COMMITTEE II.2
DYNAMIC RESPONSE

COMMITTEE MANDATE

Concern for the dynamic structural response of ships and floating offshore structures as required for safety and serviceability assessments, including habitability. This should include steady state, transient and random response. Attention shall be given to machinery and propeller exciting forces. Uncertainties associated with modelling should be highlighted.

COMMITTEE MEMBERS

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KEYWORDS

Dynamic Response, Slamming, Whipping, Springing, Damping, Vibration, Noise, Underwater Noise, Explosion, Shock, Excitation, Propeller, Vortex Induced Vibration, Prediction, Natural Vibration, Forced Vibration, Linear Response, Non-Linear Response, Countermeasures, Troubleshooting, Measurement, Segmented Models, Hydro-Elasticity, Resonance, Monitoring, Model Tests.

CONTENTS

For the sake of saving space, the Committee used the following abbreviations:

BEM	- Boundary Element Method	FE(M)(A)	- Finite Element (Method) (Analysis)
(M)DOF	- (Multi) Degree of Freedom		
RANSE	- Reynolds-Averaged Navier-Stokes Equation	CFD	- Computational Fluid Dynamics
		LF	- Low Frequency
HF	- High Frequency	UNDEX	- Underwater Explosion(s)
H1/3	- Significant Wave Height	VIV	- Vortex Induced Vibration
GRP	- Glas Fibre Reinforced Plastics	CAD	- Computer Aided Design
3D	- 3-dimensional		

1.   INTRODUCTION

In the recent years two contradicting goals were in the focus of the world community: to ensure availability of sufficient energy for the growing world population and to slow down climate change by a reduction of carbon dioxide emission. This also put two major challenges on the marine industry: To explore and exploit resources in ever greater water depths and to reduce energy consumption of the world shipping fleet. With regard to the dynamic response of ship and floating structures this implied the following trends:

- increase in ship sizes to benefit from economy of scale, seemingly implying a greater risk of relevant sea way excited vibration,
- development of structural design and propulsion concepts mainly aiming at light structures and reduced fuel consumption rather than vibration and noise avoidance,
- the development of larger vessels for polar regions and shipping routes,
- an intensified interest into low underwater-noise ships being capable of exploring resources in areas of deep water
- and extensive R&D activities into flow induced vibration of offshore pipelines and risers.

Of course these trends were also reflected in the publications concerning the dynamic response of ships and floating offshore structures in the review period. This report is subdivided into four Sections representing different fields of applied structural dynamics in the marine field: ship structural vibration, ship acoustics, shock response and offshore applications. Most space is devoted to the Section on ship structural vibration since it has been the main topic of research of the past years. To account for the difference between low and high frequency response regarding the theoretical methods and the measurement techniques a separate Section was devoted to the field of ship acoustics. The Section on shock response treats the field of transient ship response to military impulse excitations, whereas slamming excited transient ship response is covered in the Section on ship structural vibration. The abundance of publications on offshore riser and pipeline vibration connected with the sophistication of the hydro-elastic theory behind it made a complete review virtually impossible. Therefore, the committee selected those publications for review which treat these topics in a more general sense and provide some overview. The implementation of an expert group covering this topic is recommended for ISSC 2012.

2.   SHIP STRUCTURAL VIBRATION

Despite considerable progress in the theoretical and experimental treatment of ship

vibrations, questions about the accuracy of vibration analysis and prediction methods are as topical as ever. This Section examines state of the art methods and techniques and assesses the progress made in this field.

At the beginning a separate subsection was dedicated to the topic of wave induced vibration because of the depth and integrated nature of this field and the vast amount of recent publications on this subject. Further subsections were devoted to ship structural vibrations which are excited by other excitation sources as, e.g., the ship's propeller. The respective subsections treat the modelling of the excitation sources, the analysis methods used for the prediction of ship vibrations, potential vibration damping and countermeasures, full scale vibration measurement and monitoring campaigns and, finally, standards and regulations related to this field.

The committee selected those publications for review which treat validated theoretical and experimental topics and which promise practical progress in the field of ship structural vibration.

2.1   *Wave Induced Vibration*

Wave induced vibration of ships might occur in two different forms denoted as springing and whipping. Whereas springing represents a resonant periodic vibration response to high frequency harmonic wave excitation components, whipping is characterised by transient vibration response caused by slamming impulses. Researchers further differentiate between linear springing, where the encounter frequency of sea way components with short wave lengths is in resonance with the natural frequency of the basic hull girder mode, and non-linear or sum frequency springing where the periodic vibration excitation forces act with a higher order of the wave encounter frequency. The former can be predicted by linear hydrodynamic theories, while the latter depends on second order hydrodynamic effects as, e.g., superposition of different wave systems. In spite of the periodic sum frequency excitation forces having a smaller intensity per unit wave height than the linear forces, they may become of greater importance because they originate from considerably longer waves with a higher energy content.

Normally, for all kinds of wave induced vibration the hull response is dominated by the basic hull girder natural vibration modes, especially the 2-node vertical bending mode. So, depending on the size and type of ship, the typical frequency range of hull girder response to wave excitations is abt. 0.3 Hz to 3 Hz. Storhaug (2007) presented a comprehensive literature review on wave induced vibration that shows references on the subject as far back as the late 19th century. Theoretical research efforts on the subject is reported to have started in the 1960s for vessels trading on in-land water ways which were prone to springing due to their small longitudinal strength compared to ocean-going vessels.

Wave induced vibration causes additional accelerations and loads acting on the global

hull structure. An overview of the consequences of wave induced vibrations is presented in Figure 1. Solid and dashed line arrows indicate whether the respective whipping and springing vibration response is likely to be of practical relevance or tends to be more of academic interest, respectively. Wave induced vibration will occur simultaneously with the ship's rigid body motions and quasi-static hull girder deformations and, naturally, these phenomena can not always be clearly separated. This part of the ISSC-2009 report primarily deals with the global vibration response to wave excitation, whereas the local pressures and stresses and the integrated impulsive hull girder loads caused by stern and bow slamming events are discussed by Committee I.2 and V.7, respectively.
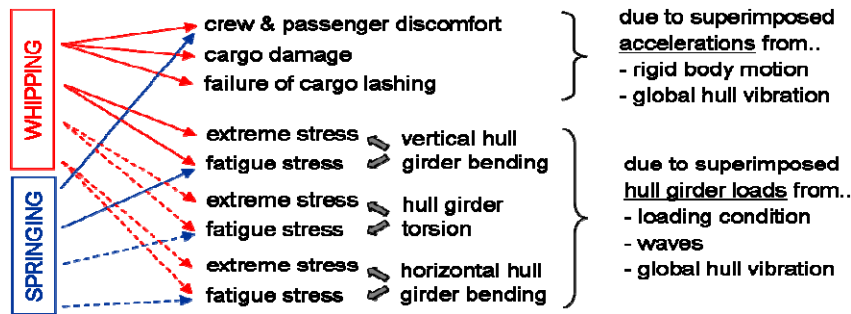

Figure 1: Overview of consequences of wave induced hull response

In recent years many publications on ship structural vibration were dealing with wave induced vibration. Research aimed to improve the knowledge on the grade of magnification of the extreme quasi-static hull girder loads due to whipping vibration, to quantify the contribution of wave induced vibration to the accumulated fatigue damage of critical structural details and to better predict the additional accelerations passengers and cargo are exposed to. The investigations included full scale measurements, model basin tests and theoretical analyses.

In the following the abbreviations HF (High Frequency) and LF (Low Frequency) will be used to differentiate between the LF quasi-static ship response to wave loads and the HF part resulting from wave induced hull vibration.

2.1.1   Full Scale Measurements

Many full scale measurement campaigns have been reported over the last period, covering a variety of ship types, ship sizes and trades. The majority of the measurement campaigns focus on the fatigue performance of the vessels, and only a few papers discuss the effect of dynamic response on the extreme loading of the hull girder.

Moe *et al* (2005) have reported a comprehensive measurement campaign carried out on a 220.000 dwt bulk carrier built in the late nineties engaged in iron ore trade in the North Atlantic between Canada and Northern Europe. Measurements were taken over a

period of three years and represent 840 days of operation in either full load or ballast condition. The vessel has been subject to an earlier measurement campaign reported by ISSC (2006), but the vessel has later been significantly strengthened because extensive fatigue damages occurred in the original configuration. The hull girder section modulus has been increased by up to 47 % as a result of the strengthening thereby increasing the natural frequency of the two node hull girder vibration mode by abt. 10 % (from 0.5 to 0.58 Hz for ballast and from 0.46 to 0.51 Hz for loaded condition). HF signals associated with hull girder vibrations are reported to be almost continuously present in the measured signals. To separate the LF part the signals were low pass filtered (0.35 Hz) and the fatigue damage was calculated for the total and LF part, the difference being the HF contribution to the damage. The authors point out that the HF damage must not be calculated from the high pass filtered signal because the magnification of the stress envelope would not be captured by such a procedure. The fatigue damage was calculated by a Rainflow counting method combined with a Miner-Palmgren damage accumulation model. The total fatigue damage decomposed into its LF and HF contributions given as function of significant wave height from 2 m up to 9 m is presented in Figure 2. Obviously the HF response is significantly less important for the full load condition compared to the ballast draft. The authors suppose that this might be explained by the much higher probability of head sea encounters on the westbound ballast route. They also mention differences in the magnitude of excitation forces, the higher draft and increased damping as possible causes for the lower relative importance of the HF part in loaded condition, but the measurements do not contain evidence for this. Following seas are reported to cause insignificant levels of vibration.

The authors presented the measurement results in terms of fatigue damage rather than as a distribution of the monitored strains/stresses. It is, however, possible to deduce some information regarding the relative magnitude of the monitored LF and HF stress ranges by reversing the fatigue damage calculation. Assuming a single sloped SN-curve (m=3) the ratios between total and LF stress range can be calculated, see Table 1. From such an estimate it can be concluded that in ballast condition the HF stress magnification amounted to 10 % to 26 % of the LF stress for a significant wave height ranging from 9 m to 5 m, respectively. Due to the simplification of a one sloped SN-curve the HF stress may be somewhat overestimated in this estimate.

Table 1
Estimated average increase of LF stress range BY HF Vibrations

Wave height (m)	Ballast		Cargo	
	D_{tot}/D_{LF}	$\Delta\sigma_{tot}/\Delta\sigma_{LF}$	D_{tot}/D_{LF}	$\Delta\sigma_{tot}/\Delta\sigma_{LF}$
9	1.33	1.10	-	-
8	1.36	1.11	1.06	1.02
7	1.40	1.12	1.10	1.03
6	1.64	1.18	1.22	1.07
5	2.00	1.26	1.11	1.04

The authors also reported that the 10% increase of the natural frequency of the 2-node hull girder vibration by strengthening did not much affect the relative HF contribution

to the overall damage for the critical structural detail. This is further illustrated by the figures listed in Table 2.

Table 2
Relative contribution of HF Damage to 20-year overall damage

	Total Damage	LF Damage	HF Damage
Before hull strengthening	3.12	1.76	1.36 (44%)
After hull strengthening	0.68	0.42	0.26 (38%)

Storhaug *et al* (2006) reported on full scale measurements on a Capesize bulk carrier trading from Canada, Brazil and South Africa to Europe. Measurements were carried out from November 2003 to May 2005 corresponding to 277 days at sea. The influence of wave induced vibrations on the fatigue utilisation of the hull was investigated and compared with the findings from the slightly larger vessel discussed in Moe *et al* (2005). In this case the relative fatigue damage from high frequency response amounted to 56% of the total damage for a typical structural detail arranged in the mid ship main deck region. However, due to the moderate wave climates which this vessel typically encounters, the total damage to be expected for a 20-year life time was predicted to be 0.23 only.

Mathiesen *et al* (2007) have utilised the measurements on the two ore carriers to evaluate whether HF vibration has a significant effect on the extreme hull girder response. The evaluated measurement period is 5 years for the first vessel and 2 years for the second. Long continuous time series representing reasonably stationary conditions in terms of sea state and operational parameters have been assembled to increase the confidence in the response statistics. Time series have been selected to include the most severe conditions encountered during the measurement period, as well as less severe conditions to investigate trends. The authors conclude that the HF increase of the hull girder response is approx. 10 percent for the most severe conditions characterised by close to head sea conditions, large significant wave heights and strongly reduced speed but that the HF part increases for more moderate sea states. The authors emphasise that there remains some uncertainty because none of the vessels has encountered conditions with significant wave heights up to 15 m, as normally used as a design condition for such vessels.

In recent years emphasis in whipping and springing monitoring has turned from bulk carriers to container ships. This can be most likely attributed to the extreme development steps which can be observed in container ship design since decades and the ever growing importance of this ship type for world trade.

Okada *et al* (2006) reported on a 3-year monitoring campaign onboard a 6700 TEU vessel from 1998 to 2001. Wave height, ship motion, deflections and strains were measured in the LF and HF range. The vessel operated in the trade between Japan and Europe. To separate the LF part from the HF part the authors did chose a low-pass filter at 0.5 Hz (hull girder natural frequency of the 2-node vertical vibration mode approx.

0.7 Hz). Based on an extrapolation of the measurement data to a 20-year life time they also calculated the cumulative damage ratio which might be experienced by a main deck butt weld. Under consideration and omitting the HF part they obtained values of 0.105 and 0.050, respectively. So for this detail fatigue damage doubled due to the HF-part, however, with respect to the permissible value of 1.0 the damage still being on a very low level. Consequently, the authors recommend the avoidance of stress concentrations and suggest further research on the fatigue contributions from whipping and springing vibration.

Toyoda *et al* (2008) reported on measurements on a similar container vessel trading between Japan and Europe between 2002 and 2004. The 2-node natural frequency of this vessel is abt. 0.67 Hz and the low pass filter was set to 0.33 Hz. The highest measured longitudinal stress in the main deck showed a significant HF contribution which caused that the design stress range according IACS UR 11S of 200 N/mm^2 was exceeded by 15 N/mm^2 in that event. The authors also extrapolated the measured stress ranges to calculate the HF contribution to the fatigue damage of a butt weld in the main deck during 20 years lifetime. As they found a contribution of 0.04 only, they concluded that this is easily covered by the implicit design margins. Nevertheless, they recommend to follow-up the relevance of HF contributions to LF extreme and fatigue loads for new ship designs.

Also smaller vessels have been subject to investigation by full scale measurements. Aalberts and Nieuwenhuijs (2006) reported on measurements carried out in 2002 onboard a 9900 tonne ice class general cargo/container vessel of 130 m length and 15.9 m breadth for general hull monitoring purposes. During the one year measurement the vessel was operating in the Baltic Sea, the North Atlantic and made one voyage to the Mediterranean. With approx. 1.0 Hz and 1.3 Hz, respectively, the natural frequencies of the 2-node horizontal and vertical bending vibration mode turned out to be rather low for such kind of ship and indicated that the hull girder horizontal bending stiffness is much smaller than its vertical bending stiffness. This might be explained by the vessel's small breadth to length ratio and represents supposedly the major reason that this represents one of the rare cases were also significant horizontal bending vibration is reported, even for head sea conditions. Nevertheless, the vertical bending vibration dominated the HF response. Based on the measured cumulative rainflow distributions the authors estimated that the amplitude of the extreme total vertical hull girder moment can be expected to exceed the moment solely induced by the LF part by approximately 20 %. There occurred situations where the HF vertical bending response amplified the LF part by up to 50 percent, however, not for the most severe environmental situations, which in turn could be related to speed reduction in heavy weather conditions. For constant ship speed it is concluded that the short term extreme HF load / response is linear with the LF load / response for varying wave heights.

With regard to the fatigue of structural details primarily exposed to vertical hull bending loads the authors estimate an increase of the accumulated damage by 30 % due to the HF part. However, they do not assess this as critical because additional HF

fatigue loads of such magnitude are said to be implicitly considered in the design safety margins of the applied rules.

Storhaug and Moe (2007) publicised on an ongoing monitoring campaign which started in 2005 onboard a 2003-built 4,400 TEU container vessel operating in the North Atlantic. Mainly because of their advantages in signal transfer and robustness, optical sensors were used for the strain measurements. To separate the LF and HF parts the authors used band pass filters ranging from 0.01 to 0.30 Hz and 0.45 to 100 Hz, respectively. The natural frequency of the vertical hull bending vibration mode is not given. The authors focus on the quantification of the HF-contribution to the overall fatigue damage. For the monitored winter season roundtrip Europe-Canada-Europe it is concluded that the fatigue damage was 4.8 times worse than the assumed design value. However, it is emphasized that the variation from trip to trip is significant and that such extreme damage ratios must not be expected for operation in other seasons. For this voyage the HF-part increased the total damage by 23 %, but during the worst half-hour record this value increased to 43 %. The authors note that the fatigue damage experienced by the vessel during the most severe day corresponded to approx. two month of average fatigue loading and, therefore, advocate the use of monitoring systems for better routing guidance. Moreover, they reported that the measured damping ratio of 1.2 % was about two times higher than the values typically obtained in full-scale measurements on bulk carriers and those deduced from model tank measurements with segmented models. The HF-components were found to be significant for vertical bending vibration only, i.e. no significant stress amplitudes could be attributed to horizontal or torsional vibration of the hull girder. The observation of fatigue damage is not reported for these vessels.

A 2800 TEU vessel sailing in a similar route as the 4400 TEU vessel described by Storhaug and Moe (2007) was measured by Storhaug et al (2007). One reason to select this vessel for monitoring was the presence of cracks after less than 8 years of operation. The authors report that a simplified fatigue life prediction which does not account for the warping deflection of the hull could not explain the crack appearance and, therefore, recommend consideration of warping deflections in design stage calculations for such kind of ships. For a North Atlantic crossing during the summer season they found that the total fatigue damage corresponded to 20% of the assumed design value and that the contribution from wave induced vibration to this damage amounted to 43 %. For this ship the authors report a damping ratio of abt. 1% and a natural frequency of the 2-node bending vibration mode of abt. 0.8 Hz.

Storhaug and Heggelund (2008) report on measurements on a sister vessel of this described in the previous paragraph. A total of 23 North-Atlantic crossings from August 2007 to May 2008 was investigated. Based on comparisons with sea state observations and buoy measurements it was concluded that the used wave measurement system overestimated the wave heights by approx. 30 % for higher sea states. Taking this into account the authors found that the encountered wave climate was less harsh than given in the wave scatter diagrams for North-Atlantic and World Wide service and conclude that some routing must have been applied by the ship's

master. Based on the monitored ship speed and revolution rate also an estimate of the voluntary and involuntary speed reduction in head seas is provided. To illustrate that considerable fatigue damage can be generated in comparatively short time the authors defined the 'fatigue damage rate' as the ratio of the actually measured half hour damage to the average half hour damage corresponding to a 20-year design life time. During the worst half hour record the fatigue damage rate reached approx. 60 (HF contribution 25 %) and on average during the monitoring period a value of 1.6 was obtained. Acknowledging that the latter value does not contain measurements during the summer period the authors do not conclude on an exceedance of the design life time but, nevertheless, reemphasise the need for a proper design of structural details on ships intended to operate in such harsh environments. Also the influence of LF and HF torsion and horizontal bending of the hull girder is addressed in this reference. The HF damage contribution from the corresponding global vibration modes turned out to be negligible.

Regarding the nominal extreme minimum (sagging) and maximum (hogging) stresses at mid ship section the authors estimated that for smaller wave heights the HF amplitudes are approximately 25% of the total stress amplitudes and that this magnitude increases for larger waves. During the measurements on head sea courses a maximum significant wave height of approx. 6.5 m occurred. The maximum measured total hogging stress amplitude was 98 N/mm^2 (LF part 75 N/mm^2) and the minimum total sagging stress amplitude was -109 N/mm^2 (LF part -68 N/mm^2). For the aft quarter length the corresponding values were 53 N/mm2 (LF part 36 N/mm^2) and -57 N/mm^2 (LF part -41 N/mm^2), respectively. From an analysis of the time series of the ship's operational parameters, motions and stresses it was found that the event with the highest HF stresses was characterised by large pitching motions of the vessel (approx. 4°) and that the ship speed reduced by 2 knots within 25 seconds due to the added resistance caused by the larger waves.

The authors also report on measurements from November 2007 to May 2008 on the 4400 TEU vessel operating in the North Atlantic for which a previous measurement period is described in Storhaug and Moe (2007). The average fatigue rate during this measurement period was 2.2 and the contribution from the HF part about 25%. As these results are tentatively very similar to those obtained on the 2800 TEU vessel operating in an identical trade the authors suppose that a significant influence of the ship size on the HF contribution to fatigue damage is not given. For this vessel also the local and global stresses of a side shell longitudinal stiffener were measured corresponding to a total accumulated damage of 0.35 for the measurement period. However, because the vessel is in service since five winter seasons and no damage has been observed so far the authors have doubts on the applied method for superposition of the global and local stresses. The most extreme whipping event occurred in a sea state with a significant wave height of approx. 8.5 m and the maximum mid ship hogging stress amplitude was 128 N/mm^2 (LF part 83 N/mm^2) whereas a minimum total sagging stress amplitude of -146 N/mm^2 (LF part -83 N/mm^2) occurred. In summary, the authors conclude that container ships in comparison to blunt form vessels seem to be less severely exposed to additional HF fatigue damage but that stress magnification due to HF vibration in extreme events tends to be more pronounced.

Kahl and Menzel (2008) reported on the first results of a monitoring campaign which

started in 2007 onboard a 4600 TEU container carrier operating in the North Atlantic and the Pacific Ocean. From the first results they report on a similar trend as reported in Okada *et al* (2006) on the contribution from LF and HF components to the overall longitudinal bending stress amplitudes. They further report that no relevant HF contributions were observed with regard to the global stresses resulting from horizontal hull bending and hull torsion.
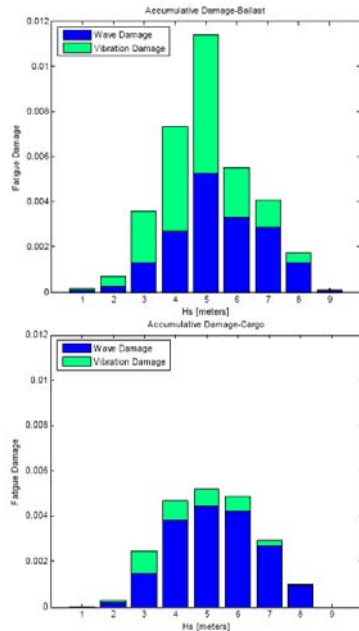

Figure 2: LF and HF fatigue damage in ballast (left) and cargo (right) loading condition, Moe *et al* (2005)
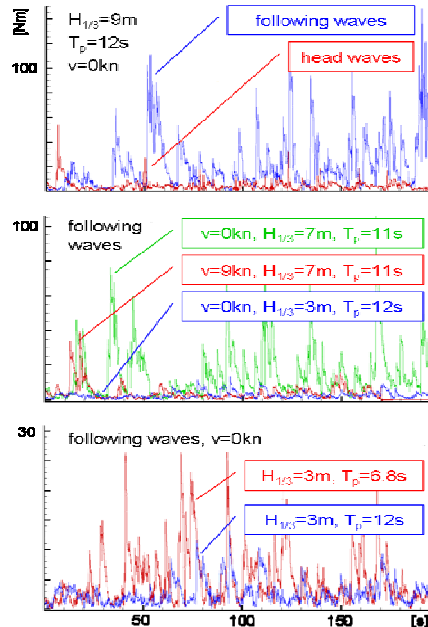
Figure 3: HF bending moment time series for a cruise ship in irregular sea, based on Dessi *et al* (2007)

2.1.2   Model Tests

Full scale measurements show that significant wave excited hull girder vibration may occur and that it can be important for the hull performance, in particular for vessels operated in harsh environmental conditions. Full scale measurements do, however, provide limited information about the physical phenomena causing the vibration and the parameters of the ship design, ship operation and the environmental conditions that affect it. This, along with the uncertainties related to the measurement of environmental conditions and operational parameters, limits the usability of full scale measurement results for the development and validation of numerical simulation tools which can be used for the prediction of sea way excited ship vibration in the ship design phase.

Model tests are perhaps the most important and reliable means for scientific examination of the mechanisms causing the response of ships in waves, although the

uncertainties can be large also here. For hull girder vibrations model tests are carried out with a flexible ship model. This introduces an additional element of uncertainty in the scaling of the experiment, as it is important to maintain the dynamic relationship between the excitation and the structural response between full scale and model scale.

Researchers use continuous elastic models, segmented models with continuous elastic backbones, or models consisting of stiff segments connected by flexible joints allowing for the deformation at discrete positions along the hull. Discrete stiffness segmented models and elastic backbone segmented models appears to be the most popular options, e.g. Dessi and Mariani (2006), Cusano *et al* (2007b), Drummen (2008), although some model tests based on continuous elastic models have also been encountered in the literature (Kinoshita *et al* (2006), Minami *et al* (2006)). ISSC (2006) gives a summary of the benefits and the limitations of the various model types.

Model tank tests on the whipping response of a 290 m cruise vessel have been conducted by Dessi and Mariani (2006) and Dessi *et al* (2007). The tests were carried out in head and following sea conditions with a segmented model with an elastic back bone tuned to the natural frequency of the 2-node vertical bending vibration mode of approx. 0.9 Hz (full scale). The beam bending moment was measured in ten sections whereas one point allowed direct sensing of the shear force. Irregular sea conditions with significant wave heights ($H_{1/3}$) between 1.5 m and 9 m and peak wave periods (T_p) ranging from 6.8 s to 12 s at ship speeds from zero to15 knots were considered. The tests confirmed full scale observations that for such kind of vessel strong whipping vibration can be excited by stern slamming caused by following seas at ship speeds below 6 knots and wave heights above 3 m. Bow slamming induced whipping for head seas is reported to increase for higher speeds and wave heights. The authors found that the exceedance of a certain threshold value of the dynamic hull bending moment correlated with the measured maxima of the local draft variations at the stern and the bow and that the total sagging bending moment resulting from LF and HF wave contributions was larger than the corresponding hogging moment.

By applying a high pass filter to the measured vertical bending moment time series the authors obtained the HF contribution to the envelopes of the vertical bending moment as presented in Figure 3. Some interesting conclusions can be drawn from these evaluations: i) for a severe sea state with $H_{1/3}$= 9m and T_p=12s the HF part due to stern slamming in following waves is much larger than this caused by bow slamming in head seas (upper graph), ii) for zero speed and following waves the induced bending moment abruptly increases if $H_{1/3}$, and thus wave steepness, reach a critical level (centre graph blue/green), iii) increasing forward speed drastically reduces the risk of stern slamming (centre graph blue/red) and iv) for comparatively small wave heights the probability of stern slamming due to short waves is larger than for long waves (lower graph). The occurrence of springing phenomena is not reported.

Also Cusano *et al* (2007a, 2007b) conducted model tests for a large cruise ship. They used a 4-segment model with discrete flexible joints. The main purpose of their

investigations was to gain experimental data which could be compared against the results of theoretical methods. They considered the effect of slamming induced HF vibrations on the LF mid ship bending moment and on the LF acceleration levels in the aft part of the vessel for head seas and following waves, respectively. With respect to the midship bending moments in head seas they found that the LF and the HF contributions can be relatively well described by a Weibull-type distribution. However, this turned out to be not true for the total moment superimposed from both parts because the phase lag between LF and HF part was found to be varying, i.e. the total measured moment was in most, but not all, cases considerably smaller than the algebraic sum of the LF and HF components.

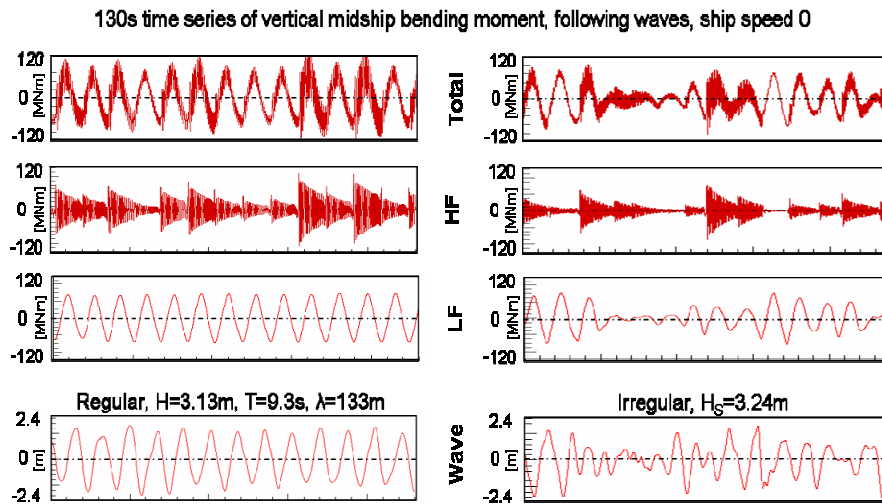**130s time series of vertical midship bending moment, following waves, ship speed 0**

Figure 4: LF and HF part of stern slamming induced bending moment in regular and irregular sea way, based on Luo *et al* (2007a)

A 2-segment model of a vessel of 121 m length and 17.4 m breadth with very flat aft body lines and practically no bow flare was subject to experimental investigations by Luo *et al* (2007a). Tests were conducted in regular head and following waves of varying height for ship speeds of 0 and 5 knots. Based on comparisons with calculations with a linear strip method the authors concluded that stern slamming is not significantly affecting the ship motions but that a magnification of the mid ship bending moment will occur. To quantify this effect they separated the measured bending moments into their LF and HF part by filtering and concluded that the magnification could amount to 44 % in regular waves and to 43 % in irregular waves, see for instance the excerpts of the time series presented in Figure 4 (measured model scale values have been transformed to corresponding full scale data). The authors concluded that the effects from stern slamming induced whipping vibration can become relevant with regard to structural fatigue and crew comfort for vessels with such flat aft body shapes. Different to Cusano *et al* (2007a, 2007b) they found that the total bending moment corresponded to a Weibull distribution and, therefore, they are deeming it justified that

such model tank test data can be used for the prediction of the short-term extreme value of the vertical bending moment.

Lavroff *et al*. (2007) report on their scaling procedure regarding the stiffness, mass and damping properties and the natural frequency (2.1 Hz for the full scale 2-node vertical vibration mode) for a 3-segment model of a 112 m high speed catamaran. In contrast to their experience from full scale tests they found in the experiments an increase of the damping ratio from zero to full speed of approx. 1%. However, for full speed they obtained a damping ratio about 2.5% in the model tests what corresponded quite well to the full scale value of 3% as measured on shorter hulls.

Storhaug and Moan (2006, 2007a, 2007b) reported on extensive model tests which were conducted for the same 220.000 dwt ore carrier that has been subject to the full scale measurements described by Moe *et al* (2005). The main objective of their investigations was to identify the vibration excitation mechanism causing the comparatively high vibration levels as measured in full scale ballast draught condition, i.e. whether impulsive slamming loads or periodic linear or non-linear wave excitations were the origin of the measured hull vibrations, respectively. They opted for a small model scale (1:35) in order to limit the uncertainties regarding the simulation of short waves typically exciting linear springing response. The authors pointed out that several non-linear effects might influence the strength of the vibration excitation from waves: i) the steady wave elevation at the bow due to the forward speed of the vessel ii) the amplification of the height of the incident waves due to wave reflection at the blunt bow, iii) bow slamming forces and iv) the generation of periodic sum frequency springing excitations due to superposition of incident and reflected waves in the bow region. Consequently, the testing program was chosen to consider all these effects. As the vibration characteristics during full scale measurements had shown significant differences in ballast and cargo condition also the model tests were conducted for both conditions.

From measurements in regular waves the authors concluded on the relative importance of linear, non-linear and HF contributions to the midship bending moment. For ballast draught it was found that i) the 1st to 5th harmonic of the wave encounter frequency could be observed in the measured response (corresponding to full scale wave lengths of 28, 112, 253, 450 and 703 m, respectively), ii) HF contributions dominated the response in 1st, 2nd and 3rd order with the 2nd order giving the strongest excitation, iii) linear springing could not explain the pronounced vibration levels which had been measured in full scale, i.e. there must have been contributions from periodic sum frequency excitations and slamming impulses iv) the clearly highest response originated from the LF wave loads which included significant non-linear contributions and v) ship motions were not affected by the HF vibrations.

The authors also introduced a criterion to differentiate whether the measured response was of springing or whipping type, which is based on the response amplitude growth of the first vibration cycles. For this vessel springing appeared to dominate the hull girder

vibration for wave heights below approx. 4 m but the importance of whipping clearly increased for higher waves. However, the criterion is said to need further improvement.

Interaction between the stationary wave and the incoming waves can also cause unsteady waves at the front of the bow that may break and cause stem flare slamming. The stem flare slamming was shown to cause little vibration in ballast condition, but to be significant in full load condition. Generally, the authors relate the differences regarding the vibration behaviour between the ballast and the loaded condition mainly to the difference in draft. They demonstrated the sensitivity of the vibration characteristics to the draft condition by performing further experiments with a moderate forward trim provided to the model and could achieve by this measure considerably reduced vibration response.

Interestingly, whipping excitations in ballast condition are reported to be associated with bilge flare slamming events. This kind of impulsive excitation is said to become effective during upward motion of the bow or downward flow of the water, respectively, both creating low pressure areas below the bow integrating to an impulsive downward pull rather than an upward slamming force. This effect appears to be amplified by the presence of the stationary wave around the vessel, as the water line at the sides of the bow will be lowered compared to the still water level.

For the estimation of the fatigue damage of a butt weld located in the midship area the authors simulated representative North Atlantic sea way. Considering involuntary speed reduction due to the added resistance in head waves by estimation formulae they succeeded to achieve reasonable agreement of the predicted fatigue life with that observed in full scale. They performed also a comparative investigation regarding the effect of the bow shape on the predicted fatigue life. For this purpose the experiments where repeated with two other bow shapes: bow no. 2 without bulb and stem flare and bulb no. 3 with a sharp vertical stem and vertical sides. For bulb no. 2 the HF fatigue damage slightly increased and for bulb no. 3 a reduction of fatigue due to waves of small height was observed. From this the authors concluded that even for the slender bow 3 some sum frequency springing was present and that whipping was also not significantly reduced.

Minami *et al* (2006) and Kinoshita et. al (2006) studied the loads acting on a post panmax container vessel in freak waves of up to 39 m height by means of an elastic continuous model. Interestingly, whipping vibration was not excited by the impact of the wave front on the bow but by the combined bottom and bow flare slamming for the ship pitching in the following wave trough. A significant influence of the ship speed and of the height of the regular waves on the whipping response could not be identified. Only the sagging midship bending moment increased proportionally with the height of the freak wave, whereas the hogging moment remained more or less constant, independently of the freak wave height and shape.

Drummen *et al* (2006) and Drummen *et al* (2008a) reported on model test for the same

4400 TEU vessel for which full scale measurements were described in Storhaug and Moe (2007) and Storhaug and Heggelund (2008). They were using a 1/45 scale 4-segment model of the vessel tuned to the natural frequency of the 2-node vertical hull girder vibration (0.56 Hz in full scale). The test program included 16 irregular sea states with the ship operating in head waves with significant wave heights between 3 m and 9 m. The main fatigue damage occurred for sea states with a peak period of around 14 s and a significant wave height of 5 m. This was true for the LF as well as for the HF damage. From the decaying curve of the 2-node bending vibration the authors measured damping ratios of approx. 0.2 % and 0.6 % in air and water, respectively. Because this was significantly less than observed in full scale (1.7 %) they studied the sensitivity of the HF fatigue on the damping ratio by nonlinear time-domain computations. For a structural detail arranged in the midship area it was estimated that the HF fatigue predicted based on the model tests would reduce by 35 % if the full scale damping value would have been reflected in the experiments. They attribute the higher damping in full scale mainly to the container loading and lashing. On the other hand they found that a 10% global stiffness increase would result in a HF fatigue damage reduction of 3 % only.

Drummen and Moan (2007) used the same model to investigate the application of response conditioned waves for the experimental prediction of the long term statistical distribution of the vertical hull girder bending moment. Compared with the results obtained for random irregular waves good agreement was found for a rigid model but for a flexible one the conditioned wave method gave approx. 15% smaller vertical bending moments. It is reported that hull flexibility was found to increase the 1 in 10.000 years vertical hogging bending moment by approx. 20%. However, the authors pointed out that his value should be considered as an upper limit because it was based on experiments with a vessel operating in head seas only.

Dallinga (2007) performed model tests with a flexible model of a modern container vessel to obtain an impression of the effects of wave condition, ship heading and speed on the statistical character of the joint statistics of the LF and HF accelerations. The author claims that these effects can be estimated reasonably well with a single parameter: the product of forward speed and the square of the root mean square of the relative wave elevation at the bow. Depending on the prudence of the ship's master, whipping is said to increase the vertical acceleration level resulting from LF loads by 25 to 50% and that voluntary speed reduction is quite effective in this respect.

El Moctar *et al* (2008) described model tests performed for a 13000TEU vessel. The 4-segment 1:73.2 scale model was tuned to reflect the natural frequencies of the 2-, 3- and 4-node vertical hull vibration mode (full scale 0.34, 0.84 and 1.23 Hz, respectively). Global bending moments, shear forces, accelerations and hull pressures were measured for comparison with numerical predictions. Test results are presented for test runs in regular waves for following wave height / ship speed / heading combinations: 19.6 m / 0 kn / following waves, 28.2 m / 0 kn / head waves, 18.0 m / 16.7 kn / head waves and 19.0 m / 17.3 kn / head waves. Between measurements and computations good and

reasonable agreement was found for LF and HF response, respectively.

2.1.3 Analysis Methods

In contrast to full scale or model scale measurements, in numerical analyses whipping and springing vibration can be clearly distinguished. Because the only difference between whipping and springing originates from the excitation mechanism, the hull girder model used for the prediction of the dynamic hull response can be the same for both kinds of wave induced vibration.

Hull structural response model

Generally, the structural model must reflect the hull's dynamic properties in the frequency range of interest, i.e. the natural frequencies, the associated mode shapes and, last but not least, the damping characteristics. Depending on the application a variety of methods is used for this purpose:

a)   dynamic amplification factors in combination with quasi-static calculations,
b)   analytical formulae valid for impulsively or harmonically excited vibrations of 1- or 2-DOF systems,
c)   Timoshenko beam FE models reflecting one or several hull girder vertical bending modes,
d)   as c) but extended to simulate also torsional and horizontal bending vibration modes,
e)   3D FE models of the complete hull for more complex hull structures,
f)   as e) but with local FE mesh refinements for specific assessment purposes, e.g. stress concentration effects or local deck panel vibration.

The prediction methods for wave induced vibration can be classified in coupled (hydroelastic) and decoupled approaches. Decoupled methods neglect the influences of hull elasticity and hull vibration displacements on the magnitude of the impulsive or harmonic hydrodynamic pressure excitation forces. The independent computation of wave excitation forces and the resulting structural response allows the use of conventional, effective and well established methods but might compromise the prediction accuracy in some cases. It should also be observed that only linear, but not higher-order springing excitation forces, can be captured by decoupled methods.

Coupling between hydrodynamic analysis and structural response calculation is normally realised by integrating the structural model into the hydrodynamic excitation force analysis. Recent publications report on using hull models according c), d), and e) for this purpose. In most applications the hull FE models are replaced by modal representations for the sake of computation efficiency. Normally, only a few of the first natural modes of the hull girder are required to describe the dynamic response of the hull with sufficient accuracy.

Cusano *et al* (2007b) compared different methods for the prediction of bow slamming excited whipping vibration with experimental results for a cruise vessel. In a first step the bow slamming loads were calculated by a 2D BEM method. For a typical bow slamming event they obtained a maximum slamming force of approx. 5000 kN with an impulse duration between 1.2 s and 1.5 s. Then they applied the impulse to vibration analysis models of varying sophistication: i) an analytical one DOF approach ii) a Timoshenko beam FE model and iii) a full 3D FE model. While the former models were used for decoupled calculations also a hydroelastic analysis was performed with the ship hull represented as a Timoshenko beam. A comparison of the calculated time histories with the experimental results is presented in Fig. 6. Apparently, the simplest approach gave the best agreement.

The authors also compared the results obtained for a stern slamming event with i) a hydroelastic analysis using a Timoshenko beam model of the hull and ii) a decoupled analysis with the ship represented as a 3D FE model. As the predicted midship bending moments differed insignificantly only, they concluded that the application of decoupled methods for the sake of calculation efficiency is justified for the analysis of slamming excited whipping vibration. It is worth to note that this reference also reports on transient wave induced vibration of deck panel structures. These were predicted by applying the calculated stern slamming impulse to a hull 3D FE model which also reflected the dynamic properties of the deck panel structures. The magnitude of the excited transient deck vibrations with respect to passenger comfort was assessed based on a crest factor method.

Senjanovic *et al* (2007) reported on their method to model hull girder torsional and horizontal bending vibrations by means of a FE beam model. The sectional properties of the beams were calculated with analytical formulae which the authors deduced for ship hulls with large deck openings. For verification they compared the natural vibration characteristics they obtained with their beam approach and a 3D FE model for a pontoon-like structure with container ship cross section and achieved excellent agreement. The authors also qualitatively studied the effect of shifting the superstructure arrangement from the aft quarter length to midship section, as encountered for the latest Ultra Large Container Carrier (ULCC) designs, for instance. They reported a significant increase of the natural frequency of the basic torsional hull vibration mode due to the suppression of warping deflections in way of the superstructure and conclude that the risk of resonance between torsional vibration modes and wave encounter frequencies with relevant energy content is reduced compared to container vessels of conventional design. In Senjanovic *et al* (2008a) it is reported how the previously described beam modelling method was applied to a flexible barge. By this investigation the authors confirmed that their beam model is realistically reflecting the natural vibration characteristics of coupled horizontal bending / torsional modes even for extremely flexible hull girders with full scale natural frequencies in the range of 0.1 Hz (this value was estimated from the given natural frequency of 0.8 Hz of the 2-node vertical bending vibration of the dry barge model and assuming a reduction of the lowest natural frequency due to hydrodynamic

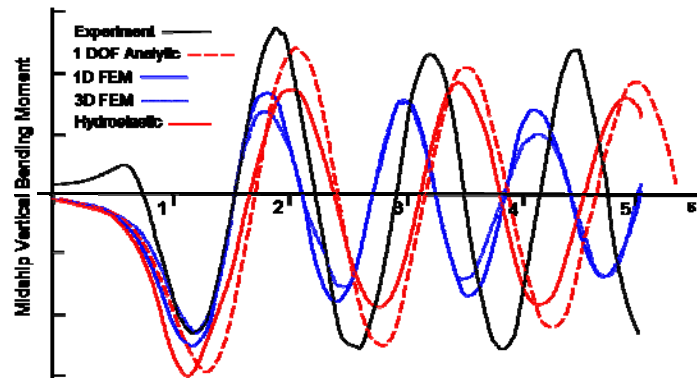mass by 30% and a geometric scale factor of 36).

Figure 5: Time histories of vertical bend. mom. of cruise vessel in head waves, based on Cusano *et al* (2007)

Whipping vibration analysis

The magnitude of hull whipping response to slamming impulses mainly depends on the strength and location of the slamming impulse. Methods for the calculation of slamming impulses are discussed in Committee V.7 and, therefore, will be not addressed in depth here.

The ratio of the impulse duration to the natural period of the relevant hull girder vibration mode and the shape of the time history of the impulsive force determine the grade of dynamic magnification of the quasi-static response. In structural dynamics the dynamic amplification magnification factor (DAF) is often used to estimate the dynamic response from the quasi-static response to avoid elaborate time domain transient response computations. Theoretically, the DAF can reach a maximum value of 2.0, but for slamming excited hull girder vibration values between approximately 1.1 and 1.4 are more typical. However, the DAF does not consider the phase relationship between LF hull deflections due to waves and the HF elastic hull vibrations. This can only be accomplished by computing the hydrodynamic wave and slamming pressures simultaneously and applying the resulting pressure time histories to the structural response model. In order to point out the difference to the fully decoupled approaches, as, e.g., the DAF concept, some researchers use the term '1-way coupling' for such a procedure as long as hydrodynamic and structural response analysis are performed independently, and '2-way coupling', if also these computations are performed simultaneously, i.e. a hydroelastic analysis is performed.

Also the calculation of ship motions, wave loads and slamming pressures can be performed with simple and efficient or more elaborate and accurate methods. In combination with the different ways to perform the structural response prediction and to couple hydrodynamic and structural analysis, there exists a number of options for the

definition of a meaningful overall analysis procedure, see Table 3. In practice, there will be always a need for a compromise between the accuracy of the respective approach and its computational efficiency. It goes without saying that the choice of the most suitable overall procedure will depend on the objective of the analysis, e.g., it must be differentiated between extreme load scenarios, the computation of stress range spectra or the prediction of design values based on long-term statistics.

Table 3
Alternative methods/models used in the hydrodynamic and structural Analysis of Hull whipping vibrations

Hydrodynamic Excitation Load Analysis		Coupling	Structural Response Analysis
Ship Motions & Wave Pressures	Slamming Pressures		
A Non-linear 2D strip methods	**A** Analytical acc. impact and momentum theory	**A** decoupled	**A** FE Timoshenko beam reflecting vertical vibration modes
	B Non-linear 2D strip methods	**B** 1-way	**B** FE beam reflecting vertical, torsional and horizontal modes
B 3D potential Flow BEM methods	**C** 3D potential flow BEM methods	**C** 2-way	**C** 3D FE model
	D 3D RANSE methods		**D, E, F** Modal coordinates deduced from **A**, **B** and **C**, respectively

Considering the diversity of whipping analysis methods it does not surprise that there remains large uncertainty regarding the accuracy which can be obtained. This issue was also addressed in MAIB (2008) reporting on the collapse of the 4400TEU container vessel 'MSC Napoli'. Amongst others, the report addressed the question, whether the occurrence of whipping vibration might have been a decisive contributing factor for the incident and followed that no clear conclusion could be drawn from the several expert opinions:

'However, from the different results obtained......it is apparent that whipping effect is currently very difficult to reliably calculate or model. Classification societies are therefore unable to predict its magnitude or effect on a ship's structure, with any confidence, and as a consequence they are not generally calculated during the structural design process.'

Considering that this refers to a case where whipping response had to be calculated for reasonably well known environmental and operational parameters, one might imagine the challenge to compute the whipping response for a large number of environmental and operational conditions as required for the determination of design values based on long term statistical distributions. For this computational effective methods are required, however, due to the involved simplifications they might miss important physical effects,

as increased damping due to viscous flow effects, for instance. Also this issue is addressed in MAIB (2008):

'Possible whipping effects …....have not been assessed theoretically.....because reliable methods and analyses tools for such evaluations are not available. This status is mainly due to the complexity and randomness of the problem and the theories and software tools are accordingly not fully developed and tested.'

It appears that the numerical determination of design values for fatigue and extreme loads under consideration of whipping effects and their validation by full scale measurements can be supposed to remain a challenge for the coming years or even decades. The following discussion of recent publications on whipping vibration analysis should be seen in this spirit.

Park *et al* (2005) presented a decoupled time domain method for the estimation of hull whipping response in irregular seas. They used a strip theory hydrodynamic model, a FE Timoshenko beam and a momentum slamming theory for the calculation of the impact loads. The method was applied to the example of a 175.000 dwt ore carrier operating in head seas ($H_{1/3} = 12$ m). At the instance of maximum wave height (19 m) the total midship bending moment was predicted to amount to two times and 80 % of the rule based design moment for ship speeds of 15 kn and 0 kn, respectively. By neglecting the terms for consideration of the slamming forces and hull flexibility the authors obtained a reduction of the predicted bending moment by 50 %. The authors pointed out that their method gave reasonable results but needs further validation by comparison with other numerical methods or experiments.

Fonseca *et al* (2006) used a decoupled method to study the whipping response of a frigate (length 125 m, displacement 4655 tons) advancing in large regular head waves. The full scale natural frequencies of the vessel were 1.4 Hz, 2.7 Hz and 4.5 Hz for the 2−, 3−, and 4-node vertical hull vibration mode, respectively. In a first step they calculated the rigid body motions of the vessel by a nonlinear time domain strip method. Afterwards they computed the bottom slamming loads by impact theory, the bow flare slamming loads by the momentum method and the resulting transient response by using the vibration mode shapes as determined with a FE Timoshenko beam representation of the hull girder. To validate their method they made comparisons with experimental results which had been conducted with a flexible model. Regarding the pitching motions very good agreement was achieved but the combined LF/HF midship bending moment was significantly overestimated by the numerical approach. The authors suppose that the negligence of 3D flow effects, nonlinear influences of the forward speed or viscous effects might have been reasons for this and recommended further research for improvement of the calculation method.

El Moctar *et al* (2006) used a 3D potential flow BEM approach, a RANSE method and a 3D FE model of a 13000TEU vessel to compute the hull response in extreme whipping scenarios with a 1-way coupling method. At first they selected approx. 25

design wave scenarios from several thousand load cases which they analysed by an efficient potential flow BEM method in the frequency domain. Subsequently, a RANSE solver was applied to obtain for the selected design wave scenarios the nonlinear ship motions and the corresponding pressure distributions in the time domain. The authors claim that in this way all significant wave load effects could be considered: bottom and bow flare slamming loads, the effect of green water on deck and 3D- and viscous flow influences. Most severe bow flare slamming occurred for the vessel advancing with 2/3 design speed in regular design head waves of 400 m length and 19 m height. For illustration, the time point of the simulation when the forecastle deck is entering the wave surface is presented in Figure 6. As the hydrodynamic simulations were performed assuming a rigid ship, the computed pressures represent an upper bound of the real ones but, on the other hand, the magnification of the hull sectional loads due to HF hull vibration is disregarded in such a decoupled approach. Therefore, in a 3rd step, the hydrodynamic pressures were integrated to nodal forces and applied time step-wise to the 3D FE model of the hull for transient vibration analysis. A maximum relative deflection between amidships and the bow of approx. 1.0 m was computed for this extreme scenario. The corresponding distribution of the van Mises stress levels in the shell elements representing the hull structure at this time point is presented in Figure 7 (highest stresses with red shading). By comparison with results from quasi static analyses the authors concluded that the midship bending moments and the related stresses in the longitudinal hatch coaming increased by up to 20 % due to hull girder whipping vibration.
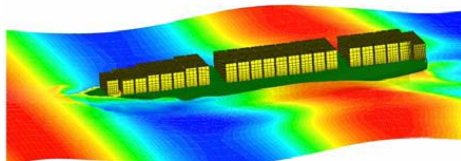

Figure 6: Time step of RANSE time domain analysis of 13000TEU vessel advancing / pitching in regular head waves, from el Moctar *et al* (2006)
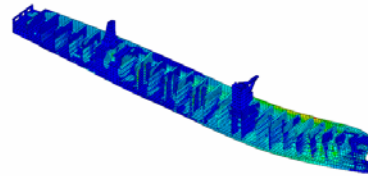
Figure 7: Time step of maximum vertical bow deflection with shading acc. v. Mises stress distribution, from el Moctar *et al* (2006)

Luo *et al* (2007b) used a decoupled approach to calculate the whipping response to stern slamming impulses from regular waves and made comparisons to the experimental results which they had obtained for the same vessel, see Luo *et al* (2006). They justify the usage of a decoupled method by arguing that the coupling between rigid body ship motions and the elastic hull vibration is small for conventional monohull ships. In a first step they used a nonlinear strip theory to calculate the rigid body ship motions and the slamming forces per ship length unit in the time domain. Then they applied these force histories to a beam FE model of the hull and performed a transient vibration analysis using the lowest the vertical hull vibration modes as generalised coordinates. It is reported that the calculated total vertical bending

moments were found to be in good agreement with the experiments. From parameter investigations the authors did draw following conclusions: i) the damping factor did not have much influence on the maximum vibratory response, ii) the period of the slamming impulse was between 0.26 and 0.36 s corresponding to dynamic amplification factors between 1.25 and 1.40 for the given natural period of the 2-node hull girder vibration iii) whipping response to stern slamming reduces drastically with increasing ship speed and iv) aft trim will reduce stern slamming and, hence, whipping vibrations.

Malenica and Tuitman (2008) reported on their hydroelastic method which couples a 3D BEM hydrodynamic formulation to a 3D FEM structural response model. The code is capable of linear springing analysis in the frequency domain, and non-linear whipping simulations in the time domain. The 3D FEM model of the ship allows for direct analysis of the response of structural details in the hull structure. For both, the linear frequency domain and the non-linear time domain analyses, the total response of the hull structure is separated into a resonant HF part and a quasi-static LF part that are solved separately. A modal superposition method involving some of the fundamental natural modes of the hull girder is used to obtain the dynamic response. As an example the method was applied to whipping simulations of a 11400TEU container ship advancing in irregular head waves ($H_{1/3} = 5.0$ m, characteristic wave period 16.0 s) with 19.5 knots. For a typical stress time history during a whipping event in this moderate seaway the LF stress range of approximately 40 N/mm^2 was found to be superimposed by a HF stress range of maximum 11 N/mm^2. The authors concluded that their method produced realistic results but that further validations and calibrations are necessary before it can be used in practice.

De Jong and Huther (2008) presented computational methodologies for i) the calculation of extreme hull girder loads resulting from superimposed LF wave loads and whipping vibration in extreme wave scenarios, ii) the estimation of whipping induced fatigue damage, iii) the prediction of springing vibrations and for iv) the estimation of fatigue damage resulting from springing vibrations. They applied their methods to large and ultra large container vessels and concluded that the magnification of the extreme LF vertical bending moment due to whipping vibration is expected to be 20 to 30% and that the increase of fatigue damage due to whipping and springing vibrations might be expected between 3% to 5% and 4% to 10%, respectively. In order to prepare for the development of even larger container ships they recommend further validations by full scale measurements, continued enhancement of the prediction capabilities and greater attention on hydroelastic wave load effects.

El Moctar et al (2008) reported on their validation of a hydroelastic CFD method for the prediction of whipping response by comparison with model experiments for a 13000TEU vessel. Simulations and experiments were conducted for the vessel exposed to equivalent regular design waves in stern and bow slamming scenarios. For the computations they used the same RANSE solver as in El Moctar et al (2006) but extended it by a Timoshenko beam model for consideration of vertical hull bending

vibrations. Hull girder accelerations, shear forces and bending moments were computed for a flexible and a rigid hull structure and compared with model test results. The calculated vertical acceleration at the bow and the midship bending moment for the vessel advancing with 16.7 kn in a regular wave of 18 m height and 306 m length are presented in Figure 8 for the rigid and the flexible case. Apparently, bow acceleration response is more affected by the 3- and 4-node hull vibration mode than the midship bending moment which is modulated by the 2-node mode only. It can also be seen that the dynamic magnification of the LF response by HF vibration is much more pronounced for the bow acceleration than for the midship bending moment.

Also Drummen *et al* (2008a) compared numerical and experimental results. They used a non-linear 2D strip theory code coupled to a Timoshenko beam model predict the dynamic response of a 4400TEU container vessel in a natural sea way. They focussed on the fatigue damage rather than on extreme wave scenarios. They splitted the computed stress response into its LF and HF part and further separated the linear HF contribution from the non-linear ones. They found that the HF fatigue damage was strongly overestimated by the calculations in comparison to the experiments. They attribute this to i) an overestimation of the slamming force by the 2D calculation method, ii) additional damping effects which have not been captured in the applied structural damping value of 0.2% and iii) the effect of the steady wave elevation at the stern and bow due to the ship's forward speed. To quantify these influence the authors performed parameter investigations from which they concluded that i) 2D methods overestimate the slamming force by approx. 20%, ii) increase of structural damping from 0.2% to 1.0% (as in the author's opinion typically encountered in full scale measurements) resulted in a 37% decrease of HF fatigue damage and iii) consideration of the steady wave elevation reduced HF fatigue damage by approx. 40% but increased LF fatigue. Another parameter study demonstrated that a 10% increase of hull girder bending stiffness resulted in a reduction of the HF fatigue of 3% only.

Springing vibration analysis

The magnitude of hull springing response mainly depends on following parameters:

- the strength of the periodic excitation forces, as normally expressed in terms of harmonic excitation orders and corresponding force amplitude and phasing,
- the degree of coincidence between the excitation frequencies and the natural frequencies of the relevant hull girder vibration modes,
- the similarity of the deflection shapes caused by the excitation forces with the mode shapes of natural hull girder vibration,
- and last but not least, the overall damping of hull vibration resulting from fluid damping, material damping, structural damping and cargo damping.

Springing excitations are normally classified according to the generated excitation frequency in multiples of the corresponding wave encounter frequency, $f_e = \omega_e/(2\pi)$, which can be calculated from the ship speed, the angle of wave encounter and the wave

length. Assuming, for instance, a container vessel advancing with 25 knots in regular head waves of 320 m length, the 1st to 4th order springing excitation frequencies will amount to 0.11, 0.22, 0.33 and 0.44 Hz, respectively. Assuming further that the lowest hull girder natural frequency is 0.33Hz there will be a resonance with the 3rd-order excitation frequency of the sea way. On the other hand, head waves with a length of approx. 70 m would generate for the given ship speed and heading 1st order springing excitation forces being in resonance with the natural frequency. This example illustrates that even for the largest vessels presently in service 1st order resonance with sea way of relevant energy content is very unlikely to occur, but that resonant vibration might be excited by the higher (non-linear) excitation orders. In this regard it should be considered that linear hydrodynamic methods can, without specific correction terms, at best predict 1st order excitations. Consequently, non-linear methods must be employed for the calculation of higher order springing excitation effects.

There are existing different hydrodynamic effects which might cause higher order excitations but whether they are practically relevant is yet not fully understood:

- 2nd and 3rd order excitations resulting from non-vertical side walls of flared hull sections,
- higher order excitations due to the occurrence of higher order wave contours compared to the linear Airy wave,
- 2nd order excitations due to the variation of the sectional hydrodynamic mass for sections in wave troughs and crests, respectively,
- 2nd order excitations due to the superposition of incident waves with reflected waves or other wave systems

Almost all researchers use coupled approaches for the analysis of springing vibrations although for linear springing also decoupled approaches could be used.

Hirdaris *et al* (2007) focussed on the effect of springing and whipping vibrations on the midship vertical bending moment (VBM) of a 35.000 tons Great Lakes bulk carrier of 225 m length which was built in 1978. They defined a 'service factor' (SF) representing the ratio of the midship VBM calculated under consideration and omission of hull springing and whipping vibration. Springing response was calculated with a 2D strip method combined with a Timoshenko beam representation reflecting the lower 4 vertical hull vibration modes (2-node mode at 0.62 Hz.). For the sea ways investigated, the authors found a maximum SF of 37% ($H_{1/3}$ = 7.1 m, characteristic wave period 7.0 s, head sea, speed 6 knots), which originated almost totally from springing vibrations. Although for other headings and wave periods considerably smaller SF were found, the authors deemed it appropriate to use this value as a design margin for the possible sea states that might be expected. Also a comparison with full scale measurement results from 1990 was conducted. As can be seen from Figure 10, measurements and calculations agree quite well. However, it should be noted that the damping assumed for these calculations was assumed to be 3 times as high as for the previously described analysis of the SF. It should also be observed that the investigated sea state contained relatively short waves with a wavelength of approx. 50m. Such waves will not create

large LF response for a 220m vessel and so the HF response becomes of relatively more importance.
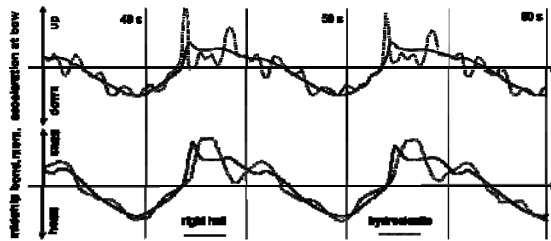

Figure 8: Time histories of 13000TEU vessel in head waves, , based on results of el Moctar *et al* (2008)
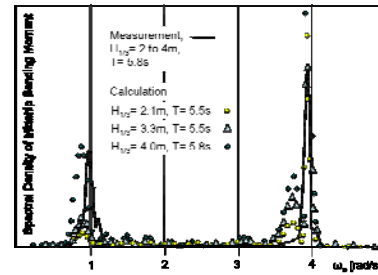

Figure 9: Comparison vib. analysis - full scale measurement, acc. Hirdaris *et al* (2007)

Jung *et al* (2007) investigated springing vibrations with a linear strip theory and a Timoshenko beam representation of the ship hull, a method intended for the application in the initial design stage of a ship. They predicted the effect of springing on the fatigue utilisation of three crude oil tankers of different size (2-node mode natural vibration frequencies ranging from 0.46 to 0.82 Hz) trading either on a World Wide or a North Atlantic route by spectral fatigue analysis. Differently to conventional approaches they used a fatigue damage calculation approach which is said to be applicable to wide band bi-modal stress spectra as typically induced by springing vibration. For a upper deck longitudinal stiffener with a stress concentration factor of 2.2 they predicted fatigue damage ratios (damage due to HF springing vibration divided by damage to LF wave loads) of maximum 3.3% and 3.6% for World Wide and North Atlantic route, respectively. Their results show that the effect of springing is very small for the considered vessels but non-linear effects have not been accounted for in the hydrodynamic model.

Based on their investigations described in Senjanovic *et al* (2007), Senjanovic *et al* (2008b) outlined a methodology for the hydroelastic analysis of wave induced LF and HF ship response. In the first step the dry vibration modes of the ship structure are calculated by FEM, either by using a beam or a full 3D representation. Then the computed mode shapes are used as generalised coordinates in the 3D hydrodynamic analysis, e.g. all quantities of the governing matrix differential equation for ship motions and vibrations are related to the dry mode shapes of the hull. The linear hydrodynamic analysis is performed in the frequency domain for unit wave amplitudes. The analysis method was verified by applying it to an extremely the flexible barge for which model test results were available. As the response amplitude operators (RAOs) obtained from simulations are demonstrated to be in good agreement with those obtained from the experiments, the authors concluded that their methodology and mathematical model are reliable enough to be applied for the hydroelastic analysis of ship structures. On the other hand it should be observed that the validation was performed for a structure with a full scale natural frequency of approx. 0.1 Hz or a

natural period of 10 s, respectively. From this follows that first order springing excitation forces from head seas with wave lengths between 200 m and 300 m are dominant for the dynamic response of the structure, which is not typical for present ship designs.

In Senjanovic *et al* (2008c) it is reported on the hydroelastic analysis of the springing vibrations of a 7800TEU container vessel The natural frequencies of the 2-node vertical vibration mode and the 1-node torsional mode in full load condition were predicted to be 0.48 Hz and 0.32 Hz, respectively, and the hull was represented by a beam model. The calculations were carried out for a situation with the ship advancing with 25 knots in quartering sea. The calculated RAO of the hull torsional moment was found to be much higher in case of resonant springing vibration than in resonant ship motion and the authors strongly advocate further investigations on this issue.

Iijima *et al* (2008) reported on their hydroelastic analysis procedure which is using a 3D modal representation of the lower vertical hull vibration modes as well as of the lower hull torsional and horizontal bending vibration modes. The 3D hydrodynamic analysis was performed with a linear potential flow panel code including non-linear corrections. Slamming forces were not considered. To avoid errors in mapping the hydrodynamic pressures to the structural model the hydrodynamic panels were chosen according to the FE-mesh of the shell surface. Based on time domain simulations in regular head waves and oblique waves the modal response was calculated. From the time histories of the 1st and 2nd modal coordinates (i.e. the basic torsional and 2-node bending vibration mode) the time instances were identified where maximum global response could be expected. For these time steps a quasi-static analysis was performed by mapping the pressure and inertia loads to the 3D FE model. By this the behaviour of structural details under dynamic wave loading could be evaluated. The authors applied their method to the S175 container ship and a 5250TEU vessel and obtained plausible results.

2.2   *Periodic Excitation Sources*

This Section covers periodic excitation sources of stationary ship vibration with the exception of hull springing excited by sea way, which is discussed in Subsection 2.1. The impulsive excitation forces of transient ship vibration are discussed in Section 3 (shock) and in Committees I.2 and V.7 (slamming), respectively. References primarily concerned with the excitation of structure borne noise are discussed in Section 4 and those specific to offshore applications are reviewed in Section 5.

2.2.1   *Propeller Excitation*

When developing a new propeller a designer has to choose between focussing on improved efficiency or on reducing vibration and noise excitation forces. To resolve this dilemma as far as possible Jung *et al*. (2007) used a genetic algorithm to increase the propulsion efficiency of a stock propeller by 1.8% without compromising its

cavitation behaviour. In successive generations the propeller efficiency and cavitation volume were predicted using a vortice lattice method and optimum pitch and camber distributions were found after about 30 generations. Model tests were performed to verify that the gain in propulsive efficiency had been achieved without an increase in propeller pressure pulses. Chen and Shih (2007) also used a genetic algorithm for propeller optimisation. They limited their study to B-Series propellers and used simple analysis methods to minimise computation time. Beside optimisation of efficiency and cavitation behaviour they aimed to minimise fluctuations in propeller shaft forces and moments.

Young and Liu (2007) validated a coupled BEM-FEM method for hydroelastic tailoring of composite propellers. They conclude that for both, steady and unsteady inflow, the composite propeller achieves higher efficiencies than a conventional one, especially at off-design operating conditions, and this can be achieved in combination with an increased cavitation inception speed. The strongly increased 2nd harmonics of vertical and horizontal shaft torque are considered acceptable because their absolute magnitude is reported to be negligible.

Lee *et al* (2006) report on an integrated simulation software for the computation of a ship's wake (CFD method), the propeller pressure pulse excitation forces (panel method) and the associated vibration levels (FE method). They demonstrate the use of such an integrated approach on the example of a vibration level prediction for a VLCC and confirm that beside the magnitude of predicted pressure pulses also the immersion condition of the shell area above the propeller and the phasing of the pressure pulses at different locations must be considered to obtain realistic results.

Van Wijngaarden (2006) addresses the shortcomings of common contract specifications which normally specify maximum values for the pressure amplitude acting on the shell above the propeller to limit the magnitude of propeller vibration excitation forces. The pressure amplitude consists of a contribution due to propeller cavitation dynamics and one due to the blade loading and thickness. The former contribution results in a hull pressure field which is approximately in phase across the after-body and the latter causes a pressure field which shows strongly varying phases. From this the author concludes that the maximum pressure can hardly be used as a measure for the integral excitation force and proposes to use the propeller source strength as a better assessment parameter. He presents an acoustic boundary element inverse scattering method for the determination of the propeller source strength from a set of hull-pressure model tank measurements and proposes to model the source strength by a multiple monopole and dipole distribution for the calculation of the complete pressure field and the corresponding excitation force acting on the hull.

Buannic *et al*. (2007) recommend a similar approach to consider the propeller higher frequency harmonics as well as broadband excitation more realistically. They also use one or several monopole and dipole sources to model the propeller excitation in the analysis. In comparison to Van Wijngaarden (2006), their regression formula for the

source strength is determined from full scale measurements rather than from model tank measurements. The authors separate the propeller excitation into narrow band parts and a broad band contribution and demonstrate a method to predict the vibration response separately and then combine the results to predict an overall vibration level according ISO6954, ed. 2000. However, using deck vibration on a cruise ship as an example they conclude that the contribution of propeller broadband excitation is insignificant compared to the contributions from 1st and 2nd propeller blade harmonic.

Ligtelijn *et al* (2006) studied the reliability of model tests and numerical analyses for the prediction of full scale propeller pressure pulses. They illustrate the (non)correlation for single screw vessels, twin screw arrangements, controllable pitch and ducted propellers and conclude that model tests for the prediction of hull pressures will only be reliable in large test facilities and that even those tend to overestimate the hull pressures for single screw ships by a large amount. On an example of a 5000 TEU container vessel they also conclude that the influence of the stern wave and ship sinkage should be considered in numerical predictions to achieve better agreement with model tank test results, see Figure 10.

The authors expect potential flow lifting surface and panel computational methods to provide reliable results for the propeller blade rate but not for the higher excitation harmonics because vicious flow effects in the propeller tip region are fundamentally important for the formation of the tip vortices. Therefore, they see a large potential for improvement in the growing use of CFD-methods in general practice today. These methods also allow for the realistic prediction of the full scale wake fields. The authors demonstrate the importance of this by comparing pressure pulses calculated for the propeller operating in the model tank wake field and in the full scale wake field with those measured in the model tank test and full scale trials, respectively. As presented in Figure 11 the differences can be quite significant. The authors investigated also some influences of different propulsion arrangements on the propeller excitation. For a fast twin screw passenger ferry it is reported that relocating the rudder 0.23 times the propeller diameter into aft direction reduced the excitation with the 6th blade harmonic but increased the 4th order harmonic. Furthermore, for the same vessel the influence of the direction of rotation of the propellers was studied. For the 1st blade harmonic inward turning propellers generated practically the same excitation level as outward turning ones but for the 2nd and 3rd harmonic an 80% reduction was achieved, however, at the expense of reduced propeller efficiency. Also these authors emphasise that the practice to guarantee a certain pressure fluctuation level does not provide any information on the actual excitation level of the ship, particularly in case of moderately cavitating propellers.
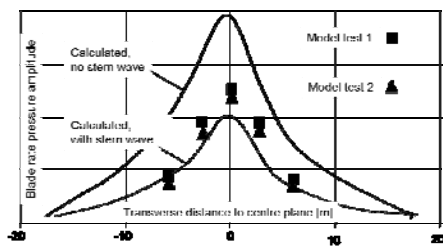
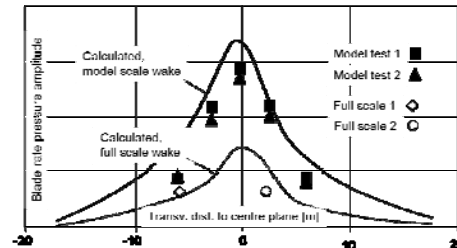Figure 10: Influence of Stern Wave on Predicted Pressure Pulses, Ligtelijn, J.Th. *et al* (2006)

Figure 11: Scale Effect of Wake Field on Pressure Pulse, Ligtelijn, J.Th. *et al* (2006)

Acknowledging the high uncertainty in predicting cavitation patterns by model tank tests, Bodanac *et al* (2005) promote full scale cavitation observations by means of a newly designed observation system. Compared to conventional systems it uses a prism to minimize the camera space needed. A three-level sealing arrangement ensures water tightness and observation and lightening windows can be installed by a diver to avoid dry docking. Two applications of the system on ships with cavitation critical propellers are described.

2.2.2   *Machinery and Shafting Excitation*

Kurth (2006) reports on design concepts for resilient mountings of medium-speed engines for diesel-mechanic and diesel-electric propulsion plants. For very demanding requirements, e.g. of the naval and mega yacht sectors, the engines are supported by very soft elements (Shore hardness 43 to 45) leading to natural frequencies of 2 to 5 Hz, i.e. below the lowest relevant excitation frequency of such engines. However, under these circumstances specific measures must be taken to limit the relatively large motions of the engine caused by large wave induced ship motions. Moreover, the engine crankcase must be designed comparatively stiff because in such an arrangement the ship foundation can not suppress engine deflection effectively. To circumvent these drawbacks a simple solution is to use stiffer mounting elements (Shore hardness 55 to 65) leading to natural frequencies between 4 and 16 Hz, however this is at the expense of much less effective isolation of the 1st order mass moment excitation. The author also demonstrates, using an example of a V12 engine, that the 1st order vertical forces introduced into the ship foundation can be considerably reduced by suitable dynamic balancing. Finally, he illustrates the influence of the aging process of the rubber elements on the example of the Queen Elizabeth II. The rubber elements of her nine propulsion units were exchanged after 10 years operation time and the ship response at several points were measured before and after the retrofit. The improvement is presented in Figure 12. The author concluded from these measurements that the rubber element stiffness increased with time resulting in an undesirable increase in the transfer of the engine forces into the ship foundation.
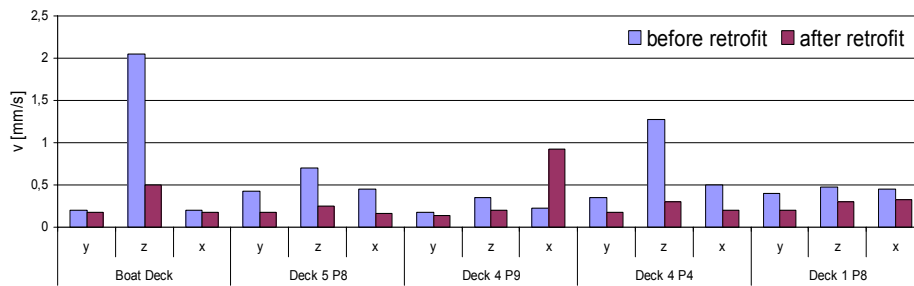
Figure 12: Effect of retrofit of elastic elements (after 10 years) on vibration levels, acc. Kurth (2006)

Martens (2006) reports on the vibration and noise characteristics of mega yacht gears of the medium power range (about 1 to 7 MW). He focuses on measures to reduce the introduction of vibration and noise from the gear into the ship foundation and reports that in this power range semi-elastic and hard elastic rubber elements can be used without compromising the transfer of propeller thrust by a thrust bearing integrated into the gear housing. It is emphasized that attention must be paid to avoid the natural frequencies of the gear vibration on its foundation coinciding with the natural frequencies of the torsional engine crankshaft and propeller shafting vibration. Also resonance between the natural frequency of the gear vibration on its foundation and the rotation frequencies of the gear input and output shaft is recommended to be avoided. The author also suggests that in case of foundations with low stiffness it is not sufficient to consider the ship structure by using discrete stiffness values but that it should be represented by means of a finite element model. Finally, he recommends to avoid revolution rates with impedance minima of the gear's resilient mountings (to minimize the transfer of foundation deflections from the engine to the gear), to properly align the gear with the propeller shafting and to optimise the gear toothing geometry to achieve an optimum vibration and noise characteristic.

Murawski (2007) gives an overview on the influence factors of lateral and axial propulsion shafting and crankshaft vibration for slow-running propulsion plants. Concerning whirling vibrations of the propeller shaft he concludes that the stern tube bearings must be modelled as continuous supports, i.e. not as a point support as is often the case, to obtain reliable results. To illustrate this he computed the oil film pressure distributions in the aft stern tube bearing for three different revolution rates. As presented in Figure 13, the effective supporting point of the propeller shaft moves from the aft bearing edge at slow rpm to a point of about 40% of the bearing width at nominal running condition and this can be supposed to clearly influence free and forced propeller shaft vibration. He reports a similar coupling phenomenon from the thrust bearings integrated into two-stroke engines which may become relevant in predicting the axial force fluctuations introduced by coupled axial-torsional shafting/crankshaft vibration into the ship hull. He points out that in crankshaft resonance condition the oil film damping and stiffness characteristics turn out to be completely different to the

nominal operating condition and that this might lead to oil undersupply. Moreover, he emphasises that use of an axial crankshaft vibration damper might not necessarily result in a decrease of the axial force exciting the ship hull because an axial force will be transmitted also via the support of the damper in the engine's bedplate.
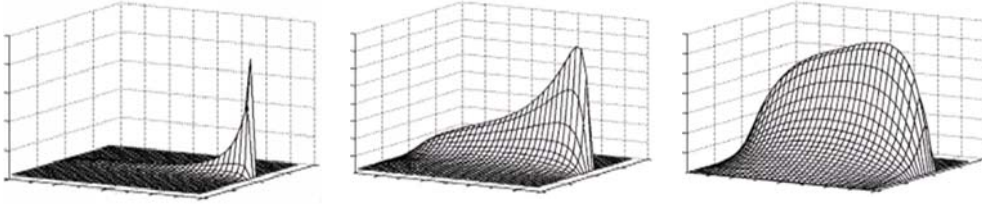

Figure 13: Oil film pressure distribution in aft stern tube bearing for 30, 50 and 100 rpm, Murawski (2007)

2.2.3   Other Excitations

In shipbuilding, in contrast to the offshore industry, flow induced vibration is not investigated in the standard design process. This might be explained by the facts that the environmental and commercial risks arising from pipeline or riser vibration are much higher than those arising from flow induced ship vibration, that relevant flow induced ship vibration seldom occurs and that it can be normally cured by localised countermeasures at reasonable cost. So the publications in the field of shipbuilding normally relate to trouble-shooting cases.

Menzel *et al* (2008) report on such a case where an annoying transverse vibration appeared onboard a mega yacht which could not be assigned to an excitation frequency generated by the propulsion plant. Initial measurements showed that the excitation frequency was dependent on the ship's speed, occurred in a particular rpm-range and coincided with the natural frequency of a transverse or torsional hull vibration mode, see left hand picture of Figure 14. So it could be clearly concluded to be dependent on some kind of flow induced excitation phenomenon but its origin still had to be identified. The authors report on the challenge to set up a program for a second measurement where the sensors had to be placed at those hull appendages or openings which most likely produced the observed vortex induced vibration (VIV) in the frequency range of interest. The combination of CFD and forced vibration simulations made this possible. The approximate vortex shedding frequencies and the corresponding magnitude of pressure pulses acting on the hull were computed by CFD. The forced vibration calculations revealed at which points the hull could be most easily excited by pressure fluctuations integrating to transverse or vertical excitation forces. As can be seen from the left hand picture of Figure 15, the CFD calculations revealed the likelihood of periodic vortex separation at the propeller shafting hull outlets with a frequency of about 3.5 Hz at the critical ship speed. Additionally, the vortices were predicted to be strong enough to create pressure pulsations of relevant magnitude and

the forced vibration analysis indicated a high sensitivity of the first hull torsional vibration mode to vertical excitation forces acting at the outer two of the four shaft outlets. Consequently, it was decided to focus on this area in the second measurement with the arrangement of the pressure and acceleration sensors. Indeed a clear correlation of the pressure fluctuation resulting from periodic vortex shedding and the annoying vibration was found and thus confirmed that a fairing of the shaft outlets would most likely resolve the problem. As can be concluded from the right hand pictures of Figs. 15 and 14 this was confirmed by CFD analysis of the modified arrangement as well as during the full scale vibration measurements after conversion of the vessel, respectively.

Navigation in ice has become a topic of intensive research in recent years but there have been only few publications on ice induced vibration (IIV). Huang and Liu (2006) report on resonant ice induced vibrations of a pile exposed to drifting ice and give a thorough and clear description of the excitation mechanism which might also provide some insight into similar effects for ship applications. The authors also present a new method aiming at the simulation of the ice force and the structural response for small relative velocities between structure and ice (i.e. ductile rather than brittle ice failure). They further demonstrate that resonance occurs if the periodicity of the ice failure process coincides with the natural period of the structure. The authors demonstrated by computation of the time histories that the response amplitude has a 'beating' character whereas the ice exciting force is of constant amplitude. This is reported to result from the fact that the structure vibrates in the same or the opposite direction of the ice drift, so it decreases or increases the relative velocity between ice and structure and consequently tends to suppress or support ice failure, respectively. The authors further demonstrate that their method is able to capture the lock-in effect of the vibration frequency to the natural frequency of the structure which was observed for a certain range of relative velocities also in field measurements. They also emphasise the stochastic nature of such processes depending on the thickness, the material properties and the drifting velocities of ice by quoting full scale measurements which showed that the maximum acceleration amplitudes of 0.07g were present during almost the whole day but increasing to 0.7g when a state of resonance was obtained for some minutes.
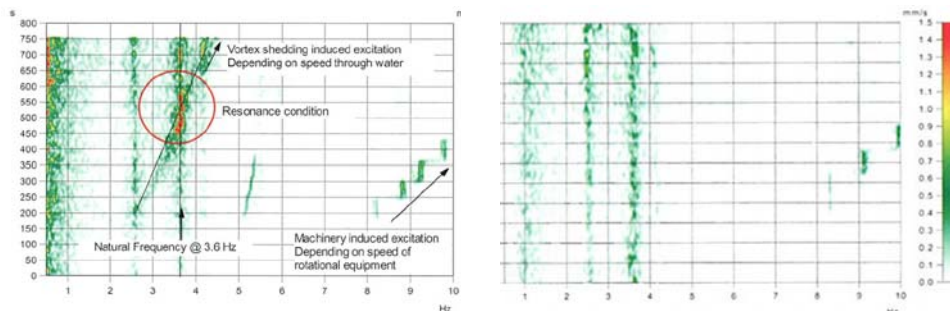

Figure 14: Vibration level before and after redesign of shaft outlets, Menzel *et al* (2007)
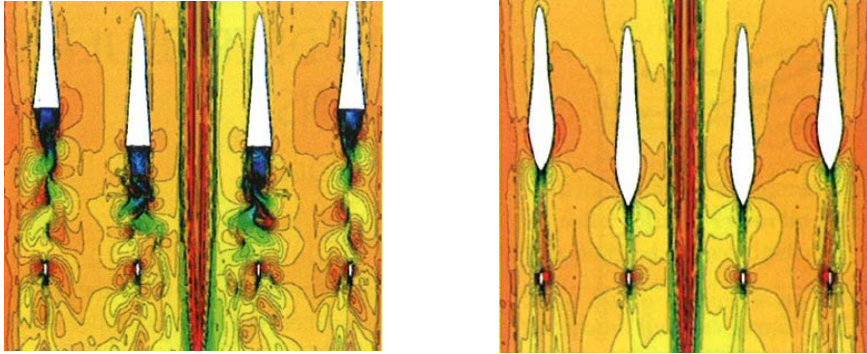
Figure 15: Top view on computed flow at shaft outlets before (with vortices) and after fairing of outlet shape, Menzel *et al* (2007)

2.3   Analysis and Prediction

Regarding the analysis and prediction of ship structural vibration; in recent years the marine industry focussed on the following topics:

- the development of analysis and prediction methods for hull girder whipping and springing vibration and related assessment criteria (see Section 2.1),
- the automation and standardisation of pre- and post-processing procedures used for the prediction of propeller and main engine excited global ship vibration,
- the improvement of the effectiveness and accuracy of the methods used for the natural vibration analysis of local ship structures and
- the investigation of vibration phenomena related to the development of new ship types or unconventional propulsion arrangements.

The finite element method (FEM) has become the most popular method for the analysis of ship vibration, however, its benefits of high accuracy and robustness are compromised by the considerable amount of time and labour which is needed to generate and assess the results for the evaluation of vibration improvement measures. Mumm (2007) qualitatively compared the effectiveness and uncertainty of numerical and analytical approaches for global and local vibration problems. For merchant vessels typically having high superstructures arranged close to major vibration excitation sources he splits the vibration design process into three tasks: forced vibration prediction of the global vibration levels in the low frequency range from 0 to about 20 Hz, natural vibration analysis of the accommodation tower deck structures (abt. 10 to 50 Hz) and natural vibration analysis of the tank wall structures located in vicinity of the main engine or the propeller (approx. 10 to 50 Hz). The author estimated for each working step the required amount of work and the accuracy of the obtained results. He concluded that FE methods are optimal for the prediction of global vibration levels but that the comparatively large effort required for the preparation of FE models which can

be used for vibration calculations of local deck or tank structures restricts their economic application in the higher frequency range. Taking also into consideration that at the design stage, when the vibration analysis of the local structures is performed, only limited input data for the preparation of the FE model is available the author strongly advocates the use of simple and fast analytical methods for such purpose.

Cho (2007) also aimed at a reduction of the amount of work inherent to the FE analysis of ships. He focussed on the 2- to 6-node hull vertical bending vibration mode and proposed a calculation method using the derivatives of the natural frequencies and mode shapes to determine the sensitivity of the ship to changes in its structure or its mass distribution, respectively. The method is based on the premise that the free vibration analysis of the hull girder is performed not by FEM but by the transfer matrix method. The governing sensitivity equation is derived from the direct differentiation of the state vector and the transfer matrix and the derivatives of the natural frequencies and mode shapes are determined after two trial calculations of the resulting equation. Using the example of a bulk carrier the author demonstrates that the computational effort required for the natural vibration calculation in the case of ship design modifications or loading variations is largely reduced without significantly affecting the calculation accuracy.

Kim (2006) proposed to better automate the FE-modelling process, as it represents the most labour intensive working step, to improve the effectiveness of global ship structure vibration analyses. He addresses the dilemma that normally no CAD data of the hull structure is available at the time when the FE model for global vibration analysis is prepared, see Figure 16, and suggests using the hull information which is available in the initial design stage of a vessel for this purpose. Consequently, the author proposes an automatic quadrilateral mesh generation algorithm utilizing the hull form offset data, the compartment data and the mesh constraints as given from the initial structural design for the generation of the global FE model of the hull. The author briefly demonstrates the practicability and robustness of the approach on the example of a LNGC, a ULCC and a CV and reports that he achieved time savings for FE modelling of about 50%.

Even if the same analysis tools are used, quality consistency in the ship vibration analysis process is not easy to achieve because the analysis accuracy depends on several other factors: the method of FE modelling, the methods to distribute light ship weight, loading and ballast masses, the accuracy of hydrodynamic mass calculation, the damping models underlying the calculations, the ability to simulate the excitation forces realistically and so forth. Wu *et al* (2006) describe their work flow of hull structure vibration analysis and report on the introduction of a vibration analysis procedure guideline scaled back from numerous full scale measurements. By introducing the guideline the authors aimed to standardise the vibration analysis and thus improve the quality and comparability of vibration predictions of merchant vessels.

In 2000 ISO 6954, which provides guidance for the measurement and assessment of

vibration levels on ships with respect to habitability, was revised. Since then the measured vibration levels shall be assessed not any more in terms of the 'maximum repetitive amplitude' of the single frequency components of the amplitude spectrum but by the 'weighted overall sum value' of the vibration velocity in the frequency range 0 to 80 Hz. This certainly improved the assessment of human perception of vibration and harmonised marine measurement standards with those of other industries but on the other hand the change created an additional challenge on theoretical vibration level prediction. Whereas formerly the predicted vibration level could be compared with the respective limit value for each excitation source and harmonic excitation component separately and independently, this is naturally no longer possible for the limit value being given as a sum value of the response resulting from different excitation sources and harmonic excitation orders, respectively. Besnier *et al* (2007) and Buannic *et al* (2007) suggest a method to cope with this problem using the example of propeller excited cruise ship vibration and the related comfort evaluation. They considered the multiple narrow-band propeller blade rate frequency components of pressure fluctuations as well as its broadband component resulting from propeller blade tip vortex cavitation effects. They calculated the structural response to these deterministic and stochastic parts of propeller excitation with different methods. While the deterministic harmonic part was evaluated with the classical frequency response analysis the broad band random part was computed from the power spectral density of the excitation. In the following step they superimposed the harmonic and broad band contributions for each 1/3 octave band by quadratic summation and finally computed the overall response according ISO 6954 ed.2000 by quadratic summation of all 1/3 octave band r.m.s. levels taking into account the frequency weighting curve as defined in ISO 6954 ed. 2000. In Figure 17 the superposition of the overall vibration response from the harmonic and the broadband contribution is illustrated for a deck panel structure of the cruise vessel investigated. Apparently, in this case, the response to the deterministic excitation with 1st and 2nd propeller blade rate clearly dominates the overall vibration level.
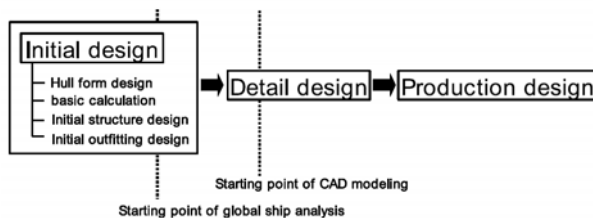

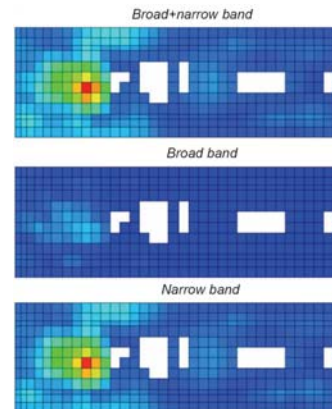Figure 16: Global vibration analysis in ship design work flow, acc. Kim (2006)

Figure 17: Deterministic and stochastic response, acc. Buannic *et al* (2007)

Smogeli (2007) followed a similar approach to compute the response to narrow and broad band propeller pressure pulse excitation for a cruise vessel. For an accommodation deck he predicted a weighted rms-response level to propeller broad band excitation of 1.2 , 0.8 and 0 mm/s aft of frame 22, between 22 and 48 and in front of frame 48, respectively, and superimposed these levels with the response to 1st and 2nd propeller blade rate and the structural response to the excitation originating from the 1st order mass moment generated by the medium speed main engines. He also addressed the uncertainty of vibration predictions resulting from the unavoidable inaccuracy inherent to the calculation of the natural frequencies of deck panel structures and recommends the response be predicted over a wider frequency range and to anticipate a potential natural frequency error of +-10% in result evaluation.

Still today the evaluation of the most effective structural counter measures against ship vibration is to a certain extent an intuitive process. Therefore, in recent years research tried to develop post-processing methods which enable the automatic identification of the most effective vibration counter measures. One promising way to examine the dynamical behaviour of a ship structure is to analyse the vibratory power flow. Kim *et al* (2006) carried out structural intensity analyses as well as a conventional vibratory response analyses for a 4,100 TEU container carrier. From the numerical studies for the original design and four modified designs of the ship structure, the engine room internal longitudinal bulkhead was identified as one of the dominant transmission paths of vibration energy from the excitation sources to the superstructure. This was true for the propeller surface force acting at the ship's stern as well as for the thrust variation force acting on the main engine's integrated thrust bearing. By applying appropriate reinforcements in way of the engine room longitudinal bulkhead, the authors achieved a reduction of the structural intensity in this area by a factor of abt. 40, see Figure 18. In addition the authors note that modifying the structure in way of the dominant energy transmission path was apparently one of the most effective methods to change the superstructure's natural frequency and its resonant response. From this they concluded that the structural intensity analysis can be usefully applied to determine proper countermeasures against troublesome ship structural vibrations.

Original Design
Max. Structural Intensity 0.0082 W/m

Modified Design
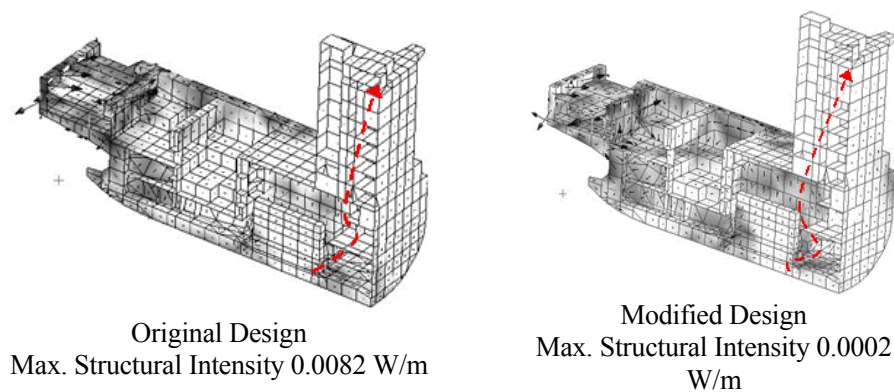Max. Structural Intensity 0.0002 W/m

Figure 18: Distribution of structural intensity for container vessel, acc. Kim *et al* (2007)

Kwun *et al* (2008) carried out a similar analysis but with the focus on the resonant vibration of a large deck panel arranged in the aft body of a RoRo Ferry. By means of a structural intensity analysis they identified the port side and starboard internal longitudinal bulkheads as the main transmission path of the vibratory energy from the propeller to the deck panel. They compared the effectiveness of reinforcing these bulkheads with several alternative measures: thickness increase of the shell above the propeller, increase of the size of the web flanges of the deck panel girders and arrangement of a pillar below the deck panel centre. The reinforcement of the longitudinal bulkheads turned out to be less effective than the arrangement of a pillar and than the increase of the scantlings of the webs of the deck panel girders however, it was much more effective than to reinforce the shell structure above the propeller. Also these authors concluded that the analysis of the structural intensity might be a useful method to determine the optimum arrangement of structural reinforcements.

To cope with stricter vibration specifications in new building contracts and to prevent tank structure damage, the development of efficient and more precise methods for the analysis of local structures continued. Cho and Kim (2006) presented a free vibration analysis method applicable to stiffened tank walls which can account for the hydrodynamic mass for full and partial tank filling states. The analysis is based on the assumed mode method and uses characteristic polynomials of the Timoshenko beam, which are able to reflect any elastic boundary condition along each edge of the tank wall. The fluid inertia is considered by a semi-analytical method using the fluid velocity potential as derived from the boundary conditions for the fluid and structure domain. By comparison with the results of numerical calculations with a coupled FEM-BEM code the authors demonstrate the accuracy of their approach for tank walls as typically used in shipbuilding industry. As presented in Figure 19 the agreement of their method with the elaborate numerical approach is very good which is true for the natural frequencies as well as the natural vibration modes. Kim *et al* (2008) applied the same method, except the boundary conditions assumed for the fluid domain, to a typical tank bottom structure. Also they achieved good agreement with FEM-BEM computations and thus confirmed that the semi-analytical method represents a suitable and effective approach for anti-resonance design of tank structures in contact with fluid.

The same is reported by Lee *et al* (2006) who presented a similar method, except the used mode functions, for the analysis of tank wall natural vibration. They also studied the influence of the distance between a vibrating tank wall and its opposite wall, i.e. the tank length, by FEM-BEM computations for a tank in model scale. As shown in Figure 20 the natural frequencies of the tank wall decrease for decreasing tank length whereas the influence is the more pronounced the lower the vibration mode. However, for tank lengths in the same order of magnitude as the filling height, this effect is reported to become negligible.

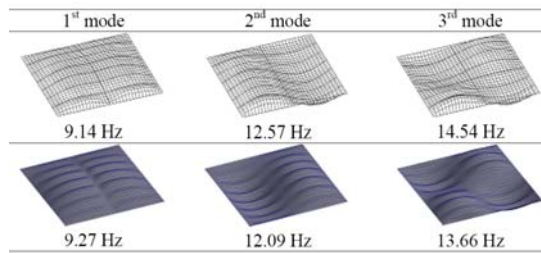| 1st mode	2nd mode	3rd mode
9.14 Hz	12.57 Hz	14.54 Hz
9.27 Hz	12.09 Hz	13.66 Hz

Figure 19: Tank wall natural vibration – comparison semi-analytical to FEM-BEM results, Cho and Kim (2006)
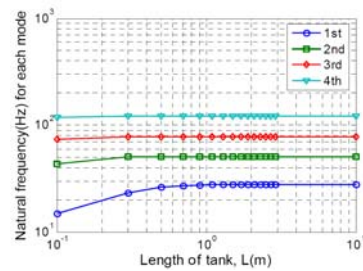

Figure 20: Effect of opposite tank wall distance on natural frequency, Lee *et al* (2007)

The effect of fluid exposure of thin walled structures on the natural vibration characteristics was investigated also by Uğurlu and Ergin (2006), however, they dealt not with plane structures but with the elastic vibration of a cylindrical shell subjected to internal or external axial flow. They assumed that the structure in contact with the flowing fluid will vibrate in its in-vacuo eigen-modes that were obtained by FEM analysis. The fluid domain was modelled by a BEM approach and the relationship between the dynamic fluid pressure on the elastically vibrating shell and the fluid velocity potential was established by Bernoulli's equation. The fluid - structure interaction forces were calculated as generalized added mass, hydrodynamic damping and hydrodynamic stiffness coefficients and subsequently the eigenvalue problem associated with the generalized equation of motion was solved to give the dynamic characteristics of the structure in contact with fluid. The applicability of the proposed method was demonstrated by the analysis of a finite length cylindrical shell in axial flow and comparison with analytical results from literature.

Hanke *et al* (2006) report on their numerical and experimental investigations on the influences of welding stresses and pre deflections on the natural vibration characteristics of a stiffened deck panel structure. They compared the natural vibrations of the panel according to

i)   a linear FEM analysis,
ii) a non-linear FEM analysis under consideration of the welding stresses and pre deflections (as measured),
iii) a measurement at the untreated panel exhibiting welding stresses and pre-deflections,
iv) a measurement at the stress-relieved panel (by heat treatment) exhibiting pre-deflections only.

The measurements were performed at seven geometrically identical deck panels that were fabricated with different welding methods. Naturally, the welding methods with the highest heat introduction resulted into the structures exhibiting the largest imperfections. The authors report on a natural frequency decrease of the plate fields

attributable to the fabrication imperfections of up to 40%. They showed also that far better agreement between the FEM analysis and the measurement at the imperfect structure could be achieved by considering the fabrication imperfections in the numerical simulation. They confirmed their hypothesis that fabrication imperfections might have considerable influence on the natural vibrations of deck panels by comparing the experimental results between iii) and iv). It turned out that the natural frequencies of iv) exceeded those of iii) and correlated far better with the results of the standard FE analysis, in which the fabrication imperfections were neglected.

Extensive research aiming at achieving realistic input parameters for the analysis of vehicle deck vibrations of RoRo ships was reported by Jia and Ulfvarson (2006). They reviewed various levels of phenomena influencing the dynamic structural behaviour of vehicle-deck systems: the vehicle vibration itself, the effect of the vehicles on the deck vibration and the dynamic interaction between tire and deck surface. They concluded from their analytical, numerical and experimental analyses that the natural frequency of the fundamental car vibration is rather low (abt. 1 Hz) and so will not be influenced by a higher frequency, e.g. propeller excited, deck vibration. On the other hand with regard to the car's influence on deck vibration the vehicles might act comparable to dynamic absorbers in a frequency range of abt. 10 to 15 Hz, i.e. forced deck vibration will be magnified or decreased depending on the stiffness properties of the car's tires and suspension. The authors confirm that placing more and heavier cars on the deck will decrease the natural frequency of its basic natural vibration mode. It is also reported that higher frequency deck vibration does not affect the frictional behaviour between the tires and the deck plating and that the car's exposure to vibration increases the closer it is placed to the deck panel centre.

Due to the increase of propulsion powers and the related increase of shaft diameters, whirling vibration of the propulsion shafting has received some attention in recent years but there exist no standardised calculation procedures regarding this topic. Mumm (2007) assessed the relative importance of the parameters influencing whirling vibration. He emphasised that whirling and shaft alignment calculation can not be seen independently from each other, e.g., because the stiffness properties of the hydrodynamic bearing's lubrication film depends on the magnitude of the static bearing load. He also recommends that hull girder flexibility in the alignment calculation should be accounted for in installations with high power and relatively flexible hull structures because it might have a significant influence on the bearing loads, i.e. on the bearing stiffness, and on the location of the effective tail shaft supporting point in the aft stern tube bearing. He illustrated the effect of hull flexibility on the loads acting on the aft and fwd stern tube bearing and the intermediate bearing for a practical case, as presented in Figure 21. The author points out that the hull flexibility should be accounted for in the whirling analysis too. On the other hand he deems it justified to neglect gyroscopic effects and the influences of hull structural vibration on shaft whirling vibration. The author also suggested the dynamic part of the bearing load caused by whirling vibration should be predicted/calculated in order to deduce a rational assessment criterion regarding vibration severity.

2.4   *Damping and Countermeasures*

The uncertainties regarding the damping characteristics of ships and their surrounding fluid have always been one major reason for the inaccuracies encountered in the theoretical prediction of dynamic ship response. As summarised in this committee's 2003 report, various methods are available for modelling ship vibration damping but research in this field is still going on because the commonly used approaches vary considerably and the most suitable method might depend on the type of ship and frequency range of interest.

Takeda *et al* (2007) investigated the damping characteristics of the superstructure vibration of a 280,000 DWT VLCC analytically and experimentally. They used a Rayleigh damping approach in the calculations, i.e. the damping matrix **B** is formed as a linear combination of the mass and stiffness matrix: $\mathbf{B} = \alpha\,\mathbf{M} + \beta\,\mathbf{K}$. They provide an iterative procedure for the determination of the coefficients α and β from full scale exciter tests and demonstrate the accuracy of their approach by comparison of the measured and calculated mobility curves in a frequency range from 4 to 8 Hz. Using tests with the exciter arranged on the compass deck they identified two natural vibration modes showing a longitudinal deflection pattern of the upper superstructure at about 5 and 7.2 Hz. From the measured response curves they determined α and β and with knowledge of the natural frequencies they calculated the damping ratio ζ for the two mode shapes. They obtained 1.1 % and 0.9 % for ballast and 1.3% and 1.6% for full load condition. However, the authors emphasise that agreement between analysis and measurement can only be achieved if the finite element model reflects the structural arrangement sufficiently well. Also, in this case, some details of the deckhouse structural design had to be considered, e.g., the presence of the main mast representing a coupled vibration system or the door openings in the deckhouse side walls and the corrugation of the internal longitudinal walls reducing the overall longitudinal stiffness of the accommodation tower.

Thomas *et al* (2008) focussed on the identification of the damping characteristics of transient whipping vibration of large high speed aluminium catamarans. For two vessels, full-scale measurements of slam events were conducted and anchor drop tests, whilst the vessels was stationary in calm water, were also performed. Beside the two-node vertical bending vibration mode, which is also typically excited by slam events for monohull vessels, the authors identified a second significant mode characterised by a counter- phase pitching vibration of the catamaran's port and starboard hulls causing a torsional deflection of its cross deck structure. Depending on the loading condition the natural frequencies were about 1.4 Hz for the torsional mode and 2.7 Hz for the bending mode. In determining the damping characteristics the authors focussed on the bending mode for better comparison with data published for monohull vessels. They also examined the components that contribute to the damping of the system and estimates were made of the relative magnitude of the various hydrodynamic components including: wave making damping, viscous damping and acoustic damping.

The total predicted damping due to hydrodynamic effects ($\zeta < 0.05\,\%$) and material damping ($\zeta \approx 0.1\,\%$), was found to account for only a fraction of the measured total damping of $\zeta \approx 1.7\,\%$. The authors presume this shortfall to be due to additional structural damping which is estimated to be one order higher than the inherent material damping. As in the case of monohull vessels, a clear physical explanation for this phenomenon is still missing.

Also the prediction and experimental determination of the damping properties of local ship structures is still a challenge to the marine industry. The damping that originates from the hysteresis of materials and from their capability to dissipate mechanical energy into heat is often used as passive countermeasure for noise and vibration control. Although all materials exhibit a certain amount of damping, this of steel and aluminium is so small that they might become quite effective sound radiators in states of resonance. By bringing such materials into intimate contact with a highly damped material, it is possible to control these resonances. Of the common damping materials in use, many are viscoelastic; that is, they are capable of storing strain energy when deformed while dissipating a portion of this energy through hysteresis. Several types are available in sheet form. Some are adhesive in nature and others are enamel-like for use at high temperatures. The simplest method of material application is the free layer configuration: the viscoelastic material is simply attached to the surface of the structure. In this case, energy is dissipated as a result of extension and compression of the damping material under flexural stress from the base structure. A more effective configuration is represented by constrained damping layers: where a composite is formed by laminating the base layer to the damping layer and adding a third constraining layer. When the system bends during vibration, shear strains develop in the viscoelastic layer. Energy is dissipated through shear deformation, rather than extension, of the material.

A finite element calculation procedure to predict the dynamic behaviour of plane structures with total or partial viscoelastic coating in free layer damping configuration was proposed by Curà *et al* (2005). Starting from the master curves of Young's Modulus and the damping loss factor, η of the respective viscoelastic material, they assemble the complex stiffness matrix which considers the damping layer properties. However, because the required input data for the characteristic values of E and η of the viscoelastic material in many cases might not be available it is sometimes more straightforward to determine the damping characteristics of the whole configuration experimentally from the outset. Ferrari and Rizzuto (2007) proposed a method to perform such tests on specimens of constrained layer configuration for this purpose. They describe and discuss alternative approaches for the determination of damping coefficients from experimental results: half power bandwidth, circle fitting and logarithmic decrement method. The authors also report on the effects related to the environmental conditions during the tests (constraints, temperature) and the phenomenon of material ageing is also discussed. Moreover, a standard experimental procedure is proposed for measuring the loss factor of monolithic viscoelastic materials by means of forced vibration tests.

Beside the research on the determination or prediction of the damping characteristics for given local or global ship structural elements several publications report on the endeavours to reduce vibration and noise levels by active vibration control (AVC) systems. This is most likely a consequence of the insight that traditional passive damping treatments are not capable of fulfilling extreme anti-vibration and noise requirements as demanded by the naval, geo-physical or high-luxury ship building sector. On the other hand, the transfer of mass product AVC systems as developed in the automotive industry, for instance, might, in the future, offer a cost advantage compared to passive systems also in ship building.

While viscoelastic damping treatments are typically used to control vibrations in the receiver room and along the vibration path, AVC systems are normally designed to bound the vibration energy directly at its source. Another strategy is followed by Keir *et al* (2005), who investigated how to actively reduce the vibration transmission along its path. For their analyses and experiments they used a plate T-joint as typically encountered in ship structures, see Figure 22. They excited a forced vibration my means of an excitation force F_p and counteracted it with an actuator force F_s in a frequency range up to 400 Hz. Magnitude and phase of the actuator force was controlled by the measured response at sensor position e1. Multiple actuator force and control sensor positions were tested and the authors could demonstrate that considerable vibration attenuation could be achieved provided the actuator and sensor arrangement is chosen carefully according to the individual structural arrangement.

Following a concept from Goodwin from 1960, Dylejko *et al* (2007) directly manipulate the vibration excitation source in order to reduce the introduction of vibratory forces into the ship structure. They use a resonance changer (RC) or dynamic vibration absorber (DVA) to control the resonance frequencies of the axial vibration of a submarine's propeller shaft in a frequency range from 0 to 100 Hz. The RC consists of a piston running along a cylinder which is connected to an oil reservoir through a pipe. It works as a spring-mass-damper system located in series between the thrust bearing and the supporting foundation. As the RC's inertial, elastic and damping properties depend on the physical properties of the oil and the geometry of cylinder, pipe and reservoir, the authors optimised these parameters by applying a genetic algorithm. They report that for an optimisation of the device the stiffness properties of the ship structure must be considered too. Depending on the choice of the optimisation parameters they achieved reductions of the force transmissibility and the power transmission between 95 % and 99.8%. Consequently, such a device can be supposed to be very effective as long as the prevailing vibration level is governed by the response to the axial propeller thrust force.

Most of recent publications on AVC are concerned with the attempt to attenuate the dynamic forces introduced by equipment and machinery into the ship structure via the connecting elements by producing counter forces of suitable magnitude, phasing and direction. Niu *et al* (2005) developed an analytical model of a 2-stage active floating

raft isolation system. In this system active actuators are arranged between the machinery components to be supported and a rigid raft and between the rigid raft and the ship structure. The active elements are supposed to act in parallel with conventional passive resilient mounts and horizontal translations and forces are disregarded because typically the vertical translations and forces dominate the dynamic behaviour of resiliently mounted equipment and machinery. The authors use the impedance matrix technique to derive the mobility for each subsystem, i.e. machines, active and passive mounts, raft and ship structure, and then perform the dynamic analysis of the overall system. The actuator forces are optimized by minimizing the cost function expressing the power transmission to the ship structure. Three different control modes have been investigated: the '*machine control*' with just the actuators below the machinery working, the '*raft control*' with just the actuators below the raft switched on and the '*full control*' mode with all actuators in operation. From the simulations with their analytical model the authors concluded that in the low excitation frequency range machine control is good enough, for middle and high-excitation frequencies raft control should be considered and for random disturbances in a very broad frequency range full control should be employed.

Annicchiarico *et al* (2005) developed and tested, at laboratory scale, an active control system to reduce the vibrations introduced by a 25 kW, 3600 rpm centrifugal pump to a base frame supported by a double stage of passive mounts. Numerical simulations with a simple lumped mass dynamic model have been used to choose the best configuration and appropriate size of the actuators arranged in parallel to the passive mounts. The test rig comprised the pump, mounted by standard anti-vibration elastomeric supports on an intermediate base frame; a digital signal processor, providing an adaptive feed forward control for the actuators; four mono-axial voice coil actuators controlling the vertical translations of the base frame and accelerometers to measure the vibrations of the pump and the base frame. The test results demonstrated the good performance of the system: the first two harmonics at 60 Hz and 120 Hz could be attenuated by maximum 12 dB and 23 dB, respectively. Also Schirmacher (2007) reported on significant attenuation of the vibrations introduced by machinery into the ship structure by an active vibration isolation system. He describes the realisation of such a system at the German Navy's acoustical test facility for a submarine machine room platform with a weight of 60 tonnes, see Figure 23. The platform was mounted on six passive spring packages and electro dynamic actuators were used to counteract platform vibration at each support in all three translational directions for frequencies up to 250 Hz. The vibrations were significantly reduced and it was demonstrated that the controlling system was also capable of handling unsteady excitation characteristics as originating, for example, from a machine with increasing revolutions.

In addition, the attenuation of the vibrations introduced by high power diesel propulsion engines into their foundations by means of an AVC system was demonstrated to be technically feasible. Noé (2007) reported on laboratory results which were obtained for a high-speed main engine as typically installed on naval vessels, high speed craft or mega yachts as a main propulsion unit. The standard

resilient mounting was replaced by an active system in which, at each mount, all three translational degrees of freedom were controlled. The system proved to be effective on the first eight harmonics of the engine's vibration excitation spectrum in the frequency range from 60 to 240 Hz. Attenuation between 20 and 35 dB were achieved with required actuator forces ranging from 30 N to 300 N.

AVC can be realised also by means of magnetorheological elastomers (MRE). MRE mounts are similar to conventional rubber mounts but due to inclusion of iron particles their stiffness and damping properties can be controlled by the variation of an external magnetic field. Choi *et al* (2007) proposed the use of an MRE core embedded sandwich beam as a controllable mounting element for the support of machinery rafts in ships. By numerical response analysis of a sandwich beam provided with MRE elements they demonstrated the feasibility of this new technique.
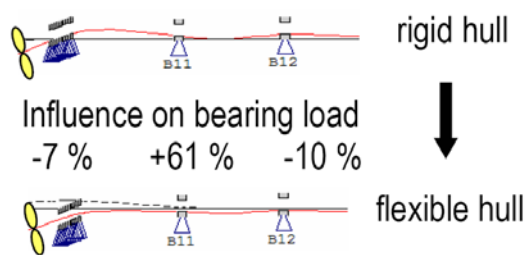

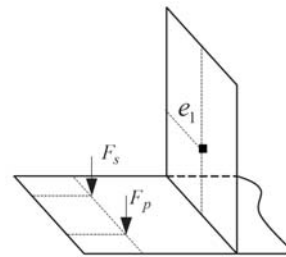Figure 21: Effect of hull flexibility on bearing load, acc. Mumm (2007)

Figure 22: T-Joint connection with excitation and actuation force and control sensor, acc. Keir *et al* (2005)

2.5 Measurement and Monitoring

In recent years research on the measurement of ship structural vibration focussed on the improvement of the measurement methods used for the identification of the natural frequencies and mode shapes, the introduction of reliable systems monitoring the vibration stresses acting on local and global structures and on the measurement of ice-induced vibration.

Experimental modal analysis (EMA) deals with the estimation of modal parameters of a structure by measurements. Classical EMA techniques use defined excitation sources to achieve proper vibration parameter estimates. In operational modal analysis (OMA), also known as 'natural input or output only' modal analysis, the extraction of the modal parameters is achieved by measuring the response of a structure to an unknown stochastic excitation as, e.g., natural seaway. This testing method is cheaper and faster than classical EMA techniques because no elaborate excitation equipment is needed. A comparison of the classical and the OMA technique on different ships was presented by Rosenow *et al* (2006). Maximisation of the signal to noise ratio was achieved by using seismic accelerometers with a high sensitivity and overloads were eliminated by use of a data acquisition unit with a high dynamic measurement range. For the deterministic excitation of the ship structures a 600 kg impulse exciter was used. The authors

obtained similar results with both methods for the ships lying in the harbour and demonstrated in this way that with regard to result quality the OMA technique is tantamount to classical approaches but has the advantage that no excitation equipment must be handled. This becomes, of course, extremely useful for ship natural vibration measurements in sea-going condition.

Generally, the estimated eigenvectors have to be large enough to permit a reliable distinction of the mode deflection shapes. This is especially important for closely spaced and similar modes, as in the case for, e.g., superstructure vibrations. When a small number of sensors is available only it is necessary to perform several measurement runs to gain a sufficiently reliable eigenvector estimation. Zierath *et al* (2006) implemented algorithms based on QR-Factorization and genetic algorithms to find the best sensor positions for OMA and thus to minimise the number of measurement runs. The algorithms were tested for a 2500 TEU container vessel and the authors demonstrated that all excited mode shapes could be distinguished properly.

Rosenow *et al* (2006) compared the most frequently used analysis techniques in OMA: Enhanced Frequency Domain Decomposition and Stochastic Subspace Identification. They applied these methods to in-service measurements of superstructure vibrations of a 2500 TEU container vessel and the hull girder vibration of a RoRo ship. Global vibration modes with natural frequencies below 5 Hz could be well identified, however, mode shapes with natural frequencies exceeding 15 Hz could hardly be distinguished due to the poor signal to noise ratio and the high modal density. Between 5 Hz and 15 Hz the main problems in the application of OMA techniques was represented by the masking of the stochastically excited vibration by the harmonic vibrations stemming from the propulsion system. However, Rosenow *et al*. (2007) could demonstrate that it is possible to eliminate the bias induced by the harmonic vibration components by applying the algorithm developed by Jacobsen *et al* (2006).

Recent advances in sensor technology, data acquisition, processing, transfer and storage have opened new possibilities in real time measurement and monitoring of ship responses. Combined with the monitoring of the ship's operational parameters and the ambient conditions, especially the prevailing wave climates, such measurement values are extremely useful for the validation of load assumptions and analysis methods as applied in ship design and, moreover, might be used also for damage detection and maintenance planning.

Budipriyanto *et al* (2007) developed a method to quantify the extend of structural damage of a cross stiffened panel by means of vibration measurements. As a typical structural detail of ship side structures exposed to stochastic fatigue loads, they tested their method on the example of cracks developing at the intersections between longitudinal side shell stiffeners with transverse bulkheads and shell web frames. They built a model scale test set-up and applied cracks of varying length at the intersection points. Then they performed vibration acceleration measurements using a shaker device to stochastically excite the model. They fed the measurement results into a neural network algorithm and thus succeeded in defining a damage index D_n sufficiently

sensitive to conclude from the measured response on the crack length. The authors demonstrated that neither the natural frequency nor the damping ratio worked as reliable damage indicator but that D_n had to be deduced as a function of the measured natural frequency, the damping ratio and the vibration response amplitude. In Figure 24 the variation of the damage index with time is presented for 5 different cases where the crack length was incrementally increased by 2.5 mm from 0 mm (case #0) to 10 mm (case #4). The damage index D_n grows consistently with the crack length. Based on their finding that this holds true also for non resonant vibration the authors concluded that their new technique is particularly suitable for on-line crack growth monitoring of ship side structures.

In recent years the focus regarding the monitoring of ship vibrations was on the design and installation of robust systems being capable of measuring the contribution of hull girder springing and whipping vibration to structural fatigue and the detection of possible extreme events. In monitoring campaigns extending over several years the challenge is to separate the overall acceleration or strain signal into a LF and HF-part omitting the need for storage of the time histories of the signals. Therefore, the settings for filtering frequencies, sampling rates, pre- and post-triggering criteria etc. must be chosen carefully in order to obtain meaningful results which can be handled with reasonable effort.

References concerning full scale measurements with the focus on whipping and springing vibration are described in subsection 2.1.
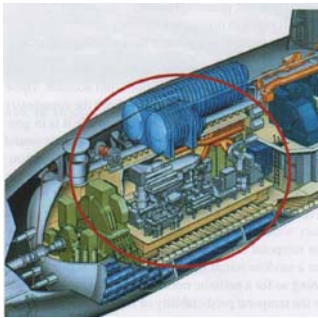
Figure 23: Machine platform supported by AVC system, Schirmacher (2006)
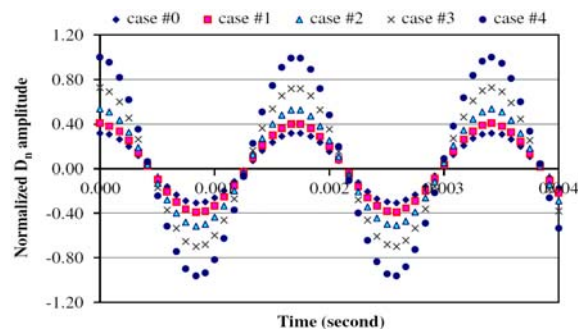
Figure 24: Correlation of damage index and crack length in ship side structure, Budipriyanto *et al* (2007)

Yu *et al* (2007) introduced a comprehensive full-scale measurement system to measure the wave environment, the ship motions and the ship's response for a 8000 TEU carrier operating between Asia and Europe. LF as well as HF components are registered. Moreover, the vessel is equipped with a hull stress monitoring system displaying the cumulative fatigue damage ratio for guidance of the ship's master.

Kahl and Menzel (2008) reported on the first results of a monitoring campaign which started in 2007 onboard a 4600 TEU container carrier operating in the North Atlantic and the Pacific Ocean with a total of abt. 60 measurement channels registered. Beside

the standard arrangement for the measurement of ship motions, global hull stresses and waves, they also arranged sensors to monitor the seawater pressures acting on the shell, strain gauges in selected hatch corners, at a critical side shell longitudinal stiffener and at the breakwater. Moreover, the local vibration levels in way of the propeller, the main engine and some selected tank structures are recorded. To cope with this high number of channels to be registered they opted for a decentralised data acquisition system which used 5 data acquisition units gathering the data from the close-by sensors and transferring the signals via optical cabling to the main switchboard. They implemented algorithms for the separation of global stresses into vertical bending, horizontal bending, torsion and axial stress as well as a procedure to differentiate between the different kinds of loads acting on a side shell longitudinal stiffener. Analysing the HF contribution to the stresses acting in the shell side longitudinal stiffener the authors report that relevant stress amplitudes were found due to the global vertical hull vibration only. i.e. no HF-effects due to local loads or deflections were observed.

Due to the increased economic interest into shipping in polar areas, several research projects have been conducted aiming to improve the knowledge on the impact loads which ship structures are experiencing when advancing through (partly) ice covered waters. As in the case of slamming excited whipping vibration the dynamic hull girder response to ice impacts will be a transient vibration dominated by the 2-node vertical hull vibration mode. As the stress levels imposed on the hull girder by such impacts largely depend on the magnitude of the impulse the studies focussed beside the determination of the local ice pressures acting on the shell structures on the estimation of the maximum overall impact force and the associated impulse. In Canada an extensive measurement program for the determination of local and global ice loads was conducted with the 6800 tonnes CCGS ice-breaking vessel Terry Fox. More than 150 controlled collisions with small pieces of glacial ice or bergy bits at ship speeds up to 12.8 knots were provoked and simultaneous measurements with three different systems were made. A qualitative comparison of the measurement results is provided in Johnston *et al*. (2008a). Two systems primarily served for the measurement of the local ice pressures and the effective contact areas and one system for the quantification of the overall impact forces, see Figure 25. Gagnon (2008) designed a 3.5 m^2 externally mounted acrylic panel for measuring the ice impact pressures by optical means with a high spatial resolution. He presented the load history for a 6-knot collision with a 240 t bergy bit, see Figure 26, and concluded that the overall impulse is caused by the combined action of crushed and hard ice in the first and second phase of the impact, respectively. With 290 kN the maximum force is rather small, however, due to the small effective contact area of abt. 600 cm^2 , the local pressure was found to be rather high (abt. 10800 kPa). Ritch *et al* (2008) confirmed the local pressure results by using a strain gauged shell area of 5.4 m^2 but they ended up with higher maximum impact forces, i.e. 5100 kN in comparison to 1200 kN. Johnston *et al* (2008b) report on their MOTAN system which computes the maximum impact force from the measured rigid body accelerations caused by the impact. The generalized mass matrix, the added mass and damping coefficients and the restoring force coefficients are needed as an input for these calculations and if the location of the impact can be well estimated, the

calculation accuracy is reported to be improved. With this system the authors ended up with impact forces from 17% lower to 33% higher compared to the force estimates from the strain gauge measurements.

Alexandrov *et al* (2007) introduced a strength and vibration monitoring system which they claim to provide guidance for optimum navigation through ice covered waters. The system consists of modules for measurement and data processing, situation analysis by finite element analysis and decision support by fuzzy and neural network models.

Bjerkås *et al* (2007) describe their work on a signal processing method which is capable of identifying ice induced resonance vibrations by the use of the measured ice pressures. They tested their method on a lighthouse located in the northern Baltic Sea. As opposed to the commonly used spectral analysis techniques, they used continuous wavelet transform (CWT) algorithms to analyse the measured ice pressures and the related accelerations of the structure. They differentiate between the ice failure mode of ductile, brittle and intermittent crushing and they suppose that only intermittent crushing will excite resonance-like response histories of the structure. Consequently, they focussed on the detection of intermittent crushing in the measured ice pressure signals by choosing appropriate CWT parameters. Despite their success in identifying short periods of strong lighthouse vibration from long time ice pressure histories they conclude that further research is needed to make their new signal processing method more robust and generally applicable.
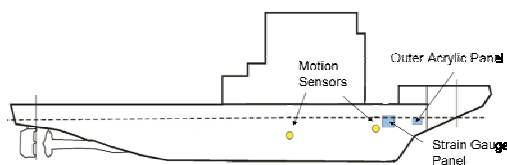
Figure 25: Location of measurement systems on board CCGS Terry Fox, Johnston *et al* (2008a)
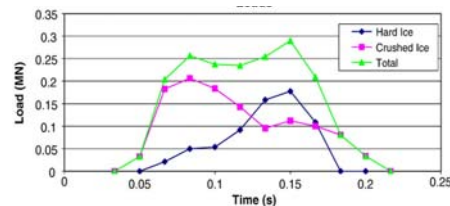
Figure 26: Time history of overall impact force during ship ice berg collision, Gagnon (2008)

2.6   Standards and Regulations

In the last years Subcommittee 2 ('Measurement and evaluation of mechanical vibration and shock as applied to machines, vehicles and structures') of ISO's Technical Committee 108 ('Mechanical vibration, shock and condition monitoring') continued its work on standardising the performance of vibration measurements on ships. It is intended to establish one ISO standard (ISO 20283) which contains five parts covering the different kinds of vibration measurements as typically applied during the design or acceptance measurements of ships.

Part 1: General guidelines
- – Part 2: Structural vibration
- – Part 3: Pre-installation vibration measurement of shipboard equipment
- – Part 4: Guidelines for the measurement and evaluation of ship propulsion machinery vibration
- – Part 5: Habitability (continuation of ISO 6954, ed. 2000)

Parts 1, 4 and 5 have not been yet finalised but ISO 20283, Part 3 became effective in 2007 and ISO 20283, Part 2 was released in 2008.

ISO 20283-2 (2008) covers the measurement of structural vibration and cancels and replaces ISO 4867 ('Code for the measurement and reporting of shipboard vibration data') and ISO 4868 ('Code for the measurement and reporting of local vibration data of ship structures and equipment') of which it constitutes a technical revision. It aims at a unified description of the vibration characteristics of ship hulls and the excitations originating from the propulsion plant in order to provide a basis for improved vibration engineering, i.e., systematic comparison against theoretical predictions, other ships and vibration reference levels. Because the guideline focuses on the standardisation of measurement methods rather than on the fixation of vibration limits it does not define acceptance levels. However, recommendations are given with regard to the ambient conditions during the measurements, the arrangement of measurement sensors, the manoeuvres to be performed, signal acquisition and processing parameters and the extend of documentation. For the determination of the global natural vibration mode shapes and the operational deflection shapes reference to the results of the theoretical global vibration analysis is recommended in order to arrange the measurement sensors in a meaningful way. If such an investigation is not available at least following measurement positions are suggested for typical merchant vessels with high accommodation towers:

- – At the transom / main deck port side corner: transverse and vertical direction.
- – On the forward port side navigation bridge deck corner: longitudinal, transverse and vertical direction.
- – On the forward starboard navigation bridge deck corner: longitudinal direction.
- – At the centre line of the superstructure front wall / main deck intersection: vertical direction.
- – At main engine top at the aft cylinder frame: transverse vibration.
- – At main engine top at the fore cylinder frame: longitudinal and transverse direction.
- – Optionally at main mast top: longitudinal and transverse direction.

Guidance is also given for the arrangement of pressure transducers serving for the measurement of propeller pressure fluctuations acting on the shell above the propeller. Recommendations for the performance of speed-up manoeuvres for propulsion arrangements with fixed and controllable pitch propellers are given as well. With

respect to signal acquisition and processing it is required that the vibration transducers and the signal processing equipment shall be capable of measurement from 1 Hz to 80 Hz with an amplitude accuracy of at least +/- 5% and a frequency resolution of at least 0.125 Hz. For standardisation of reporting some samples of plotted documentation of speed-up manoeuvres are presented as well.

ISO 20283-3 (2007) covers the topic of pre-installation vibration measurements of shipboard equipment. As such measurements mainly concern the introduction of structure borne sound into the ship structure this reference is discussed in Section 4 in more detail.

3.   SHOCK RESPONSE

Although shock response also represents a form of ship structural vibration, a separate Section was devoted to this topic because the methods used for the prediction of the excitation loads, the consideration of fluid-structure interaction effects and the calculation and assessment of the local or global hull response tend to be rather specific. Recent publications in this field mainly address the enhancement of underwater explosives, the improvement of methods for the prediction of shock loads and response and the development of hull designs and materials with increased shock strength.

3.1   Explosives and Bubble Dynamics

Massoni et al (2006) developed and tested a numerical procedure for the prediction of the blast wave effects of spherical in-air and underwater detonations of aluminized explosives. They used a detonation tracking method in combination with an ALE formulation in order to simulate the interface between detonation products and their environment as well as shock propagation. The numerical results were compared to large scale experiments conducted in air and in water involving charges of varying mass and aluminium content. Comparing the peak pressure evolution versus distance of the charge, the specific impulse, the specific energy and the positive duration phase, they obtained very good agreement between calculations and experiments.

The prediction of the gas bubble pulsation loads due to underwater explosions (UNDEX) still implies some uncertainty and so research in this field continues. Geers and Park (2005) modified the former 'Geers and Hunter' model for the analysis of underwater bubble dynamics by introducing an artificial drag to achieve better correlation with empirical formulas for the motion of a spherical bubble.

The interaction between an oscillating bubble, the free surface and a floating structure was investigated by Klaseboer et al (2005). They used a potential theory boundary integral method to describe the bubble's behaviour. The free surface was modelled by a negative image of the bubble and the submerged part of the structure. The authors demonstrated the complex interaction of the attraction of the bubble towards the solid

surface, its repulsion by the free surface and the influence of gravity. Also Kan *et al* (2005) studied UNDEX bubble dynamics. They investigated the bubble collapse under a submersed flat plate with a numerical approach and obtained good agreement with experiments regarding the pressure distribution on the plate. Their results illustrate the phenomenology of the jet impact, the formation of a radial hydraulic jump and its complex interaction with the collapsing toroidal bubble.

Krieger and Chahine (2005) investigated the dynamics of UNDEX bubbles close to a free surface and a sea bottom experimentally. They found that the first period of bubble pulsation is lengthened or shortened for increased or decreased suppression of the fluid motion around the bubble, respectively, and concluded that bubble distortion weakens the first bubble pulse but appears to increase the bubble size and the duration of the second cycle.

Xie *et al* (2006) investigated UNDEX phenomena close to a free surface constituting an explosive gas–water-air system with a shock and free surface interaction and the presence of a region of bulk cavitation. A simplified Modified Ghost Fluid Method (MGFM) was applied to simulate the explosive gas-water and water-air interface and the deformation of the free surface. The MGFM was also employed to analyse underwater close-in explosions, see Xie *et al* (2008).

UNDEX can also cause surface waves that might be another kind of load acting on the ship. A new modelling approach for real-time optimization applications was developed by My-Ha *et al* (2007) to compute bubble dynamics and the shape of the free surface. The authors claim that this method can be applied to optimize the UNDEX parameters so that a desired free surface shape is generated.

3.2   *Fluid-Structure Interaction*

Beside the influence of the presence of the hull on bubble dynamics, UNDEX are affected by the interaction of fluid pressures and structural flexibility/deformation. For the numerical solution of such problems often combined BEM/FEM methods are used which are based on Doubly Asymptotic Approximations (DAA). In practice, a so-called 'augmentation' is frequently applied to the DAA equations which might lead to a singularity of the structural mass matrix. De Runtz (2005) described a numerical method using a generalized inverse technique to circumvent this problem. Liu and Wu (2006) modified the DAA to account for the fact that the assumption of complete reflection of the incident pressure wave at the wetted hull surface, which is justified for metallic hull plates, must not necessarily hold true for composite materials. Their Acoustic DAA (ADAA) accounts for the partial transmission of the shock wave into the hull plating which might occur if the acoustic impedance of hull plating and water are of the same order of magnitude. By comparative DAA / ADAA analysis of the shock response spectrum of a 1 m x 2 m GRP plate of 5 cm thickness exposed to a charge of 5 kg TNT in 3.5 m distance, the authors found that i) between 1 and 25 Hz the same displacements were computed with both approaches, ii) between 25 Hz and

3 kHz the ADAA method gave 20% higher velocities and iii) between 3 kHz and 10 kHz slightly smaller accelerations were obtained with the ADAA approach.

Normally, the energy of an underwater shock wave is expressed in terms of a 'shock factor' which is calculated from the charge size and its distance to the target. However, the stress magnitude induced by UNDEX in the ship hull plating also depends on the degree of energy-coupling between the shock wave and the hull plate response. To take this effect into account, Rajendran and Narasimhan (2006) calculated coupling factors for circular and rectangular plates with elastic and plastic deflection behaviour. They introduced a so-called '*effective shock factor*' which is obtained by multiplying the shock factor to the power of 1.03 with the square root of the respective coupling factor. They verified their method by comparison with experiments carried out for a 300 x 200 x 8 mm high tensile strength steel plate. In Rajendran, Paik and Kim (2006) the same concept is used to predict permanent hull plate deformation due to UNDEX with analytical and empirical methods.

3.3   *Hull Plating Response*

To avoid flooding of the hull, the shell plating has to withstand the UNDEX shock wave. Moreover, highly damped materials can be used to absorb as much shock wave energy as possible to reduce the global effects of the impulse. So it is not surprising that the search for shell plating materials and design concepts with improved shock characteristics is further pursued.

Librescu *et al* (2004) proposed a methodology to deal with sandwich structures considering such effects as anisotropy and heterogeneity of face sheets, geometrical non-linearity and initial geometric imperfections.

With respect to shock loads all-metal sandwich plates have distinct advantages over conventional monolithic plates. Deshpande and Fleck (2005) investigated the one-dimensional shock response of sandwich plates subjected to an underwater pressure pulse. They used an analytical and a FE model to analyse sandwich plates with identical face sheets separated by a compressible foam core. They found that different to the monolithic plate case, cavitation does not occur at the fluid-structure interface and that weak sandwich cores may enhance the underwater shock resistance of sandwich plates. Hutchinson and Xue (2005) extended the mechanics of dynamically loaded sandwich plates of Deshpande and Fleck (2005) and developed a method to find the minimum weight design. They report that optimally designed sandwich plates can sustain water shocks that are two to three times larger in comparison to monolithic plates of the same mass and material.

Wei *et al* (2006) reported on their model experiments regarding the mitigation of UNDEX shock wave pressures and impulses by means of multi-layered all-metal sandwich panels. Based on the experimental results and FE simulations an analytic method was developed, which is reported to reflect the response characteristics of such

sandwich panels to water blast with reasonable accuracy.

Of course, also composite materials might be considered to increase the shock resistance of hull plates. However, the failure modes and the related amount of dissipated energy can hardly be predicted solely by theoretical methods. Langdon *et al* (2005) reported on their experiments for the determination of the failure modes of fibre-metal laminates (FMLs) exposed to blast shock loads in air. The results indicate that FMLs may show potential for use in blast-resistant structures, due to their ability to absorb blast energy through delamination, debonding, petalling of metallic face layers and bending and stretching of the glass fibres. The tests also highlighted the difference in response between panels constructed from woven and unidirectional fibre arrangements.

Batra and Hassan (2007) developed a 3D FE code for the prediction of UNDEX induced damage in fibre-reinforced composites (FRCs). Fibre-matrix debonding, fibre breakage, matrix cracking, delamination, relative sliding of adjoining layers and the interaction among the different failure modes are reported to be considered. Also the energy absorbed in each one of the failure mode is compared. The authors claim that their method is enabling to maximize energy absorption and hence blast resistance in the design of FRC structures.

Kalavalapally *et al* (2006) investigated the potential of using composites for the design of light weight torpedo hull bodies that can withstand UNDEX explosions on the same safety level as conventional metallic ones. In the first step they optimised the hull structure stiffening concept for a metallic and a composite design option. Then FE-models of both designs were prepared and exposed to UNDEX fluid loads. By assessing the computed response with reference to the relevant failure criteria the authors concluded that a composite torpedo hull can be designed stronger and lighter than a conventional one for a given standoff distance.

3.4   *Global Hull Response*

Global hull integrity is normally not jeopardized by the primary UNDEX shock wave because the duration of the related pressure peak is extremely short and can not produce an impulse of significant magnitude. However, due to the much longer periods of the following gas bubble pulsation cycles, considerably larger impulses are created, which might be additionally amplified if their period matches with the natural period of the hull girder. This means that during the design phase of a ship following aspects must be considered:

– machinery and equipment have to be selected, arranged and mounted in such a way as to withstand the base excitations resulting from the superimposed rigid body motion and elastic vibration of the hull girder,
– in structures contributing to the longitudinal strength of the hull, stress concentrations should be avoided as far as possible to keep plasticised and

      collapsed areas at a minimum,
–   ultimate hull girder strength must be sufficient to withstand the specified maximum dynamic global loads.

Recent references in this field mainly describe the endeavours to enhance the computation methods for the prediction of UNDEX excited transient hull vibration.

Zong (2005) used a beam hull girder model to study the formation of a plastic hinge at the midship section due to UNDEX bubble pulsation loads. He concluded that longer beams tend to sustain larger plastic deformations than shorter ones, and that on the other hand, shorter beams undergo larger rigid-body motions than longer beams.

Sprague and Geers (2006) used a 31,000 DOF 3D FE model of a ship-like structure coupled to a 10^5 to 10^7 DOF model of the fluid domain to demonstrate the practicability of UNDEX bubble analyses in combination with relatively large FE models. The fluid model employs spectral elements of various order and models the cavitating fluid as a nonlinear acoustic medium with a bilinear bulk modulus. The method is reported to become feasible thanks to the exploitation of several simplifications in the numerical procedure, e.g., a dramatic mesh truncation in the fluid domain.

Also Gong and Lam (2006) took bulk cavitation, as typically occurring for UNDEX close to the free surface, into account. Differently to Sprague and Geers (2006) they employed a FE model of the hull structure coupled to a BEM representation of the fluid domain. The authors investigated also the shock attenuation which can be obtained by measures increasing the structural damping of the wetted hull surface. Based on their simulations they concluded that an additional light weight damping layer with a high modulus and a high loss factor applied to the outer shell surface appears to work effective in this respect.

Liang and Ta (2006) used a non-linear FE model coupled to a BEM representation of the fluid domain to analyse the transient response of a 2000 ton patrol boat subjected to an UNDEX with a keel shock factor of 0.8. From the simulations they identified the structural areas likely to plasticise and computed the time histories of the shock loading along the keel and the resulting accelerations, velocities and displacements at different positions of the hull girder.

3.5   *Other Applications*

Grenestedt *et al* (2008) reported on a hybrid ship hull model with a steel truss and composite sandwich panels. One of the ideas of the concept was to mount the sandwich panels to the steel truss such that they can be blown out in a controlled fashion to ventilate a large internal blast and that there remains sufficient strength for the ship to reach a port safely even after such extensive damage. Experiments were carried out with a model consisting of a steel truss and 60 composite panels. After nine panels had

been removed, from all the different areas of the hull, it could still carry the design load, although with considerable plastic deformations of the hull. The hull was eventually loaded to final failure, which occurred at 25% above the design load.

Viada *et al* (2008) reported on the effect of UNDEX for the removal of offshore structure on sea animals and provide a review on the protection requirements and ongoing research in this field.

4.   SHIP ACOUSTICS

A separate Section was devoted to the topic of ship acoustics because noise control represents a critical issue in ship design and operation for several types of ships. Some recent research work is described and relevant problems are highlighted.

The typical objective of ship acoustical design is to maintain the airborne noise level limits which have been specified for living and working spaces. For this purpose specific analysis and measurement methods and assessment criteria are needed, which are discussed in the Subsection on interior ship noise.

For naval, geo-physical and fishery research vessels underwater noise emission often represents a mayor design parameter and in recent years a growing concern on the effects of underwater noise on the marine fauna can be observed. Therefore, in the second subchapter references addressing underwater noise phenomena are reviewed.

4.1   Interior Ship Noise

Sound can travel via the air and/or via the structure from the noise sources to the receiver spaces. Acoustic design methods must consider both transmission paths. Airborne noise is most significant within the respective source space but may also be transmitted to adjacent compartments. Structure-borne sound represents a mid to high frequency (approx. 20 Hz and 20 kHz) vibration of local ship structural elements. It is not audible but generates airborne noise due to the vibration of the decks, bulkheads, ceilings and linings of the receiver spaces. Disturbing noise can also be created by the airflow inside HVAC ducts and air conditioning outlets. However, this topic is not addressed here because no relevant references were found.

4.1.1   Prediction and analysis

Transfer Path Analysis (TPA)

TPA is based on experimental results and simulation and is widely used in ship and offshore structure design. Janssen and Buiten (1973) presented the first well-known paper. The prediction procedures in TPA estimate ship machinery source levels and calculate structural vibration transmission losses through the ship hull. SNAME's

(2001) 'Design Guide for Shipboard Airborne Noise Control' presents the transmission losses of intersections between source and receiver rooms. However, nowadays vibration transmission losses are usually calculated by Statistical Energy Analysis (SEA) methods and therefore TPA is not further discussed here.

Statistical Energy Analysis

SEA is well established as an efficient tool to estimate structural vibration and noise levels in the high frequency range. In order to obtain accurate results some restrictive assumptions make SEA difficult to apply to structures of complex geometry, such as ships. At first according to SEA assumptions, the SEA model has to be divided into reverberant and weakly coupled subsystems. Totaro *et al* (2004) proposed an automatic structural partitioning technique to identify proper SEA subsystems. Their method is based on the analysis and classification of energy transfer functions simulated by FEA, with the structure being successively excited by point forces. Application of the technique on a part of a high speed train is presented. Secondly, input parameters of the SEA model, such as Coupling Loss Factors (CLF) and Damping Loss Factors (DLF), have to be calculated or estimated using suitable methods. For complex structural junctions it becomes particularly difficult to derive a complete and accurate analytical description for this purpose. Baldanzini *et al* (2004) presented a new approach for the calculation of CLF of structural junctions for SEA models which combines the analytical model of a travelling wave in semi-infinite panels and a FEA model of the junction.

Power Flow Finite Element Analysis (PFFEA)

PFFEA is a more recently developed method for the prediction of the vibration behaviour of structures in the mid and high frequency range. Thanks to the fact that in PFFEA the governing equation for energy distribution in the steady state condition is a spatial differential equation, the spatial variation of vibration intensity in a structural component and the energy density can be computed. Compared to SEA methods, which give only an average value of energy density for a subsystem, this represents a significant advantage. Another advantage is that FE models of the structure can be used for PFFEA analysis, for instance the same model as used for the strength analysis of the vessel. However, also in PFFEA the effects of junctions between structural components is difficult to model because the power transmitted by the bending, the longitudinal and the shear waves has to be considered and redistributed to the adjacent structural components in a realistic way. Also here CLFs are used but there calculation is more challenging than the CLF calculation for SEA analysis.

Gilroy *et al* (2005) reported on the feasibility of using PFFEA methods for structural acoustic applications in ships and marine structures. Validation of the PFFEA for the steady-state power flow capabilities has been demonstrated using simple models (e.g. beams) and more complex models (e.g. stiffened box structures). The authors proposed a hybrid system of PFFEA and SEA for the analysis of ship noise. PFFEA was applied

to analyse the structure borne energy flow in and around the sources and the SEA approach was used to calculate the transport of energy away from the machinery space.

Park *et al* (2005) reported on an application of PFFEA for the vibration analysis of built-up structures on the example of a DDG-51 type destroyer. They presented the flexural energy distributions of the complete PFFEA model consisting of abt. 16000 nodal points and 5000 elements. Further information on the application of PFFEA to built-up structures by using CLFs derived for SEA analysis was provided by the same authors in Park *et al* (2007).

Hybrid Methods

For built-up structures SEA can provide accurate predictions in the high frequency range but not in the mid-frequency range. Also FE methods can not be employed effectively because a prohibitively large number of DOFs would be required to establish a suitable analysis model.

The use of SEA methods can be justified as long as the structure to be investigated is large compared to the length of the sound wave propagating through it. Typical shipbuilding structures are built up from plates and stiffening beams. Whereas plates are large compared with the length of the sound wave, this can not be said for the stiffening beams, i.e. energy transfer can be well reflected for the high frequency plate vibration but not for the mid-frequency beam vibration. This is often referred to as 'mid-frequency' problem. Several approaches use hybrid methods to solve this problem.

Shorter *et al* (2005), Langley *et al* (2005) and Cotoni *et al* (2007) presented a hybrid method which combine FEA and SEA for the prediction of the steady-state response of vibro-acoustic systems in the mid-frequency range. Subsystems with long wavelengths (or with few vibration modes) were modelled deterministically with FEA, while those with short wavelengths (or with many vibration modes) were modelled statistically with SEA. Comparisons between numerical and experimental results for the mean squared velocity response in the plates were presented for a framework panel structure. However, further validations will be required in order to demonstrate that the method works well also for more ship-like structures, i.e. complex geometrical arrangements with numerous joints and partly coupled to acoustic fluids.

Mace *et al* (2006) presented a mode-based hybrid method in which the short wavelength components are described statistically by effective mass and loss factors added to the long wavelength components. The power transmitted from the long wavelength component (e.g. beam) to the short wavelength component (e.g. plate) was computed with the aid of a component mode synthesis method.

Numerical and experimental examples for a beam-stiffened plate structure were presented by Hong *et al* (2006). They used a hybrid FEA method, where the plate bending vibration was analysed by PFFEA and the plate in-plane vibration by

conventional FEA. In a first step the power flow from in-plane to bending vibration was computed by FEA and, subsequently, the amount of energy of plate bending vibration was determined by PFFEA. The hybrid approach considerably reduced computational cost compared to conventional FEA.

4.1.2 Applications in the Marine Field

Also in ship and offshore industry we can observe an increasing use of commercial SEA tools interfacing with CAD software. Eco-compatibility and comfort are keywords in today's maritime business. Low noise levels are one important pre-requisite for the comfort of passengers and crew. Particularly on cruise ships and ferries high standards are demanded and several classification societies offer corresponding voluntary notations for the certification of categorized grades of comfort. So noise control, i.e. predictions and countermeasures, has become an essential part of the whole design and construction process for high value vessels. Strict noise criteria may not only require a large amount of noise control measures, but may also force the designer to opt for cost intensive solutions with regard to room arrangement, structural design and the selection of the machinery. Abrahamsen (2005) presented a noise control strategy for noise critical vessels, as luxury yachts, cruise vessels and research vessels giving an overview of the variety of interdependent steps involved in this process.

Annicchiario *et al* (2005) presented a computational noise prediction method based on SEA. The authors suggested an iterative approach, i.e. in the design phase the effects of parameter or geometry changes are investigated in a comparative way, and later in the construction phase, the analysis model is improved by using DLFs and CLFs which are determined experimentally for the actual ship. In this way prediction accuracy can be enhanced with progressing design and construction stage.

Also Spence *et al* (2005) and Boroditsky *et al* (2007) presented a noise prediction program for surface ships. It uses a SEA algorithm in combination with conventional civil engineering acoustical approaches. Improvement of calculation speed and accuracy is reportedly achieved by using specific pre- and post-processing tools, choosing large size elements to extend the analysis range to low frequencies and by using measured sound pressure and machinery foundation vibration accelerations as source level input. They demonstrated the application of their method on the example of the Fisheries Research Vessel 'NOAA FRV-40 OSCAR DYSON' and carried out measurements for the validation of their method. In Fischer (2005) for the same vessel the comprehensive noise control plan is described which was established to meet the specified criteria regarding noise and vibration habitability, sonar self-noise and underwater radiated noise.

4.1.3 Noise Criteria for Interior Ship Noise

This Committee's report of ISSC 2006 contains an extensive description of the commonly noise criteria related to interior ship noise. IMO Resolution A.468 (1981) is

most frequently applied to specify maximum airborne noise levels for the living and working spaces of merchant ship crews, e.g. 60dB(A) and 65dB(A) for accommodation spaces and offices, respectively.

Also the voluntary comfort notations of some classification societies provide noise level limits. E.g. the limits according DNV's Comfort Class Conf-V(1) applicable to cruise ships are 55dB(A) for public spaces and 44-49dB(A) in passenger cabins. As presented by Hamalainen *et al* (2005) the world's largest cruise-ferry 'M/S COLOR FANTASY' (75,000GT) was designed to meet the highest DNV comfort rating for noise and vibration. In a similar way the North-Atlanic Cruise Liner 'Queen Mary II' was designed to meet LR's highest comfort rating, see Blanchet *et al* (2005).

Carlton *et al* (2005) highlighted some topics involved in the determination of suitable noise acceptance criteria. The biggest challenge is that the perception of noise is highly subjective and varies significantly between individuals. For passenger ships, people's expectations focus on hotels as being the nearest equivalent but this expectation is difficult to achieve on board. Often ship builders and owners agree on a classification society's guideline to define a desired comfort level. Such comfort notations not only specify airborne sound level limits but also requirements concerning the sound insulation properties of walls and decks, to avoid as much as possible the disturbance created by human activity, i.e. the sound and impact sound insulation between adjacent cabins and below sport decks need careful attention. Also the noise levels in open deck recreation areas, as emitted from the funnel of diesel engine exhausts for instance, must be reduced until acceptable levels. All these requirements will cause additional weight, cost and a careful noise control strategy including extensive on board measurements. In Blanchet *et al* (2006) a rough estimate of the cost increase resulting from the strict noise requirements as normally encountered in cruise ship industry is presented.

In recent years, a tendency for tighter environmental regulations with regard to ship noise emissions could be observed. E.g., some Scandinavian ports defined airborne noise limits which must not be exceeded in vicinity of cruise ship terminals and also during passage of ferries closely to densely populated areas. However, no specific publications concerning this topic were found.

4.2   *Underwater Radiated Noise*

Underwater-radiated noise from ships includes both, noise radiated from the ship's vibrating shell and noise radiated by hydrodynamic noise sources, including propellers, cavitations, vortices and turbulent boundary layer flows. Many investigations concerning underwater noise are classified and publications on this topic are rare. So the review presented here can only highlight some aspects of recent research and development on underwater sound phenomena.

4.2.1   *Propeller Noise*

Despite modern experimental and numerical methods the prediction of propeller noise is still a very challenging task. Recent research activities addressed several complementary aspects of this topic and the advances in numerical software seem promising. Seol (2005) applied a panel method coupled with a time-domain acoustic analogy to calculate the noise generated by a propeller in non-uniform flow. Lee *et al* (2006) reviewed methodologies and the state of the art of the computational tools. Van Wijngaaden (2005, 2006) used a BEM method for the determination of the propeller source strength from a given set of hull pressure measurements by means of an inverse computational technique. Ligtelijn *et al* (2006) demonstrated the importance to correlate theoretical or experimental predictions with full-scale measurements by a number of case studies and highlighted the difficulties in predicting high order propeller harmonics. Raestad (2006) emphasised that the broadband pressure field generated by the propeller tip vortices has to be considered to obtain a realistic estimate of the propeller source strength. Friesch (2007) summarized the challenges regarding the experimental prediction of propeller generated noise and reported on parameters which might improve this situation. In Fischer (2008) it was briefly reported on practical solutions regarding the phenomenon of singing propellers.

4.2.2   *Applications in the Marine Field*

For underwater noise sensitive vessels Fischer (2005) and Abrahamsen (2005) emphasised the importance to establish and follow a systematic noise control plan during the whole design and construction process. To obtain a very low under-water acoustic signature, corresponding to an increasing demand for quieter ships, numerical and experimental results appear complementary to achieve an accurate prediction. Frechou *et al* (2004) proposed a methodology for the prediction of propulsor noise signatures. The authors addressed the complexity of the phenomena and summarized the numerical and experimental state of the art. A study dedicated to pod systems was presented by Crepel *et al* (2004). They used BEM, FEA and SEA tools and adapted each method to a specific task and frequency bandwidth. While there is no universal method or formula to predict underwater sound radiation, several methodologies appear complementary. For example, Zou (2003) and Tong (2007) studied the fluid-structure transmission with a combination of BEM and FEA models. All authors emphasize that theoretical noise predictions are very helpful for the decision making in the early design stages. For instance, different room arrangements can be simulated, guidance can be obtained whether additional damping measures are required and also the most suitable positions for underwater acoustic systems can be determined.

In recent years a growing concern on the effects of ship generated underwater noise on the marine fauna can be observed. Possibly IMO will introduce regulations requiring low acoustic signatures, especially in sea areas frequented by marine mammals So it is rather likely that in future the underwater sound radiation characteristics will become a relevant design parameter also for cruise ships and merchant vessels.

Hazelwood (2005) reported on a cable laying ship for which authorities required an

environmental impact study concerning the effect of underwater radiated noise on the local fish species. Acknowledging the considerable uncertainties related to the strength of the noise source, the underwater noise transmission in shallow-water coastal areas and the frequency dependent sensitivity of different fish species, the author suggested a simple first principle method to judge the related environmental impact.

Kim *et al* (2008) developed an Energy Flow Analysis method for the prediction of underwater noise and applied it exemplarily to a LNG carrier. The authors computed the underwater sound pressure levels radiated by the propulsion unit, the regasification plant and the stern thruster and compared them to comply with the given limit criteria. The calculated underwater sound pressure levels in a horizontal plane 20 m below the surface and the vertical CL plane are presented in Figure 29. Naturally, maximum levels were obtained in vicinity of the noise sources.

4.2.3   *Noise Criteria for Underwater Radiated Noise*

Abrahamsen (2007) reported that underwater noise pollution is currently an important topic with considerable ongoing research. To date the criteria have been contractual rather than statutory, and two recent examples illustrate this observation. For a vessel dedicated to deep sea oceanographic expeditions and hydro-graphic missions Babin *et al* (2006) investigated the propulsion characteristics of a new family of electric converters and compared underwater noise levels against the stringent criteria. Also Bahtiarian and Fischer (2006) presented interesting results concerning the ICES requirements, see Figure 29. The authors highlighted the measuring difficulties encountered while assessing these stringent noise level criteria. They ultimately recommended further investigations into the technique for underwater noise measurement and analysis. Pescetto *et al* (2006) developed a light-weight system for the measurement of underwater sound and used it to measure the sound pressure levels radiated by a cruise ship, see Figure 28.

It appears that presently there exists no standardised method for the measurement of underwater noise originating from ships. Since this considerably complicates the comparison and assessment of measurements from different facilities, the Committee encourages standardisation of such measurements.
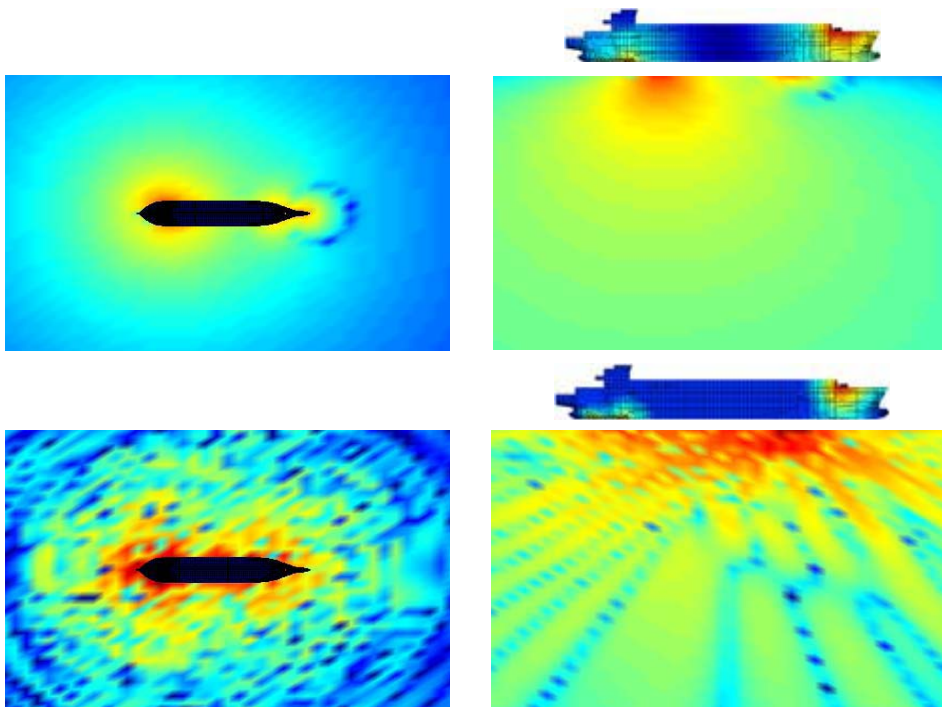
Figure 27: Underwater sound pressure levels at 31,5 Hz (upper pictures) and 500 Hz (lower pictures) radiated by LNG carrier, adapted from Kim *et al* (2008)

Figure 28: Cruise ship during underwater noise measurements, Pescetto (2006)
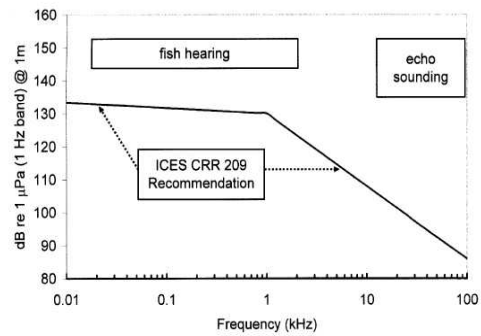
Figure 29: Recommendations of ICES CRR 209 (1995)

5.   OFFSHORE APPLICATIONS

The offshore oil and gas industry is very actively pushing into ever deeper water, and the high price of oil and gas has only increased the incentive for such pursuits. As a

result, an extraordinary quantity of current technical papers are of interest to the dynamic response committee due to the number of offshore projects under development, as well as the magnitude of the investment, rewards, and challenges. The number of papers, particularly the number of papers concerning the dynamic response of risers, umbilicals, and pipelines, precludes an adequate review within the constraints of report space and committee resources. Consequently, this review of dynamic response as it applies to offshore applications is little more than a survey.

The following review is organized into three major offshore topic areas. The first and most significant, topic area concerns risers, umbilicals, and pipelines. The third topic area concerns damping and countermeasures. And the second concerns all other offshore topics that fall under this committee's mandate. Each of these three topics is further sub-categorized according to the dominant mode of investigation represented by the pertinent literature: full-scale, model testing, or analytical.

5.1   *Risers, Umbilicals and Pipelines*

Marine risers, umbilicals, and exposed pipelines are all hydroelastic structures that share some common characteristics. Foremost is the propensity towards vortex induced vibrations (VIV). Much of the concern is for fatigue, but risers may also interact with the wakes of other risers and/or clash. The spacing of risers to avoid clashing can be a major factor driving the cost of a new production project. Through the application of generous factors of safety it has been possible to design reliable riser systems in water depths of less than 1,500 m, but both technical and economic challenges act to reduce the practical factors of safety as the depths increase towards 3,000 m and beyond.

There are a great variety of marine riser types. A partial list includes: steel catenary risers (SCR), top tensioned risers (TTR), flexible risers, free standing risers (FSR), and highly compliant risers (HCR). To this partial list must be added tendons and umbilicals. As regards dimensions of typical risers, aside from the overall depth, a partial listing of typical diameters would include: 4 inch (10.2 cm) flowline SCR, 4 inch flowline SCR with 9 inch (22.9 cm) buoyancy, 21 inch (53.3 cm) drilling riser, 48 inch (121.9 cm) drilling riser with buoyancy, 18 inch (45.7 cm) export SCR, 120 inch (304.8 cm) aircans, and 8 inch (20.3 cm) tieback risers.

5.1.1   *Full-scale (including large sub-scale in natural environment)*

Full-scale instrumentation and/or tests with sub-scale (but still large) models in the natural environment is an important mode of investigation for marine risers and umbilicals. Considerable proprietary full-scale data has been and continues to be collected by offshore operators. The Deepstar JIP project is an important source of data that has been at least partially exposed through professional publication. Earlier Deepstar sponsored tests, primarily in uniform flow, were conducted in Lake Seneca and have been addressed in this committee's report to ISSC 2006. The recent significant Deepstar sponsored tests now appearing in the technical literature were

conducted in the Gulf Stream current and are known as Miami1 and Miami2. Table 4 summarizes the salient characteristics of these tests.

Table 4
Test characteristics of deepstar jip Miami 1 and Miami 2 experiments

	Miami 1	Miami 2
Year of experiment	2004	2006
Inner Diameter	26.7 mm	24.9 mm
Outer Diameter	35.6 mm	36.3 mm
EI	488 Nm2	613 Nm2
Modulus of Elasticity (E)	1.586 x 1010 Nm2	x 1010 Nm2
EA	3.78 x 106 N	3.322 x 106 N
Weight in Seawater (flooded)	1.75 N/m	1.93 N/m
Weight in Air s/trapped water	12.11 N/m	7.45 N/m
Density	1.47 g/cc	1.383 g/cc
Effective Tension (submerged bottom weight)	3225 N	3225 N
Material	Carbon fiber-epoxy	Glass fiber-epoxy
Length	147.3 m	152.4 m

The Miami1 and Miami2 tests were towing tests conducted in the Gulf Stream, with velocity profiles ranging from nearly uniform to highly sheared in speed and direction. Reduced velocities ranged from about 3 up to about 7. Swithenbank and Vandiver (2007) observed that reduced velocities of 4.5 to 6.5 were associated with the regions with the largest RMS strain responses. Experiments were conducted with bare pipes, and also with various arrangements of VIV suppression strakes.

Examples of papers based on the Deepstar Miami1 and Miami2 field experiments include Constantinides and Oakley (2006), Vandiver *et al* (2006 a,b), Swithenbank and Vandiver (2007), Marcollo *et al* (2007), and Cheng *et al* (2007). Many other papers make use of Deepstar results or otherwise cite Deepstar.

Other full-scale riser papers include Srivilairit and Manuel (2007) and Maeda *et al* (2007). Srivilairit and Manuel (2007) applied proper orthogonal decomposition (POD) to field measurements of a deepwater (1000 m) drilling riser. Maeda *et al* (2007) compared full-scale measurements on a 175 m long, 1.0 m diameter ocean nutrient enhancer riser pipe supported by a floating spar in Sagami Bay, Japan, with forced oscillation model tests in the deep-sea basin of the National Maritime Research Institute using finite element calculations.

Lie *et al* (2007) report on towing tests in the ocean basin at MARINTEK of a 20 m section of a 120 mm diameter full-scale umbilical intended for the Ormen Lange field. Tests were conducted over a speed range from 0.3 to 2.5 m/s, with speeds up to 0.9 m/s

corresponding to anticipated current velocities at the site. The higher towing velocities were included to support studies of multi-mode response and critical Reynolds number responses. The umbilicals were tested both bare and with different strake configurations.

5.1.2   Model tests

Small scale model testing in the laboratory is the primary mode of investigation reported by a number of researchers. Braaten *et al* (2007) investigated the cross-flow response, and in a few cases the combined cross-flow and torsional rotation response, of a two pipe bundle, over a reduced velocity range from 1 to 20. They described the responses up to a reduced velocity of about ten as VIV and the responses at higher reduced velocities as galloping.

Kuiper *et al* (2007) performed experimental studies of the dynamic instability to axial internal flow of a model cooling water intake pipe cantilevered from above the free surface. An intake pipe, such as that modelled, may have application in ocean thermal energy conversion (OTEC), or as a source of cooling water for a floating offshore plant engaged in liquefaction of natural gas. Their experiments clearly showed that above a critical velocity, the pipe flutters with amplitudes of a few pipe diameters. Two response phenomena were identified. The first was a nearly periodic orbital motion with amplitude of a few pipe diameters. The second is described as quasi-chaotic with small amplitude. It is concluded that offshore water intake risers could experience instability due to internal flow at realistic flow velocities.

Prastianto *et al* (2007) towed three different model tensioned risers oriented horizontally beneath the carriage in a conventional towing tank, hence corresponding to a non-sheared velocity profile. Two risers orientations were tested, one normal to the direction of towing and the other at a 5-degree inclination to the towing tank axis. Responses at lower towing speeds were narrow banded; but, at the highest towing speed, multi-mode responses developed, including the 8th, 9th, and 10th modes.

Lee and Allen (2007) made use of a large rotating arm test facility to conduct VIV experiments of model marine risers in highly sheared flow. This was accomplished by arranging the model risers horizontally, aligned underneath the radial arm. The velocity then varied linearly along the model riser in proportion to the radius from the rotational axis of the rotating arm. Model risers were tested in bare conditions and with a variety of distributions of smooth strakes, rough strakes, and fairings.

Wu and Larsen (2007) also made use of a rotating arm facility to conduct VIV experiments of a long flexible cylinder. Their arrangement was slightly different in that the test element was arranged in the plane underneath the rotating arm, but inclined from the vertical rather than being arranged horizontally as in Lee and Allen (2007). Only cross-flow vibrations were studied, though the authors claim that the method would also be suitable for in-line vibrations. Identification was carried out variously for

either the first 11 or 13 cross-flow response modes. Added mass and lift force coefficients were identified for a single frequency case, dominated by the third and fourth modes. One interesting phenomenon observed was a variation of the dominant response frequency over time. The authors report that the dominant frequency can be quite stable for extended periods of time and then shift to some other value, a phenomenon the authors dubbed 'time sharing'.

Sha *et al* (2007) tested a model of a flexible submarine pipeline in both regular and irregular waves. The 60 mm diameter model pipe was located at various elevations above a flat bottom. Nominal gap to diameter ratios were varied between 0.4 and 0.8. The water depth was 66.7 pipe diameters. Wave periods were between 0.8 and 2.0 seconds, and Keulegan-Carpenter numbers were less than 8.0, with subcritical Reynolds numbers. Responses in modes 1, 2, and 4 were identified.

5.1.3   *Analytical*

As might be expected, the greatest number of technical papers concerning risers, umbilicals, and pipelines were primarily analytical in character.

Cheng *et al* (2007) has already been mentioned in connection with the Deepstar Miami experiments. They report on a blind comparison between time-domain analytical code, ABAVIV, the full-scale Deepstar JIP results in a sheared velocity field, and laboratory model tests in a stepped current profile.

Fontaine *et al* (2007) studied a SCR both analytically and with model tests. They identified three different regimes, which they describe as stable, unstable, and critical, depending on the intensity and effects of the Wake Induced Oscillations (WIO). Large WIO with displacement amplitudes on the order of tens of diameters were observed in the unstable regime, as well as low frequency (below first natural frequency) oscillations. In the critical regime there was a high frequency of upstream and downstream risers staying close together, and a high probability for clashing.

Many different analytical methods are represented in the technical literature. Huang *et al* (2007) used CFD, with Chimera overset grids to model VIV. A majority of their modelled VIV was of the 2S pattern, but they also observed coalescence of vortex (C pattern). They did not detect a 2T pattern in their 2-DOF model. Da Silveira *et al* (2007) used a van der Pol type wake oscillator model and Hilbert-Huang spectral analysis techniques, and eigen-analysis through the first 50 modes, to help distinguish mode jumps of the 'simultaneous modes' and 'switching modes' types. Furnes and Sørensen (2007) used coupled nonlinear van der Pol wake oscillators to study the VIV of a free-span pipeline. De Lima *et al* (2007) used a master-slave approach with message passing interface (MPI), and a 2-D discrete vortex method to study a top-tensioned SCR. Finally, Santillan *et al* (2007) modelled highly flexible risers as elastica.

Chatjigeorgiou *et al* (2007) studied extreme bending moments in a catenary riser due to

heave motion of the top support point. The extreme bending moment occurs near the bottom touch-down point. They used the WKB approximation method to solve the eigen problems. Semi-analytical solutions for the bending moment were compared with several numerical methods, including the finite-element program RIFLEX.

Morooka *et al* (2006) studied top-tensioned risers subject to wave forcing using a Morison equation. They applied linearized Morison equations in the frequency-domain and compared those results to quadratic Morison forcing in the time-domain. Josefsson and Dalton (2007) modelled a riser as an Euler-Bernoulli beam with variable top tension, and determined the excitation using large eddy simulation (LES) methods. Their LES implementation was valid for high Reynolds numbers. Riser response was solved using the Newmark method. And Rustad *et al* (2007) studied active top tension as a means of avoiding deep water riser clashing and minimizing the required spacing between risers.

Using an invariant manifold procedure, Sanches *et al* (2007) examined parametrically excited riser response due to vortex-induced motions (VIM) and FPU slow drifting. They concluded that nonlinear modes can be an effective means of reducing the degrees-of-freedom of a riser model. And, at the other end of the frequency range, Vidic-Perunovic and Nielson (2007) examined the high-frequency response of a free hanging flexible riser to FPSO springing. They examined bending moment near the touch-down point and tension, due to FPSO wave-induced motions and springing. They also found evidence of Mathieu-type subharmonic resonance.

Blevins *et al* (2007) studied wake interference of jumpers in the case of a hybrid riser tower. Though the problem considered is not a clear example of dynamic response, it is notable for the extreme complexity of the hydrodynamic interactions and for the inherent nonlinearity.

Finally, Yao *et al* (2006) and Olunloyo *et al* (2007) modelled the vibrations of pipelines excited by the internal fluid flow. Of particular interest is the accumulation over time of irreversible longitudinal extensions of the pipe, variously known as 'ratcheting' or 'pipe walking', which has been known to result in buckling failures.

5.2   *Other Offshore Topics and Applications*

Apart from the topics of risers, umbilicals, and pipelines, there are offshore dynamic response issues of interest to Committee II.2, but they are much less represented in the current technical literature. Furthermore, many of the topics represented have been addressed by Committee II.2 reporting to previous ISSC congresses. Following are a few interesting examples of some of these current 'other' offshore dynamic response topics.

5.2.1   *Full-scale*

The only significant example of full-scale offshore response in the 'other' category is a paper by Rijken and Leverette reporting on the heave acceleration and dynamic tension in the tendons of SeaStar tension leg platform in the Gulf of Mexico in response to two separate seismic events. The first was a 10 February 2006 earthquake with an approximate magnitude 5.2, and the second was a 10 September 2006 earthquake with an approximate magnitude 6.0.

5.2.2   Model tests

Zhuang *et al* (2007) report on model tests of circular and square moonpools over a range forward speed Froude numbers from 0.2 to 1.15. Resonance, acoustics, sloshing, and pressure distributions were of interest. Square moonpools were studied at relative heading angles from 0° to 45°. Model tests were analyzed using Hilbert transforms and empirical mode decomposition (EMD).

5.2.3   Analytical

Ergin and Uğurlu (2006) present a linear boundary element method suitable for the general problem of dynamic response of the boundary elements to internal axial flow (fluid conveyance). An example is given for the first five response modes of a simply supported cylindrical shell, analogous to a pipeline. Except for the generality of the method, this could also have been classified as an analytical pipeline dynamic response paper.

Huang and Liu (2006) present an interesting paper analyzing the ice-induced vibration (IIV) due to relative velocity between a structure and an ice sheet. They identify a resonant frequency lock-in for this process, which they describe as analogous to VIV lock-in. Their model accounts for discrete ice failure, the dependence of crushing strength on the ice velocity, and the randomness of ice failure. They claim that their method reasonably explains the resonance, is supported by laboratory and field observations, and represents an improvement over previous models.

Taghipour *et al* (2006) present a validation of hydroelastic analysis using the method of generalized modes as implemented in standard software. They show good agreement between three methods through the first ten modes.

Teigen *et al* (2006) examined the heave and pitch of a tension leg platform (TLP) subject to irregular seas. Quadratic transfer functions (QTFs) were obtained from a second-order diffraction code. Stochastic response was estimated using a Volterra series. The method presented is appropriate to springing, but not to tendon 'ringing'.

5.3   Damping and Countermeasures

Elements of damping and countermeasures have already been addressed above in the guise of top-tension control for risers and strakes, or fairings applied to risers. In this

sub-section, the recent technical literature on damping and countermeasures as applied offshore will be briefly assayed.

5.3.1   Full-scale

In another paper based on the Deepstar Miami experiment series, Jaiswal and Vandiver (2007) report on natural environment sub-scale measurements of damping measurements for a triple helix strake design with a pitch to diameter ratio of 17.5. Varying amounts of strake coverage were tested. In the seven test runs reported, damping ratios varied from 0.06 to 0.1 with a mean value of 0.08. Of interest, the measured responses revealed characteristics of traveling waves. Limitations of mode superposition are demonstrated, and an alternative method based on Green's functions is proposed.

5.3.2   Model tests

Schaudt *et al* (2008) report on a tests comparing several VIV suppression devices, including the AIMS Dual-Fin Flow Splitter (ADFS), 5d and 15d pitch strakes, conventional teardrop fairings, all benchmarked against bare pipe. Testing included high-mode number in-situ tests, both low and high Reynolds number tests, and wind tunnel and in-water Particle Image Velocimetry (PIV). Reductions in VIV response of over 90% are reported for both the strakes and the ADFS. Increases in fatigue life of four to six orders of magnitude are found for the example SCR.

5.3.3   Analytical

Smith *et al* (2007) and Tan *et al* (2007) both addressed bending hysteresis of flexible risers. Aside from analysis, Tan *et al* also presented some full-scale experimental data from tests of a 4-inch pipe at three different internal pressure levels.

Chatjigeorgiou *et al* (2008) investigated the damping of catenary risers using both analytical methods and 12.5:1 scale experiments.

6.   CONCLUSIONS

Ship Structural Vibration

Considerable progress has been achieved to better understand the mechanisms of wave induced ship vibration. Tank test methods and numerical approaches have been significantly improved and a variety of full scale measurements has been performed or is underway. So presently, reasonably accurate prediction of hull whipping and springing response appears possible as far as some selected scenarios with given wave and ship operation parameters are to be considered. However, due to the stochastic nature of the sea way, the non-linear character of the response transfer functions and

the impossibility to exactly predict the ship's operational parameters, there still remains some uncertainty in combining the available prediction tools with long-term statistical methods in a meaningful way. Presently, researchers use simplified approaches for this purpose, e.g. the design wave concept in numerical analyses or response conditioned waves for tank tests. Design methodologies based on 1st principle calculations and explicitly reflecting the enhancement of extreme and fatigue loads due to wave induced vibration in connection with long-term statistics are considered not yet mature enough for industrial application. Also more endeavours should be made for the determination of suitable strength assessment criteria and reasonable limit values for such kind of calculations. For this purpose realistic assumptions concerning the expected wave climates and ship operation parameters are indispensable, and so the Committee encourages enhanced full scale monitoring of hull stresses in combination with registration of the encountered wave environments and actual navigational parameters. Only in this way it can be validated whether new 1st principle design methodologies will enable the development of economic ships without compromising structural safety and integrity.

The prediction of propeller vibration excitation forces still appears to be rather uncertain, especially in the higher frequency range where viscous flow effects become more important. The increasing advance of CFD methods into the standard engineering design process might improve this situation in the coming years. Not only because more realistic propeller flow computations will be possible, but also because CFD codes can be supposed to better predict full scale wake fields. Last but not least also the analysis of flow induced vibration is believed to benefit from the enhanced integration of CAD and CFD tools in the design process.

Naturally, the determination of vibration excitations originating from the main engine, gear or shafting requires the consideration of machinery–hull interactions. Some new knowledge has been gained in this field but no general conclusions are evident.

Damping tends to remain a somewhat uncertain parameter in ship vibration analysis. Until today a generally applicable approach for damping estimation has not been found and recent publications confirmed that measured damping constants can vary significantly depending on the type of ship and the vibration frequency and mode shape. It appears that reliable data can only be obtained experimentally, provided that test objects with similar structural arrangement are available.

No mayor progress has been made in standardising methodologies for ship structural vibration analysis. This is not surprising because the diversity of ship types, propulsion plants and comfort requirements makes the definition of generally applicable standards or guidelines virtually impossible. Moreover, with the exception of torsional shaft vibration, there exist no regulatory requirements from class or flag state authorities to perform vibration analyses. Last but not least the levels of technological expertise to perform such calculations are somewhat different. With regard to experimental investigations better progress was made through continuation of the development of the

ISO 20283 guideline on ship vibration measurements. Nevertheless, it is suggested to verify whether the addition of a specific part on the effect of transient hull vibration, as excited by slamming impulses or ice impacts for instance, on the crew's health and working performance and on passenger comfort might be worthwhile.

Apparently, also in shipbuilding industry active vibration control becomes increasingly important. First applications have been realised on naval vessels and luxury yachts. Provided that more cost efficient systems will be developed there is seen significant potential for merchant ship applications too.

Further progress has been made in the development of economic, fully automatic measurement systems which are capable to handle a large number of measurement channels with on-line signal processing features. It is believed that such means to systematically track ship vibration characteristics through the whole life time, including different loading, propulsion and environmental conditions, will significantly facilitate comparisons with the results of theoretical investigations. Also vibration measurements for the purpose of crack growth monitoring appear mature enough for prototype installations.

Shock Response

The effectiveness and accuracy of numerical methods for the simulation of gas bubble dynamics and transient hull vibration under consideration of fluid structure interaction effects has been further improved. Nevertheless, it seems that more development work is needed until the numerical codes are mature enough to be used as standard CAE tools by ship yards or design offices.

Great potential for improving the shock strength of local and global hull structures is seen in the application of composite materials and a variety of references reported on corresponding investigations. In this connection knowledge of the damping properties and specific failure mechanisms of the individual composites is of utmost importance and so material tests in connection with numerical simulations are believed to be indispensable to achieve reliable designs.

Ship Acoustics

There have been continued endeavours to improve the accuracy and effectiveness of noise predictions. Depending on the frequency range of interest SEA, FEA, BEM, PFFEA methods and combinations thereof were developed. Despite this progress, complementary experimental investigations are still indispensible to achieve realistic prediction results. This is particularly true for the determination of the strength and frequency content of the sound sources.

Regulations on ship noise are expected to be enforced in the near future. First of all the adverse effect of long exposure to high noise levels on the working ability of the crew

is increasingly acknowledged. Secondly, also the sound emitted by ships into the environment is of growing concern. This is true for airborne noise, for which several harbour authorities tend to define limit criteria, as well as for underwater sound, for which regulations are presently discussed at IMO. From these trends it is concluded that in future the availability of accurate and effective noise prediction methodologies is of significant importance and, therefore, the Committee encourages further research activities in this field.

Offshore Applications

There has been a compelling amount of new publications on VIV of risers, umbilicals and pipelines. Due to continuing exploitation of natural resources in steadily increasing water depths this trend is believed to continue for the coming years and the Committee recommends to appoint an expert committee on this issue for ISSC 2012.

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