DNV·GL

CLASS GUIDELINE

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Fatigue and ultimate strength assessment of container ships including whipping and springing

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.

CONTENTS

SECTION 1 GENERAL

1 Introduction

Next to the quasi-static wave loads, the ship's hull is subjected to wave-induced vibrations. The stiffness of the ship structure and the ship mass distribution can be regarded as a simple mass-spring system, which is associated with a natural frequency. Exposed to wave loads, the hull girder may vibrate with its natural frequencies. The vibration may be excited by nonlinear impulsive wave loads such as bow flare-, bottom- or stern slamming, which leads to sudden vibrations. This is referred to as *whipping*. The vibration may also be excited by oscillating wave loads, which may lead to resonance vibrations. This is referred to as *springing*. Springing can be caused by both linear and nonlinear excitation, where the encounter frequency or the sum of two encounter frequencies coincides with a natural frequency of the hull girder.

The damping is typically low, so after a whipping event the vibration decays slowly and may last for many seconds and even minutes. Low damping may also cause significant vibration levels even though the resonance excitation is low. Whipping and springing may therefore occur more or less continuously and simultaneously and can therefore be difficult to distinguish. Their relative importance may also depend on design (flexibility and shape), loading condition (low or high draft and trim) and wave condition (sea state in conjunction with ship speed and heading). The two phenomena can be referred to as wave-induced vibrations. The associated loads are often referred to as high frequency loads, which can be compared to the conventional wave loads, which are referred to as wave frequency loads.

Ships have many vibration modes and corresponding natural frequencies. The governing vibration mode is the vertical 2-node vibration mode, which is associated with the lowest natural vibration frequency in most cases. The natural frequency may be in the order of 0.4 - 1 Hz, while the wave frequency loads are an order of magnitude less (0.05 - 0.2 Hz). The 2-node vibration mode is most easily excited and gives the largest vibration bending moment amidships. The vibration mode and the corresponding distribution of the vertical vibration bending moment are illustrated in Figure 1 for a homogeneous ship with a normalized scale. It resembles the envelope curve of vertical wave bending moment. Other modes may also contribute, in particular, for very large vessels. However, based on full-scale measurements, these are considered as less relevant.

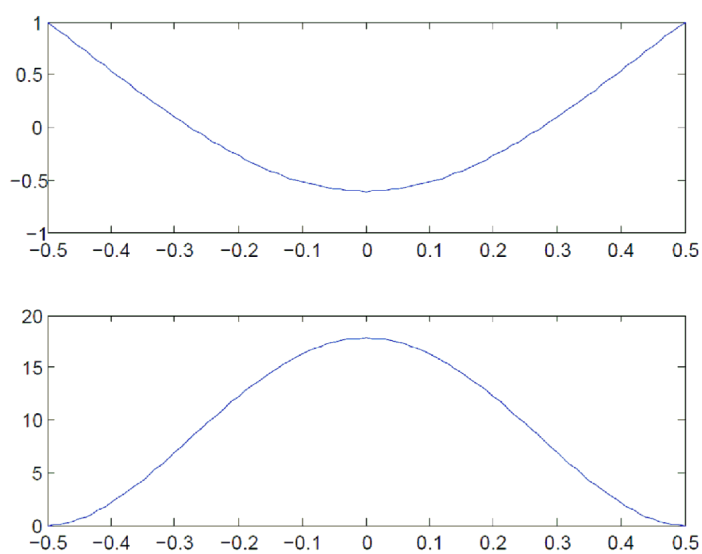

Figure 1 Vertical 2-node vertical vibration mode (top) and the associated vertical bending moment distribution (bottom)

Figure 2 presents for a Panamax containership typical sample time series for a whipping and a springing event, with stresses obtained from strain measurements amidships below the upper deck. Shown are the unfiltered signals together with their high- and low-frequency parts.

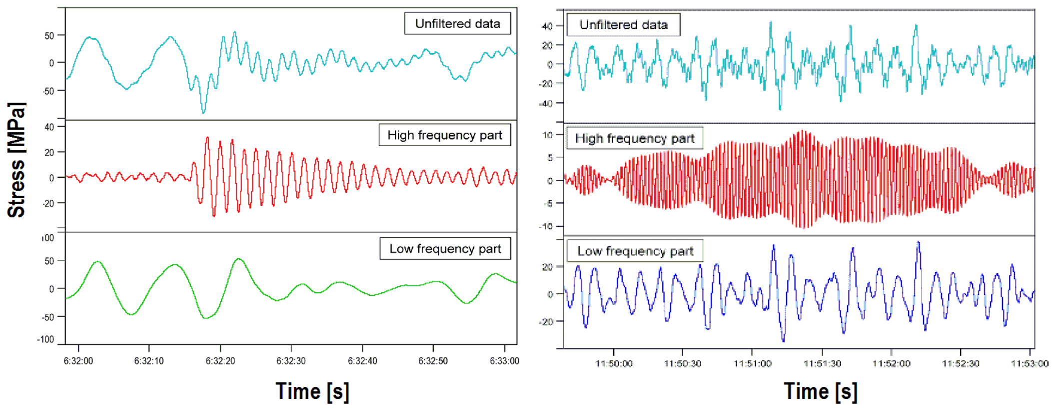

Figure 2 Typical time series of stresses in the upper deck of a containership caused by whipping (left) and springing (right)

Both, whipping and springing increase the fatigue loading, while only the whipping is considered to increase the extreme loading significantly. While the extreme loading and ultimate hull girder capacity are related to safety, the wave-induced vibrations also contribute to fatigue loading, which is next to safety mainly a concern related to maintenance and repair costs. Whipping and springing are thereby also related to economy. Both, fatigue and ultimate strength are addressed herein.

Regarding extreme loading, the whipping is superimposed on the static loading (still water loading) and the wave frequency loading. These three load components make up the total loading, which should be less than the ultimate capacity (collapse strength) of the hull girder.

Regarding fatigue loading, the springing and whipping are both superimposed on the wave frequency loading. The vibration and wave frequency response are related to two different frequency regimes, and they are widely spread on the frequency scale. For such broad banded processes, the fatigue cycles are counted by Rainflow counting, which is the recognized approach to establish the fatigue loading history. For the total stress history (wave frequency stress + high frequency stress) in a ship structure, the fatigue damage can be calculated, referred to as the *total damage*. Also for the wave frequency stress, the fatigue damage can be determined, referred to as the *wave damage*. The difference between the total and wave damage makes up the *vibration damage*. In practice it is the vibration on top of the wave frequency loading that makes up the significant part of the vibration damage. From full-scale measurements the vibration damage was found to be of comparable magnitude as the wave damage, but the relative magnitude depends on ship type, size and trade.

The basis for the herein given empirical relations has been full-scale measurements and model tests. Load histories from full-scale measurements comprise all kinds of hull girder vibrations and account for real life experience. In model tests realistic assumptions have been made in testing and evaluation of the results to avoid too conservative estimates of the effect of whipping and springing. Furthermore, damage experience from the fleet of container ships, classed by DNV GL, has been considered.

2 Application

This guideline provides a procedure to estimate the safety margin against hull girder collapse. It includes an estimate of the effect of whipping based on empirical relations, which can be used in early design to get a good first estimate of the scantlings. The procedure may also be used as basis to provide an estimate of the safety margin, which subsequently can be used in assessment of operational measures to improve the safety margin in real operation. It may for instance be used as input to hull monitoring systems, which includes measurements of static and dynamic loads including whipping.

Furthermore, this guideline provides a procedure to estimate the effect of the wave-induced vibration for the fatigue strength assessment in early design based on empirical relations. The standard procedures outlined in class guideline DNVGL-CG-0129, *Fatigue assessment of ship structures*, may be employed for fatigue strength assessment including the effect of the wave-induced vibration.

The empirical relations are restricted to whipping for extreme loading and whipping and springing for fatigue loading. In both cases the effect of the vertical 2-node vibration mode and the corresponding bending moment is superimposed on the vertical wave bending moment. Assessment of other modes requires special considerations, but can in most cases be neglected. Only the vertical bending moment with whipping in sagging and hogging should be considered and local dynamic effects can be disregarded.

Assessment of the safety margin against collapse should cover the main part of the hull girder and should be carried out at locations, selected with respect to change in scantlings, capacity and loads. In particular, locations with large changes in the longitudinal bending stiffness, e.g. aft and forward of the superstructure of a twin island design, or locations with change of the framing system, e.g. aft of the engine room bulkhead, should be considered.

The assessment of the fatigue damage with springing and whipping is restricted to the midship region and to longitudinal structural members and hatch corners. In most cases it is sufficient to consider the midship region considering the directly calculated envelope curve of the vertical wave bending moment.

Special considerations are regarded necessary for novel designs and vessels with one or more of the following characteristics, e.g., by using alternative assessment methods as outlined in Sec. 4:

— rule length > 330 m
— breadth > 47 m
— bow flare angle (see Sec.2 [3]) > 55°
— vessel contract speed (see Sec.2 [3]) > 25 knots.

Satisfying the outlined procedures will provide a vessel with ultimate and/or fatigue strength which is equal to or better than the minimum industry standard, without being too conservative. Class notation WIV can be assigned to a container ship assessed by the outlined procedures.

3 References

General assumptions and requirements for fatigue and ultimate strength assessment are given by the rules. Methods and procedures for the fatigue strength assessment of the hull structures are described in the DNVGL-CG-0129, *Fatigue assessment of ship structures*. Methods and procedures for estimation of the hull girder ultimate capacity are described in the DNVGL-CG-0128, *Buckling*.

4 Definitions

This section lists abbreviations and acronyms used in this guideline.

CSR = IACS Common Structural Rules
IACS = International Association of Classification Societies
Rules = DNV GL Rules for Classification of Ships

SECTION 2 ULTIMATE STRENGTH

1 General

The ultimate strength assessment is a check of the hull girder ultimate capacity against the hull girder extreme load. The check is carried out on the vertical bending moment only, considered to be the critical failure mode.

The hull girder ultimate capacity is reduced by a safety factor related to the variation in strength due to uncertainties in the strength assessment, e.g., material properties, imperfections and amount of corrosion. Similarly, a safety factor is associated with the loads, related to uncertainties as, e.g., special sea conditions, the effect of speed and nonlinear effects in extreme response situations, the cargo weight and distribution, and special load effects not properly accounted for in design. Furthermore, the extreme static loading is not expected to occur at the same time as the dynamic extreme loading. Consequently, ultimate strength assessment is a check of a reduced strength still being larger than an increased load, associated with a low, acceptable probability of occurrence.

2 Hull girder ultimate strength check

The formulation for the ultimate strength assessment including the effect of whipping and applicable for, both, hogging and sagging is as follows:

$$\gamma_S M_{SW} + M_{WV}(\gamma_W + (\gamma_{WH} - \gamma_W)\gamma_{dU}) \leq \frac{M_U}{\gamma_R}$$

where:

M_U = vertical hull girder ultimate bending capacity from quasi-static loading, in kNm, at the hull transverse section considered, see [4]

γ_R = partial safety factor for the vertical hull girder ultimate bending capacity

= $\gamma_M \gamma_{DB}$

γ_M = partial safety factor for the vertical hull girder ultimate bending capacity, covering material, geometric and strength prediction uncertainties as well as corrosion condition, e.g. according to [2.3]

γ_{DB} = partial safety factor for the vertical hull girder ultimate bending capacity, covering the effect of double bottom bending, e.g. according to [3]

M_{SW} = permissible still water bending moment, in kNm, in seagoing condition at the hull transverse section considered

γ_S = partial safety factor for the still water bending moment, e.g. according to [3]

M_{WV} = vertical wave bending moment, in kNm, according to RU SHIP Pt.3 Ch.4 Sec.4 [3.1.1] and at the hull transverse section considered

γ_W = partial safety factor for the vertical wave bending moment, e.g. according to [3]

γ_{dU} = partial safety factor reducing the effectiveness of whipping during collapse (dynamic collapse effect), e.g. according to [3]

γ_{WH} = partial safety factor for the additional whipping contribution according to [3] or defined as:

$$\gamma_{WH} \geq \max\left\{\gamma_W; \frac{M_D}{M_{WV}}\right\} \geq 1.0$$

M_D = design vertical wave bending moment from numerical or experimental analysis including the effect of whipping, see Sec.3 [4].

The whipping contribution can be understood to be the difference between the partial safety factor for whipping, γ_{WH}, and for the vertical wave bending moment, γ_W, including the reduction factor γ_{dU}. In reality, the maximum design moment when whipping is included often occurs for a different condition (sea state, heading and ship speed) than the condition giving the maximum wave design moment ($M_{WV} \cdot \gamma_W$), which also may have some whipping. However, to avoid assessing ultimate strength for two conditions separately, the two cases have been merged. The partial safety factor for the additional whipping contribution, γ_{WH}, is set relative to the vertical wave bending moment according to the rules, M_{WV}.

The reduction of the effectiveness of whipping (γ_{dU}) is only applied to the difference between the partial safety factor for whipping and that for the vertical wave bending moment (γ_{WH} - γ_W), and not applied to the quasi-static wave loading. Unless taken according to [3], factor γ_{dU} will have to be determined on a case by case basis through dynamic collapse studies. For γ_{dU} = 0, the formulation will be equal to the formulation in RU SHIP Pt.3 Ch.5 Sec.4 [2], DNV-OS-C102, or IACS CSR.

3 Hull girder extreme loads and safety factors

The herein given empirical relations were derived for containerships with

— block coefficient in the order of 0.6 to 0.7,
— speed in the range of 20 to 29 knots,
— length in the range of 90 to 400 meters.

Special considerations are regarded necessary for vessels that do not comply with one or more of the above characteristics, e.g., by using alternative assessment methods as outlined in Sec. 4.

The whipping contribution, which applies to both sagging and hogging, can be estimated as:

$$\gamma_{WH} = 1 + c_L[3.8 \cdot 10^{-7}(L + 1100)(V + 4.1)^2(\tan\alpha - 0.19)] \geq \gamma_W$$

where:

α = bow flare angle, in radians, according to RU SHIP Pt.3 Ch.10 Sec.1 [2.1], at 0.05 L aft of FP and between still water line at scantling draft and upper deck (hence not the local angle), refer to Figure 1
L = rule length, in m
c_L = distribution factor to be taken as:
= 1.0 for $x/L \leq 0.5$
= sin (π x/L) for $x/L > 0.5$
V = contract speed, in knots, at design draft and with 85% MCR and 15% sea margin

If the contract speed, V_d, is specified at another x% MCR and y% sea margin, it can be converted by the following formula:

$$V = 0.904\, V_d \left(\frac{1 + y/100}{x/100}\right)^{1/3}$$

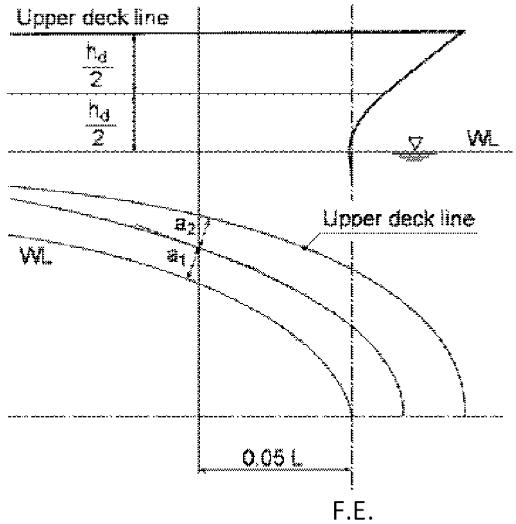

Figure 1 Determination of bow flare angle α

The partial safety factors are defined as follows:

γ_W = 1.2
γ_{DB} = 1.1 in hogging if no lateral loads are considered for the estimation of M_U, otherwise 1.0
= 1.0 in sagging
γ_S = 1.0
γ_M = 1.05 for calculation models for the estimation of M_U based on net scantlings, deducting 0.5 t_c from the gross offered scantlings, and using the multi-step model according to DNVGL-CG-0128, *Buckling*
γ_{dU} = 0.9, unless analysed for a specific ship

If nonlinear FE analysis is used for the estimation of M_U, the safety factors may be considered on case by case basis.

4 Hull girder ultimate bending capacity

The vertical hull girder ultimate bending capacity, M_U, needs to be estimated. The estimate can be based on simplified methods, analytically or semi-analytically considering the strength of individual longitudinal members (plate, stiffeners and panels), which makes up the transverse cross section, and summing up their contribution to the cross sectional collapse strength. Alternatively, nonlinear finite element analysis can be performed on part of the hull. The latter is regarded as more accurate, but the former methods are much faster and have proven to be quite accurate in many cases. For detailed description of the methods for estimating the vertical hull girder ultimate bending capacity, M_U, see DNVGL-CG-0128, *Buckling*.

5 Implication on class notations

5.1 **CSA** notation

Reference is made to RU SHIP Pt.6 Ch1 Sec.7. The **CSA** notation implies that the vertical bending moments are directly calculated as an alternative to the vertical bending moments according to the rules. Thus, for a specific ship the uncertainty in the hydrodynamic loading is reduced and the safety margin associated with the vertical bending moment, γ_W, can be reduced. Provided that the directly calculated bending moment is equal to or exceeds the rule based vertical bending moment, the partial safety factors γ_W and γ_{WH} can be reduced by 0.1, e.g. meaning that γ_W is reduced from 1.2 to 1.1. The directly calculated bending moments are then replacing the bending moments according to the rules.

5.2 **HMON notation**

Reference is made to RU SHIP Pt.6 Ch.9 Sec.4. The **HMON** notation implies that an approved hull monitoring system is installed on board. The display on the bridge gives a warning to the officer on watch when the loading is exceeding 80% and 100% of the loads according to the rules. The monitored loading includes the effect of whipping, and it increases the awareness of the actual hull loading including the effect of whipping. As a consequence the risk of overloading is reduced. Due to the reduced risk of overloading, the additional whipping contribution can be reduced by 30%, i.e.:

$$\gamma_{WH}^{HMON} = 1 + 0.7(\gamma_{WH} - 1) \geq \gamma_W$$

SECTION 3 FATIGUE STRENGTH

1 General

The vibration-induced stress is put on top of the wave-induced stress. As the vertical bending vibration modes are considered as most relevant, the vibration-induced stress is superimposed only on the wave-induced stress from the vertical bending moment. Accordingly, the vertical wave-induced stress is adjusted as follows:

$$\sigma_{WV,vib} = f_{vib} \cdot \sigma_{WV}$$

where:

f_{vib} = vibration factor, replacing the vibration factor according to RU SHIP Pt.3 Ch.4 Sec.4 [3.1.1]
σ_{WV} = wave-induced stress from vertical bending moment according to RU SHIP Pt.3 Ch.5 Sec.3 [4.1.2], for the longitudinal structure detail considered, but calculated with the vibration factor according to the rules set to f_{vib} = 1.0.

The vibration factor f_{vib} represents a correction of the wave-induced stress consistent with the additional damage from whipping and springing for the intended design area, e.g., world wide. The vibration factor assumes all wave headings of equal probability.

In general, the vibration factor is defined as $f_{vib} = f_{vib,j}$ where the subscript *j* refers to the loading condition, e.g. loaded or ballast condition:

$$f_{vib,j} = \sqrt[m]{\frac{D_{w+vib,j}}{D_{w,j}}} \geq 1.0$$

where:

$D_{w+vib,j}$ = total damage, being the sum of wave-induced and vibration-induced damage, in loading condition *j*
$D_{w,j}$ = wave-induced damage, in loading condition *j*
m = inverse slope of the S-N curve consistent with that used for the estimate of fatigue damage (for two-slope S-N curves, the value of *m* may be taken between *m* and $m + \Delta m$, as appropriate, where Δm is the change in slope of the S-N curve)

The vibration factor can be estimated based on empirical relations, see [2], numerical or experimental analysis, see Sec. 4.

2 Hull girder fatigue loads

The herein given empirical relations were derived for container ships with a length in the range of 90 to 400 meters and for World Wide or North Atlantic trade.

The f_{vib} for any loading condition can be estimated as:

$$f_{vib} = \sqrt[m]{1 + 1.2 \cdot 10^{-6}(L + 1100)(V + 4.1)^2(\tan\alpha - 0.45)}$$

where:

α = bow flare angle, in degrees, according the RU SHIP Pt.3 Ch.10 Sec.1 [2.1], at 0.05 L aft of FP and between still water line at scantling draft and upper deck (hence not the local angle)
m = 3
L = rule length, in m
V = contract speed, in knots, as defined in Sec.2 [3]

The factor f_{vib} can be considered as constant along the hull girder.

Vibration modes other than the vertical bending modes, e.g. torsional modes, are neglected in the simplified fatigue assessment. For hatch corners, the vibration factor is only applicable as a correction to wave-induced stress from the vertical bending moment and not from torsion.

3 Application

3.1 Prescriptive approach

In the prescriptive approach, useful for early design, in any position along the hull, f_{vib} in RU SHIP Pt.3 Ch.4 Sec.4 [3.1.1], is replaced with the ship specific estimate of f_{vib} according to [1]. The total stress including the effect of vibrations is not to be taken less than according to the rules.

3.2 Approach based on the directly calculated vertical wave bending moment distribution

The directly calculated vertical wave bending moment along the ship hull can be utilized to establish a reduction factor, f_d, for the bending moment towards the forward and aft part of the hull for each loading condition j:

$$f_{d,j}(x) = \frac{M_{w,j}(x)}{max(M_{w,j}(x))}$$

where:

$M_{w,j}(x)$ = directly calculated wave bending moment for position x from aft end (AE) and for loading condition j. $M_{w,j}$ should be calculated for a probability level of exceedance of 10^{-2}.

For the direct calculation of the wave bending moments, spectral analysis using linear hydrodynamic analysis should be carried out. The hydrodynamic loads should be calculated using 3D potential theory computations. For details of hydrodynamic analyses, reference is made to DNVGL-CG-0130, *Hydrodynamic assessment of wave induced loads for ships*. The following conditions should be considered for container ships:

- Ship speed: 75% of contract speed as defined in Sec.2 [3]
- Loading condition: One loading condition with 75% to 85% of permissible maximum still water vertical bending moment
- Draft: Design draft, but not less than 80% of scantling draft

— Wave directions: Equally distributed primary wave directions relative to the ship's course at intervals not greater than 15°
— Wave spectrum: Pierson-Moskowitz with cosine-squared wave energy spreading
— Probability distribution of amplitudes: Rayleigh distributed probability of amplitudes, applied to the response within each short term condition (sea state)
— Wave statistics: Scatter diagram for World Wide trade according to DNV-RP-C205, *Environmental conditions and environmental loads*, App. C, Table C-3, or, if requested by the applicant, for North Atlantic trade according to Table C-2.

Factor f_d is applied as a reduction factor to the maximum vertical wave bending moment amidships for loading condition j:

$$\sigma_{WV,vib,j} = f_{d,j} \cdot f_{vib,j} \cdot \sigma_{WV,j}$$

where:

$\sigma_{WV,j}$ = wave-induced stress from vertical bending moment according to RU SHIP Pt.3 Ch.5 Sec.3 [4.1.2], for loading condition j and the longitudinal structure detail considered, but calculated with the vibration factor and the distribution factor for the vertical wave bending moment according to the RU SHIP Pt.3 Ch.4 Sec.4 [3.1.1], set to f_{vib} = 1.0 and f_m = 1.0, respectively.

Figure 1 shows a sample distribution f_d, i.e. the normalized bending moment distribution.

The stress from vertical bending moment including the effect of vibrations is not to be taken less than according to the rules in any position along the hull.

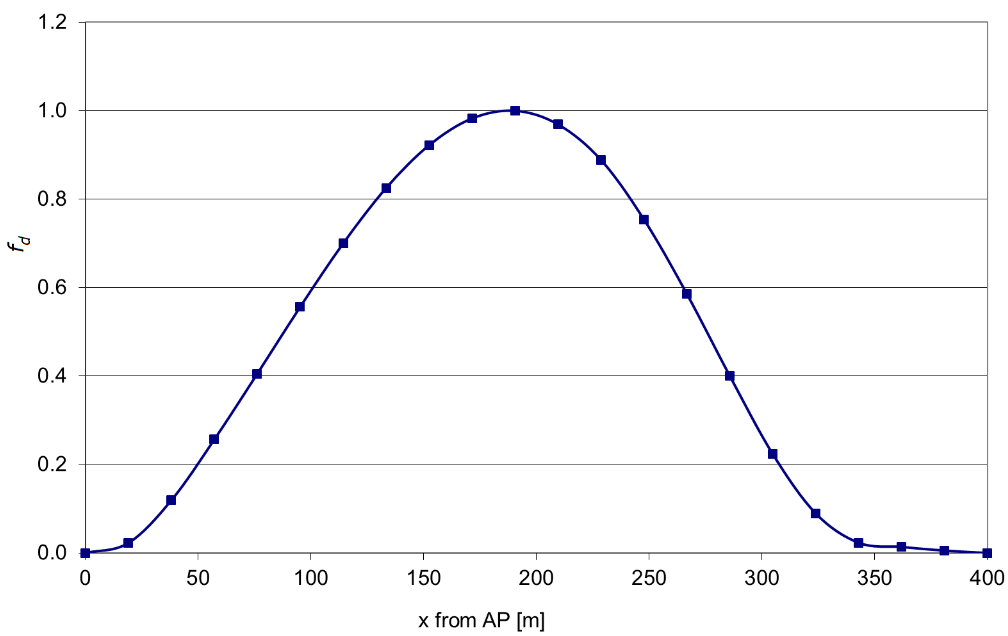

Figure 1 Example for reduction factor, f_d, over ship length

3.3 Approach based on the directly calculated vertical wave bending moment

Alternatively to the approaches described in [3.1] and [3.2], which are related to the rule vertical wave bending moment, the wave-induced stress based on the directly calculated bending moment, superimposed by the vibration-induced stress can be assessed.

For the direct calculation of the wave bending moments, spectral analysis using linear hydrodynamic analysis should be carried out as described in [3.2]. The wave bending moment should be calculated for a probability level of exceedance of 10^{-2}. The following additional conditions should be considered for container ships:

— Fatigue design life, T_{DF}: According to RU SHIP Pt.3 Ch.9 Sec.1 [1.4]
— Time at sea, f_0: f_0 = 0.85, i.e. 15% of the fatigue design life T_{df} in port or sheltered waters.

The stress including the vibration-induced stress is then derived by:

$$\sigma_{WV,vib,j} = f_R \cdot f_{vib,j} \cdot \sigma_{WV,d,j}$$

where:

f_R = factor for the consideration of weather routing and other attenuating effects
= 0.8
$\sigma_{WV,d,j}$ = wave-induced stress from directly calculated vertical bending moment for loading condition j, for a probability of exceedance of 10^{-2}, and for the longitudinal structure detail considered.

It should be observed that for the calculation of the fatigue stress range according to DNVGL-CG-0129, *Fatigue assessment of ship structures*, the environmental factor f_e should be taken equal to 1.0 as the loads are based on direct hydrodynamic analysis for a specific trade.

The directly calculated number of load cycles may deviate from that according to DNVGL-CG-0129, *Fatigue assessment of ship structures*. The number of load cycles according to DNVGL-CG-0129 should be replaced by the directly calculated number of load cycles. The directly calculated number of cycles at amidships may be taken as number of load cycles in any position along the hull.

The resulting fatigue damage is not to be taken less than that calculated according to the rules in any position along the hull. However, resulting fatigue damage in excess of that calculated according to the approach given in [3.1] or [3.2] does not need to be considered.

4 Implication on class notations

4.1 RSD notation

The approaches described in [3.2] and [3.3] can be applied to global strength analysis of container ships using EDWs from direct wave load analysis, as required for the RSD notation according RU SHIP Pt.6 Ch.1 Sec.8, and as described in DNVGL-CG-0131, *Container ships*, Sec.2.

The load cases applied in the global strength analysis with head sea condition (Φ = 180°) and an additional roll angle of φ = 0° should be assessed. The height of the design wave should be adapted to include the effect of whipping and springing.

For the approach described in [3.2] the height of the design wave has to be chosen with respect to the vertical design bending moment given by the rules, but where the vibration factor f_{vib} is replaced by f_{vib} according to this guideline.

For the approach described in [3.3] the height of the design wave has to be chosen with respect to the directly calculated vertical wave bending moment multiplied by f_R and f_{vib}.

SECTION 4 ALTERNATIVE METHODS

1 General

While empirical relations are useful in early design, a more accurate way of establishing the effect of vibrations on ship's hull girder stress response is by model tests or numerical analysis, assuming that state-of-the-art procedures and tools are applied. A further benefit of model tests and numerical analysis is that the effect of vibrations can be estimated along the whole vessel, while the empirical relations typically assume a constant value over ship length.

Numerical or experimental analyses should be sufficiently accurate and should include the relevant excitation mechanisms and an appropriate representation of hydrodynamic and structural interaction. The following influential factors have been confirmed to significantly affect wave-induced vibration and should be addressed in the analysis:

— loading condition, i.e. draft, trim and cargo distribution
— forward speed and steady ship wave system
— hydrodynamic and structural damping
— encountered sea states.

Due to their interdependency, the above issues must be handled consistently.

The experimental or numerical ship model should be flexible and tuned to accurately reproduce the vibration shape and natural frequency of the 2-node vertical bending mode. The 3- and 4-node modes should be also considered but their accuracy is less important.

For fatigue strength assessment, full-scale measurements of similar ships on similar trades can also be basis for estimation of the effect of vibrations on ship's hull girder stress response for sister vessels or similar designs.

Where model tests are chosen for estimating the effect of vibrations on the ship's hull girder stress response, a recognized testing facility should be used in order to obtain useful results with the desired quality.

In any case, the details of analyses using these alternative methods should be agreed on with the society.

2 Ultimate strength assessment

Ultimate strength is governed by loads of large amplitude where nonlinear effects become dominating. Wave impacts like bow flare slamming, inducing whipping, are most relevant.

The extremes of wave-frequency load and wave-induced vibration do not necessarily occur simultaneously or in the same short-term sea state condition, i.e. the maximum of wave loading and whipping response may be triggered by different sea state and vessel speed combinations. Therefore, investigations should assure to cover all relevant short-term conditions at realistic speeds. The extreme load relevant for ultimate strength assessment is defined as the load with one expected exceedance during the intended design life time.

This can be achieved by a procedure which starts with a few severe sea states and iteratively covers more short-term conditions until inclusion of additional sea states only marginally changes the extreme load found from summation over all short-term cumulative distributions. As a starting point, sea states which, according to linear numerical analysis, contribute most to the long-term expected value of the vertical bending moment amidships, are regarded relevant, but should be used with care for strong nonlinear phenomena such as whipping. These sea states are associated with low speeds and low expected duration. Smaller sea states with longer duration and associated with higher speeds must also be included. The expected durations of the sea states are obtained from the relevant scatter diagrams.

Involuntary speed loss should be considered to find the realistic speed in these sea states. For each investigated combination of significant wave height and peak period, an individual forward speed must be determined corresponding to the maximum achievable speed when heading against the waves.

The uncertainty in a short simulation or short measurement period compared to the relevant duration should be considered in order to achieve a reliable dimensioning value.

3 Fatigue strength assessment

Fatigue is dominated by loads of relatively low amplitude but high frequency of occurrence. Analysis must therefore in principle cover the whole range of sea state conditions of the design trade. Reasonable distributions of ship speeds and loading conditions should be used according to the intended ship operation. Typically, all wave headings are assumed equally distributed. If the specified heading distribution differs from even heading distribution, this can be accounted for. If simulations or experiments are carried out only in head seas, the results must be corrected to be applicable for the relevant heading profile.

Damping (structural and hydrodynamic) is of particular importance for the effect of vibration on fatigue. Empirical relations may be used to determine the structural damping. The hydrodynamic damping can be estimated numerically by vibration decay simulations at forward speed using a viscous flow solver for free-surface flows combined with an elastic representation of the ship structure. Also in model tests structural damping should be added.

Model tests and numerical analysis may be subject to uncertainties in the encountered wave environment in different trades. For instance the effect of routing can be significant and the assumed wave conditions can differ from real life. Thus, if available, the effect of vibrations on the ship's hull girder stress response may be estimated based on data from full-scale measurements on sister vessels or similar designs. In this case, the measurement data should:

— come from a ship design which is representative for the specific ship,
— come from a ship in a trade that is relevant for the trade of the specific ship, and
— cover a sufficiently long measurement period to represent the increase of the fatigue damage due to whipping and springing.

SECTION 5 REFERENCES

1 References

/1/ Andersen, I.M.V., *Full scale measurements of the hydro-elastic response of large container ships for decision support*, Phd thesis, DCAMM Special Report No. S166, April 2014, DTU Mechanical Engineering, Lyngby, Denmark, 2014.

/2/ Barhoumi, M. & Storhaug, G., *Assessment of whipping and springing on a large container vessel*, International Journal of Naval Architecture and Ocean Engineering, Volume 6, Issue 2, pp.442-458, ISSN (online) 2092-6782, June 2014.

/3/ Drummen, I., *Experimental and Numerical Investigation of Nonlinear Wave-Induced Load Effects in Containerships considering Hydroelasticity*, PhD dissertation, Department of Marine Technology, Norwegian University of Science and Technology (NTNU), 2008.

/4/ Drummen, I., Storhaug, G. & Moan, T., *Experimental and Numerical Investigation of Fatigue Damage in a Containership in Head Seas*, Journal of Marine Science & Technology, Japan, 2007.

/5/ Drummen, I., Storhaug, G., Moe, E. & Moan, T., *Experimental and full scale investigation of the importance of fatigue damage due to wave-induced vibration stress in a container vessel*, Proceedings of Conference on Design and operation of Container vessels, RINA, November 21-22nd, 2006.

/6/ el Moctar, O., Oberhagemann, J. & Schellin, T. E., *Free-Surface RANS Method for Hull Girder Springing and Whipping,* SNAME Transactions 119, 48–66, 2011.

/7/ Gadai, O., Naess, A & Storhaug, G. *Extreme value statistics of whipping response for large ships*, PRADS 2010, The 11th Int. Symp. on Practical Design of Ships and Other Floating Structures, Rio de Janeiro, Brazil, 19-24 September, 2010.

/8/ Heggelund, S.E., Storhaug, G. & Choi B-K, *Full scale measurements of fatigue and extreme loading including whipping on an 8600TEU Post Panamax container vessel in Asia to Europe trade*, OMAE2011- 49378, Rotterdam, The Netherlands, June 19-24, 2011.

/9/ Kahl, A., Brünner, E. & Menzel, W., *Full-Scale Measurements on a Panamax Containership*, Trans. German Society for Maritime Technology, Vol. 103, pp. 278-289, 2009.

/10/ Moe, E., Holtsmark, G. & Storhaug, G., *Full scale measurements of the wave induced hull girder vibrations of an ore carrier trading in the North Atlantic*, Proceedings of Conference on Design & Operation of Bulk Carriers, RINA, London, 18-19th October, 2005.

/11/ Oberhagemann, J., Shigunov, V. & el Moctar, O., *Application of CFD in Long-Term Extreme Value Analyses of Wave Loads*, Ship Technology Research 59(3), 4–22, 2012.

/12/ Rathje, H., Kahl, A., & Schellin, T.E., *High-Frequency Ship Response Assessment of Large Containerships*, J. Offshore and Polar Engineering, Vol. 22 (1), pp. 1-8, 2012.

/13/ Rathje, H., Kahl, A., & Schellin, T.E., *Semi-Empirical Assessment of Long-Term High-Frequency Hull Girder Response of Containerships*, Proc. 22nd Int. Offshore (Ocean) and Polar Engineering Conf. & Exhibition, Rhodos, Greece, 2012.

/14/ Söding, H., Shigunov, V., Schellin, T.E. & el Moctar, O., *A Rankine panel method for added resistance of ships in waves*, J. Offshore Mech. Arct. Eng., 136(3) 031601·1-031601·7, 2014.

/15/ Söding, H., von Graefe, A., el Moctar, O. & Shigunov, V., *Rankine source method for seakeeping predictions*, Proc. 31st Int. Conf. on Ocean, Offshore and Arctic Engineering OMAE, Rio de Janeiro, Brazil. - Paper Nr. OMAE2012-83450, June 10-15, 2012.

/16/ Steen, E., Storhaug, G., Kippenes, J., Sado, O., Austefjord, H. and Dugstad, G., *Ultimate strength assessment of the engine room area of the container vessel MSC Napoli - January 2007 Incident*, Transaction RINA, Design and operation of container vessels, UK, July, 2008.

/17/ Storhaug, G., Choi, B.K., Moan, T. & Hermundstad, O.A., *Consequence of whipping and springing on fatigue for a 8600TEU container vessel in different trades based on model tests*, 11th Int. Symp. on Practical Design of Ships and Other Floating Structures, Rio de Janeiro, Brazil, 19-24 September, 2010.

/18/ Storhaug, G., Malenica, S., Jung, B.H., Zhu, S. & Hermundstad, O.A., *Consequence of whipping and springing on fatigue and extreme loading for a 13000TEU container vessel based on model tests*, 11th Int. Symp. on Practical Design of Ships and Other Floating Structures, Rio de Janeiro, Brazil, 19-24 September, 2010.

/19/ Storhaug, G., Heggelund, S.E. & Mathisen, J., *Model test and full scale measurements of whipping on container vessels in the North Atlantic*, OMAE2009-79127, Hawaii, June, 2009.

/20/ Storhaug, G., Heggelund, S.E. & Mathisen, J., *Model test and full scale measurements of whipping on container vessels in the North Atlantic*, OMAE2009-79127, Hawaii, June, 2009.

/21/ Storhaug, G., *The 4400 TEU container vessel MSC Napoli broke its back, but did whipping contribute*", Hydroelasticity in Marine Technology, Southampton, Sept. 2009.

/22/ Storhaug, G. and Heggelund, S.E., *Measurements of Wave Induced Vibrations and Fatigue Loading Onboard Two Container vessels operating in harsh wave environment*, Transactions of RINA, Design & Operation of Container Vessels, London, UK, July 2008.

/23/ Storhaug, G., *Experimental investigation of wave induced vibrations and their effect on the fatigue loading of ships*, PhD dissertation, NTNU, 2007-133.

/24/ Storhaug, G. & Moe, E., *Measurements of wave induced vibrations onboard a large container vessel operating in harsh environment*, PRADS2007, pp 419-428, Houston, USA, 30th Sept-5th Oct, 2007.

/25/ Storhaug, G., Moe, E., and Lopes, T.A.P., *Whipping measurements onboard a midsize container vessel operating in the North Atlantic*, Proceedings of RINA, CMP & Shanghai SNAME, International Symposium on Ship Design & Construction, 28-29th of November, 2007.

/26/ Zhu, S., *Investigation of wave-induced nonlinear load effects in open ships considering hull girder vibrations in bending and torsion,* PhD dissertation, NTNU, Trondheim, Tapir, November, 2012.