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VOLUME 1

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 dynamic responses resulting from environmental, machinery and propeller excitation. Uncertainties associated with modelling should be highlighted.

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KEYWORDS

Dynamic response, slamming, whipping, springing, damping, vibration, noise, underwater noise, explosion, shock, excitation, propeller, vortex-induced vibration, ice-induced vibration, lock-in, prediction, natural vibration, forced vibration, linear response, non-linear response, countermeasures, measurement, segmented models, hydro-elasticity, resonance, monitoring, model tests.

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1 INTRODUCTION

The role of the engineer in society is to guide the wise use of scarce resources (e.g., natural resources, human resources, capital, etc.) in order to sustain and improve the human condition, while minimizing direct or collateral damage to the environment and losses from accidents (casualties, injuries, environmental damage, damage to property, etc.).

In structural design, this leads to an optimization process that seeks a balance between the use of materials and the probability of undesirable consequences arising from structural insufficiency. This process trends towards light, slender, and flexible marine structures. In some cases (e.g., deep water risers), we are near the very limits of what can be done because the structural deadweight alone dominates the demand on the structural capacity of our materials.

Light, slender, and flexible structures are inherently prone to dynamic responses. The potential exciting mechanisms are legion, and include: waves, vortex-induced vibration (VIV), ice, internal flow, machinery, propellers, and blast. Dynamic responses exacerbate fatigue, contribute in some cases to extreme lifetime loads, adversely affect habitability, and result in airborne and underwater radiated noise that degrades the environment and interferes with some marine operations.

The overarching trends observed by the 2009 II.2 Dynamic Response committee remain valid; however, there are conflicting imperatives. One challenge is to ensure sufficient energy for a growing world population along with the transportation essential to a globalized economy. The other is to reduce the impact on the environment, most especially as regards the emission of greenhouse gases. The continuing and accelerating trend towards reduction in the extent and thickness of Arctic sea ice is stimulating new interest in commerce on the Northern Sea Route, and increased activity to access and exploit offshore resources in the Arctic Ocean basin. These prime motivating forces are manifest in the maritime and offshore communities through:

- increases in ship sizes, to benefit from economy of scale,
- offshore oil and gas development in ever deeper waters, such as those associated with the Brazilian pre-salt deposits,
- renewed interest in deep water ocean mining,
- growing interest in offshore renewable energy (wind, wave, current, and thermal), especially as realized through increasing development of offshore wind turbine fields and an interest in moving wind turbines into deeper waters using floating foundation concepts,
- greater shipping and offshore activity in ice-covered and/or ice-infested waters.

These trends contribute to the continued, active research into dynamic responses that range from whipping and springing, to VIV or flow-induced vibrations as they apply to slender structures such as deepwater risers, or the ice-induced vibrations of offshore structures. Vibration is particularly crucial in the fatigue of offshore wind turbines.

The Deepwater Horizon explosion has served to remind us of the continued importance of responses to explosion and blast.

As a further means of fulfilling our mandate to elucidate uncertainties, the 2012 II.2 committee has undertaken a benchmark study regarding whipping responses.

The 2012 II.2 committee has chosen to divide this report on dynamic response, at the highest level, into ship structures and offshore structures.

The 2012 II.2 committee collected over 440 references and considered even more. This report cites and reviews 258 references.

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2 SHIP STRUCTURES

2.1 Wave-Induced Vibrations

Wave-induced vibration of ships can 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 seaway components with short wave lengths is in resonance with the natural frequency of the basic hull girder mode, and nonlinear 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; 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 higher energy content.

2.1.1 Whipping

Full scale measurements and model tests have been intensively conducted in recent years. These tests and measurements were mainly focused on unconventional larger ships, such as container ships, naval frigates, and LNG carriers. Much research focused on the effect of wave-induced vibration on fatigue performance of the vessels. Furthermore, research was conducted for the wet deck slamming for high-speed catamarans.

Hydrodynamic Excitation, Response, and Load Analysis

Ogawa *et al.* (2009) developed a practical prediction method for wave loads in rough seas that took hydroelastic vibration into account. Comparison with the experiments for a post-Panamax container ship and a mega container ship demonstrated that using this method, taking into account the time-varying sectional hydrodynamic forces, gave favourable agreement with measured wave-induced vertical and torsional moment under rough sea conditions.

Tiana *et al.* (2009) investigated the hydroelastic response of a large bulk carrier. The numerical programs THAFTS and NTHAFTS, which were developed based on linear and non-linear three-dimensional hydroelasticity theories, were used to analyze the rigid body motions and structural responses of a large bulk carrier travelling in regular and irregular head waves. The influences of the forward speed effect and the nonlinear hydroelastic actions on the motions and structural loads of this large bulk carrier were quantitatively discussed.

Mikami *et al.* (2009) extended the nonlinear strip method, which was proposed by one of the authors for an elastic body using a method of superposition of elastic mode functions, which enabled investigation of whipping phenomena due to wave impulses. Computed and measured whipping bending moments were compared. The effects of green water-on-deck were investigated through application of a practical computational model.

Miao *et al.* (2009) focused on the investigation on the failure of the *MSC Napoli* using two-dimensional (2D) symmetric (i.e., vertical bending) hydroelasticity analysis. The aim of the investigation was to assess the influence of whipping-induced loads on the structural strength of this containership. The calculations were carried out in head regular and long-crested irregular waves. Both cases included the effect of

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bottom slamming only. Global wave-induced loads were evaluated along the hull, with particular focus in the vicinity of the engine room. The investigation showed that whipping due to bottom slamming was only important for severe seas. It also showed that the keel stresses in way of the engine room could be as large as the keel stresses at amidships.

Storhaug *et al.* (2009) reported on the contribution of whipping on the 4400 *TEU* container vessel, *MSC Napoli*. Based on some measurements from a similar 4400 *TEU* vessel in both full scale and model tests, it was confirmed that whipping can increase the dynamic loading in sea states similar to those *MSC Napoli* encountered. The measurements also illustrated that it was difficult to state exactly how much the whipping contributed in a specific sea state.

Using a 2D hydroelastic method, Denchfield *et al.* (2009) predicted the vertical plane hydroelastic response (motions, loads and elastic hull girder deflections) of a Leander class frigate in long-crested regular, irregular and rogue waves. Predicted motions are compared with test measurements from a rigid model in head irregular and rogue waves. The effects of forward speed and slamming in rogue waves were examined.

Derbanne *et al.* (2010) developed an efficient hydroelastic model based on the state of the art numerical techniques. The model coupled the 3D hydrodynamics with either non-uniform beam or 3D FEM structural dynamics. This model was demonstrated to have been very efficient in terms of the CPU time. The computational results were compared with several experimental campaigns and validated.

Zhu *et al.* (2010) presented a numerical model for studying the tank wall boundary condition on the measured load effects of a backbone model. Comparisons between numerical calculations and experimental data in the towing tank showed that this kind of numerical model was efficient and reliable for evaluating the tank wall interference. It was demonstrated that the wave reflections from the tank walls substantially change the hydrodynamic forces and pressures on the backbone model, as compared to the values that would be measured in open seas.

Gaidai *et al.* (2010) proposed a method for prediction of extreme stresses measured in the deck amidships of a container vessel during operation in harsh weather using the full scale measurement data. The method opens up for the possibility to predict simply and efficiently both short-term and long-term extreme response statistics.

Lee *et al.* (2011) reported time domain springing and whipping analyses for a 10,000 *TEU* class container ship using computational tools as a part of a joint industry project. The results from the computational analyses in regular waves have been correlated with those from model tests undertaken by MOERI. It was demonstrated that the wave induced vertical bending moments with whipping vibration were reasonably well predicted by the three-dimensional, nonlinear hydroelasticity method.

El Moctar *et al.* (2011) used an implicit two-way coupling scheme between a VoF free surface RANS CFD solver and a 6-DOF motions structural FEM model, to study combined springing and whipping in regular and irregular waves of a 10,000 *TEU* 321 m containership. The effects of forward speed and directional spreading were studied. Comparisons were made with experimental results from the (2009–2010) WILS-II JIP segmented model tests. Damping was also investigated.

Kirtley *et al.* (2010) expanded the valid domain for design parameters in order to more fully encompass the validation cases for a naval frigate and the ITTC S-175 containership, and to extend the range of application beyond bottom slamming to include bow

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flare slamming. The underlying model is that of a non-uniform Timoshenko beam with infinite frequency Lewis form hydrodynamic added mass. The non-dimensional prediction equations are simple quadratic polynomials in eight independent variables. The coefficients were determined by ordinary least squares regression applied to over 25,920 parametric ship designs, for maximum midship bending moment, the two-node, 'free-free' natural frequency, and the time at which the peak whipping bending moment occurred. The independent variables are basic design characteristics such as beam-to-length ratio, block coefficient, waterplane coefficient, nondimensional midship structural inertia, and other nondimensional values available during early design. Confidence intervals for prediction were provided.

Dessi *et al.* (2010) related the slamming excitation and the whipping response to each other using a wavelet transform. The analyzed data were collected with model tests using two scaled and segmented elastic models of a fast ferry and a cruise ship. The extraction and evaluation of the excitation level due to slamming loading was clarified via the construction of the envelopes of the whipping response and the calculation of its time average.

Kim *et al.* (2011a) developed an innovative and efficient method to generate independent non-identically (*inid*) distributed phase sets in order to create irregular design wave profiles to be used as inputs to nonlinear time-domain simulations suitable for use as samples in a Monte Carlo approximation of a ship lifetime exposure. The Acceptance-Rejection (A-R) method was applied to rapidly generate an ensemble of phase sets that reproduced an *a priori* specified extreme value distribution with fidelity. The authors designated this approach the Design Loads Generator (DLG) method.

To demonstrate application of the method, the nonlinear seakeeping program LAMP2 was used to predict the combination of nonlinear wave bending and whipping bending moments for two different ships, the Joint High Speed Sealift (JHSS) and a naval combatant. Model tests of the JHSS were conducted with a segmented model, and those of the naval combatant with a structurally scaled polyvinyl chloride (PVC) model. LAMP2 was used to generate an ensemble that comprised 288 independent, five-minute simulations, which resulted in a composite 24-hour record. These simulations were compared in Weibull space with corresponding data from model tests. Extreme value distributions generated from model tests and those predicted using the DLG method were shown to compare favourably.

Wet Deck Slamming and Whipping Vibration

Davis *et al.* (2009) conducted a wavelet analysis to identify slamming events during multi-hull sea trials. Maximum loads, load impulses, and energy imparted to the structure by slams were evaluated. A segmented hydro-elastic model was developed to simulate the main whipping mode, elastic links in the model facilitating measurement of bending moments, and determination of slam loads. It was found that energy is transferred to the whipping mode, which has a damping ratio in the range 0.02 to 0.06, depending upon forward speed.

Thomas *et al.* (2009) investigated slam events of high-speed catamarans in irregular waves. Slam events experienced by high-speed catamarans in irregular waves were characterised through experiments using a hydroelastic segmented model. It was tested in irregular head seas at two speeds corresponding to Froude numbers of 0.32 and 0.60. Slams identified in the test data were analyzed with respect to kinematic parameters. Slams were found to have a large range of magnitudes; however, the majority of events

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were of relatively low severity. Immersion of the centrebow to the two-dimensional filling height of the cross-section between the centrebow and demihulls was shown to be a better indicator of slam occurrence than immersion to the top of the archway.

Fatigue Associated with Whipping

During this period, many research studies have sought to clarify the effect of whipping on both the extreme load and fatigue damage accumulation. Many of these studies, however, investigated the effect of whipping on the fatigue damage only by means of estimating the increase in the number of cycles. Most recent studies did not investigate the effects of whipping vibration on the mechanism of fatigue crack propagation. Furthermore, some studies reported large amplitude wave loads that exceeded the current criteria, such as IACS/S-11. They did not, however, necessarily clarify the effect of vessel operation, such as voluntary and involuntary speed reduction or course change. Therefore, those studies indicated certain discrepancies between real structural strength in real sea state and the evaluated structural performance.

Oka *et al.* (2009a, 2009b) examined fatigue strength of a large container ship, taking the effect of hull girder vibration into account. With few reports of hull fracture attributed to such vibrations, the resulting cyclic stress effects were unclear. In order to gain insight into the fatigue loading including hull girder vibration Oka *et al.* performed their analysis utilizing both data obtained from tank tests using a backbone type elastic model, and full scale stress measurements.

Although the long-term fatigue damage was very low, it was demonstrated that hull girder vibration due to whipping had a notable effect on fatigue damage. Therefore, it was important to accurately estimate the level and probability of occurrence of whipping stress. The effect of the operational factors on fatigue damage was also quantitatively clarified.

Storhaug *et al.* (2010a) reported on model tests in head seas of a 340 m, 8,600 TEU container ship using a new flexible model design to investigate how springing and whipping affected extreme loading and fatigue at different cross sections. Storhaug *et al.* (2010b) reported on similar studies of a 360 m, 13000 TEU twin island container ship, and Storhaug *et al.* (2011a) extended those studies to bow quartering seas with and without wave spreading.

The studies of the 8,600 TEU container ship investigated exposure in different trades. The results confirmed that fatigue damage from wave-induced vibrations could be of considerable magnitude relative to the conventional wave damage for the different trades. The maximum whipping response was also found to be relatively high. With whipping, the IACS rule levels may easily be exceeded in storms.

The results for the 13,000 TEU container ship confirmed that wave-induced vibrations dominate the fatigue damage for the East Asia to Europe trade in head seas.

Storhaug *et al.* (2011a) found from model tests that the extreme loading in bow quartering seas was higher than in head seas, which was well above IACS rule loads for vertical bending by more than 80 % in hogging. In reality, this may have been reduced somewhat by higher damping. Damping from full scale measurements needs further investigation.

Further, the effect of the vibrations was investigated on torsion and horizontal bending, as the model was also allowed to vibrate with realistic frequencies in other modes in addition to vertical bending. From comparison of extreme horizontal bending and torsion with and without whipping vibrations, it was clear that the effect of whipping

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on these modes was considerable. Even though the damping for these model tests was below the expected damping in full scale, and the vibration mode and structural response differed from a real vessel, the measurements suggested that the effect of vibration should be further investigated on real ships.

Storhaug *et al.* (2010a, 2010b, and 2011a) recommended that wave-induced vibration be considered during the design phase, and made the observation that prudent seamanship is important to the minimization of whipping during storms.

Storhaug *et al.* (2011b) used model tests to consider wave-induced vibrations during the design of the world's largest bulk (ore) carrier. It was determined that the combined effect of whipping and springing vibrations more than doubled the fatigue damage compared to that associated with the wave induced loading alone. A procedure was developed to take the vibrations into account, which resulted in reasonable scantlings based on in-service experience with similar designs and trades. A structural verification for the design loads has been performed.

Heggelund *et al.* (2010) assessed fatigue damage accumulation on an LNG carrier from full scale measurement during a period of about twelve months. Based on the measurements, the vessel was found to be in actual operation less than half the time during the period. The fatigue rate was lower than that predicted by component stochastic fatigue analysis, and the fatigue life was predicted to be longer than the design life of 40 years. The contribution from vibration was found to be large; e.g., 30–50 % of the total damage. The highest fatigue damage was obtained in rough seas and in the full load condition. Most fatigue damage was accumulated in head or following seas. It was found that speed reduction in rough seas acted to reduce fatigue damage below that predicted at constant service speed. Furthermore, measurements from wave radar were compared with scatter diagrams used for design. The measured data indicated that the sea states were shifted towards a lower mean zero-crossing period.

Heggelund *et al.* (2011) reported on the fatigue accumulation determined from full scale measurements from sixteen voyages between Asia and Europe aboard an 8600 *TEU* vessel, over a period somewhat exceeding a year. Based on the measurements, it was confirmed that the fatigue loading of critical details was dominated by the vibrations. The fatigue loading level in the deck during a storm was higher than ever before measured. Also, whipping response was found to result in high extreme loads that were above IACS rule values.

Torsional and horizontal vibrations were also observed. Vibration was found to contribute significantly to side shell fatigue, but overall side shell fatigue loading was at a comfortable level.

The full scale measurements did, to some degree, confirm previous model tests of the same vessel, but the vessel had been routinely operated at reduced speeds. The fatigue loading on this route was at a comfortable level, which was partly due to reduced speeds, but also the encountered sea states may have been less severe than those on the route-specific scatter diagram.

Matsuda *et al.* (2011) investigated fatigue crack propagation behaviour under variable amplitude loading with different frequency components, such as the wave bending moment superposing slamming loads. Numerical simulation of the fatigue crack propagation based on an advanced fracture mechanics approach with the RPG (Re-tensile Plastic zone Generating) load criterion was improved to extract the effective loading sequence for the fatigue crack growth.

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The numerical simulations of fatigue crack growth curves under the loadings superimposing damping amplitude components with high frequency like slamming loads were compared with the measurements. These comparisons validated the proposed treatment of extracting the effective loading sequences for the fatigue crack propagation from random loading sequences.

Hull Structural Response Analysis

For the analysis of hull structural response, 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 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 finite element (FE) models reflecting one or several hull girder vertical bending modes.
d) Same as Item 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) Same as Item e, but with local FE mesh refinements for specific assessment purposes; e.g., stress concentration effects or local deck panel vibration.

In recent years, various approaches were utilized to analyze the hull structural response. That research focused on unconventional larger ships, such as large container ships. Furthermore, the effect of structural discontinuity on structural response of large container ship was investigated.

Dessi *et al.* (2009) investigated the correlation of model-scale and full-scale tests aimed to determine the bending response of a navy vessel in waves. The model tests were carried out with a segmented-hull elastic model scaling the mass, the sectional moment of inertia, and the shear area of the original ship. In order to assess the reliability of the experimental technique at model-scale, 1D and 3D FE models were built, analyzed, and compared to highlight how the load segmentation and the reduction to 1D models could have affected the experimental predictions. To evaluate the correlation of model-scale and full-scale, the vertical bending moments measured on board the ship during the full-scale trials were compared with those determined with the segmented-hull tests and with the simulations using a 3D finite model of the ship structure.

Senjanović *et al.* (2009) dealt with the methodology of hydroelastic analysis based on a mathematical model, which included structural, hydrostatic, and hydrodynamic sub-models. The modal superposition method was used, and ship natural modes were determined by a sophisticated beam model based on the advanced thin-walled girder theory, which took shear influence on torsion into account, as well as the stiffness contribution of transverse bulkheads. The mathematical model was checked by model test of a flexible barge. The developed numerical procedure was applied in the case of a very large container ship. The 1D FEM model was verified by correlation of dry natural vibrations with those obtained from the 3D FEM model.

Iijima *et al.* (2009) studied symmetric and anti-symmetric modes of vibrations of a generic open ship experimentally and theoretically. First, a new design strategy for a scaled hull girder model of open ships considering the configuration of the natural frequencies of symmetric and anti-symmetric modes was developed. A backbone beam

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approach with several cut-outs was found to accommodate an appropriate torsional stiffness, as well as vertical and horizontal bending stiffness. A hydroelastically-scaled model was designed based on FE Analysis (FEA). The hydroelastic vibration characteristics in regular waves were confirmed by tank tests. Numerical calculations were also performed to investigate the hydroelastic vibrations in symmetric and anti-symmetric modes.

Kaydıhan *et al.* (2011) conducted a hydroelastic investigation into the dynamic response characteristics of two Handysize bulkers, one Handymax bulker, one Panamax bulker, and two Capesize bulkers, respectively. Detailed 3D finite element structural models were prepared to compute the dry and wet frequencies using FEA, and then compared with those calculated by using a higher-order 3D hydroelasticity theory.

Cabos *et al.* (2011) investigated a modal approach to the coupled computation of the fluid flow and the motion and elastic deformation of a floating body. Compared to earlier approaches that relied on a beam approximation of the ship, torsional elastic deformations could be considered. In this approach, the equation of motion for the ship must be solved in the moving coordinate system. This approach led to full system matrices that depended on ship translational acceleration, and rotational velocity and acceleration. The additional computational effort for these small nonlinear sets of equations was nevertheless small, since only a few modes were applied. The rigid body motion was also a result of this set of equations.

Bogaert *et al.* (2010) investigated the interaction between the NO96 membrane containment system and breaking waves. The data obtained from the full scale impact tests in the Sloshel project were used. It was demonstrated that it was not straightforward to define the interaction type by a comparison of observed load durations and predefined mode shapes. As an alternative, it was investigated whether the type of interaction could be deduced from its effect on the load, and on the response with respect to a quasi-static interaction.

Ogawa *et al.* (2011) conducted a whole ship finite element analysis system from wave loads to structural strength in a realistic sea state. Methodology for the rational analysis of structural strength by means of such a whole ship analysis, particularly from the viewpoint of loads, was examined. It was verified that the evaluation, without the effect of ship operation, may overestimate the stress induced by waves. Consequently, for the rational evaluation of strength in waves, the effect of operation on wave loads should be considered.

Senjanović *et al.* (2011) dealt with a numerical procedure for ship hydroelastic analysis, with particular emphasis on improvements of the present beam structural model. The structural model represented a constitutive part of hydroelastic mathematical model, and generally it could be formulated either as a 1D FEM or 3D FEM model.

Furthermore, the improvements included accounting for shear influence on torsion, contribution of bulkheads to hull stiffness, as well as determination of effective stiffness of the engine room structure. This procedure was demonstrated using a large container ship. In this case, validation of 1D FEM model was checked by correlation analysis with the vibration response of the fine 3D FEM model. The procedure related to determination of engine room effective stiffness was checked by 3D FEM analysis of a ship-like pontoon that had been made according to the considered ship characteristics.

2.1.2 Springing

Analytical studies of springing have primarily been conducted for unconventional larger ships such as large container ship, ultra large ore carrier, LNG carrier, and

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so forth. Different hydrodynamic effects exist that might cause higher order excitations, but whether they are practically relevant is yet not fully understood. These excitations include:

- 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.

Much research was conducted on the mechanism of springing and its effect on the fatigue stress.

Hull Excitation Load Analysis

Kim *et al.* (2009a) investigated the springing response of a flexible floating barge under an oblique wave. A time-domain Rankine panel method represented fluid motion surrounding a flexible seagoing vessel, and a finite element method was used for structural response. For accurate prediction of the structural response under an oblique wave, special attention was given to the structural model, such as the effect of warping distortion and bending-torsion coupling. The Vlasov assumption was followed for a deformable beam element to take into account the effect of warping distortion, so that the cross section of the beam deformed out of its original plane without changing its cross-section contour.

The coupled equation for both the fluid and structural domain was solved by using the implicit iterative method. Accuracy of a developed computer program was verified through the comparison with experimental data, which resulted in good correspondence between the two results.

Wang *et al.* (2010) conducted an experimental investigation of springing responses of an ultra large ore carrier in full and ballast load conditions. Springing was observed in both regular and irregular head waves. It was confirmed that springing may commence beginning from low sea states. By the analysis of springing phenomena and its mechanism, good knowledge of characteristics of wave-induced responses of large ships in short waves was examined.

Miyake (2009, 2011) presented an estimation of hydroelastic response of ultra-large container ships with verification by comparison with experimental results. Time domain numerical simulations were conducted by means of the latest nonlinear strip method. Tank tests using a 12,000 TEU ultra-large container ship in regular and irregular waves were carried out to investigate the influence of hydroelastic responses on the hull structural strength of ultra-large container ships and to verify the validity of numerical simulations. It was found that the applied nonlinear strip method was a suitable way to estimate the hydroelastic responses such as whipping and springing of ultra-large container ships. Furthermore, it was found that the influence of springing on the hull girder strength of mega-container ships may be small, since the shorter wavelengths that generate springing were not associated with maximum wave heights.

Kang *et al.* (2010) carried out the time-domain springing analyses for a 263,000 m^3 LNG carrier and a 10,000 TEU containership using a newly developed program, WISH-FLEX. Using this numerical method, the physical problem was modelled in the time

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domain by fully coupling a boundary element method for the fluid domain and a finite element method for the structure domain. The detailed method of idealizing the ship into a beam model was described, and wave elevation around ship and pressure on hull surface were compared between the cases of rigid body and flexible body. Also, the motion responses and loads were compared with the towing-tank test. It was clearly shown that hull girder loads of the flexible body were significantly different from those of the rigid body.

Kim *et al.* (2009b) conducted comparisons among different numerical approaches for the ship hydroelasticity problem, particularly focusing on linear springing phenomenon. Both the Rankine-panel-based time domain approach and wave-Green-function-based frequency domain approach were cross-compared with each other for both a flexible stationary barge model, as well as a modern merchant ship model. The motion of flexible hull was represented by Vlasov-beam based finite element method. Based on the present comparative study, the pros and cons of each method were discussed.

Fatigue Associated with Springing

In this period, considerable research was conducted on the fatigue performance associated with springing. Most, however, investigated the effect of springing on the fatigue performance only by means of counting the increase of number of vibration cycles. They did not investigate the effects of springing vibration on the mechanism of fatigue crack propagation. It seems, therefore, that there were certain discrepancies between real fatigue performance in real sea state and the evaluated fatigue performance in those studies.

Jung *et al.* (2007) investigated the influence of wave-induced ship hull vibrations on fatigue damage. The strip method was employed to calculate the hydrodynamic forces and moments on the hull. The hull was modelled as a Timoshenko beam that accounted for the rotary inertia and shear deformation.

Fatigue analysis was also performed to evaluate the fatigue life of the ship, including springing effect. Wirsching's approach was applied for the wide band bi-modal stress spectra induced from springing. Applications to recent ships were performed and the influence of springing response on the fatigue damage was examined.

Wang *et al.* (2009a) investigated the effect of springing on fatigue damage. A design-oriented procedure was presented for springing induced fatigue load. The procedure included a springing susceptibility assessment and a detailed analysis of springing induced fatigue load. In the detailed analysis, a quadratic strip theory and a non-uniform beam model were employed. The procedure was applied to a number of container carriers, and influential parameters on springing induced fatigue load were evaluated.

Boutillier *et al.* (2010) conducted a fatigue damage calculation of an ultra large container ship (ULCS) due to quasi-static wave response and springing response. The validation of this method was performed by calibration with respect to the rain flow method, which the authors generally considered to be the most accurate method for fatigue damage calculation. It also revealed that the difference in fatigue damage estimates based on the spectral method and the rainflow method increased as the stress process became more broad-banded.

Cusano *et al.* (2009) examined the effect of springing on ship design. Based on the hydrodynamic and structural analysis, it was established that the evaluation of the

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bending stiffness of large fast craft, such as high speed ferries and luxury mega-yachts, should address the risk of springing. It was also established that whipping vibrations may significantly contribute to the total bending moment of ocean-going vessels characterized by large bow-flares, such as cruise or container ships.

2.2 Machinery or Propeller-Induced Vibrations

2.2.1 Propeller Excitation

Cao *et al.* (2011) addressed the problem of propeller-induced vibrations being transmitted to shafting bearings and the surrounding structure. In particular, the authors noted the problems of lateral and axial periodic excitation of the propeller shaft due to the propeller running in a non-uniform wake, and these excitations then being transmitted to the shaft bearings and to the surrounding structure. A model scale experimental approach was used, with a relatively simple cylindrical shell structure housing a simplified propulsion system. A comparison of the experimental results to an FEA analysis of the same model, using ANSYS, is also included. While the paper does not provide full detail of the experimental procedure, the longitudinal and lateral vibration responses measured at various locations on the structure (including at shaft bearings and on the shell) were presented, and generally good agreement between experimental and analytical results was demonstrated. The principal conclusions of this study noted the significance of the effects of structure/shafting coupling, and identified the rear bearing as being the main culprit for the transfer of vibrations from the propeller-excited shafting to the shell structure.

2.2.2 Machinery and Shafting Excitation

Lin *et al.* (2009) provided a study on the vibration control of ship structures, with particular focus on the requirement to effectively control machinery-induced noise and vibration propagation at low frequency in faster and lighter ships. The example for this study was a $30\,m$ aluminium crew vessel with transverse ring frames at $1\,m$ spacing. The FEA analysis was conducted using MSC/NASTRAN. To reduce calculation time, only the engine room and the keel of the ship were included to calculate the frequency response. Furthermore, because the wavelength of the structure-borne sound in the low and medium frequency ranges (of interest) was much greater than the physical dimensions of the machine isolators (e.g. engine mounts), these excitations were idealised as point sources. Under these assumptions, the input mobility of the engine bed was calculated in the $0 - 250\,Hz$ frequency range for out-of-plane and in-plane excitation forces, and torsional and bending moment excitations with and without the hull and deck plates were included in the FEA model. These results were also compared to the results generated for the corresponding infinite and finite beam (where the length of the finite beam is taken to be that of the total length of the engine bed). Since there was no significant difference between the cases including or excluding the deck and hull plating from the frequency range of interest, it was concluded that the energy flow to the ship's structure could be estimated from the knowledge of the simple structural components alone (i.e., neglecting hull plating).

Using the established, frame-only FEA model, two approaches to passive vibration control through the vessel from the machinery excitations were examined. The first approach considered modifications to the engine bed sections in order to change the mobility at the source location. Five variations were considered and compared to the original design and, while not every case showed an improvement over the original design for the full frequency range of interest, the authors concluded that the results

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demonstrated that power injection (to the ship structure from machinery) could be reduced by proper designs. This is a useful finding for naval architects in combating noise and vibration problems on board ships.

The second approach considered the control of wave propagation through the ship structure by imposing irregularity to the ring frame spacing. In this study, the energy propagation to the forward end of the ship from the aft machinery was compared for the original frame-only hull configuration and for a new configuration in which two of the ring-frames just forward of the engine room were moved; one $0.1\,m$ aft, and the other $0.1\,m$ forward, to create a minor irregularity in the frame spacing. The results showed that the peak responses were reduced at the forward location of the ship due to this minor change. The authors concluded that the work was meaningful for the vibration control of ship structures at low frequency, where active and traditional passive control methods have had little success.

Magazinović (2010) reported on a novel regression-based method to assess the key torsional vibration responses for marine engines being installed on ships. While the method only provided statistical approximations and, therefore, should not be taken as a substitute for ordinary torsional vibration analysis, the method provided a novel procedure for fast and effective assessment of key torsional vibration responses for designers and shipyards, so that significant redesign of shafting and structure during later stages of the design or build can be avoided. It also increased computational efficiency for investigations of shafting design scenarios.

The method used a response surface methodology, representing the design space using regression fitting, applied to a limited set of fully calculated system responses over a limited set of the design space. In this case, a least-squares fitting of the data to quadratic polynomials, in 15 unknown coefficients, was used to represent the results. The input variables used were limited to shafting stiffness, propeller mass moment of inertia, and turning and tuning wheel mass moment of inertia. The results considered were for natural frequencies and mode shapes, and forced vibrations.

Although the metamodel required significant time and effort to build, the method was shown to be sufficiently accurate as a preliminary design tool. Once the metamodel was developed for a particular engine, it could be reused repeatedly for various shafting designs.

Jia (2011b) provided a limited study into using genetic algorithms (GA) to calculate the natural flexural frequencies of a ship's tail shaft. The study concluded that, for shafting systems with complex bearing supports and shafting elements, the GA solution method provided a computationally efficient means to isolate the roots (natural frequencies) of the coupled equations of motion for the discretized shaft elements and supporting bearings.

Godaliyadde *et al.* (2010) described, in detail, a novel approach to making a subjective risk assessment of the likelihood of the levels of ship hull vibration (SHV) resulting from the combined machinery, shafting, and propeller arrangement in the ship design. The motivation for investigating the method stemmed from the fact that the complete machinery and propulsion arrangement on board ships is a complex multi-component system, often with high levels of uncertainty in terms of the quantitative data for the contribution of each component to the eventual level of the SHV. Therefore, the approach adopted used a fuzzy rule base approach, combined with the uncertainty treatment method, *evidential reasoning (ER)*, which allowed both quantitative and qualitative criteria, under uncertainty, to be dealt with systematically and consistently.

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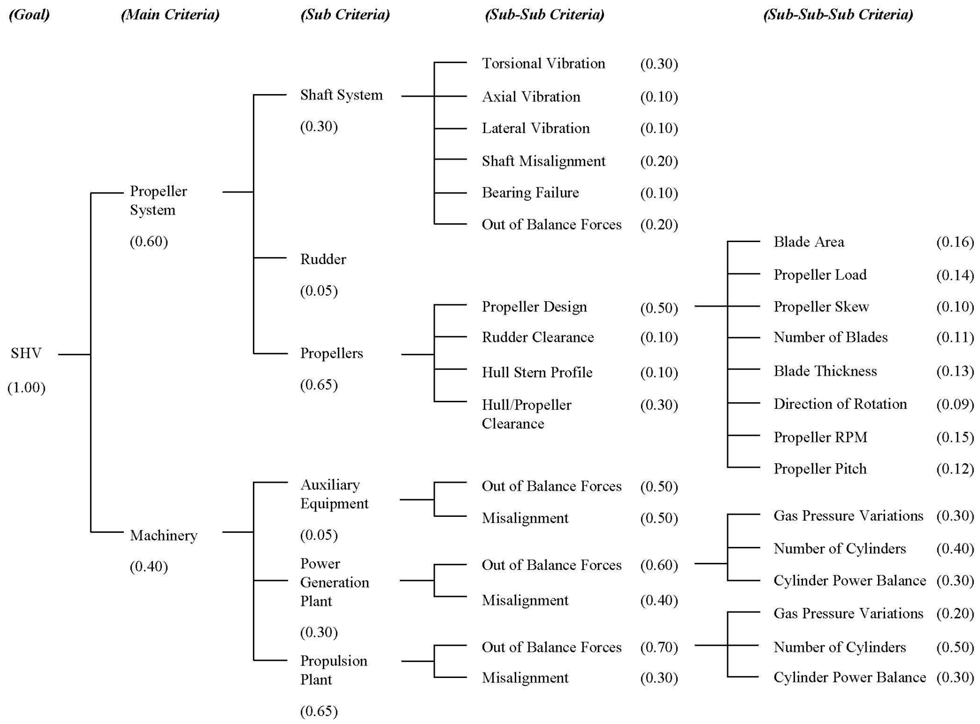

Figure 1: Hierarchical SHV model (Godaliyadde *et al.*, 2010)

Taking a bottom-up approach, in which the SHV model is broken down into subsystems contributing to SHV (i.e., machinery and propeller systems) and then further down into increasingly more detailed components levels, a five-level hierarchical model was used (see Figure 1). At each level, all subcriteria contributing to the component at the level above were assigned a weighting factor, reflecting their relative importance to that system in terms of contributing to SHV. In this study, the weighting factors were assigned using expert knowledge that was based on the opinions of two expert academics and two industry experts.

The actual characteristics of the components and subcomponents were then described in linguistic terms called assessment grades; e.g., excellent, reasonable, marginal, etc. Then, working sequentially from the bottom of the hierarchy of dependencies, a mapping process that accounted for the weighting factor of each subcomponent and the expertly-judged assessment grade for the subcomponents provided the assessment grade to the component above, until the risk estimation for SHV was determined, with the uncertainty for each outcome expressed in terms of a percentage of Belief Degree.

In this study, the method was validated using a case study where the method was applied to a real ship, for which the component assessment grades were, therefore, known, and the model results suggested that there was a "High" risk estimation of SHV with 23.38 % Belief Degree, and a "Very High" risk estimation of SHV with 65.79 % Belief Degree, and the ship itself was reported, after measurement, to have high-level SHV. The authors concluded that the method was suitable for both ship design and selection for reduced SHV.

Cho *et al.* (2010) performed a study to examine the magnitude and direction of vi-

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bration energy flow due to the various excitation sources of the propulsion system, including the unbalance force and moment acting on the crankshaft of the main engine, the longitudinal thrust variation force acting on the thrust block, and the vertical propeller forces. The study was carried out for a 4,100 *TEU* container ship and used a visualisation of a structural intensity analysis to elucidate the most dominant transmission paths of vibration energy from the propulsion system elements. For the particular ship analysed, the longitudinal engine room bulkhead was identified as the dominant transmission path, but more generally, the authors noted the usefulness of the visualisation of structural intensity for assessing (and countering) vibrations due to propulsion system components.

2.2.3 Consequences of Propeller- or Machinery-Induced Vibration

Kirkayak *et al.* (2011) report on structural dynamics model tests of the vibrational characteristics of a two-tier stack of $20\,foot$ ($6.058\,m$) ISO containers. The model experiments were performed at 1:4 scale ratio on a shaking table at the NYK-MTI experimental facility, Yokohama, Japan. Realistic mass, stiffness, and corner fitting gaps were modelled. he gaps at the corner fittings were the major source of system nonlinearity. Experiments with vertical base motion were conducted at 5, 10, 15, and $20\,Hz$ model-scale. Assuming Froude scaling, these correspond to 2.5, 5, 10, and $20\,Hz$ full-scale, which spans from frequencies associated with whipping and springing, to those associated with propeller or some machinery-induced vibrations. Good agreement was found between the experimental results and finite-element predictions. The results showed strong dependence on the corner fitting gaps and on the container payload.

2.3 Noise

Noise control is traditionally a concern for crew and passenger habitability. Recently, concern has also arisen regarding airborne noise control for the residential areas near ports and along the coasts due to heavy ship traffic. Also, reducing adverse impacts to marine wildlife due to noise radiated from vessels has become an issue. Recent research for ship noise is respectively reviewed for interior noise, and air- and underwater-radiated noise, in the next three subsections.

2.3.1 Interior Noise

Modern analysis methods to predict ship interior noise have been introduced to consider the multiplicity of sound generation and propagation mechanisms in ships, which are mainly frequency-dependent vibroacoustic phenomena. In the previous ISSC 2009 report, the advances in the analysis methods such as the transfer path analysis (TPA), the statistical energy analysis (SEA), the hybrid deterministic-statistical methods such as the combined SEA and finite element analysis (FEA), and the power flow finite element analysis (PFFEA) for mid- and high-frequency ranges were reviewed. A variety of publications since the last ISSC report can be found on the numerical methods for more accurate and efficient prediction of vibroacoustic behaviours except TPA, which are mainly aimed at extending the theoretical basis and/or reducing computational burden. Publications can be also found on the experimental and/or numerical evaluation of acoustic insulation components.

Analysis Methods and their Applications

The validity of SEA was among the main concerns in the early development of the theory due to the assumptions of a large population of modes, wide-band and uncorrelated excitations, large modal overlap, diffuse field, equipartition of energy, and

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conservative and light coupling. Many literatures already discussed each assumption for the extension of SEA. Bot and Cotoni (2010) presented a validity diagram of SEA using the four dimensionless parameters of mode count, modal overlap, the normalized attenuation factor, and the coupling strength. For a valid SEA model, the former two parameters must be high and the latter two parameters must be small.

Knowing the dominant energy transmission paths is useful to establish proper noise countermeasures. For this purpose, Guasch *et al.* (2009, 2011) presented a systematic algorithm to find the dominant energy transmission paths in a SEA model by using a modified graph theory to find the set of so-called K dominant paths. The authors demonstrated its usefulness for a SEA model of a two-floor building, with 12 identical rooms per floor.

In the case of adding details to the SEA model by the use of FEA for the analysis of mid-frequency (MF) range, one limiting factor was the calculation time for solving large FE models of industrial size. Another limitation was the loss of accuracy when only subcomponents were considered, which depends on the selection of the boundary condition at the interface of the subcomponent. Ragnarsson *et al.* (2010) proposed a wave-based boundary condition to reflect more accurate behaviour of a subcomponent model, instead of classical boundary conditions. The boundary condition for the MF analysis was derived by the wave-based substructure analysis and the modal reduction for the remaining structure for the low-frequency range via the modal analysis for a full FE model.

Cotoni *et al.* (2008) proposed a more general SEA subsystem formulation based on a combination of finite elements (FE) to represent a unit cell of section, component mode synthesis on the unit cell to reduce computation burden, and periodic boundary conditions to the unit cell. The method enables the SEA parameters to be efficiently computed for very general structural panels compared with the directly combined SEA and FEA. Langley and Cordioli (2009) extended the hybrid deterministic-statistical method using SEA and FEA to apply to coupling over the domain of a SEA subsystem, termed "area junction." This enabled, for example, a statistical structural component to be coupled to a FE acoustic volume.

Vergote *et al.* (2011) proposed a framework for coupling deterministic wave based method (WBM) with SEA models by an extension of the hybrid FEM-SEA framework, which might be computationally more efficient than the hybrid FEM-SEA framework.

For the precise prediction of vibroacoustic behaviour of plate structures in high frequency range by PFFEA, Park and Hong (2008) presented power flow models to analyze transversely vibrating Mindlin plate, whose model was derived by the energy governing equations for far-field propagating out-of-plane waves in the plate.

Kim *et al.* (2011c) presented a new energy flow governing equation considering the near field acoustic energy term and spherical wave characteristics. A numerical analysis scheme was applied using the indirect BEM for application to the analysis of the acoustic energy density and intensity of complex structures in medium-to-high frequency ranges.

A numerical example to predict the mid-frequency (MF) acoustic power radiated in water of a warship using the technique of virtual SEA instead of experimental SEA was presented by Borello (2010). In the virtual SEA approach, the dynamics of the structure was reduced to a statistical frequency response function (FRF) matrix between sets of observation nodes of the in-vacuum FE model. An algorithm reordering

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FE nodes from the FRF matrix was applied for the partitioning into subsystems with weak coupling. Then the FRF matrix computed between all nodes was compressed and the SEA parameters (modal density, mass, and coupling loss factor) were identified from the compressed FRF, which encapsulated the FEM dynamics in the MF range into the parametric SEA model. The computational expense of virtual SEA modelling was mitigated by the automation of low-level tasks.

Performance of Acoustic Insulation Components

Because of their high stiffness-to-weight ratio, lack of corrosion, high thermal insulation, and ease of assembly, lightweight partitions have been increasingly used instead of metal structures for the construction of ship cabins. Zhou and Croker (2010) presented the experimental and predicted sound transmission loss values obtained from SEA for two foam-filled honeycomb sandwich panels. Experimental modal densities, total loss factors, radiation, and internal loss factors were also presented.

Cha and Chun (2008) presented a mass-spring-plate model to predict the insertion losses of floating floor, of which upper plate and mineral wool were assumed as a one-dimensional mass-spring system lying on the simply supported elastic floor. The predicted insertion losses for seven different floating floors agreed well with that measured from the tapping machine test at a mock-up for simulating cabins of cruise ships. For a floating floor with a viscoelastic layer to reduce structure-borne noise in ship cabin, Song *et al.* (2009) proposed an analytical model that treated a viscoelastic layer and its lower and upper steel plate as a three-layered sandwich plate, and the mineral wool layer on steel deck as a single panel. The combined effects of all the layers were evaluated by the boundary conditions at the interfaces of the mineral wool and the assumed mode method. The proposed model well simulated experimental results, and structure-borne noise was reduced by $10 - 15\,dB$ at medium to high frequencies by using a constrained viscoelastic layer.

2.3.2 Air Radiated Noise

Recently, air radiated noise from ships has become a rising concern due to complaints from inhabitants of populated areas near ports, channels, and coasts. The prediction of airborne noise pollution produced by moving or stationary ships requires the information on source characteristics, sound attenuation during propagation, and an assessment method at receiving positions for time-varying and time-rated noise. Such kinds of predictions have been widely applied for the assessment of road, railway, aircraft, and industrial noise sources.

Badino *et al.* (2011) identified the main sources of airborne noise from ships as funnels, air intakes, air dischargers, and openings that put the internal sources (ventilation systems, engines) in communication with the external environment. Sound attenuation during propagation outdoors depends on propagation distance, reflecting and intervening objects, and weather conditions. Various calculation methods, such as ISO 9613-2 (1996), and numerical methods based on geometrical acoustics, have been published. Various commercial software packages have also been used for noise simulations on open deck spaces of ships.

Kumar and Nikam (2008) investigated the influence of the air-intake rain protection louver on the noise generated by the engine room air-intake of offshore supply vessels. From numerical simulation on the flow field using a finite volume unstructured computational fluid dynamics code and on-board measurement and experiment, the noise level could be significantly reduced by making the angular orientation of the louver slats in-line with the flow and increasing the spacing between the slats.

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Biot and Moro (2011) presented a prediction procedure of ship's outdoor airborne noise in harbour and numerical simulation results in nominal octave-band centre frequencies from 63 *Hz* to 8 *kHz* for a cruise ship. In the simulation, a commercial code based on a hybrid beam tracing method of geometrical acoustics was used. They considered that the outdoor noise sources of ships during harbour stay were the funnel top due to diesel generator engine exhaust, outer openings of the inlets/outlets of the HVAC (Heating, Ventilation and Air-Conditioning) system, including intakes for air supply to diesel engines and auxiliaries and outlets of extractor fans. The other noise sources, such as cargo handling, ancillary onboard devices, and activities, were also discussed, but disregarded in the simulation because of their low contribution to overall noise.

2.3.3 Underwater Radiated Noise

Underwater radiated noise (URN) is low frequency noise (under 100 *Hz*). URN travels long distance under the water surface and comes mainly from mechanical vibration (ship hulls, engine and shafting, wind/tidal turbine farm piles, etc.) and marine propeller/thrusters, either with or without the presence of cavitation. For non-cavitating cases, non-uniform wake at the propeller is the main cause of force and pressure fluctuations that lead to vibration and noise. Non-uniform wake also contributes to unsteady cavitation and all forms of cavitation can, to varying degrees, create substantial noise.

Section 2.3.3 briefly presents the recent research work and studies since ISSC2009 on URN. This section is divided in to 4 minor topics: 1) URN due to hydrodynamics of propeller in operation; 2) the URN created by mechanical vibration (such as engine, propeller shaft, air duct, etc.); 3) URN impact on marine life, and 4) recent experimental and numerical methods/models on measuring, signal processing, analyzing, and predicting URN.

URN due to Hydrodynamic Forces of Propellers, Non-Cavitation Case

Marine propulsion systems generating non-cavitation URN can also be divided into two different categories: a) mechanical noise due to fluctuation of axial forces along the propellers shaft via thrust bearings, and b) hydrodynamic noise due to the fluctuation of the fluid pressure field acting on the propeller blades and nearby hull surfaces.

The thrust of a propeller at a constant revolution speed and fixed geometry is a function of inflow velocity only. In general, the 3D detailed wake in the propeller plane (usually expressed in cylindrical coordinates by radial, axial and tangential components) is non-uniform. The nominal wake is the wake in the absence of the propeller and the effective wake includes the velocities induced by the working propeller. Depending on the propeller design, large fluctuations in thrust may occur as the blades pass through a highly non-uniform wake, resulting in significant blade frequency fluctuations in the axial bearing force. The fluctuation of the axial force along a usually long propeller shaft causes axial vibration and, in this case, the long shaft acts as a compression spring along the shaft centre, though transversal vibration may also occur.

Merz *et al.* (2009) described the situation in detail for a submarine example. The URN often is a combination of tonal and broadband random noise sources, and the prevalence of the sound sources is dependent on frequency band and speed. Here, both cavitating and non-cavitating cases were considered and, in the cavitating case, the frequency was dependent on depth as well. At a large distance from the hull, as absorption increases with the increase of frequency, lower frequency sounds are dominant. Propeller shaft speed and number of blades are the main contributors,

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and determine the frequency of the tonal components. Tonal components are also slightly dependant on the blade thrust fluctuation and thickness. Tonal components, along with the broadband noise, provide important information for the detection of the identity of the vehicle and its speed.

Depending on natural frequencies, the harmonic thrust fluctuations of a non-cavitating propeller can lead to excitation of local structural vibrations and to global ship hull vibrations.

The pressure fluctuations on the propeller blade surface create pressure fluctuations on the objects nearby (the hull, for example). This is analogous to a fluctuated force with both strength and direction, pointing at the ship hull, within a few diameters of the propeller. These kinds of fluctuating forces can be simulated and predicted by using a dipole (doublet), with strength fluctuating harmonically at blade passage frequency.

Caresta *et al.* (2010) studied how to mitigate the sound radiated by a submarine hull in bending vibration under harmonic excitation from a propeller. In the study, minimization of the sound radiation was performed using a cost function based on Active Vibration Control (AVC) and Active Structural Acoustic Control (ASAC). An array of actuators was placed circumferentially on an end plate mounted on the prow of the submarine hull. The force magnitude of the inertial actuators was spatially modulated at the fundamental frequency to actively control the excitation of the bending vibration. AVC was used to apply one or more secondary forces to reduce the structural vibration excited by the primary force. Reduction of vibration using AVC can reduce the noise level. ASAC was used to attenuate the sound radiation in one or more directions. Results were obtained and showed that, at a blade passing frequency (BPF) of $27\,Hz$, the sound pressure level was reduced substantially, from $45\,dB$, re $1\,\mu Pa$ with no control applied to $30\,dB$, re $1\,\mu Pa$ with both ASAC and AVC controls.

A numerical method was presented recently to predict non-cavitating propeller noise using a hydro-acoustic approach incorporating the Detached Eddy Simulation (DES) and the Ffowcs Williams-Hawkings (FW-H) equation (Pan and Zhang, 2010). This study paid special attention to broadband noise, instead of tonal noise. The broadband noise or the self-noise were deemed to be created by the blade surface pressure fluctuations due to the shed blade trailing edge and tip vortices and flow separation under blade heavy load condition. Predictions of the sound directivity were obtained and presented though the acoustic results were not validated.

URN due to Hydrodynamic Forces of Propellers, Cavitation Case

All forms of cavitation generate URN. Among the most common forms of cavitation (tip), vortex cavitation generates high frequency URN and sheet cavitation generates low frequency URN, usually of larger magnitude. Park *et al.* (2009) report on an experimental study, performed in a relatively small cavitation tunnel, to identify the locations of the noise sources generated by a cavitating propeller. A new signal processing technique, Matched Field Processing (MFP) was applied to the identification process. The MFP involved convolution of source signals. A virtual source was devised and included in the formulation of the method to detect the noise source of a cavitating propeller. The frequency of the virtual source was modulated from 3 to $15\,kHz$. The cavitation number for a $0.2\,m$ diameter model propeller was varied between 2.2 and 12.6. The study showed that the sheet cavitation volume was the main contribution to the propeller generated low frequency noise. It was recommended that

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the research be repeated using the same method in a large cavitation tunnel capable of accommodating a ship model with propeller.

Kehr and Kao (2011) presented an efficient numerical time-domain method for predicting the pressure on the hull, and both the near and far underwater acoustic fields, due to unsteady propeller sheet cavitation in a non-uniform wake field. Dipoles and monopole acoustic sources were distributed on the rotating propeller surface and the wave equation was solved in the presence of a rigid ship hull and reflecting free surface. In this work, the unsteady cavitation volumes were predicted using the PUF-3A unsteady lifting surface theory developed at MIT. Predictions for a container ship were compared to predictions obtained by the Laplace equation and with experimental results from the HYKAT large cavitation tunnel at HSVA.

2.4 Shock and Explosion

Explosions in air or in water produced by accidents or intentionally by hostile attacks within or immediately nearby a ship's hull can cause catastrophic damage on the ship structure and the shutdown of critical life safety systems. Explosions in air may more frequently affect local structures or sub-systems directly blasted by the shock pressure wave, while the whole ship structure can be more seriously damaged by underwater explosions: the shock wave can only locally damage the hull, but the gas bubble can induce resonance phenomena in the hull girder, up to its collapse. It becomes of significant importance to simulate the structural response to such loads in order to provide a shock- and explosion-resistant design of the ship structures.

2.4.1 Local Response

Some studies have been identified that are aimed at evaluating the local dynamic response of ship structures subjected to blast loads, from a simple stiffened steel panel, to composite plates, up to an entire mast.

Yang *et al.* (2010) investigated the dynamic response of one-way stiffened plates with clamped edges subjected to uniformly-distributed blast-induced shock loading. A singly symmetric beam model was used, based on the rigid/perfectly plastic assumption and taking into account the bending moment/axial force capacity interaction. Their relation for singly symmetric cross-section was derived and explicitly presented. The study led to assessment of the deflection condition that a plastic string response must satisfy; e.g., the linearized interaction curve and associated plastic flow rule. The dynamic response of the plate under blast loads has been calculated by means of two FE solvers (ANSYS and Abaqus/Explicit) and by the developed theoretic method to assess its functionality and suitability. The possibility of replacing an arbitrary blast load by a rectangular type pulse was assessed.

Chirica *et al.* (2011) analysed the protective capacity of ship hull structures made of composite materials (flat plates layered by two different orthotropic E-glass reinforced polyester plastic) subjected to a spherical charge explosion. The methodology, which was to apply the blast pressures caused by a close-to-surface explosion while accounting for space and time variation, was based on previous literature. By performing parametric direct-integration nonlinear dynamic analyses and assessing the structural failure of the composite layers by the Tsai-Wu criterion, the behaviour of those plates to blast loading was assessed versus explosive magnitude, distance from source of explosion, and plate thickness. Despite some criticalities in accounting for the effect of damping, the study allowed some guidelines to be drafted for the design of composite blast-resistance panels.

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Xiaobin *et al.* (2010) developed a method for the evaluation of the dynamic response of a structure subjected to air explosion and a time and space varying shock loading. The method was based on a numerical Flux-Corrected-Transport (FCT) algorithm, which was able to evaluate the shock wave propagation time history and to calculate the load in a format suitable for FE codes. The air-structure coupling was taken into consideration. The method was applied to a naval mast, which always needs an accurate design as it is essential for ship safety and combat capability. The structure of the mast was modelled by Lagrangian plate elements in the FE code LSDYNA, and loaded by the computed pressure time history on each element. The dynamic behaviour of the mast was calculated under the design blast load to verify the capabilities of the method and to provide a reference for an explosion-resistant mast design.

2.4.2 Global Response

A few publications related to the global response of ship structures to shock/explosion loads have been identified. Underwater explosion (UNDEX) analysis of ships is one of the most complicated numerical analyses, both for load calculation and for fluid-structure interacting response.

Sinha and Sarangdhar (2008) focused on a fast method to simulate and analyze UNDEX effects on ships at a design level. The FE Software ANSYS was coupled with IRUNDEX, software for the Underwater Explosion analysis of structures. The software calculated the loads by taking into account the initial shock wave, the pulsating gas bubble, and their spherical distribution; the pressure time histories calculated at any ship location were automatically applied to the FE model. By replicating already known test cases, the method capabilities were assessed versus mesh density, load discretization, time step, and nonlinearities due to the geometry and the material. Finally, the UNDEX response of a floating shock platform was calculated and analysed considering different charge weights.

Xing *et al.* (2008) presented an improvement of their Fluid – Structure Interaction Analysis Program (FSIAP), a mixed finite element method for the dynamic analysis of fluid structure interaction systems subject to earthquake excitations, explosion waves, and impact loads. The variables of acceleration in the elastic solid and pressure in the fluid were adopted as the arguments. The boundary conditions on the interface of two different fluids with different mass density were implemented, the frequency shift technique for fluid-structure interaction system was mathematically demonstrated, and a substructure-subdomain technique to solve the large vibration problem was adopted. The case of a LNG tank excited by a pressure wave was presented, which showed that a number of internal and external sloshing frequencies existed due to large area of free surface on the external water. The authors highlighted that sloshing effects induced by UNDEX pressure waves have to be accounted for in designing large LNG ships.

2.5 Damping and Countermeasures

Damping properties can be tailored in a wide range by modifying the constituent materials, lamina thickness, ply orientation, stacking sequence, or fibre fraction in multilayered structures, and also by changing skin thickness or core depth in sandwich constituents. Matter *et al.* (2011) showed that, among simple damping formulations, the hysteretic or structural model was much more pragmatic than the viscous one for the steady state vibration or modal analysis of fibre reinforced composites under normal working condition. A mixed numerical-experimental identification procedure for evaluating the elastic and damping properties of sandwich laminates with a soft core was described. It was concluded that, although the constitutive model chosen

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was based on a frequency independent hysteretic damping, the proposed identification method was generic and could be extended to alternative constituent models. Increased complexity required an improved effort on the numerical, experimental, and identification levels.

Lee *et al.* (2009) conducted experiments to investigate the structural response of a Mark III type insulation system against sloshing impact actions. A series of compressive load tests were carried out in low strain range. The temperature effect was considered. It was found that the lower the temperature, the smaller was the displacement under the same pressure. This meant that the damping effect of an insulation system, including RPUF (Reinforced Polyurethane Foam), became smaller as the temperature decreased. The damping effect was analyzed using parameters like strain rate, loading rate, and spring constant. It was found that spring constant tends to converge as the loading rate was increased and temperature decreased.

Den Besten *et al.* (2009) developed an analytical 2D mathematical model for the local structural response of a hydrodynamic impact-loaded sandwich structure with vibration isolation and structural damping properties. The structural response was determined by solving a semi-analytically hydroelastic coupled sandwich flexible core model and hydrodynamic impact model. It was found that, compared to stiffened panel hull structures, the bending stresses were low. Also, less fatigue damage sensitivity was observed because of the sandwich construction principle (absence of a stiffener-girder connection). Structural damping properties of the foam material were of reduced importance for fatigue damage sensitivity.

Passive vibration reduction methods, like elastomeric dampers or tuned vibration absorbers, were limited in their overall reduction performance. To further decrease the vibrations, Kauba *et al.* (2008) designed a complete control system for the active control of a marine engine mount. The control system was implemented and tested on a rapid control prototyping system as a Multiple Input Multiple Output (MIMO) Filtered Reference Least Mean Squares (FxLMS) algorithm. Test runs of the experimental rig with varying engine speed were conducted. After the laboratory testing, the active vibration control system was mounted to the vessel and final measurements during operation of the ship were performed.

2.6 Monitoring

Kivimaa and Rantanen (2007) presented results obtained by monitoring the dynamic hull responses for two extreme load cases. The motivating concern was that, in the development of high-speed craft, there was a lack of information on full-scale wave loads and hull beam responses in actual operational conditions. To address this deficiency, an integrated monitoring system was developed and applied to a fast monohull, SuperSeaCat4 ($LOA = 100.3\,m$, $V_s = 37\,kn$). Details of the monitoring system were provided, and a selection of the measurements from a monitoring campaign was presented. This monitoring system used stain gauges, accelerometers, a differential global positioning system (DGPS), and a wave radar, so that the dynamic global wave-excited dynamic responses of vertical bending moment, torsional moment, and shear force could be observed at four locations along the hull. The authors emphasized the importance of synchronising data streams in order to fully understand the hull behaviour in conjunction with prevailing conditions.

Grasso *et al.* (2011) also considered the development, installation, and load time-histories from hull monitoring systems. They advocated the use of a temperature-compensated, laser based optical sensors in preference to electrical stain gauges. The

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design and installation of a load monitoring system for two distinctly different vessels were considered. For a bulk carrier, the hull girder bending moments and local stresses in selected 'hot spots' were of interest, whereas for a tug boat conducting ice-breaking duties, local stress levels of the structure in the bow region were targeted. The report provided detail on the use of FEA in identifying the optimum location of the sensors for each case. The report also described the algorithms for the supporting onboard software of the systems. Because the monitoring systems were only recently installed at the time of publication, a brief strain time history from the tug boat was provided as example data.

Mathisen *et al.* (2009) principally focused on the contribution of whipping to the total hull bending stress. They analysed the time series of deck stresses in a containership (4,400 TEU, $LBP = 281\,m$) in severe stationary conditions. Six segments of data were selected with relatively severe hull stresses (in sea states, $H_s = 5.5\,m - 6.6\,m$), for which the conditions could reasonably be considered to be stationary (i.e., relevant parameters such as sea state, variability in stress levels, etc., were effectively time invariant). They observed that, in a number of cases, the extreme hogging stresses were above the design rule stress. In addition, using low-pass filtering to separate the high frequency (vibratory) stress components due to 2- and higher modes from the wave-frequency stresses, they showed that the vibratory (whipping) contributions made a significant contribution to this total stress ($\sim 35\,\%$). The authors recommended that further analysis of this type be carried out as data becomes available for container ships.

Davis *et al.* (2009) were principally interested in the slamming behaviour and subsequent whipping response of fast wave-piercing catamarans. From a monitoring perspective, while full detail of systems were not provided, the authors stated that they comprised a set of three strain gauges set to record axial stain along the length of main keel members for each demihull of the various INCAT wave-piercing catamarans used. Furthermore, they demonstrated the use of wavelet analysis of strain time histories to identify even small slamming events that were not easily identified from the strain time-history alone. From sea trials data, the subsequent transfer of the slam event to the whipping vibration mode and its subsequent decay was identified. In addition, tests on a segmented model were able to replicate the frequency and damping observed at full-scale reasonably well, and gave a clear indication of the overall mechanics of the slam process.

2.7 Uncertainties

Dessi *et al.* (2009) investigated the correlation of model-scale and full-scale tests aimed to determine the bending response of a navy vessel in waves. A preliminary analysis on the correlation of model-scale and full-scale experimental data about the global ship behaviour was carried out. Moreover, a theoretical procedure, based on the successive comparison of a chain of intermediate numerical models filling the gap between the real ship and the segmented hull, was outlined and applied in the case of regular wave excitation. These are the first steps toward the definition of a general procedure capable of tuning the model-scale data that, being affected by the load segmentation and by the reduction of the structural complexity, needed some kind of correction function depending on the space coordinates and the frequency.

Section 4 of this report also provides insight into uncertainties regarding slamming and whipping.

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2.8 Standards and Acceptance Criteria

This section focuses on noise, vibration, and shock acceptance criteria and procedures for their measurement. International standards with regard to habitability, underwater noise radiation, and shock test for ships are discussed.

2.8.1 Habitability

The ILO MLC (2006) sets minimum standards, applicable to new buildings as well as ships in operation, to address the health, safety, and welfare of seafarers. The regulations are subject to the implementation into national laws, which are expected in 2013 after the expected ratification in 2012. The convention specifies requirements with respect to preventing the risk of exposure to hazardous levels of noise and vibration. The limits for noise levels defined in IMO Resolution A.468 (XII) 1982 are commonly understood as satisfactory for compliance with the noise aspects of the convention. Because MLC 2006 does not define limit values for vibration exposure, this should be addressed by the national legislations. Most of the flag states that ratified the convention so far did not concretise the convention in this respect. To ensure clearness and avoid interpretations of the convention's compliance, the Committee encourages concretisation of acceptance criteria.

Guidance for complying with the convention's requirements is provided in the ABS guideline (2009) that gives assessment criteria and measurement methodology for obtaining a voluntary class notation. GL's Harmony Class for Cargo Ships Certificate (2009) also includes an ILO-MLC Noise Compliance certificate. In general, the new voluntary class notation introduces three harmony categories to allow for a graduation of noise and vibration levels, with different requirements valid for ships with different deckhouse positions.

In 2007, EU started an initiative to update IMO A.468(XII), with the aim to incorporate mandatory noise level limits for work and living spaces via amendments to SOLAS Regulations II-1/36. Under discussion are noise exposure levels, maximum noise levels, and limits for airborne sound insulations.

2.8.2 Underwater Noise

In recent years, a growing concern on the effects of ship generated underwater noise on the marine fauna can be observed. The Committee (ISSC 2009) encouraged standardization for the measurement of underwater noise. Two standards were published in the meantime, and one is under development.

ANSI S12.64 (ANSI 2009) described requirements for the measurement of underwater noise. The underwater sound pressure level measurements shall be performed in the far field and then corrected to a reference distance of 1 meter. The standard offers three grades of measurement, each with a stated applicability, test methodology, uncertainty, system repeatability, and complexity. The standard does not specify or provide guidance on underwater noise criteria.

With *DNV Silent*, Det Norske Veritas published in 2010 a voluntary class notation for vessel-related underwater noise radiation to ensure a low environmental impact and/or to ensure hydro-acoustic operational capability for vessels relying on hydro-acoustic equipment. The notation has five sub-notations: Acoustic (A), requirements for vessels using hydroacoustic equipment as important tools in their operation, e.g., survey vessels, ocean research vessels, pipe layers, diving vessels, naval vessels, etc.; Seismic (S), requirements for vessels carrying out seismic surveys using acoustic streamers; Fishery (F) requirements for vessels engaged in fishing; Research (R) requirements for

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research and particularly noise critical operations based on the recommendations in ICES Cooperative Research Report No. 209; and Environmental (E) requirements for any vessel demonstrating a controlled environmental noise emission. The rules specify requirements for maximum underwater noise emission for a given set of operating conditions.

ISO is currently developing a new standard, ISO 16554, *Measurement and reporting of underwater sound radiated from merchant ships*, following a request from the Marine Environment Protection Committee (MEPC) of the IMO. MEPC established a Correspondence Group to identify and address ways to minimize the introduction of incidental noise into the marine environment from commercial shipping and, in particular, develop voluntary technical guidelines for ship quieting technologies, as well as navigation and operational practices (Dambra 2010). ISO plans to publish the standard in 2012.

2.8.3 Shock

The American National Standard Institute (ANSI) published the standard ANSI S62-2009, *Shock Test Requirements for Equipment in a Rugged Shock Environment*. It defines ten thresholds of shock severity for equipment whose normal use subjects it to a certain shock. The shock severity thresholds shall be defined by drop height and half sine shock pulse or, alternatively, by velocity change and pseudo velocity shock response spectra (Lang 2010). The standard also defines criteria for compliance in case of other tests than drop shock tests producing other pulses than half sine pulses.

3 OFFSHORE STRUCTURES

3.1 Slender Structures

This section covers wave-induced loading on slender structures (risers, umbilicals, pipelines, mooring lines, and etc.) and current-induced vibrations, known as VIV. The strengths and weaknesses of the current state of the understanding of these complex fluid-structure interactions are discussed and studied by a number of institutions, and some of the many contributing authors will be mentioned in the coming subsections.

3.1.1 Wave-Induced Vibrations, Slender Structures

For preliminary design of risers and mooring lines, dynamic analysis of wave and floater-induced response is frequently based on application of regular waves with given amplitude and period. For more comprehensive concept verification, a stochastic model of the ocean surface and wave kinematics is typically applied. The corresponding dynamic response will hence also be of a stochastic nature, which implies that suitable probability distributions of local maxima and extreme values need to be identified. As the response processes in general are of a non-Gaussian nature, this may frequently become a challenging task. Such response analyses, in general, need to be repeated for multiple sea states. This implies that considerable CPU efforts are required, unless some kind of selection of important sea states is performed.

Martins *et al.* (2009) considered the fatigue and ultimate limit states of steel risers. The first part showed a comparative study of three approaches for statistical analysis of extremes. These approaches were respectively based on design storm analysis, environmental contour representation, and full long-term statistics. The contributions to fatigue damage and long-term extreme statistics from different sea states were also considered. Several full time-domain analyses were performed for two example steel catenary riser configurations located in Brazilian waters. The most important sea state blocks for each limit state were identified.

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Steinkjer *et al.* (2010) addressed the topic of how to select proper simulation lengths and blocking of the scatter diagram, which is essential to obtain reliable fatigue life estimates of non-linear riser systems. The paper presented methods for evaluating the statistical uncertainty of the fatigue damage estimate. These methods were demonstrated to be capable of describing the statistical uncertainties in short-term, as well as long-term time-domain fatigue damage estimates. Based on the obtained results, recommendations were given on simulation length in order to obtain the target reliability on the fatigue damage estimate.

Baarholm and Haver (2010) outlined a method to determine the long-term extremes by considering only a few short term sea states with application to a flexible riser configuration of the lazy-wave type, which was located in the North Sea. The sea states had a certain probability of occurrence, and were identified by a contour line in the (H_s, T_p) plane. The purpose of the contour line approach was to predict load and response levels, which corresponded to a given annual exceedance probability, without having to carry out a full long-term analysis. The advantage with this concept was that the environmental representation and response analysis was decoupled. This was very convenient if the problem under consideration were of a very non-linear nature; in particular, if characteristic values for design were to be established directly from model tests. The method seemed to give results of reasonable accuracy for most problems.

In Chen *et al.* (2009a), a methodology claiming to have a considerable potential for saving of computation time was considered. The focus was on Steel Catenary Risers (SCR), with the main response of concern located in the sagbend region near the touchdown. The methodology built on time traces of the host vessel motions, and the correlation between the vessel/porch motion and the SCR sagbend response.

Cheng *et al.* (2010) applied a methodology based on *L-moments*, which was proposed in order to a) select the probability distribution applicable to SCR response and b) estimate the extreme response during severe environmental conditions. The methodology was validated by modelling different responses for SCRs that were suspended from a deep-draft semi-submersible located at different water depths.

In Ayers *et al.* (2009), a method for qualification of polyester rope was proposed by application of test sequences representing 20 Katrina hurricanes. The test aimed at measuring both strength reduction and maximum elongation over a series of 20 hurricanes. A superior rope design would minimize cyclic wear, and an adequate splice design would demonstrate no appreciable splice slippage. A typical 20-hurricane test could be run continuously and be completed in about 4–5 days, so testing would be economical.

Spanos and Nava (2009) proposed a novel coupled six-degree-of-freedom analytical model for a Spar system with top tensioned risers. The model accounted for the interactions among spar hull motions, the riser motion, and the moonpool. This model involved six coupled nonlinear differential equations comprising nonlinearity terms associated with stiffness and damping, as well as the inertia terms. The results pointed out the importance of considering the moonpool coupling effect in the spar heave dynamics. Furthermore, the statistical linearization approach results exhibited good agreement with the nonlinear time domain analysis results.

In Naess *et al.* (2009), two methods for the prediction of extreme TLP tether tension from finite time series records were considered. The study was motivated in part by the observation that a significant part of the tether tension during storm conditions was associated with the ringing response, which was a conspicuously narrow band

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response process due to the very low damping in heave. The time series of the tether tension was, therefore, characterized by strong correlation between the neighbouring response peaks of the ringing bursts.

Felisita *et al.* (2010) performed a comparison of analysis methods in connection with installation of umbilicals (with each method corresponding to application of a different model for the wave spectral density). The objective of the study was to select the most suitable analysis method taking into account different types of installation operations. The comparison between different spectral models and different installation methods was based on several limiting criteria, including top tension, compression, minimum bending radius, and the tension at the touchdown point. Both the JONSWAP and the Torsethaugen spectral densities were applied in order to model the wave environment of the North Sea.

The large Vortex-Induced Motion (VIM) due to current acting on a circular-shaped mono column platform induces low-frequency stress variations on the SCRs connected to it. These stresses, together with stress variations associated to wave effects, must be accounted for in the fatigue analysis of these risers. Sagrilo *et al.* (2009) described a methodology for computing the fatigue damage in SCRs due to wave-frequency and VIM load effects based on a combination damage formula provided by DNV (2005). The wave frequency and VIM fatigue damages were calculated separately (by a time-domain Rainflow approach) and the combined damage was evaluated by means of the DNV formula. The results obtained in this work showed that the large low-frequency vortex-induced motions on the mono column platform caused more fatigue damage on the top of the SCRs. In the remaining SCR length, the fatigue damage was caused mainly by wave-frequency effects.

3.1.2 Current-Induced Vibrations, Slender Structures

The existing VIV prediction schemes are based on a number of hypotheses, experimental facts, and data like strip theory, energy balance, correlation length and, most importantly, the use of lift force coefficient databases. Recent advances in observing the VIV motions on experimental risers with high confidence showed that some of these assumptions may not be valid. One important source of the discrepancies between theoretical estimates and experimental observations arose from the use of experimentally-obtained lift coefficient databases. In Mukundan *et al.* (2009), they described a method to improve the modelling capability of riser VIV by extracting empirical lift coefficient databases from field riser VIV measurements. Their results showed that this method significantly reduced the cross flow response error.

Franzini *et al.* (2009) presented measurements of VIV of inclined cylinders with both circular and elliptic cross sections that aimed to check the validity of the normal velocity correction.

Vandiver *et al.* (2009) examined experiments on a tensioned long flexible riser that showed travelling wave VIV was dominant at high mode numbers as opposed to standing waves.

A full scale VIV test was carried in the Gulf of Mexico using the drilling riser standalone monitoring system. The drilling riser was exposed to excessive eddy loop current. Beynet *el al.* (2008) described the test set-up and measured VIV response. This included observations of cross flow VIV, in-line VIV, and additional response at higher frequencies not currently predicted with industry VIV analysis tools.

Liu *et al.* (2009) presented the results of a study to compute the VIV behaviour of a free-standing hybrid riser (FSHR) using a time domain analysis method, ABRAVIV,

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which had been benchmarked by field measurements and model test data for top tensioned risers and SCRs. This was the first application of the method to a FSHR. Comparisons between numerical and experimental results were presented. In addition, results from frequency domain analysis were also presented.

The gap between predicted and measured VIV fatigue damage can be very large. In an effort to more effectively understand and manage the vortex-induced vibration fatigue integrity of its drilling risers, British Petroleum (BP) instrumented a number of mobile offshore drilling units (MODUs) and offshore production platforms worldwide. Tognarelli (2008) presented several aspects of the findings from those monitoring campaigns. The measured data were used to expose some of the physical details of full-scale riser response, whose omission from predictive design tools and methods may have contributed to the observed wide gap between predicted and measured fatigue damage. To characterize the size of the gap, the data were compared to calculations using the most commonly used industry VIV analysis software, MIT's SHEAR7. The work demonstrated the inherent level of analysis over-conservatism with respect to full-scale, unsuppressed drilling risers in the field when typical analysis parameters were utilized.

Tognarelli *et al.* (2009) described BP's benchmarking of SHEAR7 version 4.5 (hereinafter SHEAR7 v4.5), and introduced the concept of time-sharing between the modes, which made spatial overlap elimination between the modes unnecessary. The concept of time-sharing was based on observations from the Gulf Stream, DeepStar-funded, slender pipe experiments. Comparisons were made between predicted and measured VIV fatigue damage for several full-scale drilling risers to demonstrate the efficacy of a calibration for the latest version. In addition, comparisons were made between VIV fatigue damage predictions using SHEAR7 versions 4.4 and 4.5 for drilling risers, as well as for a typical deepwater SCR in typical design Gulf of Mexico loop currents. The version-to-version differences were illustrated and explained. Finally, results of sensitivity studies conducted with respect to the new parameters in SHEAR7 v4.5 were presented. A key finding was that, while the predictions on average were similar from version to version, the scatter in predictions that led to requirements for large safety factors was largely unimproved.

Further, Yang *et al.* (2008) compared the measured responses from ExxonMobil's 2003 VIV model tests to simulations of the test conditions using the newest version of SHEAR7 v4.5, applying the concept of time shearing. The study indicated that the new time-sharing model, in general, generated prediction results with reasonable bias and scatter for bare risers. The prediction errors for straked risers were still high, however, even when the most favourable parameters were selected for the analysis.

Soni *et al.* (2009) presented results from a novel type of experiment. Trajectories for cross sections in a flexible beam were found from classical VIV experiments and then used as forced motions for a cylinder that was sufficiently large to obtain reliable data for hydrodynamic forces. By proper processing of the measured forces, hydrodynamic coefficients for relevant frequency components were found. The main purpose of Soni and Larsen (2009) was to compare the results as hydrodynamic coefficients and the vortex patterns found from dedicated experiments for both harmonic and periodic trajectories, and thereby investigate the potential for using coefficients from harmonic tests as the basis for empirical models.

The motion induced by vortex shedding on slender flexible structures subjected to cross-flow was considered in Violette *et al.* (2010). This phenomenon of vortex-induced

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vibration was analysed by considering the linear stability of a coupled system that included the structure dynamics and the wake dynamics.

A new form of VIV suppression device, the AIMS Dual-fin Flow Splitter (ADFS), has been developed, tested, and benchmarked against bare-pipe, 5d and 15d pitch strakes, and conventional teardrop fairings in Schaudt *et al.* (2008). Both model tests and simulations (SHEAR7) were carried out. Relevant coefficients were compared, as well as the effect on fatigue damage using various VIV suppression devices.

Another measure to reduce VIV was the use of a slit. Results of this methodology are reported in Baek *et al.* (2009).

Aglen *et al.* (2009) investigated the measured VIV for a free spanning pipeline model. The pipeline was exposed to uniform current and is free to vibrate in both cross-flow (CF) and in-line (IL) direction. The purpose of this investigation was to understand the behaviour of the free spanning pipelines with respect to response amplitudes, frequency and modal composition, and to identify characteristic cross-section trajectories.

The vibration frequency and lock-in bandwidth of tensioned, flexible cylinders experiencing VIV were studied by Lee and Allen (2010). The tests revealed that the top tension and structural stiffness (both lateral and axial) can have a significant impact on vibration frequencies.

An inverse finite element method was presented in Mainçon *et al.* (2008) for the estimation of load and response of linear dynamic structures based on measurement data. It produced load and response estimates that exactly verified dynamic equilibrium while the loads are reasonably small and the response is in reasonable agreement with the measurements. iFEM was used to process measurement data from VIV experiments on a reduced scale riser model in shear current. The technique allowed for visualisation of the distribution, and a history of hydrodynamic forces and excitation and damping zones.

Cunha *et al.* (2009) presented an analytical solution, experiments, and parameter investigation for the vibration of a simply-supported beam due to vortex shedding. The in-line and cross-flow fluid forces were coupled to the beam equation as harmonic non homogeneous terms. Experimental results of 2 DOF VIV of a flexible small scale pipe in a uniform stream were presented for perpendicular and oblique (at 60 degrees of the translation direction) pipe.

Recent advances in CFD methods and the availability of powerful computational resources have made numerical simulation a viable option for VIV prediction. In Bhattacharjee *et al.* (2009), a numerical 2-D simulation was conducted for near seabed cross-flow VIV of subsea pipelines. An advanced meshing technique, capable of handling moving boundaries, was used to discretise the fluid domain. This approach, in conjunction with a structural model, was employed to simulate the complex case of VIV in a pipeline in close proximity to the seabed.

The fatigue damage on a top-tensioned riser induced by in-line vibration and cross-flow vibration was addressed in Tang *et al.* (2009) using a statistical methodology and also the amplitude ratio.

The field data from a long flexible model riser was used to study the fatigue crack growth due to the stresses generated by high mode VIV in Iranpour (2008).

A method to estimate the fatigue damage using a monitored marine riser based on the data from a limited number of sensors was studied in Mukundan *et al.* (2009).

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The analysis showed that the travelling wave characteristics of riser VIV response were captured by the reconstruction methodologies. The effect of higher harmonic components were found to influence the riser fatigue life, however, and may or may not have been captured by the method depending on the available number of sensors and the bandwidth of the observed VIV response.

Bhalla and Gong (2008) presented a method that has been developed to identify if vortex-induced vibration (VIV) occurs in well jumper systems. A methodology to identify whether the vertical legs or all of the legs of the jumper need to be straked was presented. In addition, the technique aimed to recognize when certain legs of a given jumper system may require suppression, thus leading to a jumper design whose safety was not compromised while in the production mode, as well as minimizing downtime and identifying potential savings from probable fatigue failures.

Current induced vibration in the form of VIV and galloping produce hydrokinetic energy. The focus of the work in Lee and Bernitsas (2011) was to convert this energy into electrical energy. The experimental research supported the development of the VIVACE (Vortex-Induced Vibration for Aquatic Clean Energy) converter. The hydrodynamics of the VIVACE converter has been improved continuously since its invention in 2005.

Bernitsas and Raghavan (2008) and Bernitsas *et al.* (2008) studied the use of surface roughness to reduce VIV and break down the span wise correlation. The work was limited to studying the location of the roughness in the form of sandpaper. It was concluded that the use of roughness can reduce/suppress VIV and, to some extent, decrease the range of synchronization. The use of passive turbulence control to harness kinetic energy was studied in Chang and Bernitsas (2011). A multi-cylinder configuration was studied in Kim *et al.* (2011b). The use of multi-cylindrical configuration was an important step forward in making the VIVACE converter to a real three-dimensional device.

3.1.3 Internal Flow Induced Vibrations, Slender Structures

High frequency internal flow induced vibrations have been reported on flexible and rigid pipelines and risers, and may cause fatigue and severe damage to the pipes themselves or to the supporting structures. These vibrations resulted from the internal flow vortex shedding as mono- or multi-phase fluid was conveyed in the pipes. Severe vibration has also been reported in tie-in spools subjected to slug flows. Prediction of internal turbulent flow in LNG pipe transfer systems has also been reported as critical, especially as the maritime transportation of cryogenic LNG increases worldwide.

A collection of practical applications for vibration induced by cross and parallel flows, as well as by internal fluid flow, was presented by Kaneko *et al.* (2008). The effect of the internal flow on a vertical riser subjected to VIV was numerically investigated by Keber and Wiercigroch (2008). Guo and Lou (2008) carried out experiments to assess the effect of internal fluid flow on the vortex-induced vibration. They found that the vibration frequency decreased and the riser response amplified as the internal flow speed increased. Shang-mao and Li-ping (2009) employed DNV-RP-F105 to investigate the effect of internal flow velocity and functional loads on vortex-induced vibration response. It was found that internal flow velocity was less important for VIV response than other functional load factors, such as the effective axial force. Namba *et al.* (2009) carried out a series of experiments for modelling the dynamic interactions between the internal fluid and the structure's response on a hung-off rigid riser under axial motion. Yamamoto *et al.* (2009) developed an experimental study to address

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the interaction between the pipe dynamics and its internal mono-phase flow. Blanco and Casanova (2010) investigated the interaction between the internal slug flow and vortex-induced vibration in the fatigue life of submarine pipelines. In some cases, the riser dynamics can be influenced by the slug flow and the fatigue life may be different if each flow is considered separately.

Frohne *et al.* (2008) described a new LNG flexible pipe transfer system and reported that one of the main difficulties is to predict the turbulent internal flow behaviour and, consequently, its associated pressure drop in the corrugated configuration of flexible pipes. Pisarenco *et al.* (2009) investigated the friction factor in flexible hoses used for cryogenic LNG transport.

Analytical and numerical formulations have also been proposed. Bao and Wen (2008) employed a differential quadrature method to analyze the stability of a subsea pipeline subjected to both vortex and internal flow induced vibrations. Olunloyo *et al.* (2008) developed analytical methodology to assess dynamic stress propagation in subsea pipeline systems. Athisakul and Chucheepsakul (2009) employed a variational formulation based on the extensible elastica theory and the work-energy principle to investigate the influence of fluid conveyance on the dynamic response of marine risers. Pinto and Levi (2009) presented a numerical model to assess the dynamics of free hanging risers. The effects of external excitation due to vortex shedding as well as the effects of internal flow were taken into account. Liu and Xuan (2010) developed flow induced vibration analysis of supported pipes conveying pulsating fluids using precise integration method (PIM). In Grant's MSc dissertation (2010), a finite element analysis was employed to investigate flow induced vibrations in pipes, and to capture the critical fluid velocity that induces pipe instability. Osheku *et al.* (2010a) obtained closed form results to assess flow induced acoustic waves in offshore pipelines. Osheku *et al.* (2010b) proposed an analytical formulation to address the vibration of subsea gas pipelines. Aspects of the fluid-structure-mud interaction were included. Finally, Cheng and Vandiver (2010) presented theoretical formulation for the dynamic analysis of top-tensioned risers, which may consist of outer and inner casings and tubing, thus the response is internally coupled by the centralizers or the fluid flow (or both). If the design is optimum as far as the fluid-structure coupling is concerned, the inner pipe may absorb the vibration of the outer casing.

Multi-phase flow induced vibration continued to be a major topic for research. Bordalo *et al.* (2008) developed model tests to investigate the influence of oil and gas mixtures on the motion of slender risers in catenary configurations. The internal flow momentum may have imposed natural whipping displacements, therefore the riser fatigue life may have been compromised. The flow induced dynamic loading depended on the flow rates of the oil and gas phase distribution. Cooper *et al.* (2009) investigated the fatigue design of flowline systems with slug flow. The slug induced fatigue problem and techniques for predicting fatigue damage during design were presented. Belfroid *et al.* (2009) studied the flow induced pulsations in flexible risers, which can cause singing in offshore and subsea installations. Operational and design guidelines to mitigate this phenomenon in existing facilities and recommendations for adequate design were reported. Casanova and Blanco (2010) addressed the effect of soil properties on the vibration of pipeline spans subjected to slug flows. The results showed that the complex pipe-soil interaction is a key parameter in assessing the vibration response and fatigue life. Yamamoto *et al.* (2010) carried out experimental tests to assess the internal flow rate and pipe's oscillating frequency on the riser response. Mono-phase and bi-phase fluid of liquid and solids in suspension were used. Zhao *et al.* (2010)

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developed a methodology to predict the fatigue life of rigid tie-in spools (pipe bends) subjected to the passage of slug flows. Nielsen *et al.* (2011) presented a new carcass profile to prevent the internal gas flow vortex-induced singing behaviour in flexible risers. Under certain operational conditions, these pulsations may yield severe structural vibrations and potential fatigue failure on the top connection, such as reported on Statoil's Åsgard B platform.

3.2 Very Large Floating Structures

Very large floating structures (VLFS) technology allows the creation of land from the sea without the need for a massive amount of fill materials. These kinds of structures have been gradually appearing in many parts of the world for applications such as floating bridges, floating piers, floating performance stages, and floating storage.

In Jiao *et al.* (2009), a two-dimensional composite strategy was applied in order to couple a linear global solution with a nonlinear local analysis of pontoon-type VLFS. The effect of air cushion and resulting slamming pressure was considered. The numerical results were also compared with experimental data and other numerical solutions. The bottom slamming of a VLFS was also studied in Greco *et al.* (2009) by means of theoretical and numerical methods.

In Korogi *et al.* (2009), a new type of mobile VLFS (which is referred to as a VLMOS) for the purpose of generation of wind-energy was assessed. Towing tank tests were performed with a 1/100 scale model of the structure. Wang *et al.* (2009b) considered the hydroelastic response of interconnected beams that model a longish VLFS. The study investigated the design of the mechanical joint in order to reduce the hydroelastic response. A further extension of the analysis was provided by Wang *et al.* (2011) where one-, two-, three-, and four-line hinge connectors along two directions for a square-shaped VLFS are considered.

In Chen *et al.* (2009b), hydroelasticity theory considering the second-order fluid forces was addressed. The influence of the magnitude of the frequency increment on the computed response was analyzed. It was found that the second order response in some cases may be of the same magnitude as the first-order response.

Jin and Xing (2009) applied a mixed mode function – boundary element method for analysis of the dynamic behaviour of an integrated model of an aircraft – VLFS – water interaction system that was excited by aircraft landing impacts.

There seemed to be a steady development both with respect to VLFS concepts of practical relevance, as well as more refined computational methods. This trend is expected to continue also through the next decade.

3.3 Other Offshore Topics and Applications

Field observations from offshore installations, along with laboratory investigations and numerical simulations, continued to provide improvements to our understanding of structural dynamic response. Several of the selected research findings that follow are for fixed offshore structures, which represent a large proportion of all offshore installations. Many of the insights and observations for fixed offshore structures will also be relevant to floating offshore structures.

3.3.1 Wind-Induced Vibrations, Offshore Structures

Significant cracking has arisen in flare booms located on various platforms in the North Sea. Moe and Niedzwecki (2009) carried out an experimental investigation to examine the possible cause of vibrations for the Heimdal flare boom. Large vibrations of

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around 50 mm amplitude had been observed for tubular bracing members. The vibrations only occurred in special wind conditions where the direction was parallel to the faces of the flare boom. The observed amplitudes and frequencies of vibration were not consistent with the predicted behaviour of stand-alone members for the encountered wind speeds. It was postulated that the bracing members could be susceptible to wake-induced vibrations caused by the larger diameter chord members located upwind. A set of experiments was carried out to investigate the behaviour of an elastically supported circular cylinder located downstream from a larger cylinder of around twice its diameter. Maximum measured amplitudes of vibration in the experiments were approximately twice those of a stand-alone cylinder and occurred at higher wind speeds. A permanent solution for Heimdal was to fit shrouds and post-tensioned ties.

Jia (2011a) presented a practical approach for calculating wind induced fatigue damage for tubular structures, such as flare booms, based on nonlinear dynamic analysis. Realizations of spatially correlated random wind fields were used to generate dynamic responses in the time domain whilst taking geometric and load nonlinearities into consideration. Crosswind components of a turbulent wind field were shown to be an important contributor to the calculated fatigue damage. The damage was also shown to be sensitive to the chosen wind grid size. A larger grid size produced a more correlated wind field, leading to increased dynamic response and over-prediction of fatigue damage.

3.3.2 Wave-Induced Vibrations, Offshore Structures

Offshore platforms may be exposed to wave impacts and slamming in extreme wave conditions. Vertical wave loads on decks due to insufficient air-gap are a major concern for many in-service platforms. Improved engineering tools and methods for the prediction of loads from wave impact on FPSOs and offshore platforms in severe sea states were described in Stansberg and Baarholm (2010). A brief presentation of the tool development was reviewed, and numerical examples were demonstrated. Applications of improved engineering methods and procedures included wave-in-deck on jacket platforms, wave amplification with wave impact on large-volume platforms, green water/bow flare slamming on FPSO, and impact on columns, Stansberg *et al.* (2010).

Reliable estimates of the magnitude and duration of the impact loads are important in assessing structural and global response of an offshore platform. In Kota and Moan (2010), a Gaussian formulation of incident wave-kinematics was applied to derive a joint probability density function of deck-wetting (or exceedance) duration and its spatial extent.

Ringing vibration of dynamically sensitive offshore structures has received considerable attention in the past. A revival of practical interest in this area may be emerging in relation to the assessment of dynamic response for monopile structures supporting offshore wind turbines. Zang *et al.* (2010) investigated the loading on a bottom founded vertical circular cylinder when subjected to focussed wave groups (including breaking and non-breaking conditions), with particular attention given to the harmonic content at higher frequencies. A Joint Industry Project concerned with the dynamic response of offshore wind turbines was underway at MARIN, but its findings have yet to be published (Snieckus, 2011).

3.3.3 Current Induced Vibrations, Offshore Structures

Current induced motion (VIM), on a multicolumn structure is studied in Waals *et al.* (2007). The paper discussed the dynamic behaviour in current of multicolumn floaters and the associated complex flow patterns for both TLP and semi-submersibles.

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Holmes (2008) compared CFD simulations with wind tunnel experimental data for a towed bare spar. It was suggested that the use of wind tunnels for selected geometries might offer an inexpensive means to improve existing modelling techniques.

Roddier *et al.* (2009) investigated the effect of Reynolds number and hull appurtenances on spar vortex-induced motions (VIM) for a vertically moored 6-DOF truss spar hull model with strakes. Tests were performed at three different experimental facilities, at three different geometric scales: 1) University of California, Berkeley (scale 142.8:1); 2) Force Technology, Denmark (scale 65:1); and 3) David Taylor Model Basin, Bethesda (scale 22.3:1). Froude numbers were varied from 0.10 to 0.26 with the associated Reynolds numbers. As a result of the range of model sizes, the tests at the University of California and Force Technology were both accomplished at subcritical Reynolds Numbers, and those at David Taylor Model Basin were performed a supercritical Reynolds Numbers. Overall, the three test series covered a Reynolds number range from subcritical $4.1\text{x}10^4$ to supercritical $1.7\text{x}10^6$. In order to assess the effect of appurtenances and current heading on strake effectiveness, four different configurations were tested: 1) hull with chains and pipes, with anodes; 2) hull with anodes but no chains and pipes; 3) hull with strakes but no other appurtenances; and 4) hull with chains and pipes but no anodes. The strakes were present in all configurations. Altogether, 822 tests were performed.

3.3.4 Ice-Induced Vibrations, Offshore Structures

Offshore installations in ice-infested waters may encounter moving sea ice features. Ice actions due to these moving sea ice features will produce dynamic loads, even while the intact ice sheet moves at constant velocity and crushes against a structure. Ice-structure dynamic interaction can cause the structure to suffer severe vibration problems, which are known as ice-induced vibrations and have been reported in full scale vertical and conical structures.

Two successive research projects in the period from 1999 through 2003, known as LOLEIF and STRICE (http://www.strice.org), respectively, investigated ice-induced vibrations of vertical structures. The findings from these two projects have been integrated into the latest ISO 19906 (2010).

Yue *et al.* (2009a) observed that the ice failure mechanism associated with ice crushing depends on ice speed, and classifies as ductile, ductile-brittle transition, and brittle failure, respectively. Resulting dynamic ice forces can be classified into three modes responsible for quasi-static, steady-state, and random structural vibrations, respectively.

Xu *et al.* (2011) discussed the main factors that that influence the typical dynamic ice sheet, conical interaction process (obtained in Bohai Bay) covering ice velocity, ice thickness cone diameter slope angle and friction coefficient, and water depth, and made a comparison with other field data obtained from Kemi-1 and Confederation Bridge.

Compared to the bending failure of ice against conical structures, the crushing failure of ice on vertical faces is complex and difficult to investigate at full scale. Thus, crushing failure is best investigated in the laboratory and by theoretical analysis. In general, ice testing facilities attempt to jointly satisfy both Froude and Cauchy scaling laws. However, Palmer and Dempsey (2009) pointed out that, for ice moving slowly against stationary structures, Froude scaling was irrelevant. With only the Cauchy scaling to satisfy, it was not necessary for model ice to be made weaker and more

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ductile by contaminants such as urea. They concluded that ice was best modelled by ice, especially for ice action on a stationary structure.

Määttänen *et al.* (2011) performed near full-scale model tests to study ice crushing phenomena against a compliant stiffened plate structure. In order to have repeatable and homogeneous model ice properties, the ice blocks were manufactured by a snow ice technique with low salinity water impregnation under vacuum in the mould. Natural snow was used for ice crystal seeding, and to promote unidirectional freezing from below. Altogether, 22 ice blocks were crushed with different ice velocities and plate compliance.

Based on 11 sets of published data from vertical-sided structures, Palmer *et al.* (2010) presented a dimensional analysis of the problem, and showed that there was a correlation between the different kinds of cyclic movement and a dimensionless parameter akin to reduced velocity in VIV that can be used to link one to the other. This has been observed in both full-scale structures and models. Kärnä *et al.* (2011) pointed out that the time-varying nature of ice actions and the corresponding ice-induced vibrations should be considered in the design. The potential for dynamic amplification of the ice action effects due to frequency lock-in of ice failure and natural frequencies should be assessed. Particular attention should be given to dynamic actions on narrow structures, flexible structures, and structures with vertical faces exposed to ice action. Thus, some related models and approaches were proposed to solve the ice-induced vibrations for vertical structures.

Using advanced methods, analyses can be continued after complete material failure, and failed sections can be retained in the model without violating the law of conservation of mass. Kolari *et al.* (2009) proposed a new approach for the modelling of interaction, where the failure of material was modelled with anisotropic continuum damage mechanics (CDM) model and the failure mode was assumed to be brittle. The CDM model was used to predict direction of a crack evolution, while the proposed model update technique was applied to propagate the crack in finite element geometry. The proposed approach was used to simulate the ice failure process during the tensile test and the level ice acting on a conical structure. Gürtner (2009) and Konuk *et al.* (2009) developed a cohesive element method to study the dynamic ice-structure interaction processes when a level ice sheet moves against a vertical flexible cylinder. The model was implemented using LS-DYNA software. This cohesive element based framework could offer a reliable and rational methodology for solving ice-structure interaction problems that can capture the characteristics of the ice failure process and the ice-induced vibrations.

When an offshore installation is deployed into ice-covered waters, ice-induced vibrations can occur regardless of the water line geometry (vertical or conical), the structure dimension (narrow or wide), and the structure property (compliant or stiff). The nature of ice-induced vibration derives from the dynamic ice action, and there is no fully rigid structure in practice. Numerical simulation tools have been used in ice engineering, mainly to predict dynamic ice action effects in structure design. The dynamic ice action is still not fully understood, so further studies on this topic are still needed. Furthermore, the dynamic process of ice acting on a vertical structure is much more complex than the dynamic process of ice acting on the conical structure, especially for narrow compliant vertical structures, as the frequency lock-in phenomenon may occur under given ice conditions and structural characters.

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3.4 Noise

Noise and underwater sound due to exploration, construction, transport, drilling, and production are important for offshore activities. Identifications of pertinent mechanisms, multiple potential noise sources, and noise paths from vessels and drilling platforms are necessary for implementation of effective treatments. Hence, the subject of noise can be divided into two major branches in which specific sources and noise makers are identified, and possible treatments for those sources, mechanisms, and paths are considered.

Noise creation from various oil and gas industry activities can be summarized under the following main titles, such as seismic exploration, pile driving, explosives, propellers and thrusters of vessels, machinery noise of vessels, dredges, post trenching, hand tools, platforms, hovercraft, aircraft, and pipelines (Spence *et al.,* 2007).

An experimental study was conducted on the combined acoustic and hydrodynamic variables in a moonpool by Sadiq and Xiong-liang (2008) using a hydrophone near the free surface inside the moonpool. The acoustics were monitored during the change of the free surface due to increased forward speed, with the aim of comparing the results obtained for circular- and square-shaped moonpools. It was found that noise reached higher levels in the square-shaped moonpool in comparison to the circular-shaped one, and that changes in the in-flow wave period, the wave height, and the angle of attack for the square moonpool had significant effects in the change of the sound level.

The impact on marine species of underwater noise due to pile-driving for wind turbines was studied by Bailey *et al.* (2010). Presented in detail are the methods of research and the results regarding the background noise, the pile-driving noise, the source level, and the sound propagation model, including their potential impact on the marine mammals off the north east coast of Scotland. It was observed that the level of noise from pile-driving was detectable above the background noise levels even in a distance of 70 km, and it was suggested that higher background noise levels recorded at the turbine site are likely to be a result of the pile-driving vessel and the support ships. An impact on the behavioural habits of marine mammals due to pile-driving operations has also been observed in the form of divergence from their natural routes.

In order to cover a wide-range of frequencies, Gang *et al.* (2011) applied combined structural FEM, acoustic FEM, and SEA, in hybrid methods to predict the noise and vibration for a semi-submersible design. The applied prediction process for noise and vibration requires the definition of sources of vibration and noise and an assessment of the source spectrum for both. An acoustic finite element or boundary element model was used in both the low-and mid-frequency range. In the low frequency range, the acoustic finite element/boundary element method was hybridized with a structural finite element model. In the mid frequency range, the acoustic finite element/boundary element method was hybridized with a SEA model. Also, the SEA was used in the high-frequency range. It was found that 45-percent of the cabins' noise levels were above the noise limit established for the original design. Application of the absorption and high insulation materials resulted in reduced sound levels in the living quarters, which met the technical requirements.

Eijk and Elferink (2011) carried out an extensive dynamic analysis to achieve stringent control on noise and vibration levels for an offshore reciprocating compressor system. In their work, dynamic analysis and the efforts taken in compressor, skid, motor, piping, and deck design to meet the very stringent specified requirements that ensure a safe and reliable system for the long term operation, is presented in detail. A

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wide range of optimization has been carried out on compressors and drivers, pulsation dampers, and orifice plates in the on-skid compressor piping, anti-vibration mounts. The characteristics of the system, as optimized to meet the criteria [the API Standard 6(8)], are given in a very long list at the end of the paper.

Lloyd *et al.* (2011) presented a paper concerning the modelling of noise sources for use in the assessment of noise influence on marine life. They focused on improving noise source prediction of tidal turbines rather than the development of impact assessment techniques. Categorization of hydrodynamic tidal turbine noise sources was presented as a flowchart. This categorization was also provided in a table under the items of source, origin, importance, frequency type, and directivity. The predicted maximum far field sound pressure level (SPL) was not found to be high enough to cause threshold shift in marine animals based on standard measures. The suggestion was that it was very important to develop new comprehensive models in assessing the environmental impact of turbines for their certification, and that the present model could be a starting point.

3.5 Shock and Explosion

In offshore units, current interest is in structural response following extreme explosive loads arising from gas explosions and underwater explosion. Most explosions on offshore installations are gas explosions. Gas explosions and fires are extremely hazardous in offshore installations, which have serious consequences for health, safety, and the surrounding environment. Two examples are the Piper Alpha accident, which occurred on 6th of July 1988 (Cullen, 1990), and the Deepwater Horizon accident, which occurred on 20th of April 2010 in the Gulf of Mexico (Paik and Czujko, 2011). The latter explosion and subsequent sinking resulted in a tremendous outflow of oil.

The aim of blast analysis is to predict the dynamic response or structural damages of structural members, piping, equipment, cables and other appurtenances on offshore installations under blast loads. Both analytical and numerical methods can be used for blast response analysis due to gas explosion (Mohamed and Louca, 2008). In general, the numerical method is better suited for brief blast transient analysis, particularly involving large deformation and plasticity, which can include non-linear material and geometry, strain rate effect, and etc.

3.5.1 Internal Explosion

Internal explosions usually occur in a largely confined space, such as inside enclosed modules, or in an oil tank, or a leg of a platform. Overpressure is usually created by the expansion of gas in a confined volume as it burns and exceeds the vent capacity of the space. The presence of obstacles in the path will further enhance the overpressure generation and destruction (Pula *et al.,* 2006).

The presence of obstacles in the path will further enhance the overpressure generation and destruction (Pula *et al.,* 2006). Vik *et al.* (2011) and Middha and Hansen (2009) used the computer tools FLACS (Ultimate Strength for Offshore Structures) (USFOS, 1993–2001) and USFOS FLACS (Flame Accelerator Simulator) (FLACS) to simulate the response of platform under explosion. Yasseri *et al.* (2003) gave a reliability of explosion resistant design method. A blast wall installed on a North Sea installation was selected as a case study to demonstrate the proposed method. Lee and Yoon (2011) studied the dynamic characteristic of a simple beam under the blast load by using nonlinear dynamic FE analysis, and a safer and less costly approach was suggested.

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3.5.2 External Explosion

External explosions include gas explosion in unconfined or partially confined and highly congested spaces (Pula *et al.*, 2006), and blast wave acting on nearby offshore units.

Ignition of any vapour cloud under the process area of an FPSO (Floating Production Storage and Offloading) vessel and some offshore modules will lead to an explosion referred to as a partially confined explosion (Paik and Czujko, 2009). In this case, overpressure generation is mainly due to turbulence generated by the obstacles, such as process equipment in the path of the expanding gas. Available empirical models can be used for modelling this kind of explosion, as they have been tested and validated for these conditions. A review of all the empirical models and their comparison with experimental data has been carried out by Fitzgerald (2001).

The prediction of the received loading onto nearby structures caused by the resultant blast wave has been given attention. Louca and Mohamed Ali (2008) investigated the behaviour of a typical offshore topside structure subjected to blast loading caused by hydrocarbon explosions. Monti and Molinari (2011) presented an engineering approach to assess the structural integrity of a submarine pipeline subjected to an underwater explosion, taking in due account the loading due to the shock wave and the gas bubble pulsation. Zhang *et al.* (2006) simulated the dynamic characteristics of an underwater explosion bubble near boundaries, and solved the interaction of bubble and elastic-plastic structure by coupling with FEM.

3.6 Damping and Countermeasures

Various studies have explored the possibility of using passive and active methods to mitigate dynamic responses in the elastic modes of offshore structures. Proven systems in land-based structures have led, naturally, to the consideration of applications in fixed offshore structures. The concepts are also relevant to floating offshore structures, particularly tension leg platforms in deep water that are subject to springing and ringing responses in the elastically restrained modes of heave, pitch, and roll.

Recent interest has considered the effects of system uncertainty on the effectiveness of devices. It is possible that optimization of a device with a deterministic representation of system parameters may be lead to overestimation of performance. Taflanidis *et al.* (2009) investigated the use of mass dampers for tension leg platforms with emphasis upon reducing the response of closely spaced modes and incorporating uncertainty in the modelling of the structural system and environment. It was shown that a dual mass damper system in each hull column provided an improved performance compared to a single mass damper in reducing the heave and pitch responses, especially when the modes had different natural frequencies.

Colwell and Basu (2009) considered the use of tuned liquid column dampers in monopile structures for supporting offshore wind turbines. A significant reduction in peak responses and improved fatigue performance was predicted. It was pointed out that space limitations in the horizontal direction might make other types of damper suitable. Chakraborty and Debbarma (2011) investigated the effect of uncertain but bounded system parameters on the optimal design of liquid column vibration absorbers for the earthquake response of land-based structures. While uncertainty in the loading had exerted a dominant influence on the structural safety, it was shown that uncertainty in the system parameters could play an important role in an optimized design of the dampers.

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Lackner and Rotea (2010) developed a simulation tool for the assessment of passive, semi-active (tuneable over time), and active control of floating structures for the support of offshore wind turbines. Tuned mass damper systems were fully coupled with the dynamics of the wind turbine model. Load reductions beyond those achieved with passive systems were achieved at the expense of active power consumption. It was postulated that the load reduction potential might be better than that associated with individual blade pitch control. Stewart and Lackner (2011) investigated the effect of actuator dynamics in the control of active mass dampers for floating wind turbine systems. It was shown that control-structure interaction could strongly influence the required levels of actuator power and torque, and that lowering the control motor gear ratio could reduce the interaction. Namik and Stol (2011) investigated the use of individual blade pitch control for wind turbines on floating barge and tension leg platforms. The proposed control strategy was shown to be capable of reducing the tower loading on tension leg platforms to levels comparable with onshore turbines.

Yue *et al.* (2009b) investigated the use of tuned mass dampers for mitigating ice-induced vibrations in fixed steel jacket structures. Such damping devices were suitable for retrofitting applications, and could be oriented to align with the primary direction of the response to ice loading. Simulations indicated that the tuned mass damper could be a very effective means of reducing vibrations. An alternative approach, which is more suited to incorporation at the design stage, was to use steel rubber isolator bearings placed between the substructure and the topsides. Xu *et al.* (2009) conducted an experimental investigation into the fatigue properties of such isolators in cold conditions. Their experimental data provided some guidelines for design, and a basis for fatigue life assessment.

Ibrahim (2008) provided a comprehensive review of nonlinear passive vibration isolation in relation to the protection of structures from severe earthquake ground motion, shocks, and impact loads.

Countermeasures for slender structures can be found in Section 3.1.2.

3.7 Monitoring

The development and employment of new monitoring technologies has progressed steadily. Initially, a review is presented for its applications in large offshore structures; i.e., fixed and floating platforms for exploration and production purposes. Secondly, slender structures such as risers, umbilicals, pipelines, and mooring lines are addressed.

3.7.1 Monitoring of Large Offshore Structures

Structural monitoring can aid in assessing offshore platform structural integrity. There are two main categories of such monitoring: (i) online and (ii) offline procedures. Examples of both types of monitoring systems are considered here.

In Rijken and Leverette (2009), full-scale measurements of Vortex-Induced Motion (VIM) of a deep draft semisubmersible with four square columns were presented. A comparison between the field observations and design guidance was also provided.

Black (2009) discussed structural vibration monitoring as an aid in assessing offshore platform structural integrity. If the platform's natural period increases over time then, in the absence of a change in mass, that can indicate a loss of structural integrity. Zhang *et al.* (2009) and Zhang *et al.* (2010) described full-scale measurement for many years of platforms in Bohai Bay that were subject to ice loading. For the same area, Xu and Yue (2010) addressed the dynamic ice forces that occurred during ice and platform

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interaction. By application of ice load measurement panels and video cameras, the forces which act on a narrow cone were analyzed.

Matos *et al.* (2010) described a comparative study between full-scale measurements and theoretical predictions for the second-order pitch and roll low-frequency motions of the semi-submersible platform *P-52*. These observations were seen to contribute with important feedback to future designs.

In Peters and Adegeest (2010), online monitoring during transports of large heavy cargo, i.e., jack-up rigs or semi-submersibles, was described. This provided a valuable tool to ensure a safe and economical voyage.

3.7.2 Monitoring of Offshore Slender Structures

In-situ monitoring provides critical information during operational and extreme conditions for SCRs, flowlines, flexible pipes, umbilical cables, and mooring lines. Usually, data processing encompasses remote (standalone), hardwired, and acoustic methods. These systems can be crucial for the structural integrity management of the riser system and provide feedback information useful for establishing recommendations for new designs. They also give important phenomenological insight into the complex hydro-elastic-soil-riser interaction behaviour. More often, integrity monitoring strategy has been included as selection criteria for riser systems.

Maheshwari *et al.* (2008) reviewed existing programs and level of success for riser integrity monitoring systems and discussed the pros and cons of hardwired, stand-alone, and acoustic techniques. Job and Hawkins (2008) presented a monitoring program developed to assess the vibration and marine currents in freespan flowlines in the Gulf of Thailand. Lanan *et al.* (2008) described the Oooguruk offshore arctic flowline monitoring system. El Hares *et al.* (2011) presented an integrated pipeline integrity monitoring system. Watson *et al.* (2011) reviewed requirements for monitoring pipelines susceptible to lateral buckling and walking. Elshafey *et al.* (2011) developed an online monitoring system based on longitudinal strain measurements. Taby *et al.* (2011) discussed the employment of an on-board monitoring system capable of visualizing online the pipe static configuration during lay operations. Karayaka *et al.* (2009), Edwards *et al.* (2011), and Enuganti *et al.* (2011) presented monitoring systems employed in steel catenary risers. Legras and Saint-Marcoux (2011) described the requirements of an integrity management program for hybrid riser towers.

As far as slender structures are concerned, most of the recent developments on monitoring have been dedicated to fatigue in flexible pipes. Fibre optic sensing was the dominant technique, but the industry is seeking other solutions such as acoustics and magnetic fields.

Rabelo *et al.* (2009) presented a monitoring program to assess the annulus condition of Petrobras Marlin Sul flexible risers. Sas *et al.* (2008) described the West Africa deepwater Agbami production flexible riser system. Weppenaar and Kristiansen (2008) described existing and potential applications for fiber optic monitoring of flexible pipes, such as temperature along the pipe and strain monitoring at discrete points, as well as calculation of remaining pipe lifetime.

Real-time monitoring may also become a valuable tool for failure detection and predictive maintenance. Cour *et al.* (2008) showed how to manage the risk of fatigue failure in flexible risers with a condition monitoring technique using optical fiber technology. Soares *et al.* (2009) described an acoustic testing technique for detection of flexible pipe tensile armour wire rupture. Weppenaar *et al.* (2009) presented a real-time system for

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fiber optic gas chemical monitoring inside the annulus of flexible risers, based on the novel Quartz-Enhanced Photoacoustic Spectroscopy (QEPAS) technology. For sour service conditions, precise corrosion fatigue may be calculated combining the input from the gas monitoring system with data from strain and temperature monitoring.

McCarthy and Buttle (2009) showed an ongoing project for real-time magnetic inspection technique for flexible risers. It was non-invasive, and may be able to detect the load and broken wires as well. Corrignan *et al.* (2009) presented a joint program carried out by Technip and Schlumberger Subsea Surveillance to develop a new monitoring technology for detection of failure or conditions leading to failure of tensile armour wires in flexible pipes. The system was based on a clamped composite structure with embedded optical fibers. Morikawa *et al.* (2010) described Petrobras' ongoing inspection and monitoring programs based on acoustic emission, measurement of residual magnetic field, optical sensing, and visual monitoring with cameras. The tests conducted with optical fiber sensors provided the best results. Roques *et al.* (2010) described the measurement principle and hardware of a new system developed by TOTAL and Schlumberger to assess the flexible pipe annulus condition. Field test results for various risers in a West Africa field were shown. Dahl *et al.* (2011) discussed recently developed monitoring technology that employed embedded optical fibers for integrity management of flexible risers during their service lives. O'Brien *et al.* (2011) presented the results of the SureFlex JIP. Documents on the *State-of-the-Art of Flexible Pipe Integrity* and a *Guidance Note on Flexible Pipe Integrity Assurance* are publicly available. The paper compiled an extensive survey that included operational use worldwide, along with damage and failure incidences.

3.8 Uncertainties

Uncertainties exist in system models and system parameters. In areas involving complex physical phenomena (e.g., vortex-induced vibrations), there is a continuing need to develop improved models and work towards better comparisons with full-scale measurements. There are important areas of practical application, however, involving satisfactory system models, where an assessment of uncertainties in system parameters (e.g., mass, stiffness, and damping) could play an important role in determining the effectiveness of an engineering design. An interesting example is the optimisation of a system employing tuned mass dampers, where the uncertainty in system parameters can undermine the confidence in the ability to tune the system at the design stage. Taflanadis *et al.* (2009) incorporated uncertainty in system parameters in their investigation into the possible use of mass dampers in tension leg platforms (see Section 3.6). The subject area is often referred to as uncertainty in structural dynamics, and is a field of growing interest; i.e., Mace *et al.* (2011). Practical application of the emerging methods within the dynamic analysis of offshore structures, however, appears to be rather limited at present.

3.9 Standards and Acceptance Criteria for Ice-Induced Vibrations

Since ice-induced vibration has been found in offshore structures lately, many studies have been performed. The experiences and findings, however, have been complied into recognized design codes. The following codes provide guidelines and criteria for assessing the dynamic ice action effects in design.

- API RP 2N (1995) only provides a brief statement in Section 5.4.16.
- IEC 61400-3 (2009) provides an informative annex on ice action and effects for offshore wind turbine support structures. It noted that the offshore wind turbine

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support structure usually is a compliant structure, and it paid attention to the dynamic ice loading (see Annex E.4.6, *Dynamic Loading*). It required that the wind turbine should be checked for dynamic effects from ice loading. Further, if statistical data or measurements are not available, IEC 61400-3 suggests simplified equations for dynamic load simulation. IEC 61400-3 notes the resonant dynamic ice action scenarios, and provides a criterion for tuning. As for shock impact of ice floe, IEC 61400-3 requires it to be checked with a transient load approach and suggests a load function in a piecewise form.

- DNV OS J101 (2010) also provides guidelines on the dynamic ice loading on vertical and conical structures for offshore wind turbines, but it does not mention the shock impact loading due to ice floe (see Section 4, *Loads and Load Effects*, E 500). For vertical structures, DNV OS J101 highlights a frequency lock-in phenomenon that implied that the structure becomes excited by vibrations in its natural mode shapes. The structure should be designed to withstand the loads and load effects from dynamic ice loading associated with lock-in when tuning occurs.
- ISO 19906 (2010) provides comprehensive discussions on dynamic ice action and the effect for offshore installations, covering dynamic actions on vertical structures, dynamic actions on conical structures and fatigue accumulation due to ice actions (details see ISO 19906, Annex A, A 8.2.6).

Olav Olsen a.s., KARNA Research and Consulting, and Technical University Delft (TUDelft) has initiated a joint industry project (JIP) called "Ice-Induced Vibrations," whose objective is to establish an engineering approach to assessing ice-induced vibrations for a wide range of structures and ice conditions. The JIP aims to consider not only the typical analyses of the susceptibility to frequency lock-in, but also determination of the dynamic response to various kinds of dynamic ice actions. The project has paid much attention to the frequency lock-in and aimed at new background information for updating ISO 19906. It is focused on the development of an engineering approach to the design of vertically-sided structures against ice-induced vibrations.

4 BENCHMARK STUDY OF SLAMMING AND WHIPPING

Throughout the maritime world, considerable efforts have been spent on predicting loads associated with slamming (i.e., Kapsenberg and Thornhill, 2010, or Tuitman, 2010). Up to now, little attention has, however, been paid to the accuracy of the translation from these loads to the structural responses. The ISSC 2012 Dynamic Response committee, therefore, performed a benchmark study on this topic. The goal of this benchmark was twofold: on the one hand, the degree of variation in estimates produced by different methods and organizations was revealed; on the other hand, the absolute error made in the analyses was investigated by comparison with model test measured responses.

There were six participants: two research organizations (Marin and TNO), two class societies (GL and Indian Register of Shipping), one university (Norwegian Technical University), and a consulting company (The Glosten Associates). The benchmark was blind and consisted of three different stages. Not all participants delivered results for each stage. The tasks for each stage, as well as the results, were discussed in the subsequent sections. Participants were free in choosing methods for obtaining the results. The method they used was also described in the subsequent sections.

In order to investigate the absolute error, use was made of results from tests performed at Marin with a flexible segmented backbone model (i.e., Drummen, 2008)

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Figure 2: Model of a 173 m long ferry, courtesy of CRS

of a 173 m long ferry (see Figure 2). The data was provided by Cooperative Research Ships (CRS). In CRS, Marin brought together a group of companies with a common interest in non-competitive research. The aluminium circular backbone was instrumented with strain gauges to measure, amongst others, the vertical bending moment between the different sections. The bow was built as a separate segment and connected to the forward hull segment through a six-component force transducer. This bow segment was instrumented with 23 pressure gauges to measure the detailed pressure distribution and six accelerometers to measure the local vibrations. Of the pressure gauges, five were located on the centre line of the ship, 17 on the starboard (windward) side, and one on the port (leeward) side. Ten accelerometers were fitted inside the model, one laterally and one vertical accelerometer in the centre of each segment.

4.1 Modal Parameters

For the first stage of the benchmark, participants were asked to determine the shapes and frequencies of the two and three node, dry and wet, and horizontal and vertical flexural vibration modes. At this stage, participants were provided with details of the geometry of the model, including electronic hull description, locations of cuts, dimensions of the backbone, and mass distribution.

In order to determine the shapes and the frequencies of the dry modes, Participant A used a model consisting of a backbone, modeled with beam element and links to connect the inertia of the various segments, located at their center of gravity, to the backbone. For the purpose of calculating the wet shapes and frequencies, the structural modes were mapped onto the hydrodynamic mesh, and the 3D panel method Hydrostar was used to compute the hydrodynamic coefficients according to the method described in Tuitman (2010). Participant B used a very similar structural model. For determining the wet modes, the infinite frequency added mass was determined with a 3D panel method. This mass was subsequently added to the different sections of the structural mass, effectively diagonalizing the mass matrix. Both Participants A and B determined the dry and wet modal parameters from eigenvalue analysis. The former used MSC.MARC for the dry modes, and HEFREQ (Tuitman, 2010) for the wet modes. The wet analysis was based on the mode shapes as obtained from the dry analysis. Participant B used ANSYS for both the dry and the wet cases. Participant C built a 3D finite element model of the physical model. All plating of the model, such as bottom shell, main deck, side shell, transverse bulkheads, and aluminum beam, was modeled using shell elements. Mass elements were used to model lumped mass. The added mass was calculated for 15 sections using an empirical

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method (Mukhopadhyay, 1989). The added mass was applied to the wetted surface of the hull by adjusting the density of material. The modal parameters were obtained by performing eigenvalue analyses in ANSYS. For determining the shapes and the frequencies of the dry modes, eigenvalue analysis was done by Participant D for a 2D finite element model consisting of beam elements according to the Timoshenko approach, assuming planar shear deformation. For the present purpose, displacements in vertical direction and rotations about the horizontal transverse axis were used. The structural mass was assigned as lumped mass to the nodes of the elements. The effect of added mass was included by making an implicit two way coupling (Oberhagemann *et al.*, 2008a,b) between the structural solver and a RANS solver (Brunswig and el Moctar, 2004). In each time step, the solutions on both fluid and structure domain were found iteratively. A free vibration decay test was done for the ship in water at zero forward speed in order to determine the wet natural frequency. Participant E used a beam model in NASTRAN consisting of 25 nodes. Trapezoidal weight distributions were assumed for each of the five hull segments to produce the correct longitudinal centers. Only the aluminum beam was assumed to be effective structurally. Infinite frequency added mass was estimated using Lewis form sections. The mass, as well as the added mass, was lumped to the nodes of the beam model. Modal parameters were obtained from an eigenvalue analysis in NASTRAN. Participant F also built a 3D finite element model. Although it was less detailed then the one used by Participant C, the model consisted of shell elements. The added mass was calculated separately for each of the segments. This calculation was based on the infinite frequency value of the added mass coefficient using Lewis form sections. Subsequently, it is distributed as point masses. Here, the modal parameters were found from an eigenvalue analysis in Abaqus.

The natural frequencies obtained by the participants for the wet and dry mode shapes are shown in Table 1. Here, a comparison is also made with experimental results. These were only available for the vertical modes. The parameters in water were determined by performing hammer tests in still water. For determining the dry parameters, the fully instrumented model was suspended in air in a soft spring system. The precision error (Coleman and Steele, 1989) of the wet modal parameters was reported by MARIN to be very small. The single 95 % confidence interval of the mean value of the natural frequencies was less than 2 %. Due to the spring system, the uncertainties were larger in air, particularly for the three node mode.

Figure 3 presents the shapes of the two and three node wet vertical vibration modes.

Table 1: Natural frequencies of the global flexural vibration modes in *Hz*. Experimental results are given under 'EXP,' with numerical results from the different participants under the respective letters

mode	EXP	A	B	C	D	E	F
vertical dry 2 node	7.1	6.8	7.1	7.6	6.9	6.9	8.2
vertical dry 3 node	17.7	19	19.8	25.8	19.9	19.9	21.3
vertical wet 2 node	5.1	5.1	4.8	4.7	5.1	5.0	5.1
vertical wet 3 node	11.8	12.1	11.6	11.1	-	12.4	11.9
horizontal dry 2 node	-	6.8	6.6	7.4	-	-	-
horizontal dry 3 node	-	19	16.7	24.5	-	-	-
horizontal wet 2 node	-	6.3	6.0	4.5	-	-	-
horizontal wet 3 node	-	16.6	14.8	10.9	-	-	-

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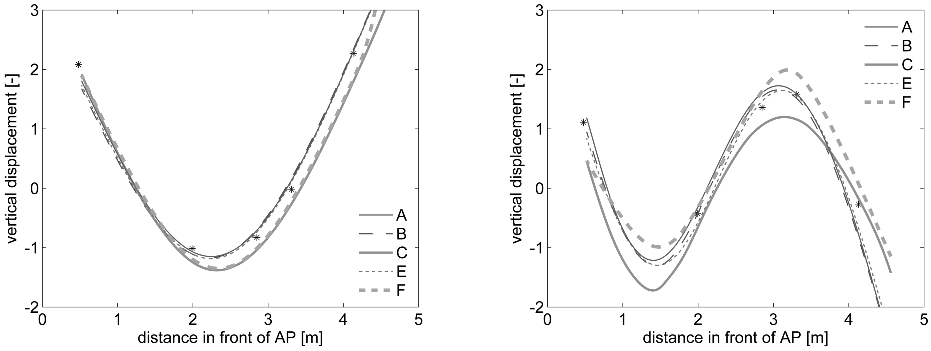

Figure 3: Shapes of the two (left) and three (right) node global vertical wet flexural vibration modes

Table 2: MAC values of mode shapes predicted by the participants compared to those measured on the physical model

mode	A	B	C	D	E	F
vertical dry 2 node	0.96	0.97	0.93	0.98	0.98	-
vertical dry 3 node	0.97	0.97	0.93	0.98	0.98	-
vertical wet 2 node	0.99	0.98	0.95	-	0.99	0.96
vertical wet 3 node	0.94	0.94	0.90	-	0.96	0.95

As can be seen from the figure, results from Participants A, B, and E were well in line with experimental results.

The modal accuracy coefficients (MAC) between the shapes determined experimentally and those determined by the participants are given in Table 2. The MAC is defined by Equation 1 as:

$$MAC = \frac{\left|\Psi_{FE}^{T}\Psi_{EXP}\right|^{2}}{\left(\Psi_{FE}^{T}\Psi_{FE}\right)\left(\Psi_{EXP}^{T}\Psi_{EXP}\right)} \tag{1}$$

where Ψ_{FE} and Ψ_{EXP} respectively denote the finite element mode shape and the one found for the test model. Two mode shapes with a MAC equal to one indicate identical modes, or full correlations. The natural frequencies determined by the participants generally agreed well with the experimental results. An exception was the dry three node mode. Results corresponded well between participants, but not with the experiments. The differences here were significantly larger than for the other modes. It was expected that the difference was related to the experimental setup, as mentioned above. No noticeable uncertainties were observed in the prediction of the added mass by comparing the performance in water and in air.

4.2 Response to Inputs

After delivering the results presented in the previous section, participants were provided with natural frequencies and damping ratios as found for the physical model. Participants were allowed to update their model. The next step in the benchmark was to apply analytical yet realistic pulses to the model. The pulses were provided to the participants as time series of the force to be applied on the model. Participants

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were asked to investigate five durations of the pulse. The investigated durations were evenly distributed between $0.5 \cdot T$ and $1.5 \cdot T$, where T was the natural period of the vibration mode under consideration. The two and three node vertical vibration modes were investigated. Each impulse was $50\,Ns$, and the time series were $60\,s$ long. The force was distributed equally in length over the most forward 10 % of the length of the model. Calculations were done for the model in water. For this stage, no comparison with experimental results was made. For each investigated pulse, participants were asked to calculate a time series of the vertical bending moment amidships.

Participant A did not update their structural model as their predictions of the natural frequencies were within 2.5 % of the experimental values. As mentioned in the previous section, the mode shapes of their structural model were mapped to the 3D hydrodynamic model. With this model, a nonlinear hydroelastic time domain simulation (Tuitman, 2010) was done. Only the first five flexural modes in vertical direction were included. The sway, roll and yaw modes were suppressed. The damping ratios for the two and three node vertical modes were set to 0.8 % and 0.7 %, respectively. The damping of the other flexural modes was set to approximately 2 %. No additional damping was added to the rigid body modes. In order to tune the natural frequencies of their model to the experimental values, Participant B adjusted the stiffness of the model. Due to this change, the natural frequencies of the wet vertical two and three node vibration modes became $5.1\,Hz$ and $12.4\,Hz$, respectively. The shapes of these modes were used together with the heave and pitch modes in a modal superposition method. Four modes were thus used to obtain the results. No damping was applied to the heave and pitch modes. The damping ratios for the two and three node vertical modes were set to 0.8 % and 0.7 %, respectively. The pulse was directly placed on the different modes in a transient dynamic analysis in ANSYS. In this way, the vertical bending moments were obtained.

Participant C changed the added mass of their model in order to tune the natural frequencies to those of the physical model. The revised model had natural frequencies of the wet vertical two and three node vibration modes equal to $5.1\,Hz$ and $11.8\,Hz$, respectively. The response was computed using Participant C's 3D wet finite element model in, again, a transient dynamic analysis with ANSYS. All the modes of rigid body motion were kept free. Distributed spring boundary conditions were applied to represent the hydrostatic stiffness. The damping ratios for the two and three node vertical modes were set to 0.8 % and 0.7 %, respectively. No damping was applied to the rigid body modes.

Participant F updated the model by adjustment of the mass distribution. The revised natural frequencies of the wet vertical two and three node vibration modes were $5.1\,Hz$ and $11.8\,Hz$, respectively. The corresponding damping ratios were 0.8 % (two node vertical mode) and 0.7 % (three node vertical mode). No damping was applied to the rigid body modes of the vessel. To obtain the vertical bending moments, an implicit dynamic analysis was done in Abaqus. The full finite element model was applied with the pulse loading distributed to the nodes along the centerline at the foremost section, which comprises 10 % of the length of the vessel.

The fatigue loading was calculated by the benchmark organizer using the Miner–Palmgren linear cumulative damage rule implemented in the form of Equation 2:

$$D = \sum_{i}^{n} \Delta VBM_i^m \tag{2}$$

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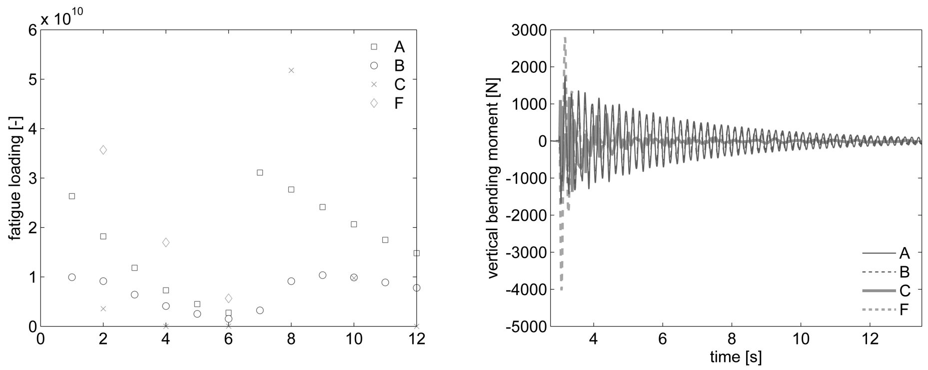

Figure 4: Fatigue loading per pulse (left) and example of time series of vertical bending moment (right). The example on the right is given for a pulse with a duration of half of the natural period of the two-node vibration mode.

where D is the accumulated fatigue loading and m is the slope parameter of the SN-curve, n is the total number of stress ranges, and ΔVBM_i is the vertical bending moment range. The slope parameter m was assigned a value of three. The bending moment ranges were determined using rainflow counting (Rychlik, 1987). The results are presented in the left plot of Figure 4. The numbers on the horizontal axis represent the type of pulse. Numbers 1 through 6 correspond to pulse durations of 0.25, 0.50, 0.75 1.00, 1.25 and 1.50 times the two node vibration period, respectively. Numbers 7 through 12 correspond to pulse durations defined by the three node mode vibration period and the same multipliers.

From the figure it can be seen that there was a considerable difference in results. Differences of up to a factor of five were seen. To investigate this difference, a typical time series of the obtained vertical bending moments is shown in the right plot of Figure 4. The figure shows that results from Participant A and B were well in line. One reason for a difference between the two could be the fact that Participant A used more flexural modes than Participant B. The available energy was then distributed over more modes, where the higher modes damp much more rapidly than the lower modes. The damping in the model of Participant F can be seen to be much higher than that of the other participants. The initial response is also significantly higher.

4.3 Response to Regular Head Waves

The third stage of the benchmark focused on a whipping analysis excited by a regular head wave with a wave height of about $5.7\,m$ and a period of $11\,s$. The model had a full scale speed of $25\,kn$. The measured wave was provided to the participants who were then asked to predict the impulsive vertical force on a defined bow area in a first step (Stage 3.1a). The measured and calculated force time histories are shown in Figure 5.

The computation of the slamming loads of Participant A was performed using twenty 2D slamming sections and a boundary element method.

Participant C applied two different approaches. Method C-1 was a hybrid approach for slamming load computation where initially vessel motions were computed using a Rankine panel method (SWAN). These motions were imposed on a model in a Reynold's Average Navier Stokes Solver (ANSYS CFX), calculating the impulsive

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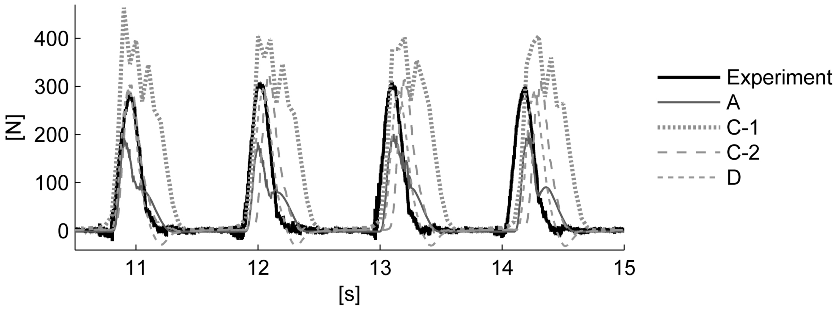

Figure 5: Measured and calculated vertical force time histories at bow area with measured wave only provided to participants

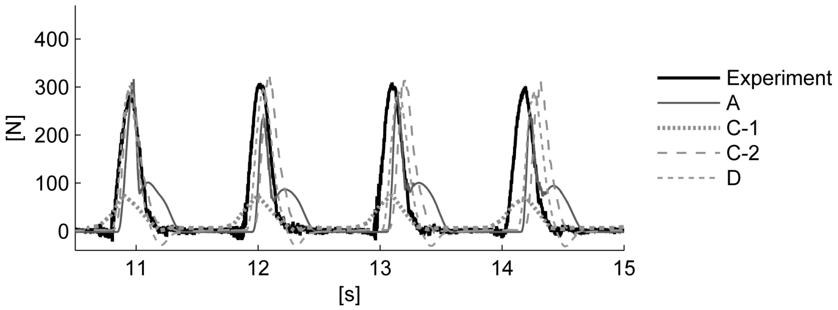

Figure 6: Measured and calculated vertical force time histories at bow area with measured wave and motions provided to participants

pressure. For Method C-2, also the motions were calculated using a RANS solver (StarCCM).

As described above, Participant D applied a fully coupled fluid-structure interaction computation based on a RANS solver (Comet) for fluid dynamics and a Timoshenko beam for structural representation.

Good agreement in amplitude and impulse shape between measured and calculated vertical force was achieved by Participants/Methods C-2 and D. To achieve better agreement, also for the two other methods, the measured motions were provided in a second step (stage 3.1b). The recalculated results are shown in Figure 6.

The agreement of the amplitude between calculation and measurement was improved by Participant A after considering the measured motions. The impulse shape still shows a second less sharp peak per wave period following the primary impulse. No improvement was achieved for Method C-1. Table 3 shows the comparison of the impulses emphasising the good agreement for method A, C-2 and D.

In a next step, the measured pressures and the impulsive force was provided to the participants. They were then asked to predict the vertical bending moment amidships. The comparison is done for the overall vertical bending moment neglecting the still

Table 3: Impulse in [Ns]

Experiment	A	C-1	C-2	D
50,8	49,7	20,8	50,3	45,6

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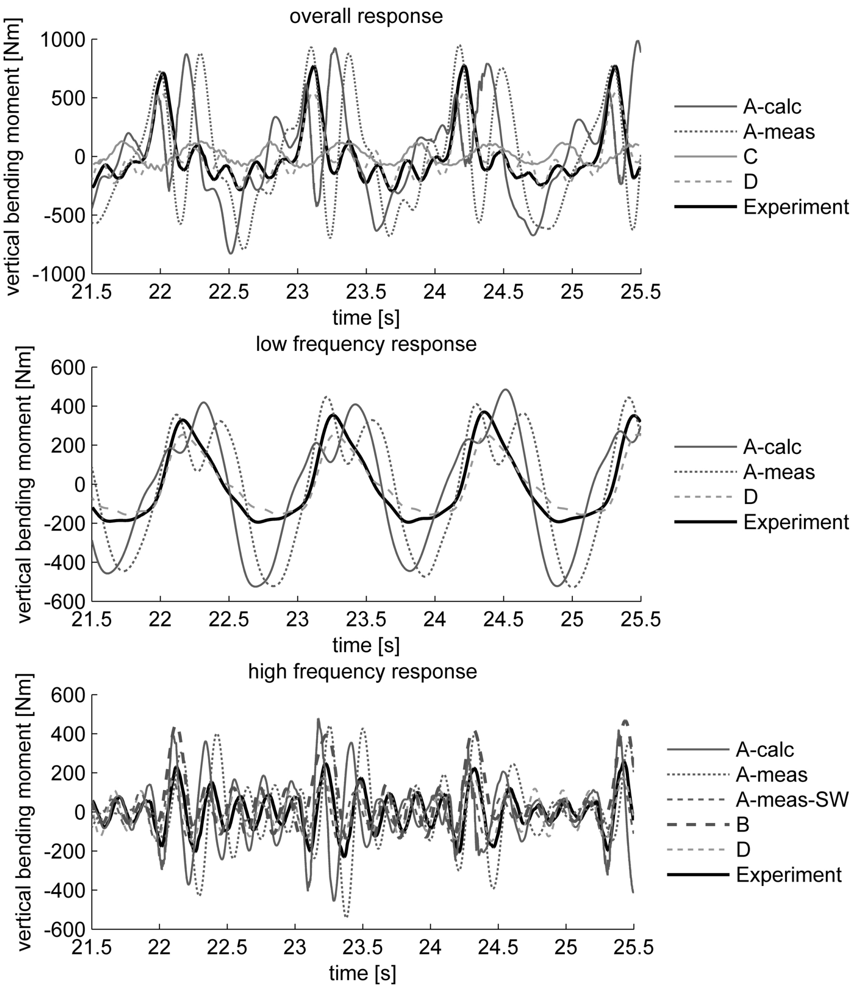

Figure 7: Measured and calculated bending moments amidships

water moment, the low frequency component representing the wave bending moment and the high frequency component representing the dynamic amplification caused by the hull girder vibration (Figure 7).

The curves A-calc and D were the bending moments directly taken from the participants' computation of Stage 3.1b. The agreement between the experiment and Participant D was good. Participant A furthermore included the measured impulse in his computation. Curve Ameas was achieved by applying the impulsive force in waves while the bending moment for curve A-meas-SW was calculated in still water. The responses for A-calc and A-meas were suffering from the second impulse following the primary impulse as shown in Figure 5. The agreement for method A-meas-SW with the experiment was good as well. Participants B and C did not include the regular wave in their computations, but applied a sequence of the measured force time history on their model. The high frequency amplitude was overestimated by Participant B. Because of the poor agreement and for better illustration, the results of Participant C were shown for the overall response only. Whereas the computation of Participant

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B took only about 3 hours, the analysis computation times for participant C and D's were more than one day.

4.4 Conclusions from Benchmark Study

From the results presented for the present benchmark, it may be concluded that the shapes and frequencies of the two and three node, dry and wet and horizontal and vertical flexural vibration modes determined by the participants, were well in line with experimental results for four of the six participants. When participants applied different realistic but analytical pulses to their model, significant differences up to a factor of five were found. On time series level, two of the four participants had results that correlated well. No experimental results were available here.

Also, the computations considering an impulse induced by a regular head wave showed significant differences between the experiment, the different participants, and applied methods. It can be concluded from the benchmark results that the more sophisticated the applied method in terms of degree of coupling and computation time, the better the agreement with the experiment.

5 CONCLUSIONS

The technical literature over the past three years has revealed that springing and whipping contribute significantly to fatigue damage of ships. Furthermore, the combination of whipping and primary nonlinear wave bending moment was found in several investigations to exceed IACS rule values. The literature also demonstrated that hull vibratory responses to waves may be important in modes other than vertical bending, specifically torsion and lateral bending. It is recommended that wave-induced vibrations be considered during the design phase.

Seamanship, both through weather routing avoidance and as applied through voluntary speed reduction, involuntary speed reduction, and heading changes, is revealed in the literature to be an important factor acting to mitigate wave-induced vibrations, particularly whipping. Therefore, lifetime exposure analyses that do not account for these seamanship effects are likely to overestimate both extreme loads and fatigue damage accumulation.

Most of the fatigue assessments in the technical literature considers only the increase in the number of cycles due to ship vibration and does not investigate the effect on fatigue crack propagation. It is recommended that future research be extended to include appropriate perspectives from fracture mechanics.

Some of the technical literature in the current period reports that extreme combined loading (nonlinear wave bending plus whipping) is, for some ships, higher in bow-quartering seas than in head seas.

Some of the researchers in the current period recommend investigation of damping based on full-scale measurements.

The 'blind' benchmark study carried out by the committee revealed both cause for encouragement and caution. Many, but not all, of the participants and methods were able to achieve reasonable correlation with the model test experiment data. Differences of up to a factor of five, however, were obtained between some predictions and corresponding experimental data. Damping assumptions were revealed to be quite important and appeared to account for much of the difference detected in fatigue loading. The benchmark study demonstrated the need for each practitioner to verify and validate their own practice, even if using industry standard tools. It is recommended that class societies and JIPs encourage validation practice by making 'truth' data available.

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It was observed in this benchmark study that the more sophisticated the applied method, the better the agreement with experiment; however, this agreement comes at a cost of increased computation time.

In both ship and offshore investigations, the literature reveals a trend towards increasing application of RANS methods and tight coupling with structural dynamics models.

Wavelet analysis methods are increasingly used to isolate transient whipping events.

The tank wall boundary condition has been revealed to have a potentially significant effect on the loads and responses measured in backbone model experiments. CFD has been demonstrated to be capable of efficiently evaluating this effect.

The first research steps have been undertake to develop a general procedure capable of correcting segmented model data for the load segmentation and the reduction in structural complexity when compared to real ships.

An efficient design loads generator has been developed that uses the acceptance-rejection method to generate an ensemble of phase sets that reproduce an *a priori* specified extreme value distribution. It has been demonstrated that this can be used to approximate the lifetime exposure for combined nonlinear wave bending and whipping.

Ratification of ILO MLC 2006 is expected soon, to be followed by national laws implementing its provisions. The convention specifies requirements to prevent exposure to hazardous levels of noise and vibration. IMO Resolution A.468 (XII) 1982 is commonly understood to establish a compliance level for noise. ILO MLC 2006 does not in itself define limit values for vibration exposure. Such limits will, presumably, be established by the national legislation implementing ILO MLC 2006. To ensure clarity and avoid interpretations of the convention's compliance, the Committee encourages concretisation of acceptance criteria for vibration.

The case of a LNG tank excited by a pressure wave from an underwater explosion has been considered. Due to the large free surface area, it was determined that a number of internal and external sloshing frequencies are excited.

When modelling machinery-induced noise and vibration, it is demonstrated that in some cases it may be possible to exclude the deck and hull plating with negligible loss of fidelity. This same study determined that non-uniform transverse frame spacing could be effective in reducing the transmission of noise and vibration.

Internal flow induced vibrations have been reported on flexible and rigid pipelines and risers, and may cause fatigue and severe damage to the pipes themselves or to the supporting structure's integrity. Internal flow is capable of inducing singing and behaviours described as whipping. Some current technical literature reports coupling between internal fluid flow and VIV induced by the external current flow.

Current induced vibrations in the form of VIV and galloping produce hydrokinetic energy. There are on-going efforts to achieve practical conversion into useful electrical energy.

There continues to be research interest into ringing in TLP tethers.

There is renewed interest in ice-induced vibrations. This report summarizes both recent research, and the standards and acceptance criteria for ice-induced vibrations.

Airborne and underwater radiated noises are both of increasing concern. Underwater-radiated noise has a potential adverse impact on marine mammals, and can also interfere with acoustic operations of the offshore oil and gas industry.

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A broad array of inspection and monitoring programs are now applied offshore, based variously on acoustic emission, measurement of residual magnetic field, optical sensing, and visual monitoring with cameras. Fiber optic technologies are showing increased application and success.

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