functions are established for each heading by analysing the FE model for a sufficient number of wave periods. This is done for all relevant headings, and each combination of wave period and wave heading is a load case in the FE analysis. This may lead to several hundred load cases for each loading condition, which requires efficient FE solvers.

The stress transfer functions from all considered FE elements are statistical processed similar to component stochastic approach in Sec.9. For FLS the stress direction perpendicular or parallel with the weld needs to be defined. Yield stress can first be estimated after stress components are estimated.

The analysis is linear in the frequency domain, but the nonlinear splash zone may be accounted for.

2 Basic assumptions

The basic assumptions is as for Sec.8 [2] for the purpose of using this approach as an alternative to the procedure in Sec.7, Sec.8 or Sec.9.

It is convenient with the full stochastic analysis to use a specific scatter diagram or a specific heading profile.

3 Global FE model

The global FE model is modelled according to DNVGL-CG-0127 .

4 Local FE models

Local FE models for ULS (yield) is given in DNVGL-CG-0127 , while local FE for hot spot models are given in DNVGL-CG-0129 .

5 Loads and load transfer

The load transfer include:

— sea pressure on wet panels
— internal tank pressure from accelerations on wet panels
— inertia forces due to accelerations transferred to mass points
— viscous damping and added mass forces on Morrison elements, e.g. from large bilge keels/appendices.

One load case is created per wave heading and wave period. Several load cases are needed to create a transfer function. Loads needs to be transferred consistently to global and local FE models.

5.1 Weather routing

The operational factor, f_R, the vibration factor, f_{vib}, and the fatigue coefficient, f_{fa}, are multiplied with the final stress range, see also Sec.2 [7]. In case of ULS, only the operational factor, f_R, would be multiplied with the dynamic stress level.

6 Modelling

A prerequisite for correct load transfer from the hydrodynamic program and to avoid unbalance is sufficient compatibility between the hydrodynamic and the global FE model:

— the mass distribution in the hydrodynamic model and structural model needs to be similar
— the wet geometry and element normals in the hydrodynamic and structural model need to be similar, e.g. ensuring similar buoyancy distribution
— the viscous damping forces and added mass loads from Morrison elements used in the hydrodynamic analysis need to be transferred to the structural model if the loads are significant.

Similar mass properties and buoyance can be ensured by using the structural FE model as mass model and panel model in the hydrodynamic analysis.

In order to ensure that the pressure loads are maintained, the local normal of each element need to be the same in the two programs, and the wet surface need to be the similar.

Having performed the load transfer, the final load equilibrium should be checked by comparing longitudinal distribution of sectional forces, e.g. still water bending moment and shear forces in calm sea, and comparing transfer functions in waves for different headings. The determination of the sectional forces in the FE model depends on integration of the stress (strain and Young's modulus) over the cross section in the FE model. Significant unbalanced forces will disturb the stress distribution and global response, e.g. false boundary conditions may easily change the stress level and cause unbalance.

In still water it should be noted that some differences should be expected, since the loading condition described in the loading manual or loading computer may lack minor physical effects compared to the hydrodynamic software, e.g. moments from external or internal end pressure giving rise to additional moments, are normally not included in the loading manual. Therefore, comparison may also be needed against the static loading from the hydrodynamic model at zero speed.

Hydrodynamic modelling is briefly described in Sec.12.

7 FLS and ULS

Calculation of fatigue damage or extreme loading are carried out statistically as for component stochastic analysis in Sec.9 [8] and Sec.9 [9], respectively.

SECTION 11 PROCEDURE FOR EQUIVALENT DESIGN WAVE APPROACH (RSD)

1 Introduction

The equivalent design wave approach relevant for the RSD notation is given in DNVGL-CG-0131 Sec.2 [1]. It includes more equivalent design waves and headings than the EDW approach in Sec.8, and each equivalent design wave is stepped through the full wave period. This results in a large number of load cases, which is reduced to about 20 final load cases by a practical procedure. The wave amplitudes are selected so that the vertical wave bending, horisontal wave bending and torsion are matching the rule level in DNVGL-RU-SHIP Pt.3 Ch.4. The element of direct analysis if focussing on the right phasing and combination of these main load components.

SECTION 12 MODELLING

1 Introduction

Modelling in this section is related to hydrodynamic modelling and load transfer. The modelling is tool dependent, but there are some general features and requirements which should be met. Proper quality in the modelling is necessary to ensure calculated response with the desired accuracy and quality. The results should not be user dependent. The recognised software tools should include a user manual based on best practise procedures, and the user should have sufficient training in understanding the objective and using the software. In the following sub-sections several of the issues are described. See also App.A for further details on modelling and transfer of loads.

2 Documentation

The following should be included in the documentation, which may be required by the Society when direct calculations have been carried out to estimate design loads:

— User manual.
— Theory manual.
— Demonstration of validation of the software if not included in the user manual or theory manual including references to public conference/journal articles.
— Description of assumptions used in the hydrodynamic calculations with reference to e.g. Sec.8 [2].
— Description of mass distribution including properties.
— Description of mesh model being used.
— Demonstration of calm water balance.
— Demonstration of convergence.
— Demonstration of transfer functions for main responses (e.g. motions, sectional loads, pressures and relative motions).
— Comparison between still water bending moment from loading manual and calculations.
— Comparison between still water and vertical wave bending moment from hydrodynamic model and structural model in case of load transfer.

Specific documentation requirements may also be specified in relation to ship specific class notations in DNVGL-RU-SHIP Pt.5 and for additional class notations in DNVGL-RU-SHIP Pt.6 Ch.1 with reference to documentation types for hull and structure (H) in DNVGL-RU-SHIP Pt.1 Ch.3 Sec.3 [2].

3 Mass model

The mass model always needs to be specified, but different formats can be used depending on purpose:

— mass model in the equation of motion for ship motions
— distributed mass model for hull girder sectional loads
— distributed mass model for load transfer.

The mass models will be described briefly in the following, but in any case the mass matrix for the equation of motion needs to be established. Balance and convergence needs to be ensured.

3.1 Mass, COG and radius of gyration

For the purpose of ship motions, it is enough to specify the mass matrix in the equation of motion. The following is needed:

— total mass of the ship, m_T, with location of centre of gravity, COG, in terms of longitudinal, transverse and vertical centre of gravity (LCG, TCG, VCG)

— radius of gyration in roll, k_r
— radius of gyration in pitch, r_{55}
— radius of gyration in yaw, r_{66}.

The total mass, m_T, should be the same as the mass from the displacement ($\rho \cdot \Delta$).

The vertical centre of gravity, VCG, is above the centre of buoyancy, VCB.

The longitudinal centre of gravity, LCG, is close to midships. If it is forward or aft of the flotation centre, LCF, the ship will trim accordingly.

The transverse centre of gravity, TCG, is often at centre line. If not, it will give a steady roll (heel) and the hydrodynamic model cannot be characterised as symmetric about the centre line.

The radius of gyration in roll, k_r, as well as metracentic height, GM, may be given in DNVGL-RU-SHIP Pt.3 Ch.4, RU SHIP Pt.3 Ch.9 Sec.4 or RU SHIP Pt.5. The GM value should be calculated by the software.

The radius of gyration in pitch, r_{55}, and yaw, r_{66}, may be estimated by the longitudinal distribution of the cross sectional mass per meter along the ship, since this is not specified by the rules.

The relation between the mass moment of inertia, I_{xx}, and the radius of gyration, r_{xx} or k_r are estimated from:

$$I_{44} = {k_r}^2 \cdot m_T$$
$$I_{55} = {r_{55}}^2 \cdot m_T$$
$$I_{66} = {r_{66}}^2 \cdot m_T$$

There may be coupling terms between the different DOF, which may need to be included.

3.2 Simplified mass point model

For hull girder sectional loads, the mass distribution must be included. For a given cross section, the integration of the inertia loads from the mass points are necessary to obtain the cross sectional hull girder loads.

The vertical bending moment should be calculated about the horizontal neutral axis. For vertical bending moment and vertical shear force (as well as axial force), the number of mass points in the longitudinal direction needs to be high enough to obtain convergence in the results. 25 mass points evenly distributed in the longitudinal direction is regarded as a minimum. In practise, the mass points along the ship may be divided into two mass points per cross section (or per meter), one on port and starboard side located at VCG and with a transverse displacement corresponding to k_r. The r_{66} becomes inaccurate, but this is not regarded important for the vertical bending moment and vertical shear force.

The torsional moment should be calculated about the base line according to the DNVGL-RU-SHIP Pt.3 Ch.4 (but in principle it should be calculated about the shear centre, which will vary along the ship hull girder). For torsional moments , a better resolution of the mass distribution is regarded necessary, not only to cover the longitudinal distribution, but also the transverse and the vertical distribution to give a representative cross sectional radius of gyration in roll. This mass distribution can be used to establish k_r for roll motion predictions in [3.1]. If the mass distribution is represented by mass points, at least 9 mass points are regarded as a minimum per cross section.

The horizontal bending and horizontal shear force need a mass distribution similar to vertical bending and vertical shear force. It is a link between the horizontal shear force and torsion moment, so to maintain consistency, the mass distribution for the horizontal shear force should be similar as for torsion. As an approximation, the torsion moment may be expressed as the horizontal shear force times the arm from the vertical location where the torsion is minimum.

3.3 FE model

If a global FE model is available, the mass points may be established based on the FE element mesh, i.e. a mass point for every mesh element available defined by e.g. mesh element area, thickness and density. This should correspond to the light ship weight when summarised. In case of deviation due to coarse modelling, the density may be tuned accordingly. The still water moment from for the light weight loading condition should be reproduced for verification. In addition, the mass points representing the cargo weights need to be included. Combined this can result in a detailed mass distribution model. To ensure balance in case of load transfer, it is beneficial to use the mass model in the structural analysis as input to the hydrodynamic analysis.

4 Panel model

The hull surface should be modelled sufficiently accurately to represent the true geometry. The panel mesh depends on the software. For linear theory, the surface up to the still water line should be modelled. For non-linear theory the surface also above the still water line needs to be included. The user manual should provide instructions on the necessary resolution and extent of the panel mesh depending on the analysed conditions (e.g. wave headings, wave periods, forward speed). The diagonal element length should be maximum 1/8 of the smallest wave length, but finer resolution may be necessary in areas with high curvature. An example of a panel model is included in Figure 1.

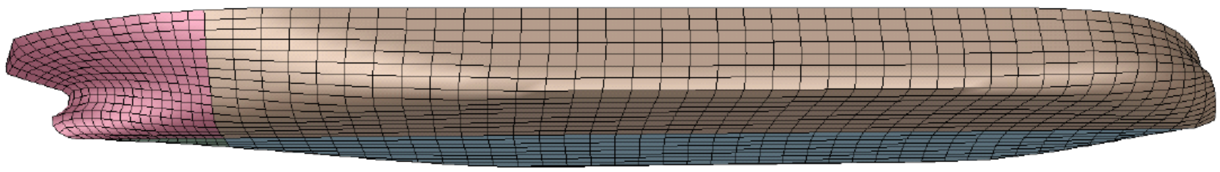

Figure 1 Illustration of panel model up to still water line without any water surface mesh

There are tree options for the mesh:

— mesh independent of subsequent structural analysis
— mesh aligned with structural FE model for load transfer
— mesh taken from structural FE model and used in hydrodynamic analysis for load transfer.

For transfer of sea pressure (or cargo tank pressure) to the structural model, it may be more convenient with a hydrodynamic mesh aligned with the web frames as illustrated in Figure 2 to ensure balance. The direction of the normal of each element in the hydrodynamic mesh compared to the structural mesh need to be aligned as well as consistency in specified wet elements in the structural and hydrodynamic model. This is especially relevant in forward speed. The wet mesh at zero speed up to still water line may not be adequate when the ship at forward speed trim and squat. If the software would require mesh up to the steady wave elevation, it should be noted that the steady wave elevation in forward speed can be significant with reference to [6].

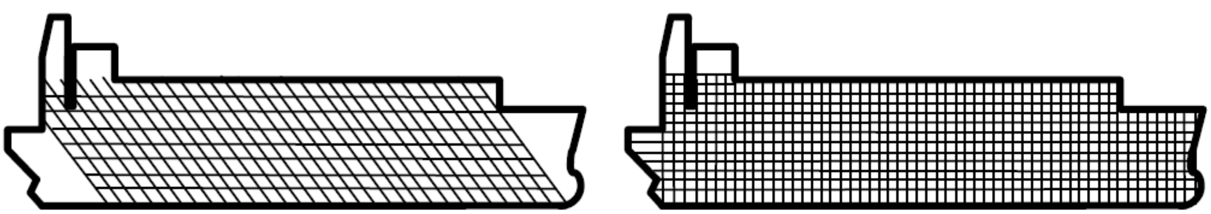

Figure 2 Convenient mesh for transfer to structural model to the right and less convenient mesh to the left

If the water surface needs to be meshed, the extent of the mesh around the ship needs to be considered versus the boundary conditions and far field conditions. Also suitable mesh at walls (channel) and bottom (sea bed) may need to be considered in special cases. The user manual should give instructions for meshing of the water surface.

At high forward speed the mesh may be simplified to avoid instable results at the transom stern, when the water should leave the hull tangentially (Kutta condition).

Balance needs to be ensured. Convergence study may be necessary, unless experience or recommendations are given by the user manual. Also animations can be convenient to justify that numerical problems are avoided.

5 Volume model

The volume model depends on the software. The user manual should suggest the necessary resolution and extent of the element mesh depending on the analysed conditions (e.g. wave headings, wave periods, forward speed).

Volume methods are supposed to cover a large fluid domain and can include the fluid domain both below and above the water surface. The extent of the volume should be considered in order to obtain reliable results. The free surface condition can be non-linear. Special considerations are necessary for the resolution and extent of the elements in the surface region, especially in relation to the non-linear incidents waves which may interact non-linearly and change the incident wave from the intended incident wave.

Balance and convergence need to be ensured.

6 Quality assurance

6.1 General

Quality assurance should be demonstrated and included as part of the documentation in [2].

6.2 Balance

Good balance should be ensured with respect to:

— mass distribution for light ship weight and cargo compared to the loading manual
— panel model versus buoyancy distribution (ship lines) from the loading manual
— total mass and displacement including location of COG which can affect trim and heel at zero speed
— still water shear force and vertical bending moment distribution between hydrodynamic program and loading manual at zero speed (and forward speed)
— mass and buoyancy at forward speed with trim and squat with reference to [6.2]
— zero still water and wave shear force and bending moment when integrating over the ship length
— transfer of loads from hydrodynamic to structural model also with respect to same properties above, but where the properties from the structural model is compared to the hydrodynamic analysis rather than the loading manual.

Balance is of particular importance in relation to non-linear damping from bilge keels. Large viscous damping forces, e.g. Morison forces, should be transferred to the FE model.

An illustration of comparison in still water shear force and bending moment is given in Figure 3 and Figure 4. In this case the software Cutres integrates the stresses from the FE model, while Wasim is the hydrodynamic tool which calculates the still water loads.

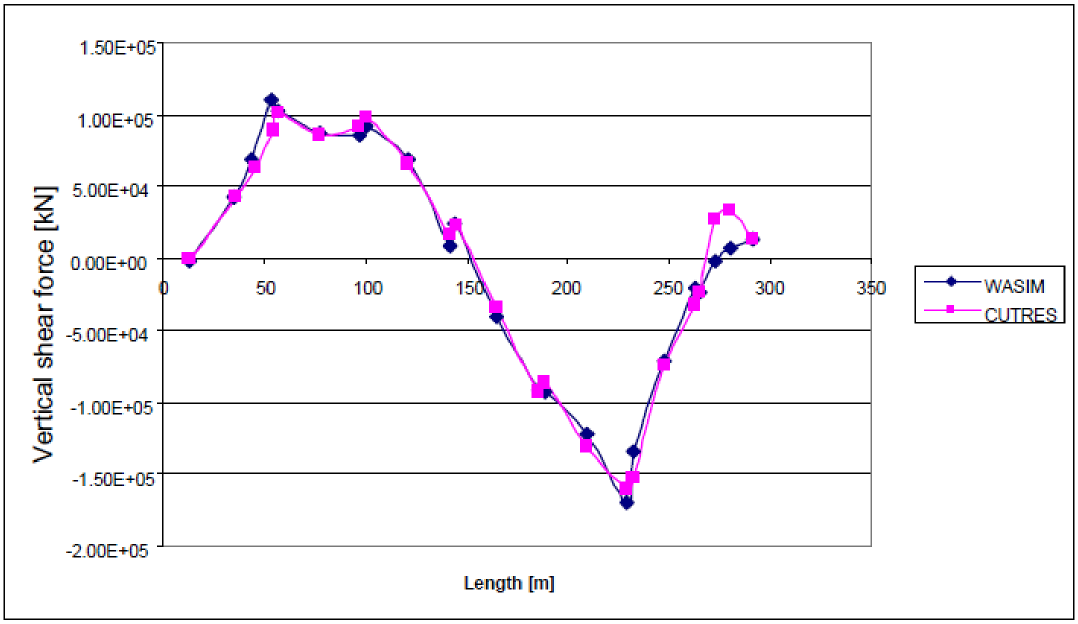

Figure 3 Example of QA for section loads – Vertical Shear Force

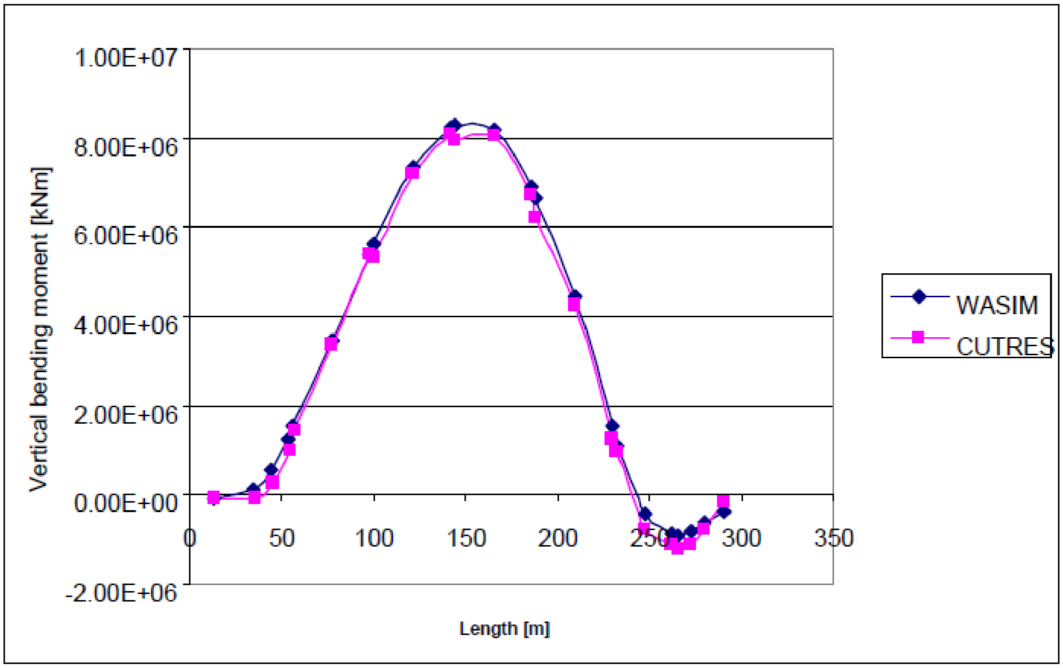

Figure 4 Example of QA for sectional loads – Vertical Bending Moment

6.3 Calm water balance

The ship may operate in forward speed, which can cause trim and squat of the ship due to the steady pressure and surface elevation around the hull. The steady surface elevation at the stem, ξ_s, for a blunt ship without bulb may be estimated as:

$$\xi_s = \frac{U^2}{2 \cdot g}$$

where U is the forward speed in m/s and g is the acceleration of gravity (9.81 m/s^2).

In calm sea at forward speed, balance should be ensured before assessing the wave loads. The wave elevation around the hull will set up a steady sagging moment, affecting the still water bending moment. The still water bending moment should not match exact what comes from the loading computer, which neglects this speed effect and neglects the bending moment contribution from the sea pressure at the ship's ends.

6.4 Convergence

Convergence implies that improving the resolution of the mesh, mass distribution and time step, the results converge towards a constant value. Further improvement in the resolution does not lead to improved accuracy of the results. Convergence should always be ensured.

6.5 Time step

Small mesh and wave periods as well as theory and solution methods may give requirements to the maximum time steps. The user manual should give instructions on time steps and if not demonstration of convergence may be needed.

6.6 Selection of wave frequencies and headings

For proper hydrodynamic assessments, it is necessary to provide RAO calculations for a suitable set of wave frequencies and wave headings. The spacing and number of frequencies and wave headings should be sufficient in order to:

— capture all relevant resonance peaks for all RAOs. The width and height of each peak should be well represented by linear interpolation between calculated points (see also Figure 1 in Sec.3 [5.2])
— the range of frequencies should also reflect the expected incident wave energy content which may affect the results
— a proper discretization is typical identified if the response spectrum is converged in terms of moments.

Depending on the software, the user manual may provide instructions on the spacing and number of the wave frequencies. In case of time domain solvers, the user manual may also give instruction on the initial part of the start up period which should be disregarded and how repeated signals in irregular sea states should be avoided.

The ship typically acts as a low pass filter. As a consequence, short waves relative to the ship length may be ignored for most RAOs.

SECTION 13 REFERENCES

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APPENDIX A TRANSFER OF LOADS TO STRUCTURAL MODEL

1 General

Structural FE analysis procedures may be based on loads from direct hydrodynamic analysis. This will require transfer of loads from the hydrodynamic calculations to the structural FE model.

1.1 External and internal loads

The principal results of the hydrodynamic calculation are:

— external loads, i.e. sea pressures onto the wet hull surface
— motions, i.e. six rigid DOF.

In addition, post-processing can provide secondary results, e.g. sectional loads. Those are not needed for the load transfer, but may be useed to confirm correct transfer of loads.

Additional external DOFs, which affect the fluid-structure domain interface may include:

— additional rigid body modes for multi-body-systems, where each additional body component induces six additional DOFs
— elastic deformation modes, when hydrodynamic interaction with structural deformation needs to be accounted for with reference to DNVGL-CG-0153.

Additional internal DOFs affect internalforces, e.g. inertia, inside the structure and the dynamics of the ship. But they do not directly affect the fluid-structure domain interface. They may include:

— free surface deformations in semi-filled tanks
— freely swinging cargo items lifted by a crane
— passive roll dampers
— active motion controls (autopilot, active roll damping).

The external and internal loads are typically handled quite differently in hydrodynamic software. The different load components will require different mapping techniques and result in different FE load components.

The spatial distribution of pressures is readily available at fluid mesh locations. Those pressure loads will be transferred to geometrically related locations in the FE model.

Within the hydrodynamic calculation, the external loads are only modelled by integrated items corresponding to the motion DOFs, e.g. the acceleration-induced loads are typically represented by a rigid body 6 × 6 mass matrix, $\boldsymbol{M}$, during the hydrodynamic calculation. On the other hand, the structural analysis needs the acceleration-induced loads spatially distributed at the proper location of the FE mesh. Therefore, those distributed internal loads need to be computed from the motions in a post-processing step, taking the modal accelerations and local mass distribution into account.

1.2 Consistently balancing loads, handling residual forces

The hydrodynamic software calculates external loads which shoud be in balance with the corresponding internal loads. It is important e to conserve this balance when transferring the loads. Otherwise, the resulting FE loads will include residual forces which may cause inaccurate deformation and stress results.

If external and internal loads are transferred independently, independent errors will yield unbalanced FE loads. There are several possible sources causing such unbalance:

— input and software bugs
— modelling errors, e.g. applying loads to FE nodes whose DOFs have been fixed
— different discretization of fluid and structural meshes
— limitation in supported modelling assumptions, e.g. if program expects symmetrical mass distribution while mass distribution is not fully symmetric

— post-processing of load components (e.g. splash zone correction to redistribute pressures from linear hydrodynamics around the still water line might cause additional forces)
— ad-hoc measures to compensate for limitations in physical models (e.g. adding roll damping to equations of motion without transferring related additional damping forces properly).

Ideally, each source of inconsistency should be addressed individually in a physically sound manner. But frequently, this is not a convenient option. The remaining residual forces then need to be compensated without disturbing the structural strength assessment in a significant manner. Without sound compensation, the residual forces would be supported by the supporting spring elements. This would cause inaccurate deformations concentrated around the supporting springs.

A generic approach is to compute an additional acceleration which yields a total reaction force compensating the residual forces. The corresponding (nodal) reaction forces from the mass model are then added to the FE loads, resulting in a finally balanced model.

In general, this will not yield the most physically sound compensation of residual forces. But if the necessary compensating acceleration is small with respect to the dynamic and gravitational acceleration, the possible impact on strength assessment can be easily estimated.

2 Transfer of external loads

2.1 Load transfer by pressure mapping

Hydrodynamic software can compute pressures at the fluid-structure interface. Many FE codes support pressure loads, i.e. loads in terms of pressures acting at element faces. An intuitive method of transferring loads consists of transferring the pressure results at the fluid-structure interface to the corresponding finite elements. The FE code will then handle conversion of pressures to forces.

The structural response to external forces does not depend on the elements orientations. Thus, from the structural analysis point of view, the finite elements at the ship's outer shell may be oriented arbitrarily. The element orientation only affects the output coordinate system for the element results. The orientation of the elements might have even been tuned for obtaining the most convenient output coordinate system.

In contrast, for transforming pressures into forces, the orientation of the pressure-loaded elements is important because orientation determines the direction of the resulting pressure-induced forces. All elements forming a fluid-structure domain boundary need therefore to be oriented consistently. FE models received from a third party need to be checked for this property. If necessary, the orientation of some elements needs to be flipped. The flipping might break an intentional choice of orientation as present in the original FE model. Therefore, additional FE post-processing steps might be required to compensate for that.

Mapping pressure from a fluid element to a structural element is obvious as long as fluid and structural meshes coincide at the fluid-structure interface. But mesh requirements for fluid and structure models may differ. Therefore, sound meshes usually won't coincide. Besides partially overlapping fluid and structure elements, fluid and structure element faces at curved areas are not exactly parallel. This might result in original forces different from the transferred forces. Internal forces which balance the original external forces (as acting in the hydrodynamic model) no longer balance the transferred forces (as acting in the structure model). Even though the FE mesh may be used in the hydrodynamic analysis, forward speed of the ship can trim and squat the ship so that the hydrodynamic mesh may be slightly changed.

As long as both models approximate the same fluid-structure boundary surface, the residual forces will be sufficiently small. Their order of magnitude will be governed by the discretization error. When compensated by standard methods, their influence on structural strength assessment will be negligible.

Remark: the latter holds for standard one-way coupling. More attention should be paid to the problem when simulating strongly coupled fluid-structure-interaction problems. The presence of residual forces might break physical conservation properties (e.g. energy or momentum), assumed by the physical models, and cause numerical instabilities.

2.2 Load transfer by force mapping

Instead of transferring pressures from the hydrodynamic to the FE model, the forces originating from each fluid panel at the fluid-structure domain boundary can be transferred directly to forces at adjacent FE nodes. The total forces and moments contributed by each panel shall be distributed to the FE nodes such that total forces and moments acting at the FE model are equal to the total forces and moments caused by the panel within the hydrodynamic model. As long as that principle is fulfilled, the resulting total forces and moments will be conserved by the mapping process, automatically. There will be no residual forces, except for the very small contribution caused by numerical round-off errors.

For conservation of total force and moment, the choice of FE nodes for distributing the panel forces is unimportant. However, for obtaining a good local spatial distribution of FE forces corresponding to the local pressure field, the distribution of pressure-induced forces needs to be governed by the local geometry. The following method will yield a sufficiently locally realistic force distribution:

— the boundary of the fluid panel and the boundaries of overlapping FE faces (corresponding to the fluid-structure domain boundary) are projected onto a plane that is orthogonal to the fluid panel's surface normal
— for each projected FE area, the intersection with the fluid panel area is computed. The centre of gravity is computed for each intersection area
— the pressure force from the panel is distributed among the centres of gravity of the intersection areas according to their share in total intersection area
— each share of force is distributed to the nodes of the corresponding finite element in a manner that conserves its total force and moment.

The direct force mapping depends on the location of the FE boundaries, only. The orientation does not matter and needs therefore no special attention.

There might be some partial fluid panel areas that don't intersect with any FE. In practice, that might occur in regions where sharp edges of fluid and structural models do not strictly coincide. At locations, where fluid and structural meshes deviate, there is no longer a geometrically motivated choice for local pressure force transfer. It is, however, still possible to maintain conservation of total force and moment. Forces contributed by above partial areas should be distributed artificially at nearby nodes in a manner conserving their total force and moment.

3 Applying internal loads

3.1 Acceleration-induced loads

Accelerations (dynamic as well as gravity) are inducing the internal loads. For many applications, they may be applied directly onto mass points as inertia loads, but may also be applied to determine internal fluid pressure.

During the hydrodynamic calculation, the dynamic properties of the ship are usually modelled by aggregated values, e.g. a mass matrix covering the relevant DOFs. The hydrodynamic code does not fully track the local internal loads. An additional post-processing step is needed to recover locally distributed FE loads from the motion results. Final consistency of loads should be ensured.

The following relates to linear constraints. The rigid body constraint is not linear, but can easily be linearized for small displacements. For small angular displacements, angular velocities will also be small and additional effects like centripetal acceleration are also negligible.

The structural FE model yields a mass matrix $\boldsymbol{M}$, which represents the internal loads resulting from accelerations. It is given in terms of the FE nodal DOF. The mass matrix does not solely consist of the FE model. It needs to account for other loads which contribute to the internal loads (but do not contribute to the stiffness). Typical additional load items are:

— additional equipment
— fluid pressure in tanks

— other types of cargo, e.g. containers, dry bulk cargo, steel coils, vehicles.

If there are N_{fem} nodal DOF, $\boldsymbol{M}$ is an $N_{fem} \times N_{fem}$ matrix which relates the vector of N_{fem} nodal accelerations, $\boldsymbol{a}$, to the vector of N_{fem} nodal force, $\boldsymbol{f}$, components as:

$$\boldsymbol{f} = \boldsymbol{M} \cdot \boldsymbol{a}$$

M_{ij} is the force induced at nodal DOF i by a unit acceleration of nodal DOF j.

The hydrodynamic model usually constrains those DOFs to a small number N_{mod} of modes. Let $\boldsymbol{q}$ be the vector of N_{mod} modal displacements. The vector $\boldsymbol{r}$ of N_{fem} nodal displacements computes as a multiplication by an $N_{fem} \times N_{mod}$ matrix $\boldsymbol{T}$:

$$\boldsymbol{r} = \boldsymbol{T} \cdot \boldsymbol{q}$$

T_{jm} is the displacement w.r.t. FE-DOF j caused by a unit displacement w.r.t. hydrodynamic model mode m.

Using $\boldsymbol{a}(t) = \ddot{\boldsymbol{r}}(t)$, where the two dots represents the second time derivative, the nodal forces induced by a modal acceleration $\ddot{\boldsymbol{q}}(t)$ can be computed as:

$$\boldsymbol{f} = \boldsymbol{M} \cdot \boldsymbol{T} \cdot \ddot{\boldsymbol{q}}$$

There is an analogous $N_{hyd} \times N_{mod}$ matrix $\boldsymbol{T'}$ which maps displacements of the modal DOFs, $\boldsymbol{q}$, to the N_{hyd} displacement components at the panel centres of the fluid-structure-boundary:

$$\boldsymbol{r}' = \boldsymbol{T}' \cdot \boldsymbol{q}$$

The local forces acting at the FE or hydrodynamic model positions can both be accumulated into generalized forces, $\boldsymbol{F}$, w.r.t. the common modal DOFs:

$$\boldsymbol{F} = \boldsymbol{T}^T \cdot \boldsymbol{f}$$

$$\boldsymbol{F}' = \boldsymbol{T}'^T \cdot \boldsymbol{f}'$$

Where the superscript T refers to the transformed matrix.

The hydrodynamic software will compute a combination of modal motions $\boldsymbol{q}(t)$ and corresponding pressure forces $\boldsymbol{f'}(t)$ such that resulting generalized forces $\boldsymbol{F'}(t)$ balance against the inertia forces as determined by a given modal mass matrix $\boldsymbol{M'}$:

$$\boldsymbol{F}'(\boldsymbol{t}) = \boldsymbol{T}'^T \cdot \boldsymbol{f}'(\boldsymbol{t}) = \boldsymbol{M}' \cdot \frac{d^2\boldsymbol{q}(t)}{dt^2}$$

When the FE model is exposed to the same modal acceleration, the same generalized force should result:

$$\boldsymbol{F}(\boldsymbol{t}) = \boldsymbol{T}^T \cdot \boldsymbol{f}(\boldsymbol{t}) = \boldsymbol{T}^{\boldsymbol{T}} \cdot \boldsymbol{M} \cdot \boldsymbol{T} \cdot \frac{d^2\boldsymbol{q}(t)}{dt^2}$$

That will be fulfilled if the modal mass matrix applied in the hydrodynamic calculation is chosen as:

$$\boldsymbol{M}' = \boldsymbol{T}^T \cdot \boldsymbol{M} \cdot \boldsymbol{T}$$

3.2 Fully hydrodynamic tank load modelling

Load from liquids in tanks can be computed by a hydrodynamic calculation applying the same methods as used for computing loads outside the ship. When the hydrodynamic code can handle such coupled ship-tanks systems directly, the tank loads can be interpreted as additional external loads. There is basically just a geometric difference: the fluid domain and the fluid-structure boundary interface decomposes into several isolated components (one additional fluid domain and fluid-structure-boundary component for each tank). Pressure-induced tank loads are transferred by means of the same methods as pressure loads acting at the outer shell. The mass matrix for computing the remaining internal loads may not include the liquids inside those tanks, only the remaining structure.

3.3 Quasi-Static load transfer

When consistent model and load transfer is used, and when all internal loads are caused by accelerations, the accelerations, for application of the internal loads, do not need to be transferred explicitly. It is sufficient to transfer the external loads, resulting in FE nodal forces $\boldsymbol{f}$, and to compute linear combination factors, $\boldsymbol{a}$, for internal loads, $\boldsymbol{a}^T$, which balance against the external loads. This is done by solving the system of linear equations:

$$\boldsymbol{M}' \cdot \boldsymbol{a} = \boldsymbol{T}^T \cdot \boldsymbol{f}$$

The final result will be the same as with explicitly transferred acceleration loads. The final result would even be the same if some remaining residual forces were compensated by additional acceleration loads consistent with the structural mass matrix $\boldsymbol{M}$.

3.4 Static loads (weight)

When equivalence of gravitational and inertial mass is recognized, weight loads are physically indistinguishable from loads caused by uniform acceleration. They can therefore be computed by the same means as dynamic acceleration loads, involving an acceleration vector which is the sum of gravitational and dynamic acceleration. There is noneed to separate those load components.

Linear hydrodynamics assumes a stationary reference state and applies linear equations of motions to the deviation from that reference state. The hydrostatic equilibrium, i.e. still-water floating state, is frequently used as the reference state. When computing the hydrostatic equilibrium, geometrical non-linearities are easy to track and might relate to a different mass model than used for the dynamic model. When different mass models are applied, loads from stationary and dynamic loads should be transferred separately, each taking its proper mass model consistently into account. The stationary and dynamic loads are finally combined linearly.

3.5 Additionally applied loads

Modelling certain load effects may be straightforward within the hydrodynamic model, but not within the structural model. E.g., for an anchored ship, the dynamics may be significantly affected by non-linear interactions of flow, ship and anchor cable. The mapping techniques described so far would require modelling the anchor cable non-linearly as a part of the structure. However, for structural assessment of the ship, a linear elastic model might be sufficient, and the structural response of the cable itself is unimportant.

An alternative method consists of modelling the anchor cable within the hydrodynamic simulation only. The hydrodynamic code needs to compute the force at the upper end of the cable. This force is finally transferred to the FE model and applied at an attachment point, in addition to the other hydrodynamic loads.

The same technique may be used when artificial roll damping is added to the equations of motion. The related damping forces are likewise only represented by the hydrodynamic model. They are not implicitly represented by the pressure-loaded structural model and may need to be transferred in addition to the pressure forces.

Viscous damping forces can be important for some ships, particularly those ships where roll resonance is in an area with substantial wave energy, i.e. roll resonance periods of 6 to 15 seconds. The roll damping may, depending on wave environment, be neglected when the roll resonance period is above 20 to 25 seconds. If torsion is an important load component for the ship, the effect of neglecting the viscous damping force should be considered.

4 Idealisation of internal loads

4.1 Physical consistency of mass models

There are several ship-specific kinds of load items, e.g. liquid tank contents, container stacks or dry bulk cargo. The physics of their internal force transfer differs. Therefore, different model assumptions are typically required to represent them in numerical simulations.

Strength assessment procedures by prescriptive rules frequently contain some formulas evaluating acceleration-induced loads. Caution is advised when such load models are used within a first principle analysis process. Those load models frequently only aim at providing some input for corresponding prescriptive strength assessment criteria. They do not necessarily model the dynamic effects physically correct.

E.g., container or dry bulk cargo that is rigidly coupled to ship should result in total loads equivalent to a point mass in the load item's centre of gravity and some additional moments of inertia. If a load model is exposed to translational accelerations, the turning moment w.r.t. the centre of gravity should be zero. This is frequently not automatically realized by prescriptive load formulas. If the hydrodynamic simulation assumes the load item's mass in its centre of gravity, inconsistent load assumptions will be present when transferring the loads to a structural FE model. If the inertia matrix used during the hydrodynamic simulation is computed from the load distribution formula, that inertia matrix might not necessarily realize a rigid body. It might even result in physically unsound mass models, e.g. none-symmetric mass matrices.

4.2 Potential theory

If tank loads are directly computed by hydrodynamic calculations, the presence of an internal fluid domain introduces additional DOFs into the system. If a potential flow is assumed inside the tanks, the flow is fully determined by the motion of the fluid boundary. The resulting loads onto the tank walls can then be represented by a hydrodynamic mass matrix. The structural inertia matrix ***M'***, as used by the hydrodynamic calculation, is computed as in [3.1]. It will provide for a physically consistent model of the interaction effects between ship motions and flow inside tanks.

For entirely filled tanks, the fluid boundary is determined by the (rigid body) displacements of the structure and needs no special treatment when solving the equations of motion. The physical effects are fully represented by the mass matrix. When tanks are partially filled, the presence of a free surface induces additional internal DOFs that are independent of the structural displacement. The hydrodynamic code needs to include them explicitly in the equations of motion.

4.3 Simplified fluid-wall interaction

For certain FE analyses, acceleration-induced loads from tank contents or sea water can be modelled by some very simplified assumptions.

A commonly used approach is to distribute the mass of the tank content as nodal masses to the FE-nodes located at the tank walls. Such a model will be able to cause physically unrealistic forces parallel to the tank walls. The rotational inertia of the tank content will be significantly overestimated by such models.

An improved method consists of applying dyadic nodal mass matrices at the tank wall's FE nodes. Those transfer forces normal to the wall surface, only. If $\boldsymbol{n}$ is the unit normal vector and m is the desired mass value, the nodal dyadic mass matrix is $m \cdot n \cdot n^T$.

Such mass models can be used to replace the full hydrodynamic mass matrix (that couples all nodes of the fluid-structure interface) by a lumped mass matrix. This might be useful for vibration analysis because the sparsity pattern of the structural stiffness matrix is retained in the generalized eigenvalue problem.

4.4 Constraint free surface modes

In principle, the presence of a free surface introduces additional DOFs, into the equations of motion, which need to be tracked explicitly. However, the deformation of the free surface might be constrained in a manner entirely dependent on the ship’s rigid body motions. Thus, the additional DOFs are no longer independent variables and can be eliminated. The equations of motions are tracked as usual. The effect of the constrained free surface deformations is tracked implicitly by the resulting mass and restoring matrices.

4.5 Simplified models and consistency of equations of motions

The acceleration-induced loads from simplified models may no longer model the physics of a rigid body. The resulting mass matrix may no longer correspond to the mass matrix of a rigid body. If the hydrodynamic code only allows for inputting rigid body mass matrices (e.g., given as mass, centre of gravity and radii of gyration), the correct mass matrices cannot be applied when solving the equations of motions. Residual forces might remain after transferring the hydrodynamic loads back to the structural model.

Properly handling non-rigid-body mass matrices is not a principle problem. Linear sea-keeping analysis involves the effects of a hydrodynamic mass matrix which does not correspond to an equivalent rigid body, either.

CHANGES – HISTORIC

There are currently no historical changes for this document.

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